Mounting Optics in Optical Instruments Second Edition



Paul R. Yoder, Jr.

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Dedication

I gratefully and lovingly dedicate the second edition of this book to Betty, my late best friend and wife for over 58 years, and to our children: David, Marty, Carol, and Alan. Over the years, they have all encouraged me to continue writing technical books and teaching short courses because I really enjoy such efforts. Further, I cannot as yet see my way clear to abandon my adopted field of optomechanics to just sit on the sun porch or in a rocking chair by the fireplace and let the rest of the world have all the fun.

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Preface to the Second Edition

This second edition of *Mounting Optics in Optical Instruments* updates and expands the prior discussions of pertinent technologies for interfacing optics with their mechanical surroundings in optical instruments. The general format of the first edition is maintained, but some topics are repositioned to fit better into the contexts of the various chapters. Two new chapters—one with expanded coverage of the design, fabrication, and mounting of metallic mirrors, and another dealing with aligning single and multiple lenses and reflecting optical systems—have been added.

The entire text of the book has been rewritten to help clarify many technical details, to correct some misleading statements in the earlier version, and to add new material. All equations that carry over from the first edition have been checked and a few corrections made. New equations have been added as appropriate to enhance the technical content of the new edition. As Jacobs¹ once said: "it is not possible to make drawings that clearly show the functioning of optical instruments without exaggeration of some details. In some cases, these exaggerations lead to technical absurdities." I also believe that, in a work of this sort, the primary purpose of a drawing can be to instruct rather than to be an exact representation of an original. For this reason, I have not hesitated to exaggerate drawing details whenever appropriate for the sake of clarity.

Specific major improvements in this edition are as follows:

- In Chapter 1 (Introduction), useful information regarding stress-induced birefringence and radiation effects in glasses has been added. Discussions of environmental effects on optics and on optical instruments are expanded. A basic procedure for tolerancing optics is outlined, and the possible effects of tightening tolerances for typical component parameters on costs of those components are indicated. Key techniques for making mechanical parts for optical instruments are summarized. The number of figures has grown ~400%.
- In Chapter 2 (Optic/Mount Interface), the important topic of centering optics in their mounts is significantly expanded. Various techniques that can be used to measure lens centration errors are explained. Basic techniques for sealing instruments statically and dynamically are illustrated. The number of pages has grown by 67%, and the number of figures has increased by ~33%.
- In Chapter 3 (Mounting Single Lenses), a new method is suggested for estimating the appropriate axial preload on lenses when those lenses are not otherwise constrained radially and are exposed to transverse accelerations. Techniques for estimating the weights and the locations of centers of gravity for lenses of different configurations are outlined and illustrated with examples. Methods for determining the annular thicknesses required in athermal elastomeric ring mountings for circular optics are outlined and the significance of the elastomer's Poisson's ratio in these calculations is explained. The size of the chapter and the number of figures have remained constant, but the number of equations has increased by ~33%.
- Chapter 4 (Mounting Multiple Lenses) now includes descriptions of hardware designs for a large astrographic objective, assemblies featuring lenses mounted in

poker-chip fashion, and optomechanical designs for high acceleration applications. Details are added regarding various photographic lenses, all-plastic lens assemblies, and mechanisms used to focus lenses and to change (i.e., zoom) their focal lengths. Page and figure counts have increased by \sim 22% and \sim 49% respectively.

- In Chapter 5 (Mounting Windows, Filters, Shells and Domes) we now include examples of designs in which the optic contours conform more or less to the skin configurations of the structure. A design for a fail-safe dual-pane window suitable for photographic use in a commercial aircraft also is referenced. The size of this chapter has increased 20%, and the figure count grew 53%.
- Chapter 6 (Prism Design) once again shows designs for various prisms and includes several types not previously included. The page count and the number of figures have increased more than 20%.
- Chapter 7 (Mounting Prisms) is basically unchanged from the corresponding chapter in the first edition.
- Chapter 8 (Mirror Design) now includes additional information on image orientation control, the layout of simple two-mirror periscopes, silicon and metallic foam-core mirrors, the adaptive secondary mirrors for the Large Binocular Telescopes, the beryllium secondary for the Very Large Telescope, and the James Webb Space Telescope segmented primary. Page count is increased by ~45%, the number of figures by ~62%, and the number of equations by ~44%.
- In Chapter 9 (Mounting Smaller Mirrors), we have added descriptions of mountings for small circular mirrors with multiple discrete bond joints to structure on the mirror's back surface and on its rim. Equations given previously for design of a 9-point Hindle mount for axial support of circular solid mirrors have been augmented to allow the nominal design of an 18-point mount. Page and figure counts have increased slightly.
- Chapter 10 (Mounting Metallic Mirrors) is expanded significantly as compared to the treatment of this subject as a section of Chapter 9 in the first edition. A much more detailed treatment of the use of single point diamond turning (SPDT) fabrication techniques is now included. Several additional examples of hardware designs are described. Many of these designs feature flexures that isolate the optical surface from forces delivered by the mounting. Published developments of platings for metallic mirror surfaces are summarized briefly and some effects of key types on mirror performance under temperature changes are indicated. Subject matter coverage, as measured by either page count or the number of figures, has increased manyfold.
- Chapter 11 (Mounting Larger Mirrors) has been reformatted to group designs into axis horizontal, axis vertical, axis variable, and space borne applications. Many of the included designs depict key developments that have allowed significant performance enhancement and size growth in astronomical telescope systems. The page count and number of figures for this important topic are both increased by $\sim 30\%$.

- Chapter 12 (Aligning Lens and Mirror Systems) is a new chapter amplifying the
 material previously in sections of Chapters 3 and 4. New topics include the use of a
 modified alignment telescope and of a Point Source Microscope^{*} to align individual
 and multiple lenses. Also added are descriptions of an extremely precise method for
 aligning very high performance microscope objectives and of a method for
 determining which components to adjust during final assembly to optimize
 performance of complex systems. Page count and the number of figures are
 increased by ~300% and ~400% respectively.
- In Chapter 13 (Estimating Mounting Stresses), previously published research leading • to the now generally accepted rule-of-thumb limit, or tolerance, of 1000 lb/in.² (6.89 MPa) for tensile stress created in a typical glass optic by applied mounting force is summarized. The effects of surface flaws, such as scratches or subsurface cracks, on this tolerance also are indicated. If the worst-case condition of the surfaces on the optic is known or can be estimated, the useful lifetime of the optic can be predicted statistically. As in the prior edition, computational methods, many utilizing equations developed by Roark² for peak compressive stresses generated in the contacting optical and mechanical members, are applied to various types of mechanical interfaces with optical components. These computations are extended in this edition by utilizing theory from Timoshenko and Goodier³ to quantify the corresponding tensile stresses in the optic. We then show how the suitability of a given optomechanical mounting design can be determined by comparison of these stress levels with the rule-of-thumb tolerance. The scope of the subject matter treatment in this edition (measured by page count and numbers of figures and equations) has slightly changed from that in the prior edition.
- In Chapter 14 (Temperature Effects), we have extended the previously published discussion of how temperature changes affect axial and transverse mounting forces. Several pertinent factors not considered in the first edition are defined. Some, but not all, of these can be quantified using available theory. In the absence of complete methodology for quantifying temperature effects on any given hardware design, we now advocate the provision of a controlled amount of compliance in the mechanical design of that hardware so as to minimize these temperature effects. Several typical practical design examples are considered. The page count of this chapter is expanded by >36% while the number of figures is increased by >46%.
- Chapter 15 (Hardware Examples) continues the practice established in the first edition of discussing the optomechanical designs of selected hardware items to illustrate many of the topics considered in the text. In this edition, twenty such examples are given while, in the prior edition, there were thirty. This, however, does not represent a reduction in the book's total technical scope because some new examples have been added to this chapter, and many of the previous examples are now discussed in the context of the pertinent technology in earlier chapters.
- Appendices A and B to the new edition provide unit conversion factors and some updated values for properties and other characteristics of the materials used in optomechanical design. As before, Appendix C derives the torque-preload

^{*} A new device offered by Optical Perspectives Group, LLC of Tucson, AZ.

relationship for a threaded retaining ring. It is often helpful to know early in the design phase how an optical component, subassembly, or complete instrument will eventually be tested to prove its suitability to withstand adverse environmental conditions. Appendix D, paraphrased from ISO Standard 9022,⁴ summarizes test methods that might be applied to simulate various environments.

• Once again, a CD-ROM is provided with this book so the reader can access Microsoft Excel worksheets that use the ~250 equations given in the text to solve the numerical examples intermingled with the technical discussions as well as to design prisms and prism assemblies that are described here. The worksheets are configured so new input data can be inserted to create new designs or to conduct parametric analyses.

I acknowledge the contributions of the many friends and associates who provided new information to this book or helped me clarify confusing matters previously presented. In particular, thanks are offered to Daniel Vukobratovich and Alson E. Hatheway who have helped me understand many pertinent intricacies of optomechanical design. On the editorial side, I sincerely thank Merry Schnell and Scott Schrum, who helped me straighten out editorial details and kept the production schedule moving at SPIE Press. While all contributors tried valiantly to help me present the technical material clearly and correctly, total responsibility for any errors that remain rests on my shoulders. Finally, I sincerely hope this book proves useful to all its readers.

- 1. Jacobs, D.H., Fundamentals of Optical Engineering, McGraw-Hill, New York, 1943.
- 2. Roark, R.J., Formulas for Stress and Strain, 3rd ed., McGraw-Hill, New York, 1954.
- 3. Timoshenko, S.P. & Goodier, J.N., *Theory of Elasticity*, 3rd ed., McGraw-Hill, New York, 1970.
- 4. ISO Standard 9022, *Environmental Test metho ds*, International Organization for Standardization, Geneva.

Paul R. Yoder, Jr. Norwalk, Connecticut June 2008

Preface to the First Edition

This work is intended to provide practitioners in the fields of optical engineering and optomechanical design with a comprehensive understanding of the principal ways in which optical components such as lenses, windows, filters, shells, domes, prisms, and mirrors of all sizes typically are mounted in optical instruments. It also addresses the advantages and disadvantages of various mounting arrangements and provides some analytical tools that can be used to evaluate and compare different optomechanical designs. The presentation includes the theoretical background for some of these tools and cites the sources for the most of the equations listed. Each section contains an illustrated discussion of the technology involved and, wherever feasible, one or more worked-out practical examples.

Two chapters deal with the fundamentals of design for optical components. These are Chapter 6 on prism design and Chapter 8 on mirror design. These topics are considered appropriate, and indeed necessary, as background for considering how best to mount these very important types of optics.

The book is based, in part, on short courses entitled *Precision Op tical Component Mounting Tec hniques* and *Principles f or Mo unting Optical C omponents* offered by SPIE-The International Society for Optical Engineering-that I have had the privilege of teaching over a period of several years. Many, but not all, of the techniques for mounting optics covered here have been presented previously in the tutorial texts *Mounting Lenses in Op tical Instruments*¹ and *Design and Mo unting of Pri sms a nd S mall Mirrors i n Optical Instruments*² as well as in my earlier reference book *Opto-Mechanical Systems Design.*³ Several recent designs for mounting optics are included here to broaden the coverage and to bring the material more nearly up to date. Coverage of window-type optics and of large mirrors has been expanded over the previous works.

Wherever possible, numerical values given in this book are expressed in both the metric or Système International (SI) units and the units in customary use in the United States and Canada. The latter are abbreviated in this book as "USC" as in some recent textbooks. Examples taken directly from the literature may be expressed only in the system used by the original author. Units can be easily changed from one system to the other through use of the conversion factors given in Appendix A.

All the designs discussed here are drawn from the literature, my own experiences in optical instrument design and development, and the work of colleagues. I acknowledge with my deepest thanks the contributions of others, including the many participants in the above-mentioned SPIE short courses and the readers of my previous books, and sincerely hope that I have accurately recorded and explained the information they have given to me. I acknowledge and thank Donald O'Shea and Daniel Vukobratovich, who reviewed the manuscript for this book and suggested many improvements. I also thank Mary Haas, Rick Hermann, and Sharon Streams for their outstanding copy editing and editorial suggestions. While these people helped me to present the material clearly and correctly, I am solely responsible for and deeply regret any errors that remain. One particularly annoying error is that the headings on even numbered pages differ from the actual title of the book!

The mounting stress theories discussed in Chapter 11 are considered to be conservative approximations. They are intended to indicate whether a given design can be judged to be adequate from a stress viewpoint or if it should be analyzed by more elaborate finite-element and/or statistical techniques. The same is true of the treatment of temperature effects on axial preload in Chapter 12. These topics would benefit greatly from further investigation, refinement, and (it is hoped) verification by other workers based on more precise computational methods, such as finite-element analysis. I would welcome comments, corrections, and suggestions for improvements in the presentations of these topics and/or in any other portion of this book.

A feature included with this book is a CD-ROM containing two Microsoft Excel worksheets that allow convenient use of the many equations given in this text to solve typical optomechanical interface design and analysis problems. Some of these equations are relatively complex, so the worksheets have been developed to facilitate equation use and to reduce the chance of improper parameter application. The 102 files included in each worksheet correspond to designs and/or numerical examples worked out in the text. Input values pertaining specifically to those examples are listed. The two worksheets on the disk are different versions of the same program. In Version 1, data inputs are in U.S. Customary units while in Version 2 inputs are in metric units. In both cases, all data are presented in both sets of units. A table of files (with hyperlinks) is provided in each worksheet to assist in finding the proper file for a specific computation. The examples in the text are cross-referenced to the applicable worksheet files. Custom solutions to problems similar to the examples in the text can be obtained by revising the input data in the file as appropriate for the case to be evaluated. The program will then automatically solve the problem using those inputs and the appropriate equations from the text. This tool should be especially useful when parametric analysis of variations of key parameters is needed to obtain an optimum design.

I sincerely wish for the users of this book and of the CD-ROM a deepening understanding of the technologies discussed and success in the application of the concepts, designs, and analysis techniques presented here.

1. Yoder, P.R., Jr., *Mounting Le nses i n O ptical I nstruments*, TT21, SPIE Press, Bellingham, 1995.

2. Yoder, P.R., Jr., Design and Mounting of Pri sms and S mall Mi rrors in O ptical Instruments, TT32, SPIE Press, Bellingham, 1998.

3. Yoder, P.R., Jr., *Opto-Mechanical Systems Design*, 2nd ed., Marcel Dekker, New York, 1993.

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Terms and Symbols

This list of the terms and symbols used in this book is intended to help the reader sort through the shorthand language of the various technical topics and the equations used to express the relationships so useful in the design process and the analysis of designs. The author has attempted to be consistent in the use of symbols for variables throughout the text, but there are occasions where the same symbol has more than one meaning. To some extent, customary usage in the field of optomechanics has dictated the use of a specific term or terms, The symbol α is a good example since it is used to represent the coefficient of thermal expansion for a material when the common abbreviation CTE is not appropriate, as in equations. Subscripts are frequently used to identify the specific application of a symbol to a specific material (as in the use of α_M to designate the CTE for a metal as distinguished from α_G for a glass). We list here fundamental parameters and their units, frequently used prefixes, Greek symbol applications, acronyms, abbreviations, and other terms found in the text. Symbols representing variables are italicized in the equations.

Parameter	SI or metric	U.S. and Canadian
	1	0 J
Angle	rad, radian	°, degree
Area	m^2 , square meter	in. ² , square inch
Conductivity, thermal	W/mK, watt/meter-kelvin	Btu/hr-ft-°F, British
		thermal unit per hour-
	2	foot-degree Fahrenheit
Density	g/m ³ , gram per cubic meter	lb/in. ³ , pound per cubic inch
Diffusivity, thermal	m ² /s, meter squared per	in. ² /s, inch squared per
	second	second
Force	N, newton	lb, pound
Frequency	Hz, hertz	Hz, hertz
Heat	Btu, British thermal unit	joule (J)
Length	m, meter	in., inch
Mass	kg, kilogram	lb, pound
Moment of force (torque)	N/m, newton-meter	lb-ft, pound-foot
Pressure	Pa, pascal	lb/in. ² , pound per square
		inch
Specific heat	J/kg-K, joule/kilogram-Kelvin	
		unit per pound-degree
		Fahrenheit
Strain	μm/m, micrometer/meter	µin. per in., microinch
		per inch
Stress	Pa, pascal	lb/in. ² , pound per square
		inch
Temperature	K, kelvin; °C, degree Celsius	°F, degree Fahrenheit
Time	s or sec, second	s or sec, hr, second, hour
Velocity	m/s or m/sec, meter/second	mph, mile per hour
Viscosity	P, poise; cP, centipoise	lb-s/ft ² , pound-sec per
-	-	square foot
Volume (solid)	m ³ , cubic meter	in. ³ , cubic inch

Units of Measure

Prefixes

mega	М	million
kilo	k	thousand
centi	c	hundredth
milli	m	thousandth
micro	μ	millionth
nano	n	billionth

Greek Symbol Applications

α	CTE; angle
β	angle; term used in equation for shear stress in a bonded optic
β _G	rate of change in refractive index with change in temperature (dn/dT)
	shape factor for a resilient pad in a prism mounting
γ γ-	thermo-optical coefficient for a glass
γ _G δ	decentration of an elastomeric-supported optic; ray angular deviation
δ _G	glass coefficient of thermal defocus
Δ	spring deflection; finite difference (change)
$\Delta_{ m E}$	eyepiece focus motion per diopter
η	damping factor
θ	angle
λ	wavelength; thermal conductivity in Schott catalog
μ	Poisson's ratio in Schott catalog
μ_M, μ_G	coefficient of friction for metal, glass
ξ	ratio of shortest to longest dimensions of a rectangular mirror; rms acceleration
-	response
π	3.14159
ρ	density
σ	standard deviation
Σ	summation
σi	tensile yield strength of components in a bonded joint
υ_1	Poisson's ratio; with subscript representing wavelength, Abbe number
-	angle, cone half-angle
φ	ungio, cono nun ungio

Acronyms and Abbreviations

А	aperture of an optic, face width of a prism; area
a, b, c, etc.	dimensions
-A-, -B-, etc.	reference feature designation on a drawing
A/R	antireflection
A _C	area of elastically deformed region at an interface
a _G	acceleration factor (interpreted as "times ambient gravity")
ANSI	American National Standards Institute
ASME	American Society for Mechanical Engineering
A _T	annular area of a thread
AVG	as subscript, indicates average value
Aw	unsupported area of a window
AWJ	abrasive water jet (Corning designation)

AXAF	Advanced X-ray Astrophysical Facility (now Spitzer Space Telescope)
b	flat spring width, length of cylindrical pad
С	Celsius; as subscript, indicates circular shape for a bond; center of
	curvature
C, d, D, e, F, g, s	as subscripts, refer to wavelengths of Fraunhofer absorption lines
CA	clear aperture
CAD, CAM	computer aided design, manufacturing
CCD	charge coupled detector device
CG	center of gravity
C _K	mirror mount type factor used to determine gravitational effect
CLAES	Cryogenic Limb Array Etalon Spectrometer
CMC	carbon matrix composite
CNC	computer numerically controlled
C _p	specific heat
cP	centipoise (unit for viscosity)
C_{R}, C_{T}	spring constants in radial, tangential directions
CRES	corrosion-resistant (stainless) steel
Cs	compressive stress in a mechanical pad
C _T	center of curvature of a toroidal surface
CTE	coefficient of thermal expansion
CVD	chemical vapor deposited
CYL	as subscript, indicates cylindrical shape
d	major diameter of an internal thread
D	thermal diffusivity, diopter, major diameter of an external thread
D _B	diameter of a bolt circle
D_G	outside diameter of a circular optic
DIEMOS	Deep Imaging Multi-Object Spectrograph
D _M	inside diameter of a mechanical part, such as a cell
dn/dT	rate of change in refractive index with change in temperature
DOF	degrees of freedom
D _P	diameter of a resilient pad
D _R	OD of a compressed snap ring
D _T	pitch diameter of a thread
E, E_G, E_M, E_e	Young's modulus, for glass, for metal, for elastomer
Ε/ρ	specific stiffness
ECM	electro-chemical machining (process for contouring metal)
EDM	electric discharge machining (process for contouring metal)
EFL ELN	effective focal length (as of a lens or mirror) electroless nickel plating
ELN EN	electrolytic nickel plating
EROM	erasable programmable read only memory
ERO	edge run out
ESO	European Southern Observatory
EUV	extreme ultraviolet radiation
f	focal length
F	force, Fahrenheit temperature
f, f_E, f_O	focal length (see EFL), of an eyepiece, of an objective
FEA	finite element analysis
FIM	full indicator movement (replaces TIR)

FLIR	forward looking infrared sensor
f_N	natural frequency of vibration
fs	factor of safety
FUSE	Far Ultraviolet Spectroscopic Explorer
g	acceleration due to ambient gravity (see a_G)
GAP _A , GAP _R	axial and radial gaps between surfaces of an optic and its mount
GEO	Geosynchronous Earth orbit
GOES	Geostationary Operational Environmental Satellite
Gy	abbreviation for unit of radiation dose (gray)
Н	thread crest-to-crest height, Vickers hardness of a material
HeNe	helium-neon laser
HIP	hot isostatic pressing
HK	Knoop hardness of a material
HRMA	high resolution mirror assembly (in the Chandra Space Telescope)
HST	Hubble Space Telescope
i	paraxial tilt angle of a plane parallel plate; as subscript, i th component
I, I'	angle of incidence, refraction
I, I ₀	beam intensity after, before an interface
ID	inside diameter
IPD	interpupillary distance
IR	infrared
IRAS	Infrared Astronomical Satellite
ISO	International Organization for Standards
J	strength of an adhesive bond
JWST	James Webb Space Telescope
K	stress optic coefficient
k K	thermal conductivity
K, K _s	Kelvin temperature; stress optic coefficient
K_1, K_A , etc.	constant term in an equation
KAO	Kuiper Airborne Observatory
K _C	fracture toughness (strength) of a brittle material
L	length of a spring that is free to bend; width or diameter of a bond
L1, L2 LACEOS	Lagrange points 1 and 2 (Sun/Earth/Moon orbit)
LAGEOS LEO	Laser Geodynamic Satellite Low Earth orbit
LEO L _{j,k}	axial length of a lens spacer between surfaces i and j
L _{J,k} LLTV	low light-level television
$\ln(x)$	natural logarithm of x
LOS	line of sight
lp	line pair (element for resolution measurement, as in lp/mm)
LRR	lower rim ray at maximum semifield angle
m	mass, reciprocal of Poisson's ratio
MEO	middle Earth orbit
MIL-STD	U.S. Military Standard
MISR	Multiangle Imaging Spectro-Radiometer
MLI	multilayer insulation
MMC	metal matrix composite
MMT	Multiple Mirror Telescope
MTF	modulation transfer function
Ν	newton; number of springs

n, n _{ABS} , n _{REL}	refractive index, refractive index in vacuum, refractive index in air
n_{\parallel}, n_{\perp}	refractive index for parallel or perpendicular polarized light component
N	number of springs; as prefix in Schott glass name, indicates "new"
NASA	National Aeronautics and Space Administration
n _d	refractive index for $\lambda = 546.074$ nm
N_E, N_1, N_2	number of threads per unit length of differential thread
	refractive index for a specific wavelength
n _λ OAO-C	Orbiting Astronomical Observatory-Copernicus
OD OD	outside diameter
OFHC	oxygen-free high-conductivity (designation of a variety of copper)
OPD	optical path difference
OTF	optical transfer function
P	preload force; optical power
-	thread crest-to-crest pitch; linear preload
p P _F , P _S	statistical probability of failure, of survival (or of success)
\mathbf{P}_{i}	preload force per spring
-	parts per million
ppm PSD	power spectral density
PTFE	polytetrafluroethylene (Teflon)
p-v	peak to valley
-	heat flux per unit of area
q Q	torque; bond area
Q _{MAX}	maximum bond area within prism or mirror face dimension
-	minimum bond area to provide needed joint strength
Q _{MIN} r	snap ring cross sectional radius
R	surface radius
	radius of elastically deformed region at an interface
r _C RH	relative humidity
	root mean square
rms roll	component tilt about transverse axis
	radius to center of a spacer
r _s R _s	reflectance
R _S R _T	cross-section radius of toroidal surface
RT	as subscript, indicates racetrack shape for bond area
R _T	radius of a toroid
RTV	room temperature vulcanizing sealant
R_{λ}	reflectance of a surface at wavelength λ
S _{AVG}	average contact stress in an interface
S _B	stress in a bent mechanical part, such as a spring
S _{C CYL} , S _{C SPH}	compressive stress from contact with a cylindrical or spherical pad
S_{CSC} , S_{CTAN} , S_{CT}	
000, 01110, 01	mount corner
SC	as subscript, indicates sharp corner interface
S_{C}, S_{T}	compressive stress, tensile stress at an optic-to-mount interface
Se	shear modulus of an elastomer
S _f	fracture strength of a window material
SIRTF	Space Infrared Telescope Facility (now Spitzer Space Telescope)
S_i, S_k	sagittal depth of the "ith" or "jth" surface
S_{M}	tangential tensile (hoop) stress in a mount's wall
- 141	

S _{MY}	microyield strength
SOFIA	Stratospheric Observatory for Infrared Astronomy
Spad	average stress in a pad-to-optic interface
SPDT	single-point diamond turning
SPH	as subscript, indicates spherical interface
S _R	radial stress at an optic-to-mount interface
Ss	shear stress developed in a bonded joint
ST	tensile stress
Sw	yield strength for window material
SXA	a proprietary aluminum metal matrix composite
S_Y, S_{MY}	yield strength; microyield strength of a material
Т	temperature
t	thickness, as that of a flat spring
t _A	axial path length in a refracting material
T _A , T _{MAX} , T _{MIN}	assembly, maximum, minimum temperature
TAN	as subscript, indicates a tangential interface
tanh	hyperbolic tangent function
t _C	cell wall thickness
T _C	temperature at which assembly preload reduces to zero
t _E	edge thickness of a lens or mirror
te	thickness of an adhesive bond; of an elastomeric ring
TIR	total internal reflection; total indicator runout (see FIM)
TOR	as subscript, indicates a toroidal interface
tpi	threads per inch
T_{λ}	transmittance of a surface at wavelength λ
U, U'	angle with respect to the axis of a marginal ray in object, image space
ULE	Corning's ultra-low expansion glass-ceramic material
UNC, UNF	unified coarse or fine thread
URR	upper rim ray at maximum semifield angle
USC	U.S. customary system of units (the "inch" system)
UV	ultraviolet
V	volume, lens vertex
V _d	Abbe number for $\lambda = 546.074$ nm
VLT	Very Large Telescope
w W	unit applied load
	weight wall thickness of a spacer
w _s X, Y, Z	coordinate axes
Х, 1, 2 Ус	mechanical contact height on an optic (measured from the axis)
yc Ys	ID/2 for a lens mounting
13	The set of state strongering

Mounting Optics in Optical Instruments Second Edition

CHAPTER 1 Introduction

This chapter addresses general issues that typically must be considered by designers or engineers during the evolution of an optical instrument design that will meet performance requirements prescribed by the system specification, while adhering to constraints imposed by that document. Subsequent chapters delve more deeply into specific design issues involving mounting various types of optics.

We begin this chapter by reviewing some ways in which optics are used in instruments. Effective engineering design of those instruments requires advance knowledge of the adverse environments under which the product is expected to provide specified performance as well as those more severe environments it must survive without damage, so we summarize ways in which temperature, pressure, vibration, shock, moisture, contamination, corrosion, high-energy radiation, abrasion, erosion, and fungus can affect an instrument's performance and/or its useful life. We also offer some general suggestions for designing an apparatus to withstand these adverse conditions. The extreme environments to be expected on or near the surface of Earth and in space are summarized. Ways to test instruments for compatibility with those environments are reviewed. Because careful selection of materials is vital for maximizing environmental resistance and ensuring the proper operation of the product, we also review selected attributes of some of the most frequently used optical and mechanical materials. The chapter closes with brief considerations of tolerancing and manufacturing optical and mechanical components.

1.1 Applications of Optical Components

Lenses serve many functions in optical instruments. In general, they are used to form real or virtual images of large or small objects at various distances, or they redirect rays to form pupils^a of the optical system. In addition, some lenses serve as correctors that differentially refract rays over their apertures in order to modify aberrations introduced by other image-forming components.

The most common forms of lenses are objectives, relay or erecting lenses, eyepieces, field lenses, magnifiers, and corrector plates. Most lenses have polished spherical or aspherical surfaces that refract rays in accordance with Snell's law. Some lens surfaces are configured to diffract rays. Here we will limit our attention to refracting lenses since we are primarily concerned with mounting principles rather than detailed image-forming considerations. Lens diameters are limited by difficulties in making high-quality optical glass in large sizes and in fabricating precision surfaces larger than about 20 in. (0.5 m).

Windows, filters, shells, and domes usually serve one or more of the following functions:

^a A pupil is an image of the aperture (aperture stop) that limits the extent of the beam that is passing through the system.

- They separate and seal the interior of the instrument from the outside environment,
- They adapt the spectral characteristics of the transmitted (or reflected) beam,
- They correct aberrations (as in the case of the shell in a Maksutov telescope).

Shells are meniscus-shaped windows, while domes are deep shells, with apertures subtending meridional angles as large as 180 deg from their centers of curvature. Hyperhemispheres are domes that extend beyond a hemisphere.

Mirrors may have flat (plano) or curved surfaces. The latter are termed "image-forming mirrors" because they have optical power resulting from their curved reflecting surfaces. They can function similarly to lenses in the above-mentioned roles. Since refraction is not involved, chromatic aberration is absent when images are formed by mirrors.

Mirrors can be made with aperture sizes far larger than those of lenses primarily because the absence of a need to transmit light allows substantial support to be provided over the full extent of the backside of the optic rather than solely around the rim, as in most lenses. Furthermore, choosing a mirror thickness for mechanical reasons rather than optical ones can enhance the stiffness of the mirror substrate. The availability of substrate materials with suitable mechanical characteristics, such as high stiffness (i.e., Young's modulus) and/or a low coefficient of thermal expansion (CTE), also contributes to the advantage of mirrors over lenses in large sizes. The weight of the substrate poses problems for large mirrors.

The principal applications for most plano mirrors, prisms, beamsplitters, or beamcombiners, i.e., components that do not contribute optical power and hence cannot form images, are as follows:

- Deviating, i.e., bending, the system axis,
- Displacing the system axis laterally,
- Folding an optical system into a given shape or package size,
- Providing proper image orientation,
- Adjusting the optical path length,
- Dividing or combining beams by intensity or aperture sharing (at a pupil),
- Dividing or combining images at an image plane,
- Scanning a beam angularly,
- Dispersing light spectrally (as with gratings or some prisms), and
- Modifying the aberration balance of the optical system.

The number of reflections provided in a system that includes mirrors and/or prisms is important, especially in visual, photographic or video applications. An odd number of reflections produce a "left-handed" (reversed or "reverted") image that is not directly readable, while an even number gives a normal, "right-handed" image. The latter is readable even if it is inverted (see Fig. 1.1). Vector techniques, summarized by Walles and Hopkins,¹ are powerful tools for determining how a particular combination of reflecting surfaces will affect the location and orientation of an image.

$\mathbf{X} \mathbf{R} \mathbf{A} \mathbf{S} \mathbf{A}$

Figure 1.1 (a) Left- and (b) right-handed images.

1.2 Key Environmental Considerations

An essential element of any instrument-design activity is to identify the environmental conditions under which the end item is expected to perform in accordance with given specifications, as well as the extreme conditions that it must survive without permanent damage. The most important conditions to be considered are temperature, pressure, vibration, and shock. These conditions can cause static and/or dynamic forces to be exerted on hardware components that may cause deflections or dimensional changes therein. These may result in misalignment, build-up of adverse internal stresses, birefringence, breakage of optics, or deformation of mechanical parts. In some applications, "crash safety" (in which the instrument or portions of it must not pose a hazard to personnel in a violent, but otherwise survivable shock event) also is specified. Other important environmental considerations include moisture and other contamination, corrosion, abrasion, erosion, high-energy radiation, laser damage, and fungus growth. All of these conditions can adversely affect performance and/or lead to progressive deterioration of the instrument.

The intended users and system engineers should define the expected exposures of the instrument to adverse environments as early in the design process as possible so that appropriate and timely provisions can be made in the design and thereby minimize environmental effects. Potential deployment scenarios should be defined as completely as possible. Also, possible failure modes should be identified and testing planned so design weaknesses are found early and corrected.

1.2.1 Temperature

In this book, we express temperature in the Celsius (C), Kelvin (K), or Fahrenheit (F) scale as appropriate to the context of other units employed. Values in any one of these scales are converted to another scale using relationships found in Appendix A.

Key temperature effects to be considered include high and low extremes, thermal shock, and spatial and temporal gradients. Military equipment is usually designed to withstand extreme temperatures of -62° C (-80° F) to 71° C (160° F) during storage or shipment. It usually must operate adequately at temperatures of -54° C to (-65° F) to 52° C (125° F). Generally, commercial equipment is designed for smaller ranges of temperatures centered at normal room temperature of $\sim 20^{\circ}$ C ($\sim 68^{\circ}$ F). Special-purpose equipment such as a spaceborne sensor may experience temperatures approaching absolute zero (0 K,

-273.16°C, -459.69°F), while sensors intended to monitor processes within a furnace may have to operate at temperatures of several hundred degrees Celsius.

Modes of heat transfer are *conduction*, the direct communication of molecular disturbance through a substance or across interfaces between different substances; *convection*, transfer by actual motion of the hotter material; and a combination of *radiation* and *absorption*. In the latter process, heat is emitted by material at a given temperature (i.e., a "source"), transmitted through adjacent media or space, and absorbed by another material (i.e., a "sink"). All modes of heat transfer tend to change the temperatures of both the heat source and heat sink until a state of thermal equilibrium is achieved.

All three modes of heat transfer are important in optomechanical design because it is virtually impossible for any object to be totally in thermal equilibrium with its environment. Spatial temperature gradients then cause nonuniform expansion or contraction of integral or connected parts. A commonplace contemporary example is the "hotdog" effect on orbiting spacecraft structures that receive solar radiation on one side and radiate heat into outer space on the other. The hotter side expands more than the cooler side and the shape of the structure bends toward the familiar shape of a cooked sausage. Differential expansion also can occur within structures made of different materials when at a spatially uniform but temporally varying temperature.

Rapid changes in temperature occur when an electro-optical sensor on an orbiting spacecraft leaves or reenters Earth's shadow or when an amateur astronomer's telescope is taken directly from a warm room to a Vermont hillside location in February. These "thermal shocks" can significantly affect performance or even cause damage to the optics. Thermal diffusivity is a characteristic of all materials that affects how quickly parts of an optical instrument respond to temperature changes. Most nonmetallic optical materials have low thermal conductivities, so heat is not transferred rapidly through them. Many high-energy laser systems use metallic mirrors made of copper or molybdenum because their high conductivities tend to dissipate absorbed heat rapidly and hence maintain their shape.

Slower changes of temperature affect performance mainly by introducing temperature gradients, changing the dimensions of parts, or causing misalignment. Materials that have slightly inhomogeneous thermal expansion coefficients such as those sometimes used in very large mirror substrates deform under temperature changes differently at various locations within a component, which can modify the surface shape. Common effects of misalignments are deterioration of image quality as a result of focus errors or induced asymmetry of images, loss of calibration in measuring devices, and pointing errors. Gradients can degrade the uniformity of the refractive index in transmitting materials such as glass and affect performance.

Frictional skin heating of windows and domes due to rapidly flowing air may affect the thermal balance of related optical instruments in high-speed aircraft and missiles. Damage may occur at very high speeds. The use of special coatings and materials minimally sensitive to temperature in such exposed optical components tends to minimize thermal problems. Domes conforming aerodynamically to the surrounding airframe contours also reduce temperature effects. See Section 5.6.

1.2.2 Pressure

Pressure is a measure of force acting on a unit area. Typical units are the pascal (N/m^2) and pounds per square inch $(lb/in.^2)$. Fluid pressure is sometimes expressed in terms of the height in millimeters or inches of a column of water or mercury supported at a specific temperature. This latter concept is utilized in defining the normal or standard atmospheric pressure. It is the pressure exerted by a column of mercury 76-cm (29.92-in.) high at sea level, at 0°C (32.0°F); it equals 101.32 kPa (14.69 lb/in.²). Pressures in vacuum environments are frequently defined in millimeters of mercury or torr (T) where 1 T = 1 mm Hg = 0.0013 atm.

Most optical instruments are designed for use at ambient pressure in Earth's atmosphere. Exceptions are those for use in a pressurized region (such as a periscope used in a submarine) or in a vacuum (such as an evacuated ultraviolet spectrometer on Earth or a vented camera in space).

Because pressure decreases with altitude above Earth's surface, an imperfectly sealed optical instrument that is exposed to cyclic altitude changes may experience a "pumping" action in which air, water vapor, dust, or other constituents of the atmosphere seep through leaks. This may contaminate the instrument and lead to condensation, corrosion, light scatter, and/or other problems. In some instruments, the optics are sealed to the housing, but a leakage path is intentionally provided so that pressure differentials do not build up. In such cases, the leakage path is through a desiccator and a particulate filter that deter moisture, dust, and other undesirable matter from entering.

Decreasing pressure can cause some composites, plastics, paints, adhesives, and sealants, as well as some materials used in welded and brazed joints, gaskets, O-rings, bellows, shock mounts, etc., to outgas and/or offgas, especially at elevated temperatures. The effluents from these materials can be harmful to coatings, or they may deposit as contaminants on sensitive surfaces, especially in the vacuum of space. Some materials absorb water from humid environments on Earth and desorb that moisture in vacuum. This may cause contamination problems.

Low pressure in the environment can also lead to extraction of air, water vapor, and/or other gases from various cavities such as those between lenses, between the rims of lenses and their mechanical mounts, within lightweighted cores of mirror substrates, or within blind holes partially blocked by screw threads. If large cavities of this type are sealed, pressure differentials may reach sufficient magnitude to distort optical and thin mechanical surfaces. Extracted materials may also serve as a source for contamination.

Instruments moving within Earth's atmosphere or under water will experience an overpressure due to aerodynamic or hydrodynamic forces exerted on exposed optical surfaces. Fluid flow over these surfaces may be turbulent or laminar, depending on design and environmental factors such as temperature, velocity, fluid density, ambient pressure (i.e., altitude or depth), viscosity, etc.

In applications such as optical lithography used during manufacture of microcircuits, the temperature of the apparatus is usually controlled to perhaps ± 0.1 °C, but until

recently no attempt was made to control barometric pressure. Weather-induced pressure variations will change the index of refraction of the air surrounding the optics sufficiently to degrade image quality and/or to vary the system's magnification so as to introduce alignment (overlay) errors between successive mask exposures. Measurement of pressure changes and a compensating adjustment of the optical system or operation in a vacuum will minimize these adverse effects.

1.2.3 Vibration

A vibration environment involves application of periodic or random frequency mechanical forces to the instrument. We will consider each of these types of disturbance. Acceleration levels are expressed as a dimensionless factor a_G , representing a multiple of ambient gravity.

1.2.3.1 Single frequency periodic

Periodic vibrations [see Fig. 1.2(a)] are typically sinusoidal in amplitude. They tend to cause the entire instrument or portions of it to be displaced repeatedly from their normal equilibrium positions. After a short-term vibration ceases, the displaced member returns to equilibrium under the action of restoring forces that may include internal elastic forces, similar to the case of a mass attached to a spring, or gravitational forces, as in the case of a pendulous mass. Under a forced periodic vibration, the member oscillates about the equilibrium position as long as the external disturbance continues.

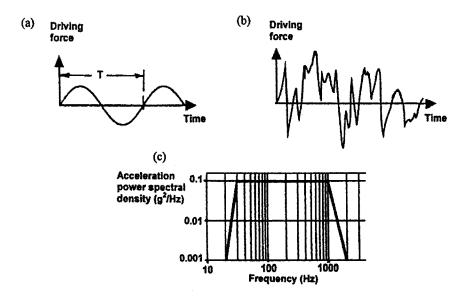


Figure 1.2 Typical vibration forms: (a) Periodic (sinusoidal), (b) random, (c) acceleration PSD.

Any physical structure has a characteristic natural or resonant frequency f_N at which it will oscillate mechanically in specific vibrational modes. Application to that structure of a driving force at or near one of these particular frequencies can cause a condition of

resonance in which the amplitude of the member's oscillation increases until it is limited by internal or external damping, or damage occurs. The following equation relates the member's approximate f_N to its mass *m* and its structural stiffness *k*:

$$f_N = \left(\frac{.05}{\pi}\right) \left(\frac{k}{m}\right)^{1/2}.$$
 (1.1)

Here, k is in newtons per meter, and m is in kilograms. To see how we might use this equation, consider Example 1.1.

Usually the designer cannot control the amplitude, frequency, and direction of the forces applied externally to an instrument, so the only corrective action possible is to make each component subsystem stiff enough so that its lowest natural frequency is higher than that of the driving forces. Preferably, f_N is higher by a factor of at least two.

Example 1.1: Estimate the resonant frequency of a prism and mount system per Eq. (1.1). (For design and analysis, use File No. 1.1 of the CD-ROM.)

Assume a prism of mass 4.85 lb (2.2 kg) is attached rigidly to a bracket having a stiffness k of 1.0278×10^4 lb (force)/ft (1.5×10^5 N/m). What is the resonant frequency of the subassembly?

Using Eq. (1.1):
$$f_N = \left(\frac{0.5}{\pi}\right) \left[\frac{\left(1.0278 \times 10^4\right) \left(32.17\right)}{4.85}\right]^{1/2} = 41.6 \text{ Hz}.$$

 $f_N = \left(\frac{0.5}{\pi}\right) \left(\frac{1.5 \times 10^5}{2.2}\right) = 41.6 \text{ Hz}.$

The prism and bracket of Example 1.1 might be mounted to a periscope that is attached to a mount that is attached to the structure of a vehicle so as to form a multicomponent interconnected system. Steinberg² advised that it is good engineering practice for such a system to be designed so the fundamental frequency of each cascaded subsystem increases by a factor of two at each interface (component to bracket, bracket to periscope, etc.) to limit resonant coupling. Some designs include a means for damping vibrations. This tends to reduce intercomponent coupling.

The key to achieving a successful design under specified vibration conditions is knowing how the instrument will react to the imposed forces. Analytical (i.e., software) tools of ever-increasing capability, using finite-element analysis (FEA) methods, are available for modeling the design and predicting its behavior under variable imposed time and spatial loading.³⁻⁶ Some of these tools interface with optical design software so that degradation of optical performance under specific adverse conditions can be evaluated directly. The effects of temperature changes, thermal gradients, and pressure changes also can be evaluated with these same analytical tools.

It is important to recognize that microscopic changes in dimension and/or contour (i.e., strains) can occur in an optical component as a result of externally applied load, by gravity or acceleration. Hook's Law requires that stresses always accompany strains. These stresses may be temporary under low (operational) loading or damaging under higher (survival) loading that exceeds the breaking point for brittle (glass-type) materials or the elastic limit^{b,c} of mechanical materials. Englehaupt⁷ indicated that stress should be at least one order of magnitude lower in optics than in conventional mechanical hardware.

Design techniques that can be used to increase the resistance of an optical instrument to vibrationally-induced strains include ensuring adequate support for fragile optical members (such as lenses, windows, shells, prisms, and mirrors); providing adequate strength for all structural members to minimize the risk of distortion beyond their elastic (or perhaps their microyield^d) limit; and reducing the mass to be supported.

1.2.3.2 Random frequencies

The vibration environment may be random in nature instead of periodic. This means that, within a given range of frequencies, acceleration of some magnitude occurs at each frequency [see Fig. 1.2(b)]. If the fundamental frequency of the instrument and its structure falls within this range, resonance could be excited.

Random vibrations are frequently quantified by the power spectral density (PSD) of their acceleration. In a simple case, this is expressed graphically on log-log coordinates as a function that rises from zero, levels off, and falls again to zero at high frequencies [see Fig. 1.2(c)]. In the frequency range of 60 to 1200 Hz, the rms acceleration response is 0.1. More complex cases have different functions occurring in different frequency regions. The acceleration PSD is quantified in units of g^2 /Hz, where g is a multiple of gravity.

The rms acceleration response ξ of a body vibrating randomly in a single degree of freedom can be approximated by the following expression:

$$\xi = \left[\frac{\pi f_N PSD}{(4\eta)}\right]^{1/2}, \qquad (1.2)$$

where the PSD is defined over a specific frequency range, and η is a factor quantifying the effective damping within that range. Vukobratovich⁸ indicated that most structural effects of a given system result from the 3-sigma acceleration that occurs, so the system should be designed and tested to a level of 3 ξ . Example 1.2 shows the use of these relationships.

Vukobratovich⁸ gave representative values for the PSDs of typical military and aerospace environments (see Table 1.1). These range from 0.001 g^2 /Hz to 0.17 g^2 /Hz

^b Defined as the level of stress that causes a strain of two parts per thousand per unit length.

^c See Appendix B for key mechanical properties of commonly used materials.

^d Defined as the stress that causes 1 part per million (ppm) of plastic (nonelastic) strain.

over the frequency range of 1 to 2000 Hz. The PSD for these and other applications are determined by measurement. Vukobratovich⁹ indicated that a reasonable design guide for random vibration acceleration level characteristic to shipping an optical instrument is $0.04 g^2$ /Hz from 60 to 1000 Hz, ramping down at -6dB per octave^e from 1000 to 2000 Hz and zero above 2000 Hz.

Example 1.2: (a) Estimate the rms acceleration response to random vibration of the prism-bracket system defined in Example 1.1. (b) To what level of vibration acceleration should the system be designed and tested? (For design and analysis, use File 1.2 of the CD-ROM.)

Assume the random vibration PSD in the range 60 to 1200 Hz is 0.1 g^2 /Hz [per Fig. 1.2(c)], and the system damping factor η is 0.055. From Example 1.1, f_N is 41.6 Hz.

(a) From Eq. (1.2),
$$\xi = \left\{ \frac{(\pi)(41.6)(0.1)}{[(4)(0.055)]} \right\}^{1/2} = 7.7$$
 times gravity.

(b) This prism/bracket subassembly should be designed to and tested at a nominal acceleration a_G of (3)(7.7) = 23.1 over the specified frequency range.

Table 1.1 Acceleration power spec	tral densities	(PSDs) for	typical	military
and aerospace environments.*				

		Power spectal
Environment	Frequency (f) (Hz)	density (PSD)
Navy warships	1–50	$0.001 g^2/Hz$
Typical aircraft	15–100	$0.03 g^2/Hz$
	100–300	+4 dB/octave
	300-1000	$0.17 g^2/Hz$
	≥ 1000	-3 dB/octave
Thor-Delta launch vehicle	20–200	$0.07 g^2/Hz$
Titan launch vehicle	10–30	+6 dB/octave
	30-1500	$0.13 g^2/Hz$
	1500-2000	-6 dB/octave
Ariane launch vehicle	5–150	+6 dB/octave
	150-700	$0.04 g^2/Hz$
	700–2000	-3 dB/octave
Space Shuttle		
(orbiter keel location)	15–100	+6 dB/octave
	100-400	$0.10 g^2/Hz$
	400–2000	-6 dB/octave

* From Vukobratovich.⁸

^e The decibel is a unit for expressing the ratio of two quantities. It equals 10 times the common logarithm of this ratio, i.e., $dB = 10 \log_{10}$ (quantity ratio). One octave doubles the frequency.

1.2.4 Shock

A shock is a force applied suddenly and briefly to a complete instrument or to a portion thereof. More specifically, it may be defined as an externally applied loading with duration equal to or shorter than half the period, corresponding to the system's natural frequency f_N . There are two effects of a shock: amplification of the input impulse and "ringing" of the excited system. Both of these effects are influenced by the shock duration and pulse shape, the system's f_N and its damping factor η . The maximum theoretical amplification is a factor of two. This is the basis for the commonly used design guideline that the system should be designed for shocks twice the worst-case applied load.⁹

Shock introduces a series of dynamic conditions into structural members. Elastic (or perhaps inelastic) deformations generally occur and inadequately supported parts dislocate relative to their surroundings. Optical alignment may be impaired temporarily or permanently and fragile components may be overstressed and fail. In the case of optics, failure is more likely if internal strain is present because of inadequate annealing or strain relief during fabrication, or if subsurface damage has been created during manufacture. The latter condition is addressed in Chap. 13.

Specifications generally define shock in terms of acceleration in multiples of gravity applied in a specified direction or in three mutually orthogonal directions. We here specify acceleration as a dimensionless multiplying factor, a_G . The shock level associated with normal manual handling of optical instruments is generally assumed to be $a_G = 3$.

Shipping often entails the worst shock conditions that an instrument will encounter. Structural loads are normally higher during transportation by truck than by rail. A vehicle with air-ride suspension will reduce the severity of the disturbance. In cases where shock during transportation is likely to be severe, the specification may distinguish between conditions with and without shipping containers and/or shock mountings. Without a container or with a direct path for force to pass from the vehicle to the instrument, the shock level may exceed $a_G = 25$. Suitably designed packaging should attenuate transportation because of wind gusts and landing forces, while more sustained forces (vibration) result from air turbulence. Note that pressure and temperature variations also occur frequently during shipping.

Defining the value for a_G is necessary, but not sufficient when preparing a shock specification for an instrument or portion thereof. Traditionally, specifications also define shock duration and pulse shape. For example, a common method for shock testing requires "3 shocks in each direction along each axis at 1 of 8 degrees of severity in the range $10 \le a_G \le 500$ with half-sine wave pulse durations of 6 to 16 msec." A space payload can encounter severe shocks during launch, stage separation, orbital changes using thrusters, activation of pyrotechnic devices, upon reentry, and during shuttle landings. Because these shocks can have drastic effects if poorly specified or designed systems are involved, especially for man-rated systems, more definitive requirements are now specified for space-borne equipment. The peak acceleration for a shock pulse

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passing through a structure typically attenuates with the distance from the disturbance and with the number of joints encountered.

Figure 1.3 shows a general case. We see that the peak acceleration here is reduced by 50% in about a 20-in. (50.8-cm) distance. Attenuation of about 40% also typically occurs at each mechanically fastened (not welded or bonded) structural joint. Attenuation becomes small after the shock passes though three joints.

Shock resistance of optomechanical systems can be increased by (1) the use of isolation subsystems (shock mounts), (2) a design that anticipates loads being spread over as large an area as possible, (3) a favorable choice of materials and fabrication processes, and (4) a design that minimizes mass of the driven body and provides adequate physical strength and rigidity in all supporting components.

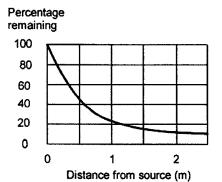


Figure 1.3 Representative attenuation of the peak magnitude of a shock pulse with distance through a structure.

1.2.5 Moisture, contamination, and corrosion

In order to maximize an optical instrument's resistance to humidity, contamination, and corrosion, it is important to assemble the instrument in a clean, dry environment; to seal all paths where leakage could otherwise occur between the instrument's interior and the outside world, and use compatible materials in the design. Techniques for sealing optical instruments are discussed briefly in Sects. 2.3 and 4.12 of this book.

Once sealed, the interior cavities of the instrument may be purged with a dry gas (such as nitrogen or helium) through valves or removable seal screws to remove traces of moisture that could condense on optics or other sensitive internal component surfaces. In some cases, the pressure of the residual gas within the instrument will intentionally be made somewhat above the ambient external pressure. Raising the internal pressure does not necessarily prevent the entry of moisture, but helps prevent internal contamination by particulates such as dust. The tendency for water to diffuse into the instrument is determined by the partial pressure differential of water between the interior and exterior regions, the permeability of the walls and seals, and the temperature. Purging the instrument with dry gas may increase the time before enough water enters the housing to become an issue. If a really dry internal environment is needed, an internal desiccator should be installed and the chamber sealed as well as possible. It then should be purged with dry gas.⁹ Evacuating the chamber and backfilling it with the purge gas is a useful technique.

McCay et al.¹⁰ indicated that the performances of optical instruments using ultraviolet (UV) light, such as lithography systems working at a 157-nm wavelength, are degraded by the presence of seemingly insignificant quantities of certain contaminants in the optical path. These may be in the form of deposits of hydrocarbons and/or silicones on optical surfaces or water vapor in the surrounding gas. Interaction of the UV radiation with molecules of the contaminant plays a strong role in these effects. The presence of small amounts of diatomic oxygen in the gas reduces the effects of some potential surface contaminants by converting them into less harmful chemicals. Excessive amounts of this oxygen can reduce transmission of the system. The quantities of all potential contaminating materials in and around the instrument need to be controlled to predetermined allowable levels to ensure the success of these instruments.

Corrosion is a chemical or electrochemical reaction between materials and their environment. Its most common form occurs when two dissimilar materials combine in the presence of water. The reaction involves oxidation, which is the formation of metallic ions and the liberation of electrons, and reduction, which is the consumption of free electrons. The electrons transfer through the fluid.

Techniques to minimize corrosion of metals include avoiding contact between noncompatible types, careful cleaning to remove corrosive residues during processing, applying protective coatings, and controlling exposure to high humidity. Certain protective coatings or platings can be applied to sensitive materials in some cases, but these may deteriorate and lose their protective nature with time or under mechanical or thermal stress.

Some of the most common forms of corrosion are *fretting*, where impact between surfaces from vibration causes the breakdown of protective coatings such as oxide layers; *galvanic at tack*, where electrons flow from one metal to a less noble (i.e., less active) metal; *hydrogen emb rittlement*, where hydrogen diffuses into a metal and makes it susceptible to brittle fracture; and *stress-corrosion cracking*, where defects such as pits in the surface of a material grow under sustained tensile stress in the presence of moisture and lead to brittle fracture.¹¹ A plausible explanation for the mechanism of stress-corrosion-accelerated failure in mechanically stressed metals, given by Souders and Eshbach,¹² is that pits or notches formed by corrosion are opened up as the metal's surface deforms under tension and becomes filled with rust or other contaminants. When the tension is released and the openings close over the foreign material, wedge action tends to increase the stress within the part and to induce additional and more severe cracks. The situation grows progressively worse until the part fatigues and failure occurs.

Metals differ considerably with regard to their inherent resistance to corrosion. For example, aluminum and its alloys are reasonably immune to corrosion if kept in a dry atmosphere. Moisture, alkalis, and salt cause these materials to corrode. Anodic oxide coatings offer considerable protection. Titanium is commonly used if corrosion resistance as well as a high structural strength-to-weight ratio is needed. Magnesium is quite susceptible to damage from atmospheric contaminants, such as salt in the presence of moisture. The stainless steels^f offer varying degrees of resistance to corrosion. Type-410 CRES, for example, forms a superficial oxide film after a few weeks of exposure to air. All other factors being equal, Type-316 CRES is best for resistance to a salt atmosphere.^{13,14}

1.2.6 High-energy radiation

Limited protection can be provided for optics exposed to high-energy radiation in the form of gamma and x rays, neutrons, protons, and electrons by shielding them with materials that absorb these radiation types; by using optical materials, such as fused silica, that are relatively insensitive to such radiation; or by using radiation-protected optical glasses. The latter contain specific amounts of cerium oxide and suffer from a slight reduction in blue-end visible- and UV-light transmission before exposure to radiation, but maintain their transmission characteristics over a broad spectral range after such exposure much better than unprotected glasses.

Several types of glass incorporating cerium oxide are available from suppliers such as Schott North America, Inc. These materials differ only slightly in their optical and mechanical properties compared with equivalent standard glasses (see Marker et al.¹⁵).

1.2.7 Laser damage to optics

The interaction of intense coherent radiation with optical materials has been the topic of numerous technical studies and publications ever since the laser was invented. The chief means for communication about this effect since 1969 is the continuing series of international conferences called the *Boulder Damage Symposia* that are held each year at the National Institute of Science and Technology in Boulder, Colorado. Recurring topics for contributed papers include definitions of laser-induced damage thresholds for onset of damage, means for predicting damage onset, methods for testing, protocols for data reduction, formats for reporting, and techniques for modeling the interactions. In addition to sessions for oral presentations and poster sessions, these symposia have, over the years, included several minisymposia on vital technology issues such as mirror contouring by diamond turning, damage to lithographic optics, contamination effects, damage to optical fibers, laser-diode developments, and optics for the deep-UV.

Through the cooperative efforts of thousands of symposia participants, immense progress has been made in improving the purity, homogeneity, thermal conductivity, surface quality, etc., of substrates and coatings (thereby reducing absorption of light and increasing laser-damage thresholds); in understanding the mechanisms for component failures; and in finding ways to evaluate materials, and increasing lifetime by minimizing damage. For example, it has been shown that surface and subsurface flaws can be introduced into optics by laser irradiation during testing or actual service. These flaws tend to grow slowly with time and can cause failure of the optic when they reach a

^f In this book, a stainless steel is abbreviated as corrosion resistant steel (CRES).

critical size. Even before they cause failure, these defects tend to increase scatter of incident radiation and adversely affect performance of the system.

The list of publications resulting from the Boulder Damage Symposia continues to grow at a remarkable rate; this indicates the importance of and interest in this rapidly advancing technology. Because the subject is so complex and not directly pertinent to the present subject of optics-mounting techniques, we will not pursue the topic in detail here. Interested readers should refer to the literature for guidance on this subject.

1.2.8 Abrasion and erosion

Abrasion and erosion problems occur most frequently in devices with optical surfaces that are exposed to wind-driven sand or other abrasive particles or to raindrops or ice and snow particles moving at high relative velocities. Usually the former damage occurs on land vehicles or helicopters, while the latter occurs on aircraft traveling at high speed [>200 m/s, (447 mph)]. Softer optical materials, such as infrared-transmitting crystals, are most often used for these applications and, unfortunately, are most easily affected by these conditions.^{16,17} Thin coatings of harder materials can afford a limited degree of protection to these optics. In the space environment, exposure to micrometeorites and orbiting debris may cause damage to optics such as unprotected telescope mirrors. Retractable or disposable covers are used in some cases to provide temporary protection.

1.2.9 Fungus

The potential for damage to optics and coatings by fungus is greatest when the instrument is exposed simultaneously to high humidity and high temperature; conditions that primarily exist in tropical climates. The use of organic materials, such as cork, leather, and natural rubber in optical instruments, or in associated items such as carrying cases, is specifically forbidden by U.S. military specifications. This is also generally good practice for nonmilitary applications unless the environment will be *very* well controlled. Organic materials, such as the natural oil in a fingerprint, may support the growth of fungus. In the long term, this growth may stain the glass and adversely affect transmission and image quality. Careful maintenance can reduce the likelihood of damage from fungus. Of course, glass and crystal surfaces must be cleaned carefully. Approved materials and procedures should be used.

1.3 Extreme Service Environments

1.3.1 Near Earth's surface

Typical extreme environments for military ground-based materiel are specified in MIL-STD-210 *Climatic Inf ormation t o Det ermine Desi gn and Test Requirements.*¹⁸ Representative environmental conditions are listed in Table 1.2. Examples are given for each category to illustrate the general type of instrumentation expected to undergo the extreme exposures. Additional data pertinent to random vibration levels that optical instruments may experience in common applications were given earlier in Table 1.1.

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More or less severe conditions may apply. These guidelines also apply, within limits, to design and testing of commercial and consumer products.

				Instrument
				Undergoing
				Extreme
Environment	Normal	Severe	Extreme	Condition
Low temperature				Cryogenic
(T _{MIN})	293 K (20°C)	222 K (51°C)	2.4 K (271°C)	satellite payload
				Spectrometer
High				cell for
temperature				combustion
(T _{MAX})	300 K (27°C)	344 K (71°C)	423 K (150°C)	study
	88 kPa	57 kPa	0 kPa	Satellite
Low pressure	(0.9 atm)	(0.5 atm)	(0 atm)	telescope
	108 kPa	1 MPa	138 MPa	Window on
High pressure	(1.1 atm)	(9.8 atm)	(1361 atm)	deep-sea vehicle
Relative				Submerged
humidity (RH)	25-75%	100%	Under water	camera
Acceleration				Gun-launched
factor (a_G)	3	100	11,000	projectile
	$200 \times 10^{-6} \text{ m/s}$	$0.04 g^2/Hz$	$0.13 g^2/Hz$	
	rms	$20 \le f \le 100$	$30 \le f \le 1500$	
Vibration	$f \ge 8 \text{ Hz}$	Hz	Hz	Satellite launch

Table 1.2 Typical values for selected adverse environmental conditions.

^{*}Adapted from Vukobrotovich⁸ and Yoder¹⁶.

1.3.2 In outer space

The environmental conditions encountered in space vary in severity, depending on spacecraft location relative to the Sun, Earth, Moon, and other celestial bodies. Table 1.3 classifies key Earth orbits while Fig. 1.4 depicts them graphically. The Low-Earth-Orbit (LEO) environment is well known, having been explored by instrumented probes and manned missions.^{11,19,20} Higher orbits also are fairly well defined. The Lagrange Points 1 and 2 are 1% of Sun-Earth separation from Earth on near and far sides of the planet. Ventures to nearby planets have revealed harsh environments that challenge payload designers to select materials and configure hardware so as to protect sensors long enough to accomplish the mission. Detailed considerations of the problems with designing optical instruments to cope with such environments are beyond the scope of this book.

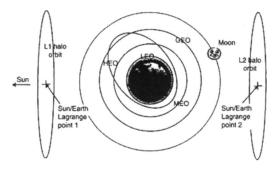


Figure 1.4 Spacecraft orbital locations relative to altitude. Note: Not to scale. (From Shipley.²⁰)

Orbit	Altitude (km)	Period	Applications
Low-Earth (LEO)	200-700	60–90 min.	Military
			Earth/weather
			monitoring
			Space Shuttle
			missions
Middle-Earth (MEO)	3000-30,000	Several orbits per	Military
		day	Earth observation
			Weather monitoring
Geosynchronous	35,800	1 day	Communications
(GEO)			Mass media Weather
			monitoring
Highly Elliptical	Perigee < 3000	Wide range in	Communications
_(HEO)	Apogee > 30,000	hours	Military
Halo about L1	$\sim 1.5 \times 10^7$	80–90 days	Solar observations
			Global observations
Halo about L2	$\sim 1.5 \times 10^7$	days-months	Scientific
			observations
			Global observations

Table 1.3	Earth orbi	t classific	cations.

^{*}From Shipley.²⁰

1.4 Environmental Testing

Environmental testing of hardware is intended to ensure that it has been designed and manufactured to withstand all pertinent environmental conditions to which it might reasonably be subjected, alone or in combination, throughout the equipment's planned life cycle. Whenever possible, the test item should be in the final configuration and interfaced with the related support structure, if any. In some cases, testing portions of the equipment or representative prototype versions may be permitted. Testing can, in some cases, be expedited by testing at more severe levels than actually required to obtain meaningful results in a relatively short time period. This testing is frequently called

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qualification testing inasmuch as, once successfully completed, it "qualifies" the design, materials, and manufacturing method and processes used in the hardware.

1.4.1 Guidelines

U.S. military specification MIL-STD-210, *Climatic Information to Determin e Desig n* and Test Requirements for M ilitary Systems and Equipment ¹⁸ provides guidance regarding expected extreme and typical natural climatic conditions for hot, basic, cold, severe cold, and sea surface and coastal regions of Earth. Conditions to an 80-km (262,000-ft) altitude also are given. ISO-10109, *Environmental Requirements*²¹ provides similar information.

1.4.2 Methods

Information pertinent to planning and conducting environmental tests to determine the ability of military equipment to withstand the anticipated climatic exposures may be found in U.S. MIL-STD-810, *Environmental Test Met hods and Engi neering Guidelines.*²² This information also may apply, within limits, to nonmilitary equipment. Another excellent source of information regarding testing of optical instruments is contained in International Standard ISO 9022, *Optics and Opt ical Instruments-Environmental Test Methods.*²³ This detailed specification defines the required types and severity of tests in a variety of methods. Parks²⁴ gives a useful summary of this document.

In Appendix D, we briefly summarize methods for testing individual optical components and complete optical instruments in 13 types of adverse environmental conditions. These considerations are based primarily on ISO 9022, because that document is directly applicable to optical instruments. In many cases, the tests specified in ISO 9022 are similar to those specified in U.S. military standard MIL-STD-810, and similar documents used in other countries. Some tests defined in the ISO standard were derived from related standards prepared by the International Electrotechnical Commission (IEC) and suitably modified to apply to optical instruments.

Each test is categorized by a nonconsecutive "method number" beginning with 10 and ending with 89; the applicable numbers are indicated in Appendix D. During each environment test, the specimen is to be in one of the following states of operation:

0: In transport/storage container.

1: Unprotected, ready for operation, power off.

2: Operating, i.e., functionally tested during exposure.

Unless otherwise required, the test sequence includes the following steps:

- *Preconditioning*: Here the test specimen is prepared for test and temperature stabilized to within ±3 K of ambient.
- Initial test: The specimen is tested per specifications and examined for conditions that could affect results of environmental tests.
- Conditioning: Exposure to environmental condition at specified severity and state of operation.

- *Recovery*: Specimen is brought back to within ±3 K of ambient temperature and otherwise prepared for final functional test.
- Final test: Examine and test specimen performance per specifications.
- Evaluation: Results are reviewed to determine pass/fail.

For space applications, MIL-STD-1540, *Product Verification Requirements for Launch, Upper St age, and Space Vehi cles*,²⁵ recommends functional testing of optical equipment as indicated in Table 1.4. The listed tests are performed during qualification testing or acceptance testing as indicated. Some tests are shown as optional. The decision as to the need for such tests is determined on the merits of the design and the application. Yoder¹⁶ observed that certain tests not listed as required in MIL-STD-1540 for optical equipment are often appropriate for many space-borne optical instruments. These include thermal cycling, sinusoidal vibration, handling/transportation-level shocks, pressure, and leak testing.

Table 1.4 Testing specified by MIL-STD-1540 for space-borne optical equipment before and after environmental exposure.^{*}

Performed during:	Qualification	Acceptance	Optional test
Test type:			
Thermal vacuum	Х	X	
Sinusoidal vibration			x
Random vibration	Х	Х	
Pyrotechnic shock			х
Acceleration	Х		
Humidity			х
Life			х

*From Sarafin.²⁶

1.5 Key Material Properties

Key terms and mechanical properties of materials in the context of optical instrument design and the symbols and units used to represent them in this book are as follows:

- Force (F) is an influence applied to a body that tends to cause that body to accelerate or to deform. The force is expressed in newtons (N) or pounds (lb).
- Stress (S) is force imposed per unit area. It may be internal or external to a body and is expressed in pascals (Pa) [equivalent to newtons per square meter (N/m²)] or pounds per square inch (lb/in.²).
- Strain (ΔL/L) is an induced dimensional change per unit length. It is dimensionless, but is commonly expressed in micrometers per meter (μm/m) or microinches per inch (μin./in.).
- Young's modulus (E) is the rate of change of unit tensile or compressive stress with respect to linear strain within the proportional limit. It is expressed in pascals (N/m^2) or pounds per square inch (lb/in.²).

- Yield strength (S_Y) is the stress at which a material exhibits a specified deviation from elastic behavior (proportional stress vs. strain). It usually equals a 2 × 10⁻³ or 0.2% offset.
- Microyield strength (or precision elastic limit) (S_{MY}) is the stress that causes one part per million (ppm) of permanent strain in a short time.
- Thermal expansion coefficient (CTE or α) is change in length per unit length per degree of temperature change. It is commonly expressed in millimeters per millimeter per degree Celsius [mm/(mm°C)] or inches per inch per degree Fahrenheit (in./in.-°F). It may also be expressed as ppm per degree.
- Thermal conductivity (k) is the quantity of heat transmitted per unit of time through a unit area per unit temperature gradient. It is commonly expressed in watts per meter Kelvin [W/(mK)] or British thermal units per hour-foot-degree Fahrenheit (Btu/hr-ft-°F).
- Specific heat (C_p) is the ratio of the quantity of heat required to raise the temperature of a body by 1 degree to that required to raise the temperature of an equal mass of water by 1 degree. It commonly is expressed in joules per kilogram Kelvin (J/kg-K) or British thermal units per pound-degree Fahrenheit (Btu/lb-°F).
- Density (ρ) is mass per unit volume and is expressed in grams per cubic centimeter (g/cm³) or pounds per cubic inch (lb/in.³).
- Thermal diffusivity (D) quantifies the rate of heat dissipation within a body. It is a derived property expressed as thermal conductivity divided by the product of density and specific heat $[k/(\rho C_p)]$.
- Poisson's ratio (v) is the dimensionless ratio of a lateral unit strain to a longitudinal unit strain in a body under uniform longitudinal tension or compression. Its maximum value is 0.5
- Stress optic coefficient (K_s) relates internal stress to the optical path difference for polarized components of light in refractive materials. It is expressed as meters squared per newton (m^2/N).

The materials of greatest importance to optical instrument design are optical glasses, plastics, crystals, and mirror substrate materials; metals and composites used for cells, retainers, spacers, lenses, mirror and prism mounts, and structures; adhesives; and sealants. Some of these materials are discussed at length by Paquin.²⁷ Appendix B contains many tables of the key optomechanical properties of selected materials. A few general comments follow.

1.5.1 Optical glasses

Manufacturers worldwide have manufactured several hundred varieties of optical-quality glass for many years. The "glass map" shown in Fig. 1.5 includes most of the glasses produced a few years ago by Schott North America, Inc. in the United States and/or Germany. Other manufacturers have produced essentially the same glasses. The glass types are plotted by refractive index n_d (ordinate) and Abbe number v_d (abscissa) for yellow (helium) light as shown. They fall into distinct groups (designated) based upon chemical constituents. A more recent representation of available glasses is shown in Fig. 1.6. Kumler²⁸ discusses the reduction in available glass types.

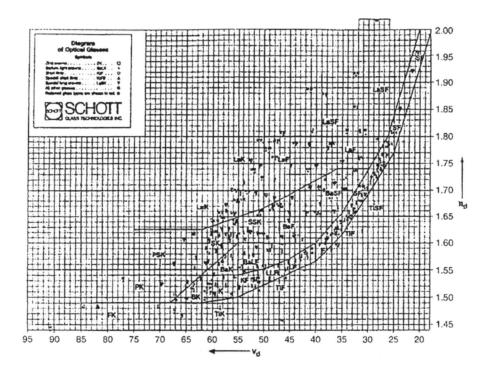


Figure 1.5 "Glass map" showing optical glasses available a few years ago from a supplier. (Courtesy of Schott North America, Inc., Duryea, PA.)

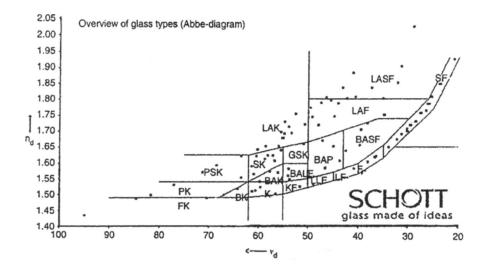


Figure 1.6 Glasses more recently available from the same supplier as in Fig. 1.5. (Courtesy of Schott North America, Inc., Duryea, PA.)

INTRODUCTION

Figure 1.7 is a reproduction of one page from the Schott optical glass catalog, which we will use to illustrate the types of information provided by glass manufacturers for optical design and engineering purposes. The optical properties of interest shown here are the refractive indices (measured values averaged for several melts) at as many as 23 wavelengths, Abbe numbers v_d and v_e , 12 relative partial dispersion values, constants B_i and C_i for an index vs. wavelength equation, temperature variation coefficients D_i , E_i , and λ_{TK} for refractive indices, and average values for internal transmittance τ_i of 10- and 25-mm thicknesses of the material at many wavelengths between 250 and 2500 nm.

Other parameters of interest from the optomechanical viewpoint are homogeneity; variations of nominal refractive indices and Abbe numbers from the catalog values; and presence of localized threadlike vitreous inclusions (striae), bubbles, inclusions, and residual stress (perhaps resulting in birefringence). These parameters are not listed on the individual catalog pages but rather in the general specifications for the various material quality levels available for purchase. Special controls on manufacturing processes and/or selected materials with specific properties may be available at premium prices.

All these properties for optical glasses are generally given in glass catalogs. For example, they are shown for a representative glass type (Schott N-BK7) in Fig. 1.7. This figure illustrates the types of technical information available about each glass for optomechanical design purposes. The $\alpha_{.30/+70}$ item is the material's CTE for the temperature range of concern in instrument design; λ (here k) is thermal conductivity, ρ is density, E is Young's modulus, and μ (here, υ) is Poisson's ratio. Resistance to humidity is indicated on a scale of 1 (high) to 4 (low) by the parameter CR. Rates of change of index of refraction with temperature Δn (here dn/dT) are of interest in temperature-compensated systems.

Not all optical glasses that can be produced are routinely used in optical design. Walker²⁹ designated 68 glasses as the types he considered most useful to lens designers in 1993. His list included glasses in "the most common range of refractive index and dispersion and [that] have the most desirable characteristics in terms of price, bubble content, staining characteristics, and resistance to adverse environmental conditions." In 1995, Zhang and Shannon³⁰ reported a study conducted to identify the "minimum number of glasses needed for 'most' lens designs." Using the double Gauss lens form as a model and three commonly used lens design libraries—CodeV Reference Manual,³¹ Laikin,³² and Cox³³—as specific design sources, they created a list of fifteen most commonly used glasses and a subset of nine recommended glasses. Many of the glasses in the list of fifteen are not included in Walker's list. This is due, in part, to Walker's omission of glasses that he felt had less desirable mechanical or environmental resistance properties while Zhang and Shannon considered only optical properties in choosing their candidates.

Table B1 in Appendix B lists the 49 glasses currently supplied by Schott as "preferred" types that appear in the combined lists of Walker and Zhang and Shannon."

^{*} BK7 glass also is included in Table B1. Residual quantities exist and many designs use it.

Data Sheet	Data	Sheet	
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SCHOTT

N-BK7			Г	nd=1.51	680	vd=64.17	$n_{F} - n_{C} = 0.0$	08054	_
51764	2.251			ne=1.51	872	ve=63.96	nr- nc= 0.0	08110	
			-				E		
Refractio	e Indices	<u> </u>				ncessel	Relative Partia		۲.
	λ [nm]			λ [nm]	t; (10mm)		Pst	0.3098	
n2325.4	2325.4	1.4892		2500	0.67	0.36	Pc,s	0.5612	
n1970.1	1970.1	1.4949		2325	0.79	0.56	Pd,C	0.3076	
n1529.6	1529.6	1.5009		1970	0.933	0.840	Pe,d	0.2386	_
n1060.0	1060.0	1.5066		1530	0.992	0.980	P _g ,r	0.5349	_
nt	1014.0	1.5073		1060	0.999	0.997	Pin	0.7483	_
n,	852.1	1.5098		700	0.998	0.996		0.207/	_
n,	706.5	1.5128		660	0.998	0.994	P's,t	0.3076	
nc	656.3	1.5143		620	0.998	0.994	P'C.s	0.6062	-
nc	643.8	1.5147		580	0.998	0.995	P'd,C'	0.2566	_
n _{632.8}		1.5150		546	0.998	0.996	P'e.d	0.237	_
nD	589.3	1.5167		500	0.998	0.994	P'g.F'	0.4754	_
nd	587.6	1.5168		460	0.997	0.993	Pith	0.7432	_
ne	546.1	1.5187		436	0.997	0.992	Destation		
nr	486.1	1.5223		420	0.997	0.993	Deviation of R		
nr	480.0	1.5228		405	0.997	0.993	Partial Dispers	IONS AP.	80
ng	435.8	1.5200		390	0.997	0.992			23
nh	365.0	1.5362		390	0.998	0.989	APC.t	0.0216	_
ni	334.1	1.5427		370	0.993	0.983	APC.s	-0.0009	-
D334.1	312.6	1.5486		365	0.991	0.977	ΔPF,e	-0.0009	_
n312.6	296.7	1.2400	20	350	0.986	0.920	ΔPg.F	0.0035	
n _{296.7}	290.7			334	0.905	0.920	ΔPig	0.0033	-
n _{280.4}	248.3			320	0.903	0.780	Other Propert	144 000 000	1.0
n248.3	240.3			310	0.574	0.320	a-30/+70°C [10-6/k		24
Constan	ts of Disp	stringw		300	0.290	0.230	a +20/+300°C [10 %	-	-
	1. S. S. S. C. S.	20.925	2223 E	290	0.060	0.050	τ ₄ [℃]	557	_
B1	1.039612	and the second		280	0.000		T1013.0[°C]	557	-
B ₂	0.231792			270			T107.6 [°C]	719	
B ₃	1.010469			260				0.858	-
C1	0.006000		-+	250			cp[]/(g·K)] λ [W/(m·K)]	1,114	-
C2	0.020017		-11	2.50			v [wi(u.v)]	1.114	-
C3	103.5606		-+				p [g/cm ³]	2.51	_
-)	1.03.3000						E [10 ³ N/mm ²]	82	_
Constan	ts of Dith	ersion	1.680	Color	a 30/0.70	2.4.4.3	"	0.206	-
dn/dT	ts of Disp	May and	820 F	λ_{80}/λ_{5}	1999 1997	33/29	K[10-6 mm ² /N]	2.77	-
D ₀	1.86 . 10	1. 1. 1. 1. 1. 1.	0000	$(*=\lambda_{70}/\lambda_5)$		33/27	HK0,1/20	610	_
D1	1.31 . 10			(HG	3	-
D ₂	-1.37 . 10		- 6	Remarke	2264C.V.	نيندوديو. <u>د</u>			
Eo	4.34 . 10		-1 1°		0.00.000000	2	B	0.00	_
E1	6.27 . 10		-1					0.00	-
λ _{TK} [μm]	0.170		-1				CR	2	
	1						FR	0	
Tempera	ture Coel	ficients	of Refr	active ind	PERMIT		SR	1	_
and past		ΔT [10-6/K			abs/AT [10-6		AR	2	_
[**]	1060.0	e	g	1060.		9	PR	2.3	-
-40/ -20	2.4	2.9	3.3		_	1.2			_
+20/+40	2.4	3.0	3.5		1.6	2.1			_
+60/ +80	2.5	3.1	3.7			2.7			_

Figure 1.7 A page from an optical glass catalog showing optical and mechanical parameters for a typical glass, N-BK7. (Courtesy of Schott North America, Duryea, PA.)

The glasses are listed in order of increasing "glass code." The listed glasses should suffice for many new designs for either commercial or noncommercial applications. Exceptional design requirements would demand the use of other, less standard, glass types. Many older glass types can be found in existing inventories or produced on special order.

INTRODUCTION

For each of the glasses listed in Table B1, the reader will find the glass name and type, glass code, the Young's modulus, the Poisson's ratio, a constant K_G that is used to estimate contact stress under mounting forces in Chap 13, the CTE, and the density. Values are given in both USC and metric units. The extreme high or low values for each parameter are identified in the table by the symbols "H" and "L." This is intended to highlight the general magnitudes of the changes that occur in these parameters throughout this group of selected glass types. At the bottom of each column, the ratio of the maximum to minimum values listed is given for each parameter. This indicates the approximate range of those parameters for this limited population. Note that most ratios are about 2. Hence, we generalize that, mechanically, all optical glasses are almost equal.

The prefix "N" in some of the glass names indicates a newer version of that glass now produced by Schott. These have essentially the same refractive properties as the older versions thereof, but their mechanical properties may differ from those of older versions. Three digits added to the glass code for Schott glasses represent density divided by 10. On nine occasions in Table B1, both older and newer versions of the same glass are listed. Mechanical differences are apparent, especially in terms of density where elimination of lead from the chemistry of the newer versions has reduced that parameter.

Stress introduced into optical glasses during cooling (a process called annealing) of the melt can cause problems during subsequent component fabrication. Typically, a piece of poorly annealed glass has surfaces under compressive stress and an interior under tensile stress. As the piece is cut or when material is removed from one surface, these stresses are at least partially relieved and the piece warps slightly. Response to the various steps in fabrication can then be unpredictable. Residual permanent stress after fabrication or temporary stress introduced later from thermal or mechanical causes may affect the optical performance of the finished component. This stress can be detected and measured approximately as birefringence by polarimetry. The resultant variation in refractive index is best measured by interferometry. Stress-related problems can be greatly reduced in magnitude by specifying the permitted residual birefringence of the raw material and of the finished optical component and minimizing external (i.e., mounting) forces applied to the component. Analysis of birefringence effects and determination of appropriate tolerance values for that attribute during optical system design can by accomplished using methods recently described.^{34–36}

Tolerances on birefringence are usually expressed in terms of the permitted optical path difference (OPD) for the parallel (||) and perpendicular (\perp) states of polarization of transmitted light at a specified wavelength. According to Kimmel and Parks,³⁷ birefringence of components for various instrument applications should not exceed 2 nm/cm for polarimeters or interferometers, 5 nm/cm for precision applications such as photolithography optics and astronomical telescopes, 10 nm/cm for camera, visual telescope, and microscope objectives, and 20 nm/cm for eyepieces and viewfinders. Lower quality materials can be used in condenser lenses and most illumination systems. In all cases, the material's stress optic coefficient K_S determines the relationship between the applied stress and the resulting OPD:

.

$$OPD = \left(n_{\parallel} - n_{\perp}\right)t = K_{s}St, \qquad (1.3)$$

where n_{\parallel} and n_{\perp} are the indexes of refraction for the two states of polarization, t is the path length in the material in cm, K_S is in mm²/N, and S is the stress level in N/mm².

Table 1.5 lists the values of K_s at a wavelength of 589.3 nm and a temperature of 21°C for the glasses listed in Table B1. All values are positive for the listed glasses. Some glasses, such as Schott SF58, SF66, and SF59 (not listed), have negative K_s values $(-0.93 \times 10^{-6} \text{ mm}^2/\text{N}, -1.20 \times 10^{-6} \text{ mm}^2/\text{N}, \text{ and } -1.36 \times 10^{-6} \text{ mm}^2/\text{N}, \text{ respectively})$. Another older Schott glass, SF57, has an extremely low K_s value of $0.02 \times 10^{-6} \text{ mm}^2/\text{N}$. Use of the latter material in critical optical components might be appropriate if externally induced birefringence must be minimized. Example 1.3 illustrates the use of Eq. (1.3).

Stress Optic Stress Optic Coefficient Coefficient $(10^{-6} \text{ m}^2/\text{N})$ $(10^{-6} \text{ m}^2/\text{N})$ **Glass Name** Rank **Glass Name** Rank N-BaF51 1 N-FK5 2.91 26 2.22 2 K10 27 N-SSK5 1.90 3.12 3 N-ZK7 3.63 H 28 N-BaSF2 3.04 2.95 29 2.28 4 K7 SF5 2.77 30 N-SF5 2.99 5 N-BK7 BK7 2.80 31 N-SF8 2.95 6 7 N-K5 3.03 32 SF15 2.20 2.93 8 N-LLF6 33 N-SF15 3.04 9 1.80 2.60 34 SF1 N-BaK2 10 N-SF1 2.72 LLF1 3.05 35 2.48 N-LaF3 1.53 11 N-PSK3 36 12 N-SK11 2.45 37 SF10 1.95 2.92 13 N-BaK1 2.62 38 N-SF10 N-BaF4 39 N-LaF2 1.42 14 3.01 40 15 LF5 2.83 LaFN7 1.77 41 16 N-BaF3 2.73 N-LaF7 2.57 2.92 42 SF4 1.36 17 F5 2.58 N-SF4 2.76 18 N-BaF4 43 19 F4 2.84 44 SF14 1.62 20 N-SSK8 2.36 45 SF11 1.33 21 2.81 SF56A 1.10 F2 46 22 N-F2 3.03 47 N-SF56 2.87

Table 1.5 Stress optic coefficients K_s at 589.3 nm and 21°C for the optical glasses listed in Table B1.

Ratio (high/low) = 5.58

N-SK16

SF2

N-LaK22

23

24

25

From Schott Optical Glass catalog (CD Version 1.2, USA)

1.90

2.62

1.82

Optical glasses in general have excellent light transmission characteristics. Not all such materials are exactly equivalent in transmission throughout the UV to near-infrared spectral regions. The crowns generally tend to cut on at shorter wavelengths than the flints, whereas the latter types tend to transmit farther into the near infrared. In the green

48

19

50

SF6

N-SF6

LaSFN9

0.65 L

2.82

1.76

to red regions, all common optical glasses are approximately equal in regard to internal transmission. In the absence of antireflection (A/R) coatings, the higher-index glasses suffer greater Fresnel losses. Simple A/R coatings such as a quarter-wave thick MgF_2 film are more efficient with flint glasses than with crowns because of the higher refractive indices of the flints.

It is well known that all standard optical glasses tend to lose transmission (i.e., to "brown") when exposed to high levels of particle or photon radiation. Exposure to 10 gray $(Gy)^g$ is sufficient to cause perceptible transmission loss in most of these glasses. It is well known that chemically stabilizing (doping) optical glasses with cerium in the form of CeO₂ inhibits darkening when the material is exposed to certain kinds of radiation.

This doping process lowers the transmission of the material slightly throughout the transmission range and significantly in the near ultraviolet region, but effectively reduces the darkening effect of radiation. Figure 1.8 shows this effect for equal thicknesses of standard Schott BK7 and Schott radiation resistant glass BK7G18.^h

Manufacturers such as Schott offer a limited number of radiation resistant optical glasses. All Schott varieties are "inquiry" types that can be manufactured for special applications.

Example 1.3: Estimate the birefringence of a stressed optic per Eq. (1.3). (For design and analysis, use File No. 1.3 of the CD-ROM)

Assume a 0.787-in. (2-cm) thick NBK7 window seals the 3.937-in. (10-cm) aperture of an aerial camera. If it were to be mechanically stressed over a significant part of its aperture during use to 50 lb/in.², would its birefringence be acceptable?

From Table 1.5, $K_s = 2.77 \times 10^{-6} \text{ mm}^2 / N$. Given: $S = (50 \text{ lb} / \text{in.}^2) [6894.8 (N/m^2) / (\text{lb} / \text{in.}^2)] = 3.45 \times 10^5 \text{ N} / \text{m}^2$.

By Eq. (1.3),

 $OPD = (2.77 \times 10^{-6} \,\mathrm{mm^2} / N) (3.45 \times 10^5 \,N / m^2) (2 \,\mathrm{cm}) (10^{-6} \,\mathrm{m^2} / \mathrm{mm^2}) (10^7 \,\mathrm{nm/cm})$

=19.11 nm for 2 cm path or 9.55 nm/cm.

From the text, a camera needs birefringence to be less than 10 nm/cm. Hence, this OPD should be marginally acceptable.

Table B2 lists mechanical characteristics for seven radiation resistant Schott glasses. These glasses may be compared with the equivalent standard types listed in Table B1.

^g The gray (Gy) is the SI unit for the absorbed radiation dose equal to the radiation necessary to deliver 1 J of energy to 1 kg of tissue.³⁸

^h The number added to the glass name is 10 times the percentage content of CeO₂.

Another concern in some applications of optical glasses is the effect of exposure to intense UV radiation. This effect is sometimes called *solarization*. Setta et al.³⁹ and Marker et al.¹⁵ discussed the effects of UV exposure on various types of standard and CeO₂ doped glasses. Some of the latter have inferior transmission characteristics, as compared to the undoped versions of the same materials, after UV exposure.

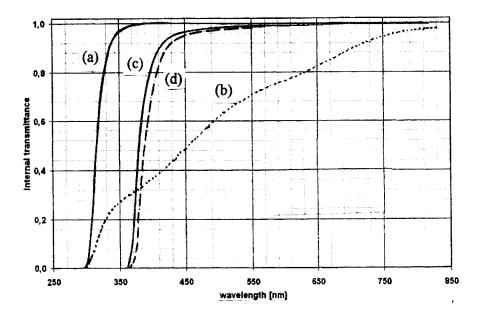


Figure 1.8 Internal transmittances of standard and protected glasses as functions of wavelength for: (a) BK7 unirradiated, (b) BK7 irradiated at 100Gy, (c) BK7G18 unirradiated, and (d) BK7G18 irradiated at 8,000,000 Gy. Note: (b) and (d) are for gamma radiation. (From Schott.⁴⁰)

1.5.2 Optical plastics

A few types of commercially available plastics are suitable for use as optical components in some applications. Key types are identified and selected mechanical properties are listed in Table B3.

In general, optical plastics are softer than glasses, so they tend to scratch easily and are hard to polish to a precise surface figure. Their CTEs and dn/dTs are larger than those of glasses and of most crystals. Plastics tend to absorb water from the atmosphere. This changes their refractive indices slightly. Their specific stiffnesses (E/ρ) are lower than those for glasses.

The biggest advantages of using plastic optical components are their low densities and ease of manufacture in large quantities by low unit-cost molding techniques. It is relatively easy and inexpensive to mold integral mechanical mounting features into plastic components such as lenses, windows, prisms, or mirrors during manufacture. This

INTRODUCTION

facilitates mounting these components and eliminates the need for some mechanical parts, thereby reducing overall costs.

1.5.3 Optical crystals

A variety of natural and synthetic crystalline materials are available for use in optics when transmission in the infrared or ultraviolet spectral regions is required. A few also transmit in the visible region, but usually not as well as optical glasses. Crystals are also used to provide special optical characteristics such as increased dispersion for some specific wavelengths. They fall into four groups: alkali and alkaline earth halides, infrared-transmitting glasses and other oxides, semiconductors, and chalcogenides. Mechanical properties are given in Tables B4 through B7 for the crystals commonly used as optics. Because they are soft, most optical crystals are harder to polish to optical quality than optical glasses.

1.5.4 Mirror materials

Generically, mirrors consist of a reflecting surface (usually a thin-film coating) attached to or integral with a supporting structure or substrate. Their sizes can range from a few millimeters to many meters. The substrates can be made of glasses, low-expansion ceramics, metals, composites, or (rarely) plastics. Tables B8a and B8b list some mechanical properties of the most common mirror materials. Table B9 quantifies structural figures of merit for most of the same materials.⁴¹ The figures of merit allow direct comparisons between candidate materials for a given application. For example, a commonly used figure of merit in mirror design is specific stiffness, E/ρ , which helps us determine which material would have the least mass or self-weight deflection for a mirror of a given geometry and size. The various figures of merit to apply in a particular case depends upon the design requirements and constraints. Tables B10a through B10d list the characteristics of aluminum alloys, aluminum matrix composites, several grades of beryllium, and major silicon carbide matrix types used in mirrors.

1.5.5 Materials for mechanical components

The materials typically used for the mechanical components of optical instruments, such as instrument housings, lens barrels, cells, spacers, retainers, and prism and mirror mounts, are metals (typically aluminum alloys, beryllium, brass, Invar, stainless steel, and titanium). Composites (metal matrices, silicon carbide, and filled plastics) may be used in some structural applications. Some of these materials are also used as mirror substrates. The mechanical properties of selected versions of the metals and one metal matrix may be found in Table B12. The general qualifications of the metals in the context of optical component mounting applications are as follows:

 Aluminum alloys: Alloy 1100 has low strength, is easily formed by spinning or deep drawing, and can be machined and welded or brazed. Alloy 2024 has high strength and good machinability, but is hard to weld. Alloy 6061 is a general-purpose structural aluminum alloy with moderate strength, good dimensional stability, and good machinability. It is easily welded and brazed. Alloy 7075 has high strength and machines well, but is not suited for welding. Alloy 356 is used for moderate- to high-strength structural castings. It machines and welds easily. Most aluminum alloys are heat treated to differing degrees of hardness, depending on the application. Their surfaces oxidize quickly, but can be protected by chemical films or anodic coatings. The latter may produce significant dimensional buildup. A black anodized finish reduces light reflections, so this type of finish is frequently used on aluminum parts for optical instruments. The CTE match of aluminum alloys to glasses, ceramics, and most crystals is not close. Table B10a compares the characteristics of several aluminum alloys used for mirror substrates.

- Beryllium is light in weight, has high stiffness, conducts heat well, resists corrosion and radiation effects, and is fairly stable dimensionally. It is relatively expensive to purchase and to process, so it is used primarily in optical instruments intended for sophisticated applications such as structures and mirror or grating substrates for use at cryogenic temperatures. It also is the material of choice in some space applications where radiation resistance or weight savings are vital and monetary costs are of lesser importance. Table B10c compares the characteristics of several common beryllium grades. Paquin has countered claims about the extreme hazards of working with beryllium by pointing out that simple exhaust systems with suitable filters for particulate material and conventional means for the collection and disposal of loose abrasive grinding and polishing slurry are very effective as safety precautions.⁴²
- Brass is used where high corrosion resistance, good thermal conductivity, and/or ease of machining are required, but weight is not critical. It is popular for screw-machined parts and marine applications. Brass can be blackened chemically.
- Invar, an iron-nickel alloy, is used most frequently in high-performance instruments for space and/or cryogenic applications to take advantage of its low CTE. It is quite dense, and machining sometimes affects its thermal stability. Annealing is advised. A version called Super-Invar has an even lower CTE over a limited temperature range. It is not recommended for use below -50°C (-58°F). To prevent oxidation, Invar frequently is chrome plated.
- Stainless or corrosion-resistant steels (here abbreviated CRES) are used in optical mounts primarily for their strength and their fairly close CTE match to some glasses. They are relatively dense, so a weight penalty must be paid to achieve these advantages. A chromium oxide layer that forms on exposed surfaces makes these steels resistant to corrosion. In general, these steels are harder to machine than aluminum alloys. Type 416 is the most easily machined and can be blackened chemically or with black chrome plating. Type 17-4PH has good dimensional stability. Stainless steels can be welded to like materials or brazed to many different metals.
- Titanium is the material of choice in many high-performance systems where a close CTE match to crown glass is essential. Flexures are sometimes made of titanium because of its high yield strength (S_Y). It is about 60% heavier than aluminum. Titanium is somewhat expensive to machine. It can be cast. Brazing is easy, but welding is more difficult; electron beam or laser welding techniques work best. Parts can also be made by powder metallurgy methods. Corrosion resistance is high.

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Some plastics, particularly glass- or carbon-fiber-reinforced epoxies and polycarbonates, are used in structural parts such as housings, spacers, prism and mirror mounts, and lens barrels for cameras, binoculars, office machines, and other commercial optical instruments. They are relatively lightweight, and most can be machined conventionally or by single-point diamond turning (SPDT). Some can be cast. Generally, plastics feature low cost. Unfortunately, they are not as stable dimensionally as metals and tend to absorb water from the atmosphere and to outgas in a vacuum. The CTEs of filled varieties can be customized to some extent.

1.5.6 Adhesives and sealants

Optical cements used to hold the refracting surfaces of lenses or prisms together as, for example, in forming cemented doublets, triplets, or beamsplitters, must be transparent in the spectral region of interest, have good adhesion characteristics, have acceptable shrinkage, and (preferably) be able to withstand exposure to moisture and other adverse environmental conditions. The most popular optical cements are thermosetting and photosetting (ultraviolet light curing) types. Some mechanical properties of interest are given in Table B13 for a generic type of optical cement.

Structural adhesives most frequently used to hold optics to mounts and to bond mechanical parts together are one- and two-part epoxies, polyurethanes, and acrylics. Most cure best at elevated temperatures and suffer some (up to 6%) shrinkage during curing. Their CTEs are about 10 times those of structural materials and glasses, while stiffnesses are about two orders of magnitude lower than those for structural materials and glasses. Some adhesives emit volatile ingredients during curing or if exposed to a vacuum or elevated temperatures. The emitted material may then condense as a contaminating film on nearby cooler surfaces, such as lenses or mirrors. A few adhesives have low shrinkage and low volatilities. The typical properties of representative types are summarized in Table B14.

During WWII, optical instruments were sealed with very messy and hard to apply polysulfide-type sealants such as EC-801 made by 3M Corporation. Although EC-801 is still available and used for other purposes, sealants used today are usually room-temperature-vulcanizing (RTV) elastomers that cure into flexible, form-fitting masses with reasonably good adherence properties. They are typically poured or injected into gaps between lenses and mounts or between mechanical components to seal leaks and/or to help hold the optics in place under vibration, shock, and temperature changes. Some sealants outgas or emit effluents (typically acetic acid or alcohol) during curing or in a vacuum more than others. The manufacturers recommend use of primers prior to application of the sealants for many of these products.

Typical physical characteristics and mechanical properties of a few representative sealants are given in Tables B15a and B15b. At least one of these (DC 93-500) is accepted by the National Aeronautics and Space Administration (NASA) as a low-volatility sealant for space applications. Unfortunately, it is relatively expensive. The curing times, colors, and certain physical properties of sealants can be modified significantly through the use of additives and/or catalysts.

1.6 Dimensional Instability

Paquin⁴³ has defined dimensional instability as "the change that occurs in response to internal or external influences." In order to create a dimensionally stable instrument, one must control these changes (i.e., strains) in optomechanical components to levels that do not compromise performance requirements. In situations requiring stability on the order of normal machining tolerances, strain must be controlled to about 1 part in 10^3 . This is relatively easy to accomplish. In high-precision applications, tolerances of 1 part in 10^6 apply and strain must be controlled to the same degree. Even higher precision is possible, but tolerances may be as small as 1 part in 10^9 , and finding materials and manufacturing processes to achieve such tolerances requires the utmost care at all stages of design and production. A considerable amount of luck would also be helpful.

The dimensional changes of concern here are due to externally applied forces that cause plastic deformation; internal (residual) stresses that relieve themselves (usually unpredictably) with time, with temperature change, or under vibration and/or shock; microstructural changes such as phase transformations or recrystallization within the materials; and inhomogeneity and anisotropy of the materials. Paquin^{42,43} and Jacobs⁴⁴ deal with these potential causes of problems at much greater length than we can here. It is important to recognize the possibility that microscopic changes will occur within the parts of an optical instrument so provisions can be made to avoid or at least minimize these changes whenever possible.

1.7 Tolerancing Optical and Mechanical Components

Closely related to the performance specifications and constraint definitions that are the starting points of part of any design effort are the multilevel budgets on allowable deviations from perfection of component dimensions and alignments relative to other components in the instrument. Tolerances on these deviations, or errors, strongly influence how an optomechanical system will perform and the total life cycle cost of that instrument. They also form a basis for inspection of parts and assemblies. We strive for a balance between overly tight tolerances that waste production and inspection time and missing or excessively loose tolerances that result in unacceptable hardware. Some minimal number of assembly adjustments should, in many cases, be factored into the tolerance budget so as to allow compensation for errors that are costly to control solely by means of tight tolerancing.

Ginsberg⁴⁵ suggests a process for developing an appropriate budget of tolerances for errors in an optomechanical design. This process, extending from the specification and mechanical constraint definitions to toleranced drawings ready for manufacture of the optics, is depicted in Fig. 1.9. Recomputation loops used to optimize the design and provide required performance are indicated. Willey⁴⁶ described additional loops that bring the valuable expertise of manufacturing, assembly, test, and maintenance personnel into play (see Fig. 1.10).

The starting point for the iterative development of a suitable error budget often is the assignment of preliminary tolerances on optical and mechanical parts parameters based

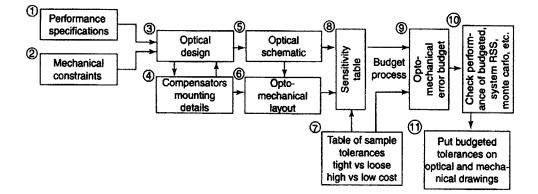


Figure 1.9 Block diagram showing an optomechanical error budgeting and tolerancing process with loops used to optimize the tolerance distribution. (From Ginsberg.⁴⁵)

Table 1.6 Dimensions and parameters to be toleranced in an optical instrument. *

Surface geometry	Surface finish
• Radius	• Quality (scratch and dig)
• Departure from nominal shape	• Roughness, scatter, etc.
Aspheric deformation	
Surface separation	Index of refraction
• Element thickness	Value at central wavelength
Axial spacing	• Total dispersion (Abbe number)
	Partial dispersion
	Homogeneity
Alignment	Transmission
Surface tilt	Optical material
• Element tilt and/or decentration	Spectral characteristics of filters
• Component tilt and/or	Coating characteristics
decentration	
 Prism or mirror angles and tilt 	
Physical characteristics	
• Thermal effects (CTE and dn/dT)	
Stability	
• Durability	
*From Smith. ⁴⁷	_

on prior experience and/or guidance from the literature. For example, Table 1.6 shows typical dimensions and other parameters of a generic optical system that usually need to be toleranced.⁴⁷

The tolerances applicable to each dimension and characteristic depend largely on the performance level of the system. Table 1.7 shows typical loose, tight, and limiting

tolerances for optics. Arriving at an optimum distribution of tolerances is a complex process. Smith.⁴⁷ describes one way to approach this task. He also warns that there are no savings from loosening tolerances beyond the point where costs level off. Further, costs climb rapidly as tolerances approach a level where fabrication becomes impossible.

		Tole	erance	Approximate
Parameter	Units	Loose	Tight	limiting value
Index of refraction		0.003	0.0003	0.00003 ^a
Radius departure from test plate	fringes ^b	10	3	1
Departure from spherical or flat	fringes ^c	4	1	0.1
Element diameter	mm	0.5	0.075	0.005
Element thickness	mm	0.25	0.025	0.005
Element wedge angle	arcmin	3	0.5	0.25
Air space thickness	mm	0.25	0.025	0.005
Decenter, mechanical	mm	0.1	0.010	0.005
Tilt, mechanical	arcsec	3	0.3	0.1
Dimensional errors of prisms	mm	0.25	0.01	0.005
Angle errors: prisms and				
windows	arcmin	5	0.5	0.1

Table 1.7 Sample tolerances applicable to optomechanical parameters.

Notes: ^aDepends on piece size.

^bOne fringe equals 0.5 wavelength at 546 nm (mercury green). Fringes are specified over the clear aperture.

^cDepends on the manufacturing process.

Adapted, in part, from Ginsberg⁴⁵ and Plummer⁴⁸ as updated by Fischer and Tadic-Galeb.⁴⁹

Any tolerance budget can be relaxed if a few, carefully chosen, adjustments are allowed. For example, focus adjustment may be acceptable so some optic mounting could be designed to allow axial adjustment. Smith⁴⁷ suggests the following other approaches:

- Adapt the design to use surface radii corresponding to available test plates.
- Adapt the design to use measured axial thicknesses of refracting optics and adjust air spaces to restore performance.
- Adapt the design to use measured refractive indices (manufacturer's melt data).
- Measure the aberrations, i.e., OPD, of a fabricated system and calculate the errors that would produce that performance by optimizing the design to have those residual errors. Then, adjust the system to eliminate those defects by introducing dimensional changes opposite to, but equaling, the calculated values.

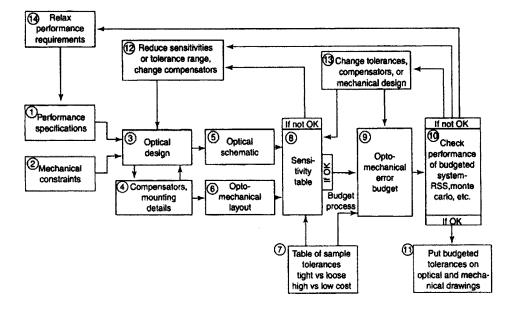


Figure 1.10 Extended version of the error budget process from Fig. 1.9 showing additional loops to ensure producibility. (From Willey.⁴⁶)

1.8 Cost Aspects of Tightened Tolerances on Optics

The cost of a lens or other optic depends strongly on the tolerances of its dimensions and other parameters. For example, if the tolerances for a given lens are loose enough that the standard fabrication and inspection methods used by a given optical shop cause no special labor, tooling, or test equipment costs, then the unit cost will be minimal. This is defined as the base cost of that optic. If, however, the tolerances were to be tightened, the cost of that lens made in the same shop would be expected to rise. The rate of increase as the tolerance is tightened is not linear; cost increases more rapidly as the tolerances are made more demanding. For many years, several workers have tried to correlate lens unit cost to the specified tolerance for a variety of lens dimensions and parameters.⁴⁸⁻⁵³ We here summarize a few of the cumulative results of those efforts, as presented by Willey and Parks,⁵⁴ to indicate how to approach a more cost-effective design in optical instruments. The interested reader should delve more deeply into this literature.

Figure 1.11 shows an empirically derived graph for the increase in cost of a lens above the base grind and polish cost as a function of glass choice expressed in terms of its susceptibility to staining when exposed to lightly acidic water during processing. The abscissa is the Schott stain code. A lower number on a scale of 0 to 5 represents a more resistant glass type so increasing the number has the same type of effect on cost as tightening a dimensional tolerance. The equation shown in the figure is a reasonably good fit to data from Fisher and Tadic-Galeb⁴⁹ and Willey⁴⁶ for FR ≤ 4 .

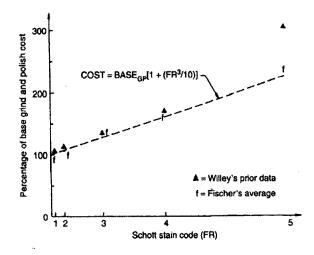


Figure 1.11 Relative cost of glass stain characteristics according to various authors. (From Willey and Parks.⁵⁴)

Figure 1.12 shows an empirically derived graph for the increase in cost of a lens above the base grind and polish cost as a function of radius of curvature tolerance expressed as sagittal depth error. Sagittal depth is measured with a spherometer or interferometer.

Figure 1.13 depicts the growth of unit cost as the tolerance on surface figure error is tightened. The curve is biased towards Plummer's data⁴⁸ rather than Willey's data⁴⁶ because average opticians obtained the former results while the latter were obtained by highly skilled specialists. The vertical error bars indicate that polishing time varies from block to block.

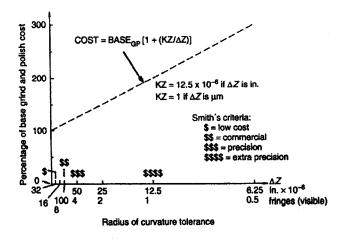


Figure 1.12 Relative cost vs. reciprocal tolerance according to various authors concerning radius of curvature expressed as sagittal error. (From Willey and Parks.⁵⁴)

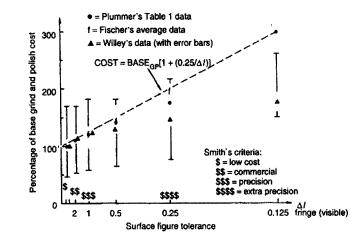


Figure 1.13 Relative cost vs. reciprocal tolerance according to various authors concerning lens surface figure irregularity. (From Willey and Parks.⁵⁴)

Figure 1.14 shows one cost vs. tolerance relationship pertaining to mechanical parts. Here, the cost is seen to vary with tolerances on concentricity (ΔCE) or runout along the length of the bore (ΔLE) created to hold a lens. Two lines are shown: the upper line assumes that the part must be removed from the machine after cutting one surface, reinstalled in the machine, and realigned before a second surface can be cut, while the lower line assumes that all machining can be done in one set up. The big difference between the slopes of the two lines indicates the extra labor (and cost) to accomplish the second setup. The desirability of designing the mechanical part so as to allow it to be machined in one setup is apparent from the graph. This same principle should be applied whenever possible in the design of other mechanical parts.

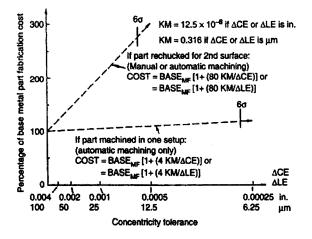


Figure 1.14 Relative cost vs. reciprocal tolerance according to various authors concerning lens mounting bore concentricity (ΔCE) and tilt and length runout (ΔLE). (From Willey and Parks.⁵⁴)

1.9 Manufacturing Optical and Mechanical Components

The process of manufacturing an optical instrument to a given design entails acquisition, storage, and handling of raw materials, parts fabrication, parts inspection, preliminary and final assembly, quality control, optics-to-product interfaces and tests, and the attendant costs, schedules, process development and control, and personnel utilization. Ideally, manufacturing, assembly, inspection and metrology, and maintenance personnel would have been involved throughout the design process because the product is not properly designed if it cannot be produced and kept operating. Ease of manufacture, assembly, and test enhances reliability of the hardware. Most instruments will experience some level of disassembly before it is finished. Ease of disassembly not only enables access to fix some internal problem that shows up late, but makes maintenance during use much easier.

Generating, grinding, polishing, edging, coating, cementing, and bonding glass or crystalline materials most often are the processes used to make optical parts. Some optical parts can be made by single point diamond turning (SPDT) methods if the materials are compatible with that process. Table 1.8 from Englehaupt⁵⁵ identifies common methods for shaping, surface finishing, and coating optical and mechanical components using most of the material categories considered in this chapter. In choosing the appropriate combination of methods, it is important that no process, including plating and/or coating, introduces excessive internal or surface stress into the finished part. Such authors as Malacara,⁵⁶ DeVany,⁵⁷ Karow,⁵⁸ and Englehaupt^{7.55} have described optical shop methods and processing materials.

The most common ways to make metal parts are machining, chemical and electrical discharge machining, sheet-metal forming, casting, forging, extruding, and single point diamond turning (SPDT). Assembly entails mounting the optics and aligning them with respect to other optical and mechanical components and mechanisms. In-process inspection and testing at various points during the overall manufacturing process play very important roles in building a successful product.

Table 1.9 elaborates on the basic ways to make mechanical parts. All are used from time to time in optical instrument fabrication, but the ones most frequently employed are machining, casting, and forging/extrusion. The latter two usually involve the first since parts roughly shaped by casting or forging generally need to be finished by machining to create interfaces with other parts or to remove excess material. Yoder¹⁶ summarizes some important aspects of these three key fabrication methods. Table B11 compares advantages, disadvantages, and applications of composite materials for manufacturing mechanical parts.⁶⁰

It is vital that all processes for manufacturing and assembly of both optical and mechanical parts are documented completely and followed. Revision of these documents based on practical experience as they are used should be allowed and encouraged. One should not perpetuate erroneous instructions simply because it takes time and effort to correct them or because it may be somewhat embarrassing to admit that there might be a better way to do some task.

		Surface finish	
Material	Machining method ^a	control method	Coatings
Al alloys	induction in the second	ELN + SPDT + PL	MgF ₂ , SiO, SiO ₂ , AN,
6061, 2024	SPDT, SPT, CS, CM,	Polish with oil	AN + Au, ELNiP and
(most common)	EDM, ECM, IM	distillates + diamond	most others
	HIP, CS, EDM, ECM,	distinates - diamond	most others
Al matrix	GR, PL, IM, CM, SPT		MgF ₂ , SiO, SiO ₂ ,
Al or Al + SiC	(difficult)	ELN + SPDT + PL	AN, EN + Au
Low silicon-Al		ELN + SPDT + PL	MgF ₂ , SiO, SiO ₂ ,
castings	SPDT, SPT, CS, CM,	Polish with oil +	AN, $EN + Au$ and
A-201, 520	EDM, ECM, IM	diamond	most others
Al silicon	CS, EDM, CE, IM,	unumonu	
hypereutectic 393.2	SPDT, SPT, GR, CM		
Vanasil + lower	(easier than composite		ELN or ELNP
silicon A-356.0	Al-SiC)	ELN +SPDT + PL	followed by others
· · · · · · · · · · · · · · · · · · ·		ELN + SPDT + PL	
	CM, EDM, ECM, EM	Polish with oil +	None (IR) or coat
Beryllium alloys	GR, HIP, not SPDT	diamond	ELN
	SPDT, SPT, CS, CM,	GR, PL with oil +	
Magnesium alloys	EDM, ECM, IM	diamond	Similar to Al, ELN
	HIP/mandrel + GR,	· · · ·	
	CVD/mandrel + GR,		
SiC	molded carbon +		
Sintered, CVD,	reaction with Silane to		
RB, carbon + Si	SiC ^b	GR + PL	Vacuum processes
Silicon	HIP/mandrel, GR,	GR + PL	Vacuum processes
	CVD/mandrel		
Steels	CM, EDM, ECM, Gr,	ELN or ELNiP +	ELN, ELNiP, and
Austenitic	not SPDT, CM, EDM,	SPDT + PL	most others
PH -17-5, 17-7	ECM, GR, not SPDT		
Ferritic 416			
Titanium alloys	CM, HIP, ECM, EDM,	PL, IM	ELN + most others
	GR, not SPDT		Cr/Au
Glass, quartz,	CS, GR, IM, PL, CE,	PL, IM, CMP, GL	Vacuum processes
Low expansion	SL	(laser or flame)	Cr/Au, CR, Ti-W, Ti-
ULE, Zerodur			W/Au SiO, SiO ₂ ,
			MgF ₂ , Ag/Al ₂ O ₃

Table 1.8 Techniques for machining, finishing, and coating materials for optical applications.

Notes: ^a AN = anodize, CE = chemical etch, CM = conventional machine, CMP = chemical mechanical polish, CS = cast, CVD = chemical vapor deposit, ECM = electrochemical machine, EDM = electrical discharge machine, ELNiP = electrolytic nickel phosphorus plate (can replace ELN), ELN = electroless nickel (usually ~11% phosphorus by weight), GL = Glaze, GR = grind, HIP = hot isostatic press, IM = ion mill, PL = polish, SPDT = single-point diamond turn, SPT = precision turn with tool other than diamond, SL = slump casting over mold.

^b An interesting new process by POCO Graphite, Inc., Decatur, Texas.

Adapted from Englehaupt⁵⁵ (revision and expansion of information from Englehaupt⁷).

Process	Description	Advantages	Disadvantages
Machining	Remove material by cutting or grinding	Can achieve many shapes and virtually any dimensional tolerance and surface finish; does not degrade material strength; can be automated with numerical control programs	Can be expensive (machining time and material waste); can require expensive tools; can result in detrimental residual stresses
Chemical milling (etching)	Remove material by immersing the part into a chemical solution	Can make thinner walls than machining; can remove material from parts with two axes of curvature	Very limited intricacy; rough surface finish; difficult to control lateral dimensions accurately; machining costs less for flat cuts
Sheet-metal forming	Form shapes by bending; usually sheet metal, but sometimes plate stock	Low cost; economical for low-quantity production	Suitable only for ductile materials; thick parts require large bend radii, which can limit applicability
Casting	Pour molten material into a mold and allow it to solidify	Versatile; many processes at different costs	Depends upon the casting process
Forging	Force hot metal into a die by pounding	High strength and fatigue resistance in the direction of the material grain	Lower strength and resistance in nongrain directions; expensive for low-quantity production
Extruding	Squeeze hot metal through a die to make a part of uniform cross section	Economical; good surface finish; many standard shapes available	Poor transverse properties

Table 1.9 Basic processes for fabrication of metal mechanical parts.	Table 1.9 Basic processes for fabrication of metal	mechanical parts.*
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Whenever hardware is assembled (or disassembled and reassembled) there is the potential for internal contamination. For example, soldering electrical connections or machining operations such as drilling and reaming holes for mechanical pins to preserve component alignment can introduce flux, solder droplets, metal chips, dust, and/or traces of lubricants into the instrument. Care must be exercised to minimize performance loss or mechanism failure from such contamination sources.

Optical instruments often include conventional light sources, light-emitting diodes, lasers, detectors, actuators, figure or image quality sensors, analog-to-digital converters, thermal control subsystems, and electric powered delivery/control subsystems. These also need to be fabricated, tested, and integrated into the instrument at appropriate times during the manufacturing process.

Verification is a very important part of manufacture. Questions such as are depicted in the flow diagram of Fig. 1.15 need to be asked and answered. Processes, individual parts, assemblies, and the complete instrument should be considered. Analyses and inspections conducted during manufacture verify fit and function. Inevitably, tests are required to *prove* the adequacy of the design from the viewpoints of engineering, environmental qualification, and acceptance of hardware for delivery. In some cases, lack of correlation between analysis results and test results or significant problems uncovered during manufacture lead to requirements for additional analyses, testing, or even redesign and retrofit of hardware. A significant responsibility of the design team is to prevent, or at least minimize, these troublesome events.

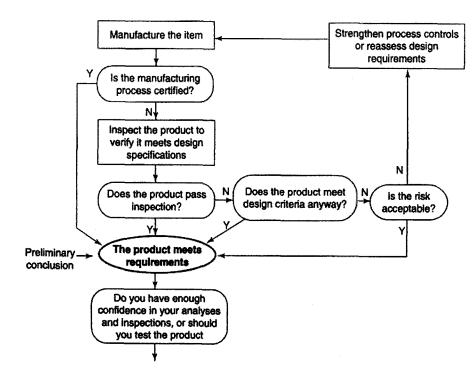


Figure 1.15 Flow diagram for design/process verification steps during manufacture of an optical instrument. The advisability of testing may be indicated. (From Sarafin.²⁶)

1.10 References

- 1. Walles, S. and Hopkins, R.E., "The orientation of the image formed by a series of plane mirrors," *Appl. Opt.* 3, 1447, 1964.
- 2. Steinberg, D.S., Vibration Analysis for Electronic Equipment, Wiley, New York, 1973.
- 3. Genberg, V., "Structural analysis of optics," Chapter 8 in Handbook of Optomechanical Engineering, CRC Press, Boca Raton, 1997.
- 4. Hatheway, A.E., "Review of finite element analysis techniques: capabilities and limitations," *Proceedings of SPIE* CR43, 367, 1992.
- Hatheway, A.E., "Unified thermal/elastic optical analysis of a lithographic lens," Proceedings of SPIE 3130, 100, 1997.
- 6. Genberg, V. and Michels, G., "Design optimization of actively controlled optics," *Proceedings of SPIE* **4198**, 158, 2000.
- 7. Englehaupt, D., "Fabrication methods," Chapter 10 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, 1997.
- 8. Vukobratovich, D., "Optomechanical design principles," Chapter 2 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, 1997.
- 9. Vukobratovich, D., private communication, 2001.
- 10. McCay, J., Fahey, T., and Lipson, M., "Challenges remain for 157-nm lithography," Optoelectronics World, Supplement to Laser Focus World, 23, S3, 2001.
- Wendt, R.G., Miliauskas, R.E., Day, G.R., MacCoun, J.L., and Sarafin, T.P. "Space mission environments," Chapter 3 in *Spacecraft Structures and Mechanisms*, T.P. Sarafin and W.J. Larson, eds., Microcosm, Torrance and Kluwer Academic Publishers, Boston, 1995.
- Souders, M. and Eshbach, O.W., eds., *Handbook of Engineering Fundamentals*, 3rd ed., Wiley, New York, 1975.
- 13. Mantell, C. L., ed. *Engineering Mat erials Handbook*, McGraw-Hill, New York, 1958.
- 14. Elliott, A. and Home Dickson, J. Laboratory Inst ruments-Their Desi gn and Application, Chemical Pub. Co., New York, 1960.
- 15. Marker, A.J. III, Hayden, J. S., and Speit, B., "Radiation resistant optical glasses," *Proceedings of SPIE* 1485, 55, 1991.
- 16. Yoder, P.R., Jr., Opto-Mechanical Systems Design, 3rd ed, CRC Press, Boca Raton, 2005.
- 17. Harris, D.C. Materials for Infrared Windows and Domes, SPIE Press, Bellingham, 1999.
- 18. MIL-STD-210, Climatic Information to Determine Design and Test Requirements for Military Systems and Equipment, U.S. Dept. of Defense, Washington.
- 19. Musikant, S. and Malloy, W.J. "Environments stressful to optical materials in low earth orbit," *Proceedings of SPIE* 1330:119, 1990.
- 20. Shipley, A.F. "Optomechanics for Space Applications," SPIE Short Course Notes SC561, 2007.
- 21. ISO 10109, *Environmental Requirements*, International Organization for Standardization, Geneva.
- 22. U.S. MIL-STD-810, *Environmental Test Met hods and Engineering Guidelines*, Superintendent of Documents, U.S. Government Printing Office, Washington.

- 23. ISO Standard ISO 9022, *Environmental Test Methods*, International Organization for Standardization, Geneva.
- 24. Parks, R.E., "ISO environmental testing and reliability standards for optics," *Proceedings of SPIE* **1993**, 32, 1993.
- 25. MIL-STD-1540, Product Verification Requirement s for launch, Upper Stage, and Space Vehicles, U. S. Dept. of Defense, Washington.
- Sarafin, T. P. "Developing confidence in mechanical designs and products," Chapter 11 in Spacecraft Structures and Mechanisms, T. P. Sarafin, ed., Microcosm, Torrance and Kluwer Academic Publishers, Boston, 1995.
- 27. Paquin, R. A. "Advanced materials: an overview," *Proceedings of SPIE* CR43, 1997:3.
- 28. Kumler, J. (2004). "Changing glass catalogs," oemagazine 4:30.
- 29. Walker, B. H., "Select optical glasses," *The Photonics Design and Appl ications Handbook*, Lauren Publishing, Pittsfield: H-356, 1993.
- 30. Zhang, S. and Shannon, R. R., "Lens design using a minimum number of glasses," *Opt. Eng.* 34, 1995:3536.
- 31. CodeV Reference Manual, Optical Research Associates, Pasadena, CA.
- 32. Laikin, M., Lens Design, Marcel Dekker, New York, 1991.
- Cox, A., A Syst em of Opt ical Design: The Basi cs of Image Assessment and of Design Techniques with a Sur vey of Current Lens Types , Focal Press, Woburn, 1964.
- 34. Doyle, K.B. and Bell, W.M., "Thermo-elastic wavefront and polarization error analysis of a telecommunication optical circulator," *Proceedings of SPIE* **4093**, 2000:18.
- 35. Doyle, K.B., Genberg, V.L., and Michels, G.J., "Numerical methods to compute optical errors due to stress birefringence," *Proceedings of SPIE* **4769**, 2002a:34.
- 36. Doyle, K.B., Hoffman, J.M., Genberg, V.L., and Michels, G.J., "Stress birefringence modeling for lens design and photonics," *Proceedings of SPIE* **4832**, 2002b:436.
- 37. Kimmel, R.K. and Parks, R.E., ISO 10110 Opt ics and Opt ical Inst ruments— Preparation of drawings for optical elements and systems: A User's guide, Optical Society of America, Washington, 1995.
- Curry, T.S. III, Dowdey, J. E., and Murry, R.C., *Christensen's Physics of Diagnostic Radiology*, 4^{th.} ed., Lea and Febiger, Philadelphia, 1990.
- 39. Setta, J.J., Scheller, R.J., and Marker, A.J., "Effects of UV-solarization on the transmission of cerium-doped optical glasses," *Proceedings of SPIE* **970**, 1988:179.
- 40. Schott, *Technical Information, TIE-42, Radi ation Resistant Glasses*, Schott Glass Technologies, Inc., Duryea, Pennsylvania, August, 2007.
- 41. Paquin, R.A., "Materials for optical systems," Chap 3 in Handbook of Optomechanical Engineering, A. Ahmad, ed., CRC Press, Boca Raton, 1997a.
- 42. Paquin, R.A., "Metal mirrors," Chap 4 in *Handbook of Opt omechanical Engineering*, A. Ahmad, ed., CRC Press, Boca Raton, 1997b.
- Paquin, R.A., Dimensional instability of materials: how critical is it in the design of optical instruments?," *Proceedings of SPIE* CR43, 1992:160.
- 44. Jacobs, S.F., "Variable invariables—dimensional instability with time and temperature," *Proceedings of SPIE* 1992:181.
- 45. Ginsberg, R.H., "Outline of tolerancing (from performance specification to toleranced drawings)," Opt. Eng. 20, 1981:175.

- 46. Willey, R.R., "Economics in optical design, analysis and production," *Proceedings* of SPIE **399**, 1983:371.
- 47. Smith, W. J., Modern Lens Design, 3rd. ed. McGraw-Hill, New York, 2005.
- 48. Plummer, J., "Tolerancing for economies in mass production of optics," *Proceedings of SPIE* 181, 1979:90.
- 49. Fischer, R.E., and Tadic-Galeb, B. Optical Syst em Desi gn, McGraw-Hill, New York, 2000.
- 50. Smith, W. J., "Fundamentals of establishing an optical tolerance budget," *Proceedings of SPIE* 531, 1985:196.
- Fischer, R. E., "Optimization of lens designer to manufacturer communications," *Proceedings of SPIE* 1354, 1990:506.
- 52. Willey, R. R. and Durham, M. E., "Ways that designers and fabricators can help each other," *Proceedings of SPIE* 1354, 1990:501.
- 53. Willey, R. R. and Durham, M. E., "Maximizing production yield and performance in optical instruments through effective design and tolerancing," *Proceedings of SPIE* **CR43**, 1992:76.
- 54. Willey, R. R. and Parks, R. E., "Optical fundamentals," Chapter 1 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, 1997.
- 55. Englehaupt, D., private communication, 2002.
- 56. Malacara, D., Optical Shop Testing, Wiley, New York, 1978.
- 57. DeVany, A. S., Master Optical Techniques, Wiley, New York, 1981
- 58. Karow, H. K., Fabrication Methods for Precision Optics, Wiley, New York, 1993.
- Habicht, W. F., Sarafin, T. D., Palmer, D. L., and Wendt, R. G., Jr., "Designing for producibility," Chapter 20 in *Spacecraft Structures and Mechanisms*, Sarafin, T. P., Ed., Microcosm, Inc., Torrance, and Kluwer Academic Publishers, Boston, 1995.
- Sarafin, T.P., Heymans, R.J., Wendt, R.G., Jr., and Sabin, R.V., "Conceptual design of structures," Chapter 15 in *Spacecraft Structures and Mechanisms*, Sarafin, T.P., ed., Microcosm Inc., Torrance and Kluwer Academic Publishers, Boston, 1995.

CHAPTER 2 The Optic-to-Mount Interface

The prime purpose of the optic-to-mount interface is to hold the component (lens, window, filter, shell, prism, or mirror) in its proper position and orientation within the optical instrument throughout its useful life, including storage and shipping. This implies the presence of mechanical constraints, i.e., external forces that limit component motion—even when the temperature changes or when external mechanical disturbances occur. The importance of these constraints, the advantages of semikinematic-type mounting techniques, and the alternatives used when that technique is not appropriate are considered here. We concentrate first on rotationally symmetric optics such as lenses and mirrors and then introduce the reader to typical interfaces for prisms and larger mirrors. This chapter concludes with considerations of ways to seal the optic-to-mount interface so as to maintain a favorable environment within the instrument.

2.1 Mechanical Constraints

2.1.1 General considerations

Under all operating conditions, it is important that each optical component be constrained so it remains within decentration, tilt, and axial spacing budgets and that induced stresses, surface deformations, and birefringence are tolerable. Both lateral and axial constraints are needed for each component. Assuming the component to be constrained is stiff, the ideal mechanical interfaces would be *kinematic*. Then, all six degrees of freedom (three translations and three rotations) would be independently constrained without redundancy. A true kinematic interface would apply exactly six forces at six points so that no bending moments can be transferred to the optic. Figure 2.1 schematically illustrates such a mounting configuration for a cube-shaped prism. The induced stress (force per unit area) at each point contact would undoubtedly be large, even for small applied forces, because the areas are infinitesimal. For this reason, it is practically impossible to mount any optic in a true kinematic manner.

A *semikinematic* interface is one with the same six forces, but each of these acts over a small area on the optic to distribute the force and reduce the stress to a tolerable level. Design of such a mounting is a compromise between the need to make the contact areas large enough to keep the stresses from getting too big and the need to make those areas small enough so moments transmitted through those contacts are not large enough to distort the optic. Reducing the magnitudes of the applied forces is generally not an option because movement under acceleration must be prevented. Semikinematic support of optics generally is practical only for prisms and small mirrors in which the thickness is at least one fifth of the largest dimension.

Rotational symmetry of most optical system apertures leads to circular symmetry in mounts for lenses, windows, and many mirrors. This usually simplifies the design of the associated mechanical housings because these components can be constrained by applying force continuously around their rims. Generically, this would be termed a *nonkinematic* or *overconstrained* mounting, i.e., one in which contact can occur at many points. Large forces

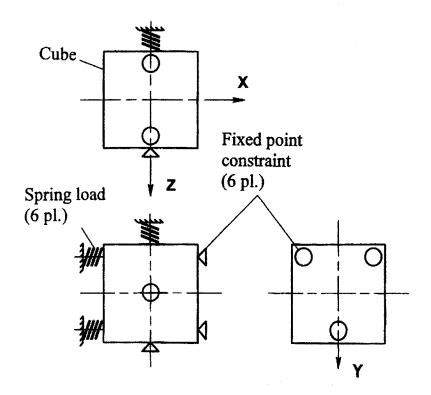


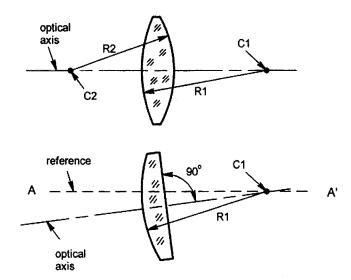
Figure 2.1 A cube-shaped prism kinematically constrained by forces acting at six points.

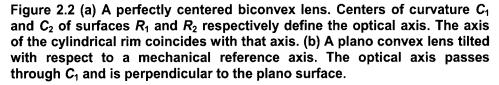
acting on such an optic may deform it and degrade its optical performance, especially if the optic is thin and hence flexible. Before we discuss this type of mounting, we should understand the process generally used to create the cylindrical rim on such optics.

2.1.2 Centering a lens element

Because of the inherent rotational symmetry of spherical surfaces and their aberrations, it is customary for an optical designer to start a lens design by defining a straight line in space and locating all surfaces having optical power, i.e., curved surfaces, symmetrically about that line. If the centers of curvature of a single component or an ensemble of such surfaces lie on this same line, we define the line as the optical axis and that system as being centered. Figure 2.2(a) illustrates a perfectly centered biconvex lens element. The centers C_1 and C_2 of the surfaces R_1 and R_2 define the optical axis of the lens. The ground rim of the lens is cylindrical with the axis of that cylinder coincident with the optical axis.

A lens with a plano (i.e., flat) surface is shown in Fig. 2.2(b). Here, the plano-convex lens is tilted with respect to an arbitrarily oriented line A-A' (dashed line). This line could well be the mechanical axis of a cell into which the lens is to be mounted. The optical axis of the lens, in this case, is designated as the line that passes through the center C_1 of R_1 and is perpendicular to R_2 . Symmetry exists only about this axis. Systems with intentionally tilted surfaces such as optical wedges or those that need asymmetry for aberration correction reasons cannot be considered to be centered or rotationally symmetric. We will not consider such cases here.





Practically all lens elements have cylindrical rims produced by grinding the edge of the element on an edging machine after the refracting surfaces are polished. This cylinder defines a mechanical axis of the lens that may or may not coincide with the lens's optical axis, depending on how the latter axis is oriented while the rim is ground. Figure 2.3 shows a biconvex lens mounted on the bell (or chuck) of one type of lens-centering and edging machine. The left surface of the lens is seated against the edge of the bell and is secured in place with an adhesive such as wax or pitch. It is held on a vacuum chuck in some machines. That surface is automatically located on the rotational axis of the spindle. By warming the adhesive to soften it or by partially releasing the vacuum and judiciously pushing the lens sideways the operator aligns the lens to the machine's rotational axis. Care must be exercised to ensure that the glass remains in uniform contact with the bell during the edging process.

Figure 2.4(a) illustrates an element that is perfectly centered. Its surface centers C_1 and C_2 are on the spindle mechanical axis so the optical axis coincides with the axis of the cylindrical rim as well as with the spindle axis of rotation. The edge thickness is uniform all around.

Three possible cases of centering errors created in the lens during edging are shown in Fig. 2.4(b) through (d). These errors are greatly exaggerated for clarity. They may be caused by an irregularity, or burr, on the rim of the bell, a speck of dust on that rim, or failure to keep the lens in contact with the bell until the adhesive solidifies. View (b) shows the lens decentered so both surface centers lie off the spindle axis by a distance d.

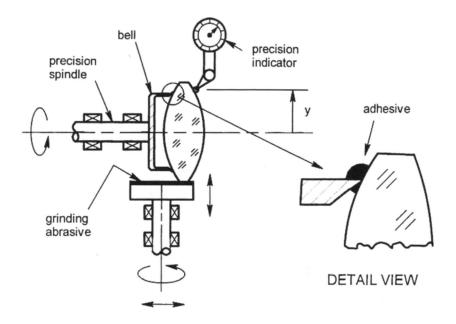


Figure 2.3 Schematic setup for centering and edging a lens element on a precision spindle. The detail view shows one means for securing the lens on the bell of the machine.

In view (c), the optical axis is tilted and the surface centers lie off the mechanical axis by approximately the same distance but in opposite directions. In view (d), the element is tilted about C_1 . R_1 is centered, but R_2 is tilted. In all the misaligned cases, (b) through (d), the exposed surface will wobble as the spindle rotates. If we can measure this wobble and reduce it to an insignificant amount, the lens will be adequately centered.

In Fig. 2.3, the device used to measure wobble of the second lens surface is shown as a mechanical dial gauge at a distance y from the rotation axis. Other types of gauges (capacitive, pneumatic, etc.) can be used. The measurement's result is called full indicator movement or FIM.¹ Optical techniques also can be used. The simplest is to observe the image of a light source reflected from the wobbling surface (see Fig. 2.5). Motion of the image indicates that C_2 is not on the spindle axis. By moving the lens on the bell this error can be reduced. This test is limited by the ability of the eye to discern small movements of the image location) to magnify this movement would help.

Another optical test technique can be used if the edging machine has a hollow spindle (see Fig. 2.6). Here, the beam from a visible laser is passed through a special lens system, through the lens being centered, and through the spindle. It then falls onto a fourquadrant detector. The special optical system is customized so the refractive effects of the

¹ FIM has replaced the formerly used term "total indicator runout" (TIR) in common usage and in ANSI Specification Y14.5, *Dimensioning and Tolerancing*.

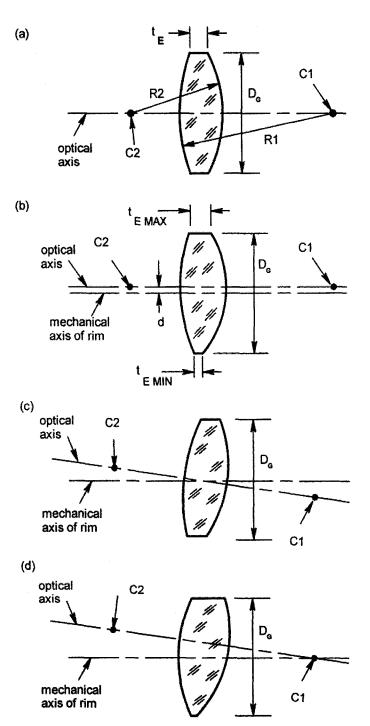


Figure 2.4 (a) A perfectly centered lens element, (b) an element decentered so its optical axis is parallel to, but off the mechanical axis by distance d, (c) an element whose centers C_1 and C_2 are decentered equally, but on opposite sides of the mechanical axis so it is tilted, and (d) an element with C_1 centered but tilted so C_2 is decentered.

lens on the bell can be compensated for and the beam focused into an appropriately sized spot on the detector. A rotating misaligned lens will cause the spot to move in a circular path as the spindle is rotated. The detector is mounted on a dual axis translation stage to allow it to be centered on the center of this path. The cyclic X and Y signals from the detector will indicate the centering error on a monitor. As the lens is moved toward its centered position, the radius of the spot path will decrease. When the lens is properly aligned, the signals will not vary significantly and the spot will stand still.

For production runs of lenses, a variation of the setup shown in Fig. 2.3 is frequently used. One is shown schematically in Fig. 2.7. Bells on two spindles rotating together on a single axis face each other. One bell can be moved axially and is spring-loaded toward the other bell. The lens element to be centered is placed between the bells and the moveable bell is moved in to capture the lens. Usually, the element will be more or less

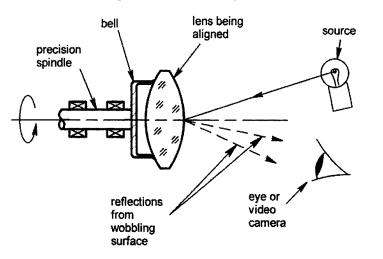


Figure 2.5 Sensing centration errors by observing a reflection from the exposed surface of the rotating element.

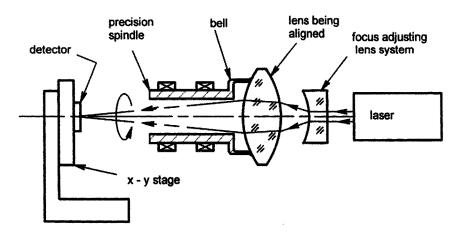


Figure 2.6 Sensing centration errors by projecting a transmitted image onto a quadrant detector.

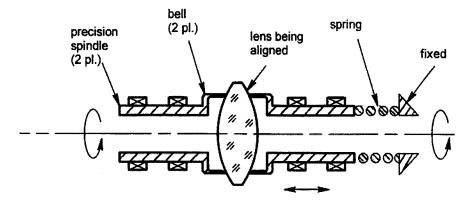


Figure 2.7 Double-bell centering a lens element.

misaligned, but the applied axial preload will tend to move the lens toward the centered condition if the curvatures of the surfaces are sufficient for the radial components of the preload against points on the curved surfaces to overcome friction at the contacts with the bells. According to Karow,¹ the following equation involving the heights of contact y and absolute radii R for each surface must be satisfied for centering to take place:

$$(2y_1/R_1) + (2y_2/R_2) \ge 4\mu.$$
 (2.1)

Here, μ is the coefficient of friction. A typical value for μ , applicable to glass sliding on polished steel, is about 0.14. The left side of Eq. (2.1) must then be at least 0.56. This value of μ may or may not apply in any specific case. Hence, the equation must be considered to be an approximation. Example 2.1 applies Eq. (2.1) to a typical case.

Example 2.1: Self-centering effect due to axial preload on curved lens surfaces. (For design and analysis, use File 2.1 of the CD-ROM) Two biconvex lenses, each with a diameter of 50.000 mm (1.968 in.) have surface radii of (a) R_1 of 175.000 mm (6.890 in.) and R_2 of -120.00 mm (-4.724 in.) and (b) R_1 of 200.00 mm (7.874 in.) and R_2 of -200.000 mm (7.874 in.). The height of contact y is 24.000 mm (0.945 in.). Should these lenses self-center under axial preload in a double-bell centering machine? Note: Use absolute values for radii. (a) Applying Eq. (2.1) for lens (a): [(2)(0.945)/6.890] + [(2)(0.945)/4.724] = 0.274 + 0.400 = 0.674This is larger than 0.56 so the lens should self-center. (b) Applying Eq. (2.1) for lens (b): [(2)(0.945)/7.874] + [(2)(0.945)/7.874] = 0.240 + 0.240 = 0.480This is smaller than 0.56 so the lens should not self-center. Long radii surfaces will not self-center by this technique. These and higher precision applications probably require the use of a calibrated set up, similar to that shown in Fig. 2.6, where the centering error can be measured. Means for establishing the desired alignment also is needed.

Some lens drawings specify the maximum edge runout (ERO) of the lens' OD. This error can be measured directly with a precision mechanical indicator in a setup as shown schematically in Fig. 2.8. Both surfaces of the lens must first be aligned on the bell so its optical axis coincides with the rotation axis of the spindle. ERO can be measured directly with a precision mechanical indicator as depicted in the figure.

Other lens drawings specify the allowable beam deviation angle δ . A decentered or tilted lens that causes beam deviation is said to have a built-in geometric wedge. The transmitted beam always tilts toward the thickest point on the wedge. Such a lens is illustrated in Fig. 2.9(a). The wedge lies between two spherical caps. For comparison, view (b) shows a properly centered lens that effectively has a plane-parallel plate of thickness t_A between the spherical caps.

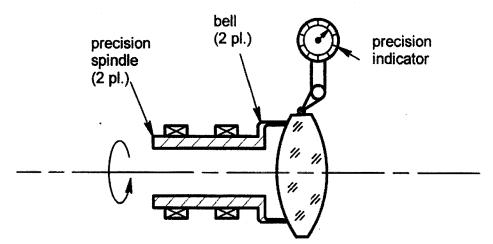


Figure 2.8 Schematic of a technique for measuring edge runout (ERO) of the rim of an imperfectly centered element.

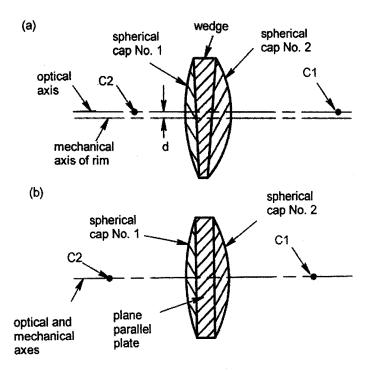
In a simple test for deviation, we define the geometric wedge as θ . The maximum and minimum edge thicknesses of the lens at a height $\pm y$ are measured. Then:

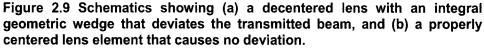
$$\theta = \frac{\left(t_{EMAX} - t_{EMIN}\right)}{\left(2y\right)} \tag{2.2}$$

and

$$\delta = (n-1)\theta, \tag{2.3}$$

where both angles are in radians. To convert them to minutes of arc, we divide by 0.00029.





Example 2.2: Deviation produced by a wedged lens. (For design and analysis, use File 2.2 of the CD-ROM) A lens has edge thicknesses of 4.000 mm (0.157 in.) and 4.050 mm (0.159 in.), both measured at heights y of 24.800 mm (0.976 in) at opposite points on the lens rim. The glass refractive index is 1.617. What are the wedge angle θ and approximate deviation δ ? From Eq. (2.2), $\theta = (0.159 - 0.157) / [(2)(0.976)] = 0.001$ radian From Eq. (2.3), $\delta = (1.617 - 1.000) (0.001) = 0.00062$ radian or 0.00062 / 0.00029 = 2.13 arcmin

A more direct method for measuring deviation would be to align the lens' optical axis to the rotation axis of a hollow spindle, pass a collimated beam through the lens as it rotates, and measure the diameter of the circular runout path of the image formed at the lens' focal plane using, for example, a traveling microscope. Figure 2.10(a) illustrates this technique for a positive lens element. The beam deviation δ is given by dividing the image runout path diameter by twice the lens' focal length.

A similar technique can be used if the lens has a negative focal length. Figure 2.10(b) illustrates one such case. A positive lens is inserted between the collimator and the lens under test to form an image to the left of the test lens at the focal point of that lens. The test lens will then recollimate the beam so the image can be observed with a telescope. The telescope is provided with a reticle calibrated in angular units. The deviation of the beam emerging from the test lens can then be measured directly while the spindle is rotated slowly. The test lens is moved on the bell of the spindle to minimize the error.

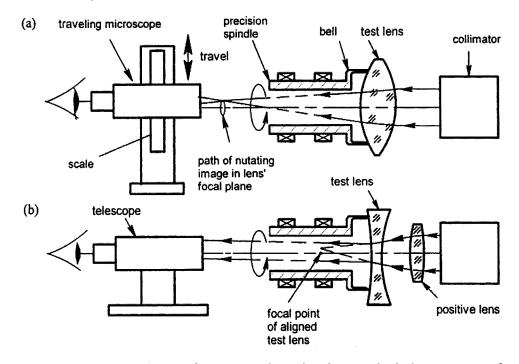


Figure 2.10 Techniques for measuring the beam deviation errors of improperly centered lenses: (a) for a positive lens, and (b) for a negative lens.

The process of edging the rim of a lens element to diameter and alignment with the optical axis is sometimes combined with production of other features such as bevels. For example, consider Fig. 2.11. The desired element configuration is shown in view (a). It has the usual cylindrical rim, a step-shaped bevel, and a 60-deg bevel on the concave side. Edging starts by mounting and aligning the polished lens blank on the bell of an edging machine [see view (b)]. As shown in view (c), the rim is ground parallel to the spindle axis and to a specified OD with grinding wheel No. 1. In view (d), we see the same grinding wheel shifted into position to grind the step bevel. This also involves producing a specified OD for the cylindrical portion of that bevel and a surface perpendicular to the spindle axis. The latter surface might be used later as a mechanical reference to clamp the lens in its mount. Grinding wheel No. 2 is then brought into position to grind the angled bevel. This bevel is usually not so critical because it removes unwanted material, but is not used mechanically as a reference. The edging machine represented here probably would operate under computer control. Such a device is termed a computer numerically controlled (CNC) machine.

Before leaving the subject of lens centration, we should mention certain documents that specify how one expresses symbolically on a drawing what one wants in the way of dimensions and functions. In years past, specifications for optical instruments procured for U.S. Government use have referred to military specifications, standards, and other government publications. These documents defined general requirements and provided guidance for the selection of materials, design, inspection, and testing of a variety of equipment items. Since 1994, use of national and international standards rather than military specifications has been encouraged in the U.S. The International Organization for Standards (ISO), headquartered in Geneva, Switzerland, spearheads the development of international optical standards through ISO Technical Committee 172, Optics and Optical Instruments. The Optics and Electro-Optics Standards Council (OEOSC) acts as the administrator of national optical standards for the U.S. and is also responsible for supporting ISO/TC 172 through a U.S. Technical Advisory Group (TAG). The American National Standards Institute (ANSI) formed a committee called ASC/OP "Optics and Electro-Optical Instruments" to develop U.S. national standards for potential incorporation into international standards.

Documents particularly pertinent to the present discussion are ANSI Y14.5, *Dimensioning and Tolerancing*, ANSI Y14.18, *Optical Parts*, and ISO 10110, *Optics and Optical Instruments - Preparation of Drawings for Optical Elements and Systems*. To help the user understand the use of the latter document, The Optical Society of America (OSA) published a very useful user's guide edited by Kimmel and Parks.²

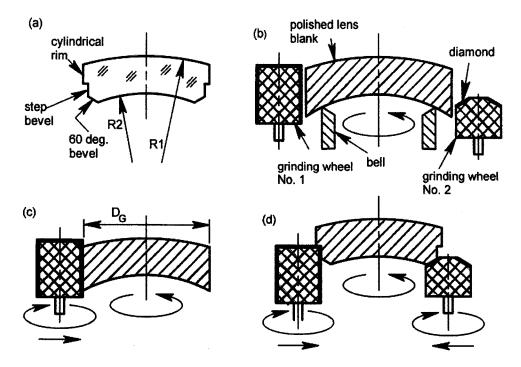


Figure 2.11 Producing various mechanical surfaces in addition to the rim on a lens element during edging.

2.1.3 Lens interfaces

2.1.3.1 The rim-contact interface

A lens element whose OD is very nearly equal to the ID of its mechanical mount is said to be a *rim-contact* configuration. Figure 2.12 shows this configuration schematically. The radial clearance Δr around the lens may be as small as 0.005 mm (0.0002 in.) and still be large enough for the lens to be installed (carefully) into the mount. If not held firmly by axial preload against some mechanical reference surface such as a shoulder in the mount, a lens can tilt within this clearance by an angle as large as approximately $2\Delta r$ / t_E in radians, or 6875.5 $\Delta r/t_E$ in arc minutes before the front and back corners of the rim touch the ID of the mount. For example, for $\Delta r = 0.005$ mm and $t_E = 5$ mm, the maximum tilt is 6.88 arcmin. This angle should be compared to the tolerance for element tilt to see if the design is acceptable. Obviously, the tilt occurring with a given radial clearance is reduced if the lens rim is longer or the radial clearance reduced. Designs with larger radial clearance can suffer larger tilts. A condition in which this tilting problem might occur is if the preload is dissipated at high temperature when the mount expands away from the lens and allows the lens to move freely. This subject is considered in more detail in Chapter 14.

Typically, when a lens is inserted into its mount, the mount is held so the axis of the hole into which the lens is to go is vertical and the lens (attached to a suction cup) is lowered carefully into the hole until it reaches a shoulder or spacer that positions the lens axially. If the radial clearance around the lens rim is smaller than 0.0002 in. (0.005 mm) and the lens is somewhat tilted, the lens may jam in place before it is seated against the axial constraint. This can chip the lens rim or otherwise damage the lens. Removal of the lens may then be very difficult—especially if other lenses are already installed, so access from below to push the tilted element upward does not exist.

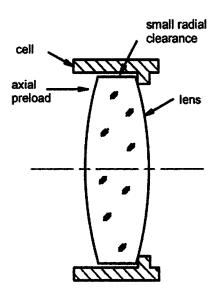


Figure 2.12 A rim-contact lens mounting.

Figure 2.13 illustrates a design feature that minimizes this potential problem. Here the lens rim is fine ground as a sphere with the radius equal to one-half the lens' OD. The rim is then a short centralized section of the surface of a ball that fits easily into the hole at any angular orientation. Ideally, the high point on the spherical rim should be in the plane normal to the axis and containing the lens' CG.

A variation of the spherical rim principle is to provide a lens with a crowned rim. Here the radius of the rim is longer than one-half the OD. The allowable range of tilt without jamming is smaller with this type of rim than with the spherical one, but considerably larger than would obtain with a cylindrical rim. Long spacers for high-precision multiple-lens assemblies also are frequently made with crowned rims.

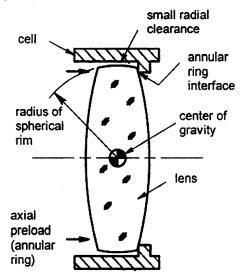


Figure 2.13 A spherical-rim lens mounted in a cell with minimal radial clearance.

Although the spherical or crowned rim requires some special optical fabrication steps, the costs of tooling and labor are increased only slightly since the contour of either surface does not need to be precise. The tolerances on the lens OD and mount ID are, of course, relatively tight. An exception to this requirement is the case of the lathe-assembled lens in which the cell ID is machined at assembly to closely match the OD of the specific lens being mounted. This technique is described in Section 4.3.

Curving the rim would be a worthwhile design feature as a means for preventing damage at assembly when the lenses have maximum value and replacement could be expensive and affect the production schedule. This is especially true if the lens is part of a matched set, i.e., lenses made to a design optimized for specific glass-melt parameters, for as-manufactured thicknesses of other components in the system, or if the cell ID has been customized for the particular lens being installed.

2.1.3.2 The surface-contact interface

To alleviate the problems we might expect with rim-contact designs, it is frequently possible to configure the lens-to-mount interface so that the interfaces of the lens to its mount provide for the mount to contact both the polished optical surfaces and not the rim.

We define this as a *surface-contact* mounting. Figure 2.14 illustrates such a design conceptually. The motion required to center R_1 is a counterclockwise rotation.

Many of the higher-precision mounting methods described in later chapters utilize this principle. One advantage of this configuration is that the mechanical interfaces are on the most accurately made surfaces on the lens, i.e., the polished ones, rather than against secondary surfaces such as the ground rim. Another advantage is that errors in grinding the rim do not affect alignment—as shown in Fig. 2.15.

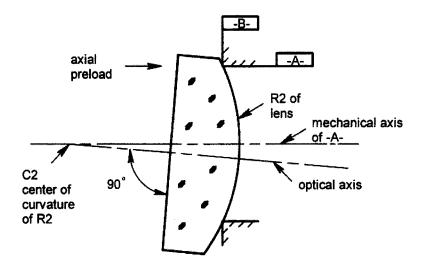


Figure 2.14 Concept for a surface contact lens mounting.

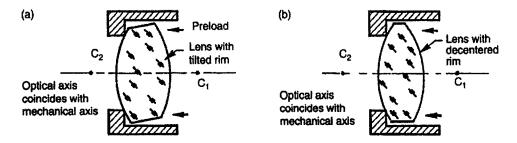


Figure 2.15 Accurate lens edging is not required with a surface contact interface.

The mechanical surfaces created in the mount to interface with a lens in either a rimcontact or surface-contact design need to be accurately machined in order for the lens to be aligned properly with other components of the optical system. Figure 2.16 illustrates schematically some things that can happen if this is not the case. In view (a), the bore and shoulder are tilted so the lens is also. In view (b), these features are decentered, as is the lens. In view (c), the lens is preloaded against a wedged spacer so it is tilted. Finally, in view (d), the lens is preloaded by a retainer that has either a very close thread fit or is piloted into the mount. Contact with the aligned lens is unsymmetrical. This can lead to excessive stress concentration.

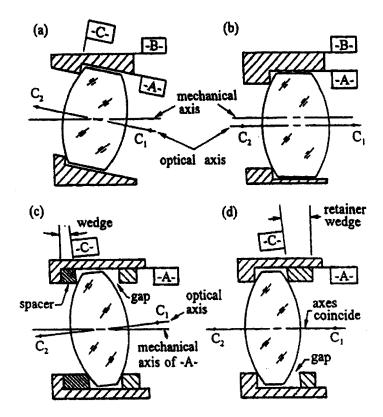


Figure 2.16 Lens alignment errors in either rim- or surface-contact mountings that result from errors in machining the mount.

2.1.3.3 Contacting flat bevels

When a lens is interfaced on secondary (ground) surfaces, such as flat bevels, a high level of care must be exerted in creating those reference surfaces so as to control tilt errors to within the allowable limits for the particular design. Achievement of the highest precision in alignment requires the use of an air-bearing spindle to obtain the minimum possible instrumental error due to spindle-axis wobble. In all such cases, precision error detection instrumentation must be provided for measuring and minimizing the misalignment of the particular element during processing.

2.1.4 Prism interfaces

Figure 2.17, indicates how the required six DOF constraints might be applied to a simple cube prism. In view (a), six identical balls are attached to three mutually perpendicular flat surfaces. The dashed construction lines indicate how the balls are located symmetrically. If the prism is held in contact with all six balls, it will be constrained kinematically. The prism base rests on the three contacts parallel to the X-Z plane so it cannot translate in the Y direction or tilt about the X- or Z-axes. Two contacts parallel to the Y-Z plane prevent translation along the X axis and rotation about the Y axis. The single contact at the X-Y

plane controls translation along the Z axis. A single force applied to the near corner of the prism and aimed toward the corner would hold the prism in place. Ideally, this force should pass through the center of gravity (CG) of the prism. This is not always feasible.

Three forces, each applied normal to one of the exposed prism surfaces and directed toward a contact point or toward a point midway between adjacent contact points also would hold the prism. Unfortunately, this particular multiple-force configuration is not very practical for an optical application since all prism faces would be at least partially obscured. By increasing the separations of some of the balls, it may be possible to clear the apertures without destroying symmetry or the kinematic condition. However, this would not help the stress concentration problem inherent with the point contacts.

Figure 2.17(b) shows conceptually how the point contacts on balls could be replaced by small areas on raised flat pads to distribute the mechanical preload forces on the prism surfaces. The design now is semikinematic. The pads may have any practical shape such as square (as shown) or perhaps circular. If the pads are machined or lapped coplanar and mutually perpendicular, introduction of stress at the contacts with a perfect cube prism is minimized. Distortion of the prism also is minimized.

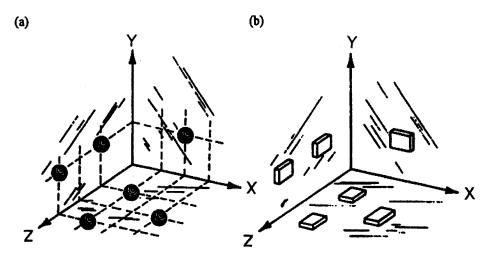


Figure 2.17 (a) Kinematic and (b) semikinematic position-defining registration surfaces intended for interfacing with a cube-shaped prism (not shown).

In practice, it is very difficult (read "expensive") to make the pads touching any prism surface as coplanar as the polished optical surface, which typically has shape imperfections that are no larger than a small fraction of the wavelength of light. If the pad surfaces are not accurately aligned to each other and the prism is forced into intimate contact with them, the prism could be distorted by introduced moments, or one or more interfaces could degenerate into line or point contacts, thereby causing stress to increase. Figure 2.18 illustrates these conditions for two possible pad-to-prism surface mismatches. Assuming that the mount is more rigid than the prism, the dashed lines in each view show possible distortions of the adjacent optical surface that are caused by imposed moments. To keep the optical performance of the prism within allowable bounds, the tolerances on coplanarity errors for the pads must be essentially the same as the allowable surface figure errors for that prism. Typically, careful lapping of the mechanical surfaces on a flat surface can achieve coplanarity within about 20×10^{-6} in. (0.5 µm). Single-point diamond turning of the interface should reduce these errors to less than 4×10^{-6} in. (0.1 µm).

The means used to clamp the prism against its reference pads in a semikinematic mount include a variety of spring types, such as cantilevered clips and "straddling" springs. These constraints are discussed in detail in Chapter 7.

Frequently it is necessary for optical subassemblies, including their mounts, to be removed from the optical instrument and replaced in the identical location and orientation. Semikinematic interfaces allow this to be accomplished with high accuracy. Figure 2.19(a) shows (schematically) one such interface as described by Strong.⁴ It consists of lower and upper plates; the optic is attached to the upper plate (thereby forming the removable subassembly) and the lower plate is permanently attached to the instrument's structure. Attached to the bottom of the upper plate are three balls, symmetrically located on a given "bolt-circle" diameter. The lower plate has two "sockets" (a "vee" and a trihedral-shaped hole) and a flat surface. This is sometimes referred to as a "Kelvin clamp." The balls fit repeatably into the sockets, establishing six positional constraints whenever the plates are clamped together. The three balls and the three sockets can be manufactured or purchased with posts that can be pressed into holes drilled in the plates. A conical socket can be substituted for the trihedral with slight loss of accuracy (because contact occurs on a line rather than three points).

Figure 2.19(b) shows a similar interface mechanism in which the three balls mate with three radially directed "vees." Kittel⁵ has shown how "vees" can be formed by pressing dowel pins into three pairs of holes drilled into the lower plate. The pins form three parallel grooves, [see Fig. 2.19(c)]. This construction is less expensive than machining the "vees" directly into the plate.

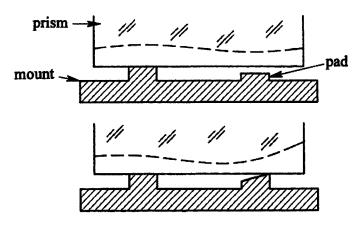


Figure 2.18 Effects of coplanarity errors on a prism forced into contact with two registration pads of a small area.

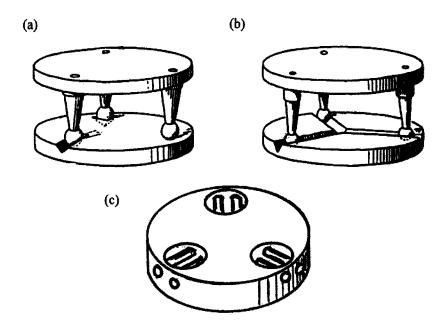


Figure 2.19 (a) and (b), Concepts for zero degree of freedom separable interfaces. (From Strong.⁴) (c) Concept for providing "vees" with multiple parallel rods. (From Kittel.⁵)

A nonkinematic technique frequently used for mounting prisms involves glass-to-metal bonds with thin layers of adhesives. These designs generally result in reduced interface complexity and compact packaging while providing mechanical strength that is adequate for withstanding the severe shock, vibration, and temperature changes characteristic of military and aerospace applications. This mounting technique is detailed in Chapter 7. Glass-to-metal bonding is also used in some less rigorous applications because of its inherent simplicity and reliability.

A stress-free technique for mounting prisms inserts a series of flexures between the prism and the mount. These flexures can be of a variety of configurations; their prime purpose is to isolate the optic from temperature expansion/contraction effects on dissimilar materials and to prevent the introduction of moments into the glass. This mounting technique is also described in Chapter 7.

2.1.5 Mirror interfaces

Semikinematic clamped and adhesive-bonded mechanical mounts typically are used to support small (i.e., stiff) mirrors, while the mounts for larger mirrors always are nonkinematic because the latter optics tend to be thin relative to their maximum dimension and therefore are relatively flexible. Multiple axial and radial supports must be provided for larger mirrors to minimize the self-weight deflection of optical surfaces between supports. These mountings are discussed in Chapter 11.

2.1.6 Interfaces with other optical components

Windows, shells, and domes are mounted nonkinematically by such techniques as potting them into their mounts with an elastomeric material or clamping them against a machined surface of the mount. A moisture, dust, and pressure seal is provided in some cases. Filters also are mounted nonkinematically using techniques similar to those used for mounting lenses and small windows. Examples of all of these optical component mountings are described in Chapter 5.

2.2 Consequences of Mounting Forces

Preloads applied to the surfaces of optical components compress (or strain) the optic and produce corresponding elastic stresses within the material. In Chapter 13, we show how to estimate the magnitudes of these stresses and to determine if they appear to be tolerable. As mentioned earlier, forces concentrated on small surface areas cause localized stresses of high magnitude. These are particularly undesirable since they can lead to excessive distortion of malleable materials such as plastics or some crystals, or breakage of brittle materials such as glass or other crystals. Reaction forces are also exerted on the mount. These can distort the mount temporarily or permanently or, in extreme cases, cause the mount to fail.

Applied forces may introduce birefringence (inhomogeneity of the refractive index) into normally isotropic optical materials. Birefringence affects the propagation speeds of the perpendicular and parallel components of polarized light passing through the material, so these components become out of phase. The magnitude of birefringence occurring per unit length in a particular sample of material under a given level of stress depends on the stress optic coefficient of the material. This parameter can be found in manufacturer's catalogs for optical glasses and elsewhere in the literature for some crystals. Birefringence is most important in optical systems using polarized light, such as polarimeters, most interferometers, many laser systems, and high-performance cameras.

Even low levels of applied force can cause optical surfaces to deform, especially if the forces are not applied symmetrically. Minute surface deformations (measured in fractions of a wavelength of light) affect system performance. The significance of a given deformation depends strongly on the location of the surface in the optical system and the performance requirements of the system containing the optical surface in question. Larger departures from perfection can be tolerated on surfaces near an image than near a system pupil. Because of these system- and application-dependent factors, no general methods of estimating surface deformations, or guidelines in regard to how much surface deformation can be tolerated, are given here.

2.3 Sealing Considerations

Optical instruments intended for military or aerospace applications and many intended for more environmentally benign commercial or consumer applications need to be sealed against entry of moisture or other contaminants from the surrounding environment. This means that minute gaps between windows and/or lenses and mechanical parts of the instrument must be sealed shut. The sealing materials include flat or convoluted gaskets, Orings, and formed-in-place elastomeric seals.

Figure 2.20 illustrates three standard techniques for sealing a lens into a cell. These are termed *static seals* because no motions are involved. In view (a), a compressed O-ring fills the radial gap between the lens rim and the cell wall. In view (b), an O-ring is compressed between a retaining ring, the cell wall, and the corner of the lens rim. view (c) shows a design with a formed-in-place gasket in which a sealant is injected with a hypodermic syringe through several access holes in the cell wall to fill an annular groove machined into the cell's ID adjacent to the lens rim. During the injection process, the lens axis should be horizontal and the injection started at the bottom. As the gap is filled, air escapes through holes at the top, and filling continues until sealant starts to emerge from all the holes. Extra radial holes may be added to the design to assist in monitoring the filling process in larger versions of this type of mounting. In some other static designs, cell-mounted optics are sealed in place with an elastomeric sealant applied through a thin hypodermic needle into a gap between the rim of the component and the mount wall before a mechanical retaining means is attached. The lens axis should be vertical while the sealant is injected and cured.

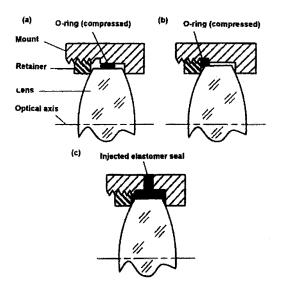


Figure 2.20 Three techniques for statically sealing a lens into its mount: (a) an O-ring around the lens rim, (b) an O-ring between the retainer, cell, and lens corner, and (c) An injected (formed-in-place) elastomeric seal.

In all the designs of Fig. 2.20, the sealant effectively fills the space between the optic and the mount with a slightly flexible material that adheres well to the adjacent surfaces. In general, the same techniques can be applied to lenses, windows, filters, shells, and domes as discussed and illustrated by several examples in Chapter 5.

Moving components such as those used in focusing mechanisms for camera lenses or eyepieces are sometimes sealed to fixed members with O-rings that roll as one component moves relative to the other [see Fig. 2.21(a)]. The configuration shown in Fig. 2.21(b) features a more or less square-shaped seal that slides axially but does not roll. Another

THE OPTIC-TO-MOUNT INTERFACE

dynamic sealing technique is shown in Fig. 2.21(c). Here, a dual-purpose rubber bellows seals the focusing lens cell to a fixed housing at the left (not shown) as well as to the innermost lens at the left of that cell. The outermost lens at the right of the cell is sealed statically with elastomer. The moving inner subassembly (cell and lenses) slides rather than turning when the focusing ring is turned. A fixed pin riding in a slot in the moving part prevents rotation of this subassembly. A portion of the bellows fits into a groove in the mounting flange to seal the entire eyepiece to the instrument.⁶

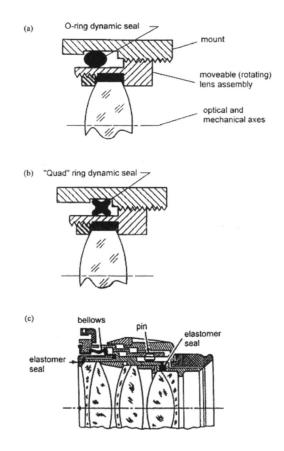


Figure 2.21 Three techniques for dynamic sealing of a moving subassembly: (a) With an O-ring, (b) With a "quad" ring, and (c) with a flexible bellows. View (c) is from Quammen et al.⁶

Castings used for housings are often porous so should be impregnated to fill pores and minute holes that otherwise can leak. Either vacuum- or pressure-applied sealants include acrylic resins that cure at elevated temperatures or anaerobically, styrene-based polyesters, epoxy, or silicate-based mixtures. Commercial and military standards define methods of application and verification.

2.4 References

- 1. Karow, H.H., Fabrication methods for Precision Optics, Wiley, New York, 1993.
- 2. Kimmel, R.K. and Parks, R.E., ISO 10110 Opt ics and Opt ical In struments— Preparation of drawings for optical elements and systems; A User's Guide, Second Edition, Optical Society of America, Washington, 2002.
- 3. Smith, W.J., "Optics in practice," Chapt. 15 in *Modern Optical Engineering*, 3rd. ed., McGraw-Hill, New York, 2000.
- 4. Strong, J., Procedures in Applied Optics, Marcel Dekker, New York, 1988.
- 5. Kittel, D., "Precision Mechanics," *SPIE Short Course Notes*, SPIE Press, Bellingham, 1989.
- 6. Quammen, M.L., Cassidy, P.J., Jordan, F.J., and Yoder, P.R., Jr., "Telescope eyepiece assembly with static and dynamic bellows-type seal," U. S. Pat ent No. 3,246,563, issued April 19, 1966.

CHAPTER 3 Mounting Individual Lenses

In this chapter, we consider several techniques for mounting individual lenses in optical instruments. These techniques are most applicable to optics with apertures in the range of approximately 0.25 to 16 in. (6 to 406 mm). Although most of the discussions deal with glass lenses interfaced with metal mountings, the same principles are generally applicable to lenses made of optical crystals and plastics. Numerous examples are included to illustrate the use of given design equations.

Our first topic deals with estimation of the appropriate axial preload applied to the lens at assembly so that it is held firmly against the mechanical interfaces under all expected adverse environments, including combined extreme temperature and acceleration—the latter directed along any of three orthogonal axes. In order to define this preload, we need to know the weight of the optic. Therefore, standard equations and numerical examples are given for calculating this parameter. We also present equations for locating the lens's center of gravity.

The discussion of lens mounting designs begins with inexpensive, lower-precision techniques. Designs with threaded retaining rings and with compliant ring flanges are considered next. Then we describe the common types of glass-to-metal interfaces: sharp corner, tangential, toroidal, spherical, and flat and step bevels. The chapter continues with descriptions of ways to mount lenses and nonsymmetrically shaped optics in elastomeric supports and on flexures. It concludes with brief considerations of mountings for plastic lenses. The all-important subject of aligning the lens in its mount is considered in Chapt. 12.

3.1 Preload Requirements

The total axial force (preload), P, in pounds, which should be exerted on the lens by any means of constraint to hold it in place against its mechanical reference surface, may be calculated as the product of lens weight W and the worst case axial acceleration. Theoretically, the latter term is the vector sum of the axial components of all the maximum anticipated externally applied accelerations, such as those due to constant acceleration, random vibration (3σ), amplified resonant vibration (sinusoidal), acoustic loading, and shock. For simplicity, frequency effects are ignored, the accelerations are expressed as a multiple a_G of ambient gravity, and friction and moments imposed at the interfaces are neglected. Because all types of external accelerations do not generally occur simultaneously, the summation does not need to be taken literally. If a_G is a single-valued, worst-case number, then

$$P_A = Wa_G . \tag{3.1}$$

If the lens weight is expressed in kilograms, Eq. (3.1) must include a multiplicative factor of 9.807 to convert units. The preload is then in newtons (N). The subscript "A" indicates that this preload is associated with the axial motion of the optic.

Another situation involving axial preload can occur in a lens mounting with surface contacts if no radial constraint (such as pads) exists around the lens rim to constrain lateral motion and the assembly experiences lateral acceleration. Then, only the radial components of the axial force exerted against the lens surface or surfaces and friction will prevent decentration. We assume that the preload P is initially applied uniformly around a circular annulus of radius y_1 to the curved surface with radius R_1 as illustrated in the sectional view of Fig. 3.1. That preload is transmitted through the lens to the interface at height y_2 on the shoulder. Acceleration a_G is applied to the instrument in the downward direction, as indicated by the arrow passing through the lens's center of gravity. The force developed by this acceleration is a_G times the weight of the lens. It is directed downward in the figure. Because the shape of the lens is effectively a wedge being driven against the two interfaces, the mount tends to resist movement of the optic. The surfaces of the optic and of the mount are compressed microscopically and locally at the interfaces, but we will ignore those effects and treat the optic and the mount as rigid bodies.

As indicated in the figure, the downward (acceleration) force must be opposed by an upward force of magnitude Wa_G if the lens is not to move downward. As shown in the detail view of Fig. 3.1, this upwardly directed force has a component tangential to the lens surface given by

Tangential component =
$$Wa_G \cos \theta$$
. (3.2)

Associated with the tangential force component is a force component normal to the lens surface. Its magnitude is

Normal component =
$$\left(\frac{Wa_G}{\mu}\right)\cos\theta$$
, (3.3)

where μ is the coefficient of friction between the glass and the metal at the interface on R_1 .

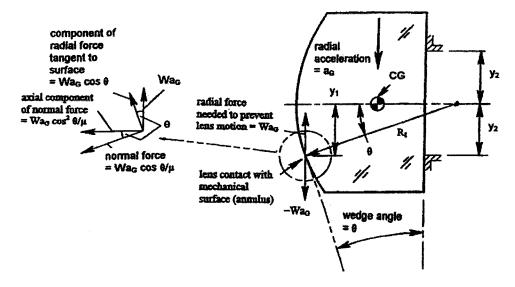


Figure 3.1 Geometry for estimating preload required to prevent a surface contact lens from decentering under lateral acceleration when not otherwise constrained from such motion.

Once again referring to the detail view of Fig. 3.1, we see that there is a component of the normal force directed parallel to the axis. Its magnitude is

Axial component =
$$\left(\frac{Wa_G}{\mu}\right)\cos^2\theta.$$
 (3.4)

Because there are interfaces between glass and metal on both sides of the lens, we can approximate the axial force (preload) that generates sufficient total frictional resistance to prevent transverse motion of the lens as

$$\mathbf{P}_{T} = \left[\frac{Wa_{G}}{(2\mu)}\right] \cos^{2}\theta, \qquad (3.5)$$

where the subscript "T" indicates preload related to transverse motion of the optic.

In all these equations, the angle θ is given by:

$$\theta = \arcsin\left(\frac{y_1}{R_1}\right) + \arcsin\left(\frac{y_2}{R_2}\right).$$
(3.6)

This equation can be evaluated for lenses of different shapes. The geometry is indicated in Fig. 3.2 for four general cases. In addition, we should include a plane parallel plate, such as a flat window or a reticle, for which the angle θ is identically zero. This result also applies to a lens with two curved surfaces if those surfaces have flat bevels that serve as the interfaces with the mount. When the mechanical interfaces are on two curved surfaces, θ is the sum of two individual angles for each surface measured relative to the central plane normal to the axis. Note that the interfaces that oppose lateral-lens motion of a meniscus lens are on opposite sides of the axis [Fig. 3.2(b)].

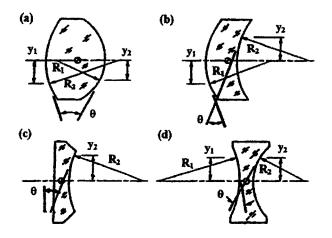


Figure 3.2 Geometry for determining the angle θ for lenses with four shapes: (a) biconvex, (b) meniscus, (c) plano concave, and (d) biconcave.

Example 3.1: Estimate the axial preload required to constrain a lens laterally when it is subjected to a lateral acceleration. (For design and analysis, use File 3.1 of the CD-ROM.)

A biconvex lens has the following dimensions: diameter 4.000 in. (101.6 mm), $R_1 = -R_2 = 6.000$ in. (152.4 mm), W = 0.683 lb (0.31 kg), $y_1 = y_2 = 1.900$ in. (48.26 mm). Assume that the lateral acceleration is $a_G = 15$ and $\mu = 0.2$. What axial preload P_T is needed to prevent translation, if that is the only means of laterally constraining the lens?

From the geometry of Fig. 3.1 and Eq. (3.6), we find that $\theta_1 = \arcsin(y_1/R_1) = \arcsin(1.900 / 6.000) = 18.461^\circ$. The angle θ_2 is the same as θ_1 , so $\theta = (2)(18.461^\circ) = 36.923^\circ$

and $\cos \theta = 0.7794$.

Applying Eq. (3.5):

$$P_{T} = \frac{(0.683)(15)(0.7994)^{2}}{[(2)(0.2)]} = 16.37 \text{lb} (72.81 \text{ N}).$$

In this case, by Eq. (3.1), $P_A = (0.683)(15) = 10.24$ lb (45.55 N). This is smaller than P_T so the lens might not be adequately constrained by friction from lateral motion under the specified acceleration unless the preload is increased to at least equal P_T .

Example 3.1 applies Eqs. (3.6) and (3.5) in sequence to estimate the preload P_T for a symmetrical biconvex lens. The basic method can be applied to any lens shape.

3.2 Weight and Center of Gravity Calculations

In general, a lens can be divided into a combination of spherical segments, right circular cylinders, and/or truncated cones. We will here refer to these as a cap, a disk, and a cone. In the following equations, the algebraic signs of all radii area assumed positive, the lens diameter is D_G , and ρ is density.

The sagittal depth and weight of a cap are:

$$S = R - \left[R^2 - \left(\frac{D_G^2}{4} \right) \right]^{1/2},$$
 (3.7)

and

$$W_{CAP} = \pi \rho S^2 \left[R - \left(\frac{S}{3} \right) \right]. \tag{3.8}$$

The weight of a disk of axial length L is

$$W_{DISK} = \frac{\pi \rho L D_G^2}{4}.$$
(3.9)

The weight of a cone is

$$W_{CONE} = \frac{\pi \rho L \left(D_1^2 + D_1 D_2 + D_2^2 \right)}{12},$$
(3.10)

where L is the axial length of the cone, D_1 is the diameter of the larger end of the cone, and D_2 is the diameter of the smaller end of the cone. Usually, $D_1 = D_G$.

Figure 3.3 shows section views of nine basic lens configurations. To determine the weight of a plano-convex lens, as shown in view (a), we would add the weights of the cap and the disk. For a plano-concave lens we would subtract the weight of the cone from that of the disk.

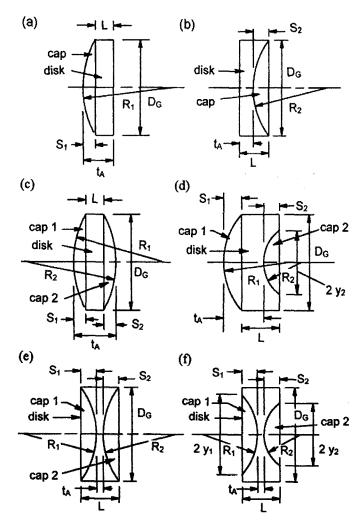


Figure 3.3 Schematic sectional views of nine lens configurations dimensioned as required to calculate their weights: (a) plano convex, (b) plano concave, (c) biconvex, (d) meniscus, (e) biconcave, and (f) biconcave with dual flat bevels (continued on next page).

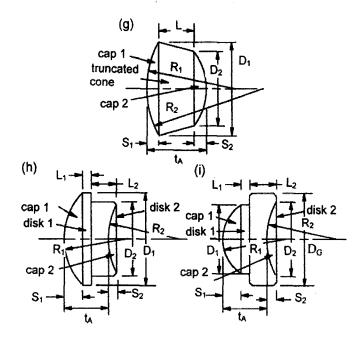


Figure 3.3 (continued) (g) biconvex with conical section, (h) cemented meniscus with larger plano-convex element, and (i) cemented meniscus with larger plano-concave element.

Weights for cemented doublets and configurations that are more complex are obtained by adding the contributions from the individual elements. We follow the principles demonstrated in the following examples.

Example 3.2: Weight of a biconvex lens. (For design and analysis, use File 3.2 of the CD-ROM.) A biconvex lens as shown in Fig. 3.3(c) has dimensions $D_G = 4.000$ in. (101.6 mm), $t_A = 1.000$ in. (25.4 mm), and $R_1 = R_2 = 6.000$ in. (152.4 mm). The lens is made of NBK7 glass with $\rho = 0.091$ lb/in.³ (2.519 g/cm³). What is its weight? From Eq. (3.7): $S_1 = S_2 = 6.000 - \left[6.000^2 - \left(\frac{4.000^2}{4} \right) \right]^2 = 0.343$ in.(8.716 mm). From the geometry of the figure, L = 1.000 - (2)(0.343) = 0.314 in. (7.976 mm). From Eq. (3.8): W_{CAP} of each cap is $(\pi)(0.091)(0.343)^2 \left[6.000 - \left(\frac{0.343}{3} \right) \right] = 0.198$ lb (0.090 kg). From Eq. (3.9): W_D of the disk is $(\pi)(0.091)(0.314) \frac{(4.000)^2}{4} = 0.359$ lb (0.163 kg). The total weight $W_{LENS} = 0.359$ + (2)(0.198) = 0.755 lb (0.342 kg).

Example 3.3: Weight of a cemented meniscus lens with larger plano-convex element. (For design and analysis, use File 3.3 of the CD-ROM.)

A cemented lens as shown in Fig. 3.3(h) has a first plano-convex element with a cylindrical rim of length $L_1 = 0.100$ in. (2.54 mm), a diameter D_1 of 1.180 in (29.972 mm), and a surface radius of 1.850 in. (46.990 mm). Its glass type is NBK7 with $\rho = 0.091$ lb/in.³ (2.519 g/cm³). The second element has a cylindrical rim of length $L_2 = 0.350$ in. (8.890 mm), a diameter D_2 of 0.930 in. (23.622 mm), and a surface radius of 1.950 in. (49.530 mm). The axial thickness t_A of the lens is 0.491 in. (12.471 mm). Its glass type is SF4 with $\rho = 0.172$ lb/in.³ (4.761 g/cm³). What is its weight?

We apply Eqs. (3.7), (3.8), and (3.9):

$$S_1 = 1.850 - \left[1.850^2 - \left(\frac{1.1802}{4}\right)\right]^{1/2} = 0.097 \text{ in. } (2.464 \text{ mm})$$

and

$$S_2 = 1.950 - \left[1.950^2 - \left(\frac{0.9302}{4}\right)\right]^{1/2} = 0.056 \text{ in. } (1.422 \text{ mm}).$$

The weight of the first cap is

$$(\pi)(0.091)(0.097)^2 \left[1.850 - \left(\frac{0.097}{3}\right)\right] = 0.005 \text{ lb} (0.002 \text{ kg}).$$

The weight of the first disk is

$$(\pi)(0.090)(0.100)\left(\frac{1.1802}{4}\right) = 0.010 \text{ lb } (0.004 \text{ kg}).$$

The weight of the second cap is

$$(\pi)(0.172)(0.056)^2 \left[1.950 - \left(\frac{0.056}{3} \right) \right] = 0.003 \, lb(0.001 \, kg).$$

The length L_2 of the second disk is

$$t_A - S_1 - L_1 + S_2 = 0.491 - 0.097 - 0.100 + 0.056$$

= 0.350 in. (8.890 mm).

The weight of the second disk is

$$(\pi)(0.172)(0.350)\frac{(0.930)^2}{4} = 0.041 \text{ lb} (0.018 \text{ kg}).$$

The total weight is

$$W_{\text{LENS}} = 0.005 + 0.010 + 0.041 - 0.003 = 0.053 \text{ lb} (0.024 \text{ kg}).$$

The above equations can also be used to estimate mirror weights. The shape of a simple, solid convex mirror is typically the same as that of a plano-convex lens while that of a simple solid concave mirror is typically that of a plano-concave lens. Techniques for estimating the weights of solid mirrors with contoured backs are summarized in Section 8.5.1. The weights of lightweight mirrors made from solid blanks with cavities of various shapes and sizes created in the mirror backs can be estimated by subtracting the total

weights of all the cavities (assuming them to be filled with the same material as the mirror) from the weights of the solid substrates. The weights of mirrors with built up construction are best estimated by dividing the structure into groups of parts of the same sizes and shapes, estimating the total weight of each group, and summing these to get the aggregate mirror weight.

In general, we treat aspheric surfaces as spherical ones unless the asphericity is strong, as in the case of a very deep paraboloid. In such a case, we can determine the volume (and the weight) by calculating the cross-sectional area of the aspheric volume and multiplying that by 2π times the height from the axis of symmetry of the centroid of the area. This technique is used in Section 8.6.1 to calculate the weight savings of contoured backs for arched mirrors. Appropriate equations for parabolic sections are given there. Most general aspherics can be approximated by conics. One can obtain the area and centroid height equations for conic sections other than the parabola from standard solid analytic geometry texts.

We next see how to locate the center of gravity (CG) for a lens or simple mirror. Figure 3.4 shows the cross sections of the three basic shapes that make up these elements. The dimensions X indicate the locations of the CGs relative to the left side. The following equations allow us to determine X for each shape using the dimensions shown in the figures:

$$X_{\text{DISK}} = \frac{L}{2},$$
(3.11)

$$X_{CAP} = \frac{S(4R - S)}{[4(3R - S)]},$$
(3.12)

$$X_{\text{CONE}} = 2L \frac{\left[\left(d_1 / 2 \right) + D_2 \right]}{\left[3 \left(D_1 + D_2 \right) \right]}.$$
 (3.13)

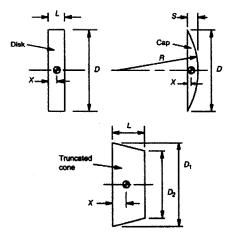


Figure 3.4 Schematic sectional views of basic solid shapes from which lenses are formed. The location of each center of gravity (CG) is shown.

Example 3.4: Locate the center of gravity of a lens. (For design and analysis, use File 3.4 of the CD-ROM.)

A lens of the type shown in Fig. 3.3(h) has these dimensions and weights:

For Cap 1, $S_1 = 0.0966$ in., R = 1.8500 in. and $W_{CAP1} = 0.0048$ lb. For Disk 1, $L_1 = 0.1000$ in. and $W_{DISK1} = 0.0098$ lb. For Disk 2, $L_2 = 0.3500$ in. and $W_{DISK2} = 0.0409$ lb. For Cap 2, $S_2 = 0.0562$ in., R = 1.950 in. and $W_{CAP2} = 0.0033$ lb. The total lens weight $W_{LENS} = 0.0522$ lb.

Where is its center of gravity with respect to the left vertex?

From Eq. (3.12):

$$X_{CAP1} = \frac{0.0966 [(4)(1.8500)] - 0.0966}{4 [(3)(1.8500) - 0.0966]} = 0.0323 \text{ in.}$$

Then,

 $X'_{CAP1} = 0.0966 - 0.0323 = 0.0643$ in.

Hence,

 $(W_i X'_i)_{CAPI} = 3.086 \times 10^{-4} \text{ lb-in.}$

From Eq. (3.11):

$$X_{\text{DISK1}} = \frac{0.1000}{2} = 0.0500$$
 in

Then,

$$X'_{\text{DISK1}} = S_1 + X_{\text{DISK1}} = 0.1466 \text{ in. and } (W_i X'_i)_{\text{DISK1}} = 1.437 \times 10^{-3} \text{ lb-in.}$$

From Eq. (3.11):

$$X_{DISK2} = \frac{0.3500}{2} = 0.1750$$
 in

Then,

$$X'_{\text{DISK2}} = S_1 + L_1 + X_{\text{DISK2}} = 0.3716$$
 in. and $(W_i X'_i)_{\text{DISK2}} = 1.520 \times 10^{-2}$ lb-in.

From Eq. (3.12):

$$X_{CAP2} = \frac{0.0562 [(4)(1.9500) - 0.0562]}{4 [(3)(1.9500) - 0.0562]} = 0.0188 \text{ in.}$$

$$X'_{CAP2} = 0.0966 + 0.1000 + 0.3500 - 0.0188 = 0.5278$$
 in. and $(W_i X'_i)_{CAP2} = 1.742 \times 10^{-3}$ lb-in.

From Eq. (3.14):

$$X_{\text{LENS}} = \frac{3.086 \times 10^{-4} + 1.437 \times 10^{-3} + 1.520 \times 10^{-2} - 1.742 \times 10^{-3}}{0.0522} = 0.2913 \text{ in.}$$

This is the distance from the left vertex to the CG of the cemented lens.

The location of the CG of any lens (or mirror) comprising N parts and weighing a total of W_{LENS} can be estimated from:

$$X_{\text{LENS}} = \sum_{i=1}^{i=N} \frac{(X_i W_i)}{W_{\text{LENS}}}.$$
(3.14)

In Eq. (3.14), X_{LENS} and all values of X'_i are measured from the same point on the axis of the lens. For example, in view (g) of Fig. 3.4, if we choose the left vertex as the reference and remember that X_{CAP} is always measured from the plano side, $X'_{\text{CAP1}} = S_1 - X_{\text{CAP1}}$, $X'_{\text{DISK}} = S_1 + X_{\text{DISK}}$, and $X'_{\text{CAP2}} = S_1 + L + X_{\text{CAP2}}$. Example 3.4 shows how to locate a lens's CG.

To check the CG location computation for any lens, we find the distances from each part's CG to the lens's CG (called moment arms) and multiply these dimensions by the respective part weights. The products are moments. Those parts whose CGs lie to the left of the lens's CG cause counterclockwise (CCW) moments while those to the right of that CG cause clockwise (CW) moments. For the CG of the lens to be properly located, the sum of the CCW moments must equal the sum of the CW moments. In the example just considered, the CCW moment sum is $(0.2913-0.0643)(0.0048) + (0.2913-0.1466)(0.0098)=2.508\times10^{-3}$ lb-in. and the CW moment sum is $(0.3716-0.2913)(0.0409) - (0.5278-0.2913)(0.0033) = 2.504\times10^{-3}$ lb-in. These moments are equal so the location of the lens's CG is confirmed.

3.3 Spring Mountings for Lenses and Filters

Optical components that do not need precise positioning and that must not be excessively constrained throughout large temperature changes, such as condensing lenses and heatabsorbing filters used in close proximity to heat sources in slide projectors are frequently mounted on springs. These low-cost designs allow free flow of air across the optical surfaces to help minimize temperature rise while maintaining adequate alignment at all temperatures. They also provide limited shock and vibration resistance.¹⁻³

Figure 3.5 illustrates one example that shows a plano-convex lens made of heatresistant glass (such as Pyrex) held in detents on three flat springs spaced at 120 deg intervals about the lens rim and cantilevered from a metal mounting ring. The symmetry of the cantilevered springs tends to keep the lens centered. A variant of this design has two such lenses held convex to convex in springs appropriately shaped with multiple detents to support both lenses with an appropriate axial separation.

Figure 3.6 shows the mounting for a heat-absorbing filter and a biconvex condensing lens as used in the Kodak Ektagraphic slide projector Model EF-2. The rims of the optics fit partway into appropriately shaped cutouts in the sheet metal base plate and are held down by a spring-loaded, notched dual clip that fits over the component's rims at the top.

Separation of the elements is maintained by the cutouts in the base plate and the notches in the clip. Notches in the sheet-metal bracket shown at left in the figure hold the optics against rotation about a vertical axis. The shape of the base-plate cutout for the lens can be designed so a lens with different first and second curvatures cannot be inserted

backward without calling attention to that fact. The spring that holds the clip over the optics is designed for ease of engagement and disengagement to facilitate servicing. Many other variations on the spring-mounting technique exist; all are intended to minimize design and hardware production cost while meeting technical goals, such as promoting cooling, preservation of low-level alignment, and ease of replacement.

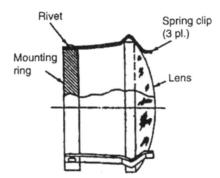


Figure 3.5 A low precision technique for mounting a lens on three leaf springs located at symmetrically around the rim.

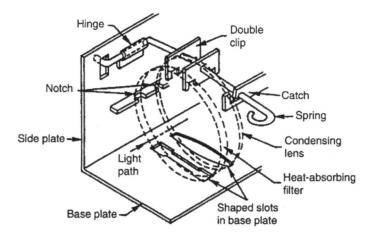


Figure 3.6 Conceptual schematic of a spring-loaded mounting for a heatabsorbing filter and a biconvex condensing lens in a Kodak Ektagraphic projector. The optics are shown by dashed outlines.

3.4 Burnished Cell Mountings

The burnished-cell technique is most frequently used for mounting small lenses in microscope objectives or for the tiny lenses used in endoscopes and borescopes, where space constraints prevent the use of separate retainers and a need for disassembly is not anticipated. This type of mounting has a cell made of malleable material such as brass or untempered aluminum alloy. It is designed with a protruding lip that is to be mechanically deformed around the rim of a lens at the time of assembly.^{1,3–4}

Figure 3.7 shows a typical example, where view (a) shows such a cell and lens prior to assembly. The optional chucking thread facilitates installation of the cell on a lathe spindle. In some designs, the cell lip is tapered slightly to facilitate obtaining more intimate contact with the bevel on the lens.

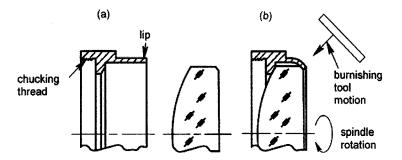


Figure 3.7 A lens burnished into a cell made of malleable metal. (a) Cell and lens configurations, (b) The completed subassembly.

The cell lip is deformed by pressing three or more, hardened rod-shaped tools or cylindrical rollers simultaneously and symmetrically against the lip at an oblique angle while the cell is slowly rotated. The lens should be held axially against the cell shoulder by some external means (not shown in the figure) during the burnishing procedure to help keep it aligned. If the radial fit between the lens and cell wall is close and the lens rim is accurately ground, this technique results in a well-centered subassembly. Burnishing the cell lip pushes the lens toward an internal shoulder or spacer, but does not guarantee the existence of any preload because the metal may spring back after pressure is removed. Once completed [see Fig. 3.7(b)], the subassembly is essentially permanent, since it would be very difficult to unbend the metal.

In a slightly different assembly method that does not require the cell to be rotated, the cell lip is deformed around the lens rim by a swaging process in which a concave conical die is pressed axially against the lip, bending it uniformly around its periphery toward the lens. Figure 3.8(a) shows this schematically. Figure 3.8(b) shows an enlarged view.

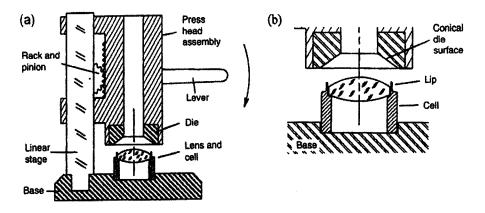


Figure 3.8 An alternate method for burnishing a lens into a cell using a mechanical press to bend the cell wall without rotating the subassembly. (a) Conceptual view, and (b) enlarged view. (Adapted from Yoder.¹)

As a minor variation on either of these methods, a thin, narrow washer or O-ring made of slightly resilient material such as Nylon or Neoprene is sometimes inserted against the exposed surface of the lens and the metal burnished over that washer rather than causing the metal to directly contact the glass. This tends to seal the glass-to-metal interface and can offer a slight spring action to hold the glass axially against the seat at higher temperatures when the metal lip tends to expand away from the glass.

Other designs may incorporate a compressed spring between the lens and the shoulder to offer a more predictable axial preload and compliance during temperature changes. Figure 3.9 shows an example. The spring ends should be ground flat to promote uniform contact all around the edge of the lens aperture. Jacobs⁴ has suggested using a thin brass tube that is partially slotted transversely as the spring. In designs of this type, we gain some of the advantages of spring mounting as well as the simplicity of the burnished-rim mounting technique. If the spring force (axial preload) is not sufficient to hold the lens against the lip under shock and vibration, rebound may damage the polished surfaces and affect alignment.

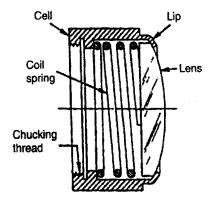


Figure 3.9 A spring-loaded version of the burnished lens mounting. (Adapted from Jacobs.⁴)

3.5 Snap and "Interference Fit" Rings

A discontinuous (i.e., cut) ring that drops into a groove machined into the ID of a cell is commonly termed a "snap ring" since it acts as a spring.^{1,2} This ring usually is made of spring steel wire and so has a circular cross-section as shown in Fig. 3.10. Rings with rectangular or trapezoidal cross-sections are less frequently used. The cut through the ring allows it to be compressed slightly while sliding into the groove; this cut usually is made wide enough for a tool to be inserted to allow ring removal. The groove cross section can be rectangular (the most popular), v-shaped, or curved.

It is difficult to ensure contact between the lens surface and the ring using this technique since the thickness, diameter, and surface radius of the lens as well as ring dimensions, groove location and dimensions, and temperature changes all affect the degree of mechanical interference, if any, existing between the lens surface and the ring. For this reason, this technique is used only where the location and orientation of the lens are not critical. It is virtually impossible to provide a specific axial preload to the lens.

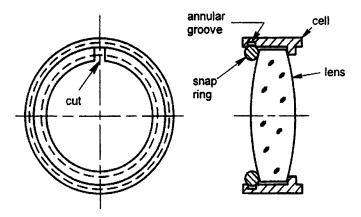


Figure 3.10 Technique for interfacing a convex lens surface with a cut circular cross-section snap ring located in a groove in the mount ID.

To assist the reader who might need to design this type of lens-to-mount interface, Fig. 3.11 illustrates (to an exaggerated scale) the pertinent geometry for a convex lens surface of radius R_L , diameter D_G , and clear aperture A with a circular ring of cross-sectional diameter 2r and (compressed) OD of D_R when mounted in a cell with an ID equal to D_M . The following equations define the contact height y_C and the axial location x of the inner edge of the groove relative to the vertex of the lens. The dimensions (width w and depth d) of the nominal rectangular groove that allow the ring to just touch the lens, while seating against both edges of the groove are also defined. This design is based on the reasonable but somewhat arbitrary specification that the angular subtense of the groove width as seen from the center of the ring cross-section is 90 deg.

$$y_c = \frac{D_G + A}{4},$$
 (3.15)

$$x_{c} = \left(R_{L}^{2} - y_{C}^{2}\right)^{1/2}, \qquad (3.16)$$

$$S_C = R_L - x_C, \qquad (3.17)$$

$$\Delta y_1 = \frac{y_C r}{R_L},\tag{3.18}$$

$$\Delta x_1 = \frac{x_C r}{R_L},\tag{3.19}$$

$$\Delta y_2 = \left(\frac{D_M}{2}\right) - y_C - \Delta y_1 = \frac{w}{2},\tag{3.20}$$

$$\Delta x_2 = \Delta x_1 - \Delta y_2, \tag{3.21}$$

$$x = S_C - \Delta x_2, \tag{3.22}$$

$$d_{\text{MINIMUM}} = r - \Delta y_2, \qquad (3.23)$$

$$d_{\text{RECOMMENDED}} = 1.25 d_{\text{MINIMUM}}, \qquad (3.23a)$$

$$w = 2\Delta y_2, \tag{3.24}$$

$$D_{R} = 2(y_{C} + \Delta y_{1} + r).$$
(3.25)

Note that $(D_R - 4r)$ should be at least equal to the required lens aperture A. If not, a new (smaller) value for the ring cross-sectional diameter 2r should be chosen and the computations repeated until this is so. Example 3.5 illustrates the use of this set of equations.

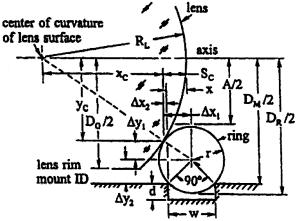


Figure 3.11 Design geometry for constraining a convex surface with a cut circular cross-section snap ring.

Figure 3.12 shows what happens if a nominally dimensioned lens is properly seated in the cell, where the groove has nominal dimensions and the ring cross-sectional diameter 2r is nominal (the ring just touches the lens), oversized (the ring contacts the groove only on its outer edge and tends to rise out of that groove), and undersized (the ring seats in the groove, but clearance exists between the lens and the ring). Only in the case of an oversized ring is the lens subject to any axial preload. Analytical means for predicting this preload are not available at present. This problem can also occur if the groove is mislocated axially, has the wrong width, or has the wrong depth.

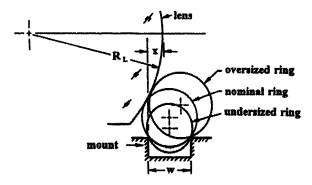


Figure 3.12 Effect of variation in ring cross-sectional diameter in the mounting configuration of Fig. 3.11.

Example 3.5: Convex surface interface with a snap ring. (For design and analysis, use File 3.5 of the CD-ROM)

Design a lens snap-ring mounting to interface with a convex-lens surface having the following dimensions: $D_G = 25.4 \text{ mm} (1.0 \text{ in.}), A = 22.0 \text{ mm} (0.8661 \text{ in.}), R_L = 50.8 \text{ mm} (2.0 \text{ in.}), D_M = 25.6000 \text{ mm} (1.0079 \text{ in.}), \text{ and } r = 1.0 \text{ mm} (0.03937 \text{ in.}).$

From Eqs. (3.15) through (3.25): $y_{c} = \frac{25.4000 + 22.0000}{4} = 11.8500 \text{ mm} (0.4665 \text{ in.}),$ $x_{c} = (50.8000^{2} + 11.8500^{2})^{1/2} = 49.3986 \text{ mm} (1.9448 \text{ in.}),$ $S_{c} = (50.8000 - 49.3986) = 1.4014 \text{ mm} (0.0552 \text{ in.}),$ $\Delta y_{1} = \frac{(11.8500)(1.0000)}{50.8000} = 0.2333 \text{ mm} (0.0092 \text{ in.}),$ $\Delta x_{1} = \frac{(49.3986)(1.0000)}{50.8000} = 0.9724 \text{ mm} (0.0383 \text{ in.}),$ $\Delta y_{2} = \left(\frac{25.6000}{2}\right) - 11.8500 - 0.2333 = 0.7167 \text{ mm} (0.0282 \text{ in.}),$ $\Delta x_{2} = 0.9724 - 0.7167 = 0.2557 \text{ mm} (0.0101 \text{ in.}),$ x = 1.4014 - 0.2557 = 1.1457 mm (0.0451 in.), $d_{\text{MINIMUM}} = 1.0000 - 0.7167 = 0.2833 \text{ mm} (0.1111 \text{ in.}),$ $d_{\text{RECOMMENDED}} = (1.25)(0.2833) = 0.3541 \text{ mm} (0.0139 \text{ in.}),$ w = (2)(0.7167) = 1.4334 mm (0.0564 in.), $D_{R} = (2)(11.8500 + 0.23330 + 1.0000) = 26.1666 \text{ mm} (1.0302 \text{ in.}).$ This is greater than *A*, therefore acceptable.

While the surface's contacting type of interface could be used with a concave lens surface, it is more common to place a flat bevel on the surface and locate the snap ring so it just touches that bevel at its midpoint. Figure 3.13 and Eq. (3.26 through 3.32) then apply.

The flat bevel width is b. The suitability of the ring cross-sectional diameter 2r should be checked by calculating $(D_R - 4r)$; this should equal or exceed A:

$$y_C = \left(\frac{D_G}{2}\right) - \left(\frac{b}{2}\right),\tag{3.26}$$

$$\Delta y_1 = \left(\frac{D_G}{2}\right) - b \tag{3.27}$$

$$\Delta y_2 = \left(\frac{d_M}{2}\right) - y_C , \qquad (3.28)$$

$$S_{c} = R_{L} - \left(R_{L}^{2} - \Delta y_{1}^{2}\right)^{1/2}, \qquad (3.29)$$

$$d_{\rm MINIMUM} = r - \Delta y_2, \qquad (3.30)$$

 $d_{\text{RECOMMENDED}} = 1.25 d_{\text{MINIMUM}} , \qquad (3.23a)$

$$w = 2\Delta y_2, \qquad (3.31)$$

$$x = S_C + r - \left(\frac{w}{2}\right),\tag{3.32}$$

$$D_{R} = 2(Y_{C} + r). (3.33)$$

Example 3.6 illustrates this type of design.

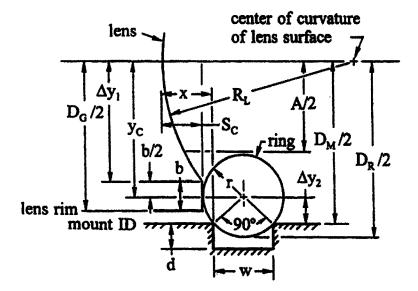


Figure 3.13 Configuration for snap ring constraint of a concave surface having a flat bevel with a cut circular cross-section snap ring.

Example 3.6: Concave surface interface with a snap ring. (For design and analysis, use File 3.6 of the CD-ROM.

Design a lens snap-ring mounting for a concave-lens surface with a flat bevel per Fig. 3.13. Let $D_G = 25.4000 \text{ mm} (1.0000 \text{ in})$, A = 22.0000 mm (0.8661 in.), $R_L = 50.8000 \text{ mm} (2.0000 \text{ in.})$, $D_M = 25.6000 \text{ mm} (1.0079 \text{ in.})$, r = 1.0000 mm (0.0394 in.), and b = 1.0000 mm (0.0394 in.).

Applying Eqs. (3.26) through (3.33): $y_{c} = \left(\frac{25.4000}{2}\right) - \left(\frac{1.0000}{2}\right) = 12.2000 \text{ mm } (0.4803 \text{ in.})$ $\Delta y_{1} = \left(\frac{25.4000}{2}\right) - 1.0000 = 11.7000 \text{ mm } (0.4606 \text{ in.}).$ $\Delta y_{2} = \left(\frac{25.6000}{2}\right) - 12.2000 = 0.6000 \text{ mm } (0.0236 \text{ in.}).$ $S_{C} = 50.8000 - (50.8000^{2} - 11.7000^{2})^{1/2} = 1.3657 \text{ mm } (0.0538 \text{ in.})$ $d_{\text{MINIMUM}} = 1.0000 - 0.6000 = 0.4000 \text{ mm } (0.0157 \text{ in})$ $d_{\text{RECOMMENDED}} = (1.25)(0.4000) = 0.5000 \text{ mm } (0.0197 \text{ in.})$ w = (2)(0.6000) = 1.2000 mm (0.0472 in.) $x = 1.3657 + 1.0000 - \left(\frac{1.2000}{2}\right) = 1.7657 \text{ mm } (0.0695 \text{ in.}).$ $D_{R} = (2)(12.2000 + 1.0000) = 26.4000 \text{ mm } (0.8819 \text{ in.})$ This is greater than *A*, therefore acceptable

Figure 3.14 shows what happens if a lens with a flat-bevel interface on a circular ring is properly seated in the cell: the groove has nominal dimensions, and the ring's cross-sectional diameter 2r is also nominal (the ring just touches the lens); oversized (the ring contacts the groove only on its outer edge and tends to rise out of that groove) and undersized (the ring seats in groove, but clearance exists between the lens and the ring). As for a convex surface, only in the case of an oversize ring is the lens subject to any axial preload. Analytical means for predicting this preload are not available.

A configuration for a snap ring-constraint design intended for a consumer application and having a different form of groove is shown in Fig. 3.15. Here a circular cross-section ring rests against a tapered or ramped inside surface of the cell wall. The ring is pressed in place and the cell is plastic, so its wall is somewhat resilient. A spring action holds the ring between the lens surface and the ramp. This design is less sensitive to dimensional errors and temperature changes than those with conventional grooves. Preload is hard to predict, but is not critical for the consumer camera application involved.

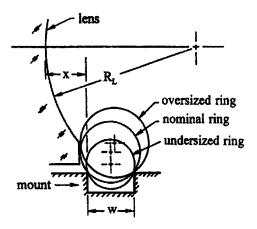


Figure 3.14 Effect of variation in ring cross-sectional diameter for the mounting design of Fig. 3.13.

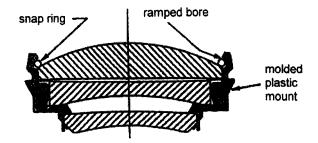


Figure 3.15 A snap-ring-loaded lens-mounting configuration featuring a ramped seat for the ring. (Adapted from Plummer. 5)

A lens can also be constrained by a continuous ring as shown in Fig. 3.16. The OD of the ring is made very slightly oversized with respect to the ID of the cell for an interference fit. After the lens is installed, the ring is pressed into the cell. It is difficult to determine exactly when the ring touches the lens surface, so it would be impossible to achieve a particular preload on the lens and hard even to ensure that contact is made.

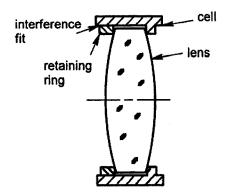


Figure 3.16 A lens constraint design with a continuous ring pressed in place with an interference fit.

A preferred assembly technique would be to heat the cell (and perhaps cool the ring) so the ring slips easily into place against the lens surface. It is theoretically possible to achieve a specific preload by calculating the dimensional changes that are due to the temperature change and by ensuring that the ring contacts the lens at the start of the temperature equalization process. In this type of design, the cell and ring materials should have similar CTEs to prevent loosening or excessive internal stress buildup at extremely low temperatures as a result of differential contraction. Publication B4.1-1967 of the American National Standards Institute (ANSI) defines the appropriate dimensions for force/shrink fits using thin sections (Class FN-1).⁶ Assembly by an interference fit technique is quite permanent, since it is virtually impossible to remove the ring without damaging it or the lens.

3.6 Retaining Ring Constraints

3.6.1 Threaded retaining rings

The most frequently used technique for mounting lenses is to clamp the lens near its rim between a shoulder or spacer and a retaining ring. The ring may be threaded or configured as a continuous annular ring flange. When mounting very large lenses, it is sometimes advantageous to use multiple cantilevered spring clips to serve as an "interrupted" constraining flange.

Manufacturing variations in the axial dimensions of lenses and cells can be compensated for with any of these constraints. Threaded retaining ring configurations are compatible with environmental sealing with a cured-in-place elastomer or an O-ring, as indicated in Fig. 2.20. This type constraint is easily incorporated into multiple-component lens systems that are separated by spacers as discussed in Chapter 4.

Figure 3.17 illustrates a typical threaded retaining ring mounting design for a bi-convex lens. Contact between the lens and the mount occurs on both polished surfaces as recommended earlier for precise centering of the lens and to minimize the need for precise edging or close tolerances on the diameter of the lens. To minimize bending of the lens, contact should occur at essentially the same height from the axis on both sides of the lens.

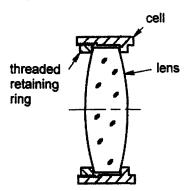


Figure 3.17 Typical configuration of a lens secured in its mount with a threaded retaining ring. (From Yoder.⁷)

The fit of the ring threads into the cell threads should be loose (Class-1 or -2 per ANSI Publication B1.1-1982) so the ring can tilt slightly, if necessary, to accommodate residual wedge angle in the lens when the lens is properly centered optically.⁹ This helps to ensure that the preload is distributed uniformly around the lens periphery. A rule-of-thumb criterion for a suitable fit of the threaded version is to assemble the ring in the mount without an optic in place, hold it to the ear, and shake it. One should be able to hear the ring rattle in the mount.

Sets of holes or transverse slots are usually machined into the exposed face of the retainer to accept pins or rectangular lugs on the end of a cylindrical wrench that is used to tighten the retainer. Alternatively, a flat plate-type tool that spans the retainer can be used as the wrench. The cylindrical wrench is easier to use and is more conducive to measurement of torque applied to the ring. It also minimizes risk of damage to the lens coating or surface that results from slippage.

An equation for the approximate magnitude of the axial preload (P) produced by tightening a threaded retainer with pitch diameter D_T (see Fig. 3.18) to a torque Q against a lens surface can be derived as shown in Appendix C. The first term within the parentheses results from the classical equation for a body sliding slowly on an inclined plane (i.e., the thread), while the second represents the friction effects at the circular interface between the lens surface and the end of the rotating retainer. This equation is

$$P = \frac{Q}{D_T \left(0.577\,\mu_M + 0.500\,\mu_G\right)},\tag{3.34}$$

where μ_M is the sliding-friction coefficient of the metal-to-metal interface in the thread and μ_G is the sliding-friction coefficient of the glass-to-metal interface.

Some designs place a thin metallic "slip ring" between the lens and the retainer to prevent rotation of the lens as the retainer is tightened. Then, μ_M is used in both terms of the equation.

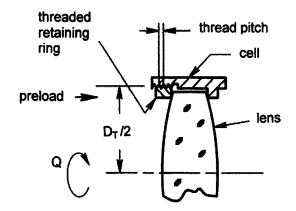


Figure 3.18 Geometry for relating torque applied to a retainer to the resulting axial preload.

Equation (3.34) is an approximation because of small factors neglected in the derivation and larger uncertainties in the values of μ_M and μ_G . The latter values depend strongly on the smoothness of the metal surfaces (which depends, in part, on the machined finish as well as how many times the thread has been tightened and loosened) and whether the surfaces are dry or moistened by water, a lubricant, or fingerprints. Laboratory physics-type measurements of the angle at which a dry anodized aluminum block begins to slide down an inclined dry anodized aluminum surface have indicated that μ_M is about 0.19, while similar experiments with polished glass and dry anodized aluminum indicated μ_G to be about 0.15. Substituting these values into Eq. (3.34), we obtain $P = 5.42Q/D_T$. This corresponds to within about 8% to the commonly accepted approximation of the *P* to *Q* relationship, which is ⁹⁻¹¹

$$P = \frac{5Q}{D_T} \tag{3.34a}$$

Example 3.7 illustrates use of this equation.

Note that similar metals (aluminum on aluminum, etc.) should never be in contact in a threaded joint without lubrication (such as a dry film) or some form of coating or plating since they will gall and possibly seize.

Example 3.7: Preload obtained from a torqued threaded retainer. (For design and analysis, use File 3.7 of the CD-ROM)

A 2.100 in. (53.340 mm) OD lens is to be clamped with a total preload of 12.50 lb (55.60 N) delivered by a retainer screwed into a cell on a thread of pitch diameter 2.200 in. (55.880 mm). Using Eq. (3.34a), approximately what torque should be applied?

Rearranging Eq. (3.34a):

$$Q = \frac{PDT}{5} = (12.50) \left(\frac{2.200}{5}\right) = 5.5 \text{ lb-in.} (0.62 \text{ N-m}).$$

An aspect of threaded retainer design that has, to the best knowledge of this author, escaped consideration in the literature on optical instrumentation design is the question of preferred dimensions for the threads and the stress developed in those threads by an applied axial preload. Intuitively, one might expect that a coarse thread would withstand axial force better than a fine one. Dimensional or "packaging" constraints might, on the other hand, require the use of fine threads in order to minimize wall thickness and overall diameter of the mount. Of course, extra care must be exercised in assembling a fine-threaded retainer to prevent "crossing" the threads and rendering the parts unusable.

Figure 3.19 shows the commonly used terminology for screw threads, while Fig. 3.20 shows the basic profile of a thread.¹¹ The dimension designations apply to a metric bolt (here, the retainer's thread) and its matching nut (here, the internal thread in the mount). The profile of a thread with inch dimensions is essentially the same as that shown in Fig. 3.20. These belong to the unified thread system, with two major series called "UNC" and "UNF" for coarse and fine pitches, respectively.

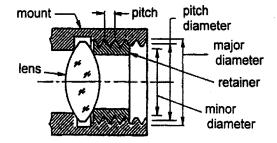


Figure 3.19 Schematic showing the terminology for retainer screw threads.

We are interested here in determining the average stress in the threads as the total axial preload divided by the annular area over which that force is distributed. We then compare that stress with the yield stress of the materials used since deformation of the threads is the chief concern. From the geometry of Fig. 3.20, the crest-to-root thread height, H, is related to the thread pitch, p, by the following equation:

$$H = (0.5)(3)^{V_2}(p) = 0.866 p.$$
(3.35)

The annular region of the thread actually in contact has a radial dimension of (5/8)H. Hence, the annular area per thread is:

$$A_{T} = \pi D_{T} \left(\frac{5H}{8}\right) = 1.700 D_{T} p, \qquad (3.36)$$

where D_T is the pitch diameter of the thread.

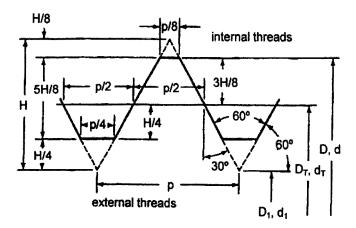


Figure 3.20 Basic thread profile where D(d) = major diameter, $D_1(d_1)$ = minor diameter, $D_T(d_T)$ = pitch diameter, p = pitch. Capital and lower-case letters represent external and internal threads, respectively.

It is well known that the first few (typically three) engaged threads on a machine screw carry most of the tensile load developed when the screw is tightened. Assuming this is also the case for a threaded lens retainer, the total annular area in contact is $3A_T$. Hence, the stress in the threads, S_T , is approximately

$$S_T = \frac{P}{3A_T} = \frac{0.196P}{D_T p}.$$
 (3.37)

It should be noted that for those common designs in which the mount has a higher CTE than the lens, the thread stress should be estimated at the lowest survival temperature since the preload is the greatest and this represents a worst-case situation. In Section 14.3, we show how to estimate the preload change in typical lens mounting configurations as the temperature drops to its minimum expected value.

Example 3.8 demonstrates the use of Eq. (3.37).

Example 3.8: Stress in retainer threads due to axial preload. (For design and analysis, use file 3.8 of the CD-ROM)

(a) Estimate the stress in the threads of the retainer in Example 3.7 and the applicable safety factor if the thread size is 32 threads per inch (tpi) (1.26 threads/mm) assuming the preload increases to 935 lb (4161 N) at a low temperature. The metal parts are aluminum 6061T6 with a yield stress of 38,000 lb/in.² (262.0 MPa). (b) What is the finest thread that produces a safety factor of 2.0?

(b) From Example 3.7, $D_T = 2.200$ in. (55.880 mm) By Eq. (3.37):

$$S_{\tau} = \frac{(0.196)(935)}{(2.200)(1/32)} = 2665.6 \text{ lb/in}^2 (18.4 \text{ MPa}).$$

This stress is far smaller than the yield stress so thread failure should not be a concern. The safety factor is

$$\frac{38,000}{2665.6}$$
=14.3.

(c) For a safety factor of 2.0:

$$S_T = \frac{38,000}{2} = 19,000 = \frac{(0.196)(935)}{(2.2)(1/p)}$$

Then,

$$p = \frac{(19,000)(2.2)}{(0.196)(935)} = 228.1$$
 tpi (8.98 threads/mm).

This thread would probably be too fine for easy assembly as thread crossing might occur.

3.6.2 Clamping (flange) ring

A typical design for a lens mounting with a flange-type retaining ring is shown in Fig. 3.21. These retainers are most frequently used with large-aperture lenses where manufacture and assembly of a threaded ring would be difficult. The functions of these flanges are very

much like those of the threaded ring described earlier, but flanges offer distinct advantages—as will be apparent from the following description.

The magnitude of the preload produced by a given axial deflection Δx of the flange shown in Fig. 3.21 can be approximated by considering the flange to be a perforated circular plate with its outer edge fixed and an axially directed load applied uniformly along the inner edge to deflect that edge. The applicable equation relating inner edge deflection to total preload as given by Roark¹² can be rewritten as follows:

$$\Delta x = \left(K_A - K_B\right) \left(\frac{P}{t^3}\right),\tag{3.38}$$

where

$$K_{A} = \frac{3(m^{2}-1)\left[a^{4}-b^{4}-4a^{2}b^{2}\ln\left(\frac{a}{b}\right)\right]}{\left(4\pi m^{2}E_{M}a^{2}\right)},$$
(3.39)

and

$$K_{B} = \frac{3(m^{2}-1)(m+1)\left[2\ln\left(\frac{a}{b}\right) + \left(\frac{b^{2}}{a^{2}}\right) - 1\right]\left[b^{4} + 2a^{2}b^{2} \ln\left(\frac{a}{b}\right) - a^{2}b^{2}\right]}{4\pi m^{2}E_{M}[b^{2}(m+1) + a^{2}(m-1)]}.$$
 (3.40)

The total preload is P, t is the flange thickness, a and b are the outer and inner radii of the cantilevered section, m is the reciprocal of Poisson's ratio (v_M) and E_M is the Young's modulus of the flange material.

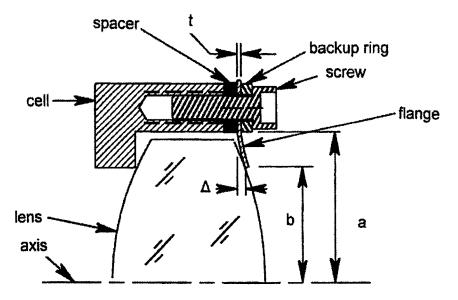


Figure 3.21 Schematic configuration of a flange-type retainer axially constraining a lens in a mount. The deflection Δ corresponds to Δx in the text.

The spacer under the flange can be ground at assembly to the particular axial thickness that produces the predetermined flange deflection when firm metal-to-metal contact is achieved by tightening the clamping screws. Customizing the spacer accommodates variations in as-manufactured lens thicknesses. The flange material and thickness are the prime design variables. The dimensions a and b, and hence the annular width (a - b) of the bent region of the flange, can also be varied, but these are usually set primarily by the lens aperture, mounting wall thickness, and overall dimensional requirements.

An important factor to be considered in designing a flange retainer for a lens is the stress S_B built up in the bent portion of the flange. This must not exceed the yield stress S_Y of the material. The following equations adapted from Roark ¹² apply:

$$S_B = K_C P / t^2 = S_Y / f_S, \qquad (3.41)$$

where

$$K_{c} = \left[\frac{3}{2\pi} \right] \left[1 - \frac{2mb^{2} - 2b^{2}(m+1) \ln(a/b)}{a^{2}(m-1) + b^{2}(m+1)} \right].$$
(3.42)

A useful relationship is obtained by solving Eq. (3.41) for t:

$$t = (f_s K_c P / S_y)^{1/2}$$
(3.43)

A typical design case is given in Example (3.9).

It is important for the end of the mount or cell to which the flange is referenced to be flat and parallel to the axial reference surface (the shoulder in Fig. 3.21). Also, the clamped annular region of the flange should be stiff enough for the deflection Δx measured between the attachment points (screws) to be essentially the same as that at the screws. A simple design adds a back-up ring between the screw heads and the flange as shown in Fig. 3.21. This ring can be made of aluminum if it is thick enough to ensure uniform clamping action. A back-up ring of stiffer material such as titanium or CRES could be thinner. If the flange is machined from a thick blank, the clamping ring can be an integral part.

A great advantage of the flange-type constraint over the threaded retainer is that it can be calibrated so we know quite precisely what preload will be delivered when the flange is deflected by a particular distance, Δx . This measurement of the spring constant of the flange can be done offline using a load cell or other means to measure the force produced by various deflections. This refines the performance prediction made during design using the above equations. Since the test is nondestructive, we can safely assume that, during actual use, the hardware will behave as measured.

Another technique for holding the flange of Fig. 3.21 against the end of the mount is shown in Fig. 3.22. Here, a threaded cap is used instead of multiple screws. The prime benefit is that the cap tends to hold the flange uniformly all around its periphery, while the screws hold it intermittently. The reference surface machined into the cap should be flat and the thread axis perpendicular to that surface. As in the case of threaded retainers, the fit of the threads in this cap should be Class-1 or -2 per ANSI Publication B1.1-1982 so the cap can square itself to the flange as necessary. Slots or holes for a wrench should be provided.

Example 3.9: Deflection of a flange ring. (For design and analysis, use File 3.9 of the CD-ROM)

Consider a 15.750 in. (40.005 cm) diameter corrector plate for a telescope with a total preload *P* of 120 lb (533.8 N) distributed uniformly around and near the edge of the plate by a titanium (Ti6Al4V) flange that limits the aperture of the plate to 15.500 in. (39.37 cm). A radial clearance of 0.010 in. (0.254 mm) is to be provided between the plate OD and the mount ID. Calculate (a) the required flange thickness *t* for a safety factor of 2 and (b) the flange inner edge deflection Δx .

Design dimensions and material properties are:

$$a = \left(\frac{15.750}{2}\right) + 0.010 = 7.885 \text{ in. } (200.280 \text{ mm}),$$

$$b = \frac{15.500}{2} = 7.750 \text{ in.} (196.850 \text{ mm}),$$

$$E_{M} = 120,000 \text{ lb/in.}^{2} (1.14 \times 10^{5} \text{ MPa}), v_{M} = 0.340,$$

$$S_{\gamma} = 120,000 \text{ lb/in.}^2 (827.364 \text{ MPa}), \qquad m = 1/v_M = 2.941$$

(a) From Eq. (3.42):

$$K_{c} = \left[\frac{3}{(2\pi)} \right] \left[1 - \frac{(2)(2.941)(7.750^{2})(2.941 - 1) \left[\ln \left(\frac{7.885}{7.750} \right) \right]}{(7.750^{2})(2.941 - 1) + (7.750^{2})(2.941 + 1)} = 0.0164.$$

From Eq. (3.43):

$$t = \left[\frac{(2)(0.0164)(120)}{120,000}\right]^{1/2} = 0.0057 \text{ in. } (0.145 \text{ mm}).$$

(b) From Eqs. (3.39), (3.40), and (3.38):

$$K_{A} = \frac{(3)(2.941^{2} - 1)\left\{ \left[7.885^{4} - 7.750^{4} - (4)(7.885^{2})(7.750^{2}) \right] \ln\left(\frac{7.885}{7.750}\right) \right\}}{(4\pi)(2.941^{2})(16.5 \times 10^{6})(7.885^{2})}$$

= 1.0556 × 10⁻¹¹ in.⁴/lb,

$$(3)(2.941^{2} - 1)(2.941 + 1)\left\{(2)\left[\ln\left(\frac{7.885}{7.750}\right)\right] + \left(\frac{7.885^{2}}{7.750^{2}}\right) - 1\right\}$$

$$K_{B} = \frac{\left\{7.750^{4} + (2)(7885^{2})(7.750^{2})\left[\ln\left(\frac{7.885}{7.750}\right) - \frac{7.885^{2}}{7.750^{2}}\right]\right\}}{(4\pi)(2.941^{2})(16.5 \times 10^{6})\left[(7.750^{2})(2.941 + 1) + (7.885^{2})(2.941 - 1)\right]}$$

$$= 1.8321 \times 10^{-13} \text{ in.}^{4}/\text{lb},$$

$$\Delta x = \left(1.0556 \times 10^{-11} - 1.8321 \times 10^{-13}\right)\left(\frac{120}{0.0057^{3}}\right) = 0.0067 \text{ in.}(0.171 \text{ mm}),$$
Fo check using Eq. (3.41): $S_{B} = \frac{(0.0164)(120)}{0.0057^{2}} = 60,572 \text{ lb/in.}^{2}.$

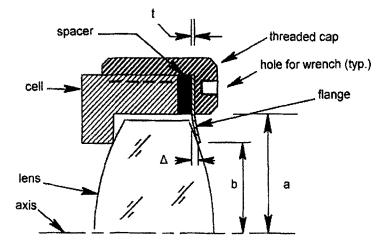


Figure 3.22 Using a threaded cap rather than a set of screws to secure a flange. The deflection Δ corresponds to Δx in the text.

3.7 Constraining the Lens with Multiple Spring Clips

A simple way to clamp a lens into its mount is illustrated by Fig. 3.23. Here, the plano surface of the lens rests against three thin Mylar pads (shown with exaggerated thickness) attached to a shoulder in the cell. The rads are located at 120-deg intervals and serve as semikinematic registration surfaces. Three metal clips that act as cantilevered springs apply preload. The outer ends of the clips are attached to the cell with screws. Spacers between the clips and the cell are machined at assembly to impart specific deflections to the clips, thereby providing preload for the lens. The clips are located so the preload is directed through the lens directly toward the pads. This minimizes bending moments that otherwise could be applied to the lens. The use of the Mylar pads reduces the need to machine the shoulder as a geometrically accurate and smooth surface. It should be normal to the lens optical axis, however. In some lens mounting designs of this type, additional clips are used; they are usually spaced equally around the lens periphery. The portion of the preload derived from each clip is reduced. This reduces the bending stress within each clip.

The following equations (adapted from Roark¹²) can be used to calculate the deflection Δx required of each of N clips from its relaxed (undeflected) condition to provide a specific total preload and the bending stress developed within each clip:

$$\Delta x = \frac{(1 - v_M)(4PL^3)}{(E_M bt^3 N)}.$$
(3.44)

$$S_B = \frac{6PL}{\left(bt^2N\right)},\tag{3.45}$$

where: v_M is Poisson's ratio for the clip material, P is the total preload, L is the free (cantilevered) length of the clip, E_M is Young's modulus for the clip material, b is the width of the clip, t is the thickness of the clip, N is the number of clips employed, and S_Y is the yield stress for the clip material. Example 3.10 shows how to use these equations.

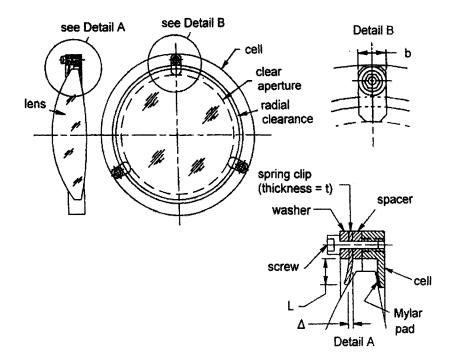


Figure 3.23 Concept for a lens mounting using three radially oriented cantilevered springs to provide preload against pads on a cell shoulder.

Example 3.10: Cantilevered spring lens mounting (For design and analysis, use File 3.10 of the CD-ROM.) A lens is to be constrained axially against Mylar pads with a preload of 60 lb (267 N) by three Ti6Al4V titanium spring clips with dimensions L = 0.312 in. (7.925 mm), b = 0.375 in. (9.525 mm), and t = 0.041 in. (1.041 mm). (a) How much should each clip be deflected, and (b) what stress safety factor f_S exists? From Table B12, $v_M = 0.340$, $E_M = 1.65 \times 10^7$ lb/in.² (1.14 × 10⁷ MPa) and $S_Y = 1.2 \times 10^5$ lb/in.² (827.4 MPa). (a) Applying Eq. (3.44): $\Delta x = (1 - 0.340^2)(4)(60)(0.312^3)/[(1.65 \times 10^7)(0.375)(0.041^3)(3)]$ = 0.0051 in. (0.129 mm). (b) From Eq. (3.43): $S_B = (6)(60)(0.312)/[(0.375)(0.041^2)(3)]$ $= 5.94 \times 10^4$ lb/in.² (410 MPa). Then, $f_S = 1.2 \times 10^5/5.94 \times 10^4 = 2.0$. With conventional mechanical measuring devices, it should be possible to determine the deflection, Δx , of an actual spring clip within 0.0005 in. (0.0013 mm). Because of the linear relationship between Δx and P expressed in Eq. (3.44), the preload should be determined to the same level of accuracy. Hence, in the case described in Example 3.10 where the nominal deflection is 0.0051 in., the preload would be known within 0.0005/0.0051 = 0.098 or about 10%. This level of predictability should be adequate for most applications. If not, a more accurate measuring technique should be employed.

Another point of interest regarding the use of cantilevered spring clips was pointed out by Roark.¹² This is that the bending stress for a given deflection in such a clip could be reduced by a factor of about three from that given by Eq. (3.45) if the clip of given thickness and free length were not perforated for a screw constraint, but rather clamped firmly in place. This added design complexity would allow a larger deflection before the yield stress would be reached and improve the accuracy of preload determination for a given deflection measurement capability.

In some optical systems, such as laser diode beam collimators, optical correlators, anamorphic projectors, and some scanning systems, the natural aperture shapes of some lenses, windows, prisms, and mirrors are rectangular, racetrack, trapezoidal, etc., because their fields of view or beam dimensions are different in the vertical and horizontal meridians. Cylindrical, toroidal, and nonrotationally symmetrical aspheric optical surfaces are frequently used in such optical systems to create the desired beam shapes or to introduce different magnifications in orthogonal directions. These lenses have nonrotationally symmetrical surface shapes and may have noncircular apertures, so they cannot be mounted conventionally in circular cells or held in place by threaded retainers since the surface sagittas at a given distance from the axis are not equal. Mounts for these optics are usually customized for the particular application.

A simple example of such a mounting design is shown in Fig. 3.24. The lens is a plano-concave cylindrical lens with a 2:1 aperture-aspect ratio. The lens is clamped with four spring clips into a rectangular recess machined into a flat plate. The plate is circular, so it can be attached conventionally to the structure of the instrument. Note that a slot is provided to align the lens to the system axis by a pin or key (not shown). The clips provide localized preload to hold the lens in the recess under the anticipated axial acceleration forces. Four Mylar pads are attached to the mount shoulder interface with the flat side of the lens directly opposite the clips. These pads must be accurately coplanar in order not to bend the lens by over constraint from the clips, so the shoulder must be accurately machined flat. With small optics, three pads probably would provide adequate support and function semikinematically. The design of the clips follows that described for circular lenses.

Lenses with convex or concave cylindrical surfaces can be registered against offcenter parallel rods as shown schematically in Fig. 3.25. The rods are pressed into accurately parallel holes bored through the mount at the appropriate distances from the mount axis. Some means of preloading would be needed to maintain axial contact between the lens and the rods under axial acceleration. This constraint might well be provided by spring clips or an appropriately shaped retaining flange. By contacting the cylindrical surface, a mounting of this type would facilitate control of the rotational alignment of the cylinder axis relative to other parts of the system.

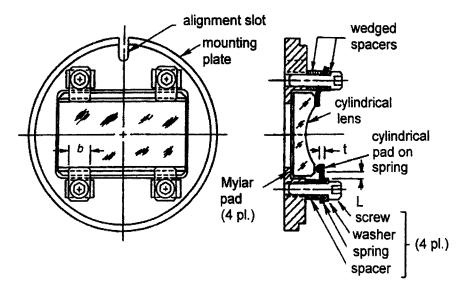


Figure 3.24 Schematic diagram of a mounting for a cylindrical lens.

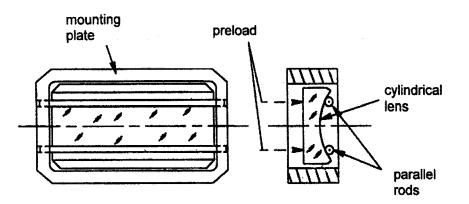


Figure 3.25 A parallel-rod type of registration interface for a cylindrical lens.

3.8 Geometry of the Lens-to-Mount Interface

3.8.1 The sharp-corner interface

The sharp-corner interface is the circle created by the intersection of a cylindrical bore and a flat surface machined perpendicular to the axis of that bore. It is the interface easiest to produce and has been used over the years in the vast majority of optical instruments. In reality, the sharp corner is not actually a knife-edge. Delgado and Hallinan¹⁴ quantified it as one, which, in accordance with standard machine shop practice, is burnished with a hardened tool to remove burrs and other irregularities. The authors reported the results of a series of tests wherein many parts made to drawings typical of a lens cell and specifying a sharp corner were manufactured and measured. The burnished corners were rounded and had an average radius on the order of 0.002 in. (0.05 mm).

This rounded surface contacts a convex lens surface at a height of y_c as shown in Fig. 3.26(a). The configuration for a concave surface is as shown in Fig. 3.26(b). The surface radius is *R*, the lens clear aperture is *A*, y_s is the height of the innermost surface on the mount. It is typically 0.5% larger than the aperture radius, as defined by

$$y_s = 0.505 A.$$
 (3.46)

 P_1 is a point on the mount located at an axial distance Δx from the surface vertex V. This distance provides a basic reference for dimensioning the mount.¹⁵ Figures 3.27(a) and 3.27(b) show variations of the design in which the interface with the glass is at an edge with an obtuse angle (135 deg). Machining this angle >90 deg tends to make the edge smoother.¹⁶

In the four cases of Figs. 3.26(a) through 3.27(b), the dimension Δx is simply the sagittal depth of the spherical surface at the contact height y_C :

$$\Delta x = R - \left(R_2 - y_C^2\right)^{1/2}.$$
 (3.47)

In Fig. 3.26(a), $y_C = y_S$, while in view (b) of that figure and in Figs. 3.27(a) and 3.27(b), the contact typically occurs at a height of y_C , where

$$y_C = \left(\frac{y_S}{2}\right) + \left(\frac{D_G}{4}\right). \tag{3.48}$$

Example 3.11 applies Eqs. (3.46) through (3.48) to each of the last four cases.

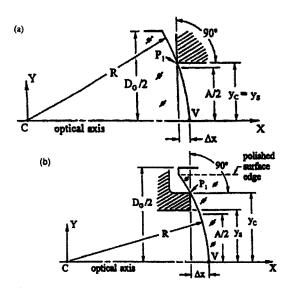


Figure 3.26 Schematics of 90-deg sharp-corner interfaces with (a) a convex lens surface and (b) a concave lens surface.

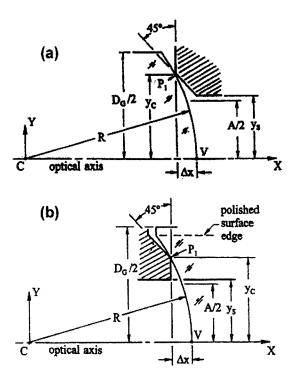


Figure 3.27 Schematics of 135-deg sharp-corner interfaces on (a) a convexlens surface and (b) on a concave-lens surface.

3.8.2 The tangential (conical) interface

If the spherical lens surface contacts a conical surface in the mount, the design is called a "tangential interface" (see Fig. 3.28). The tangential interface is not feasible with a concave-lens surface, but it is generally regarded as the nearly ideal interface for convex surfaces.

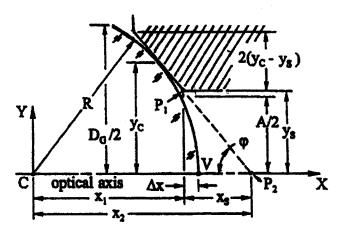


Figure 3.28 Schematic of a tangential interface on a convex spherical surface.

Example 3.11: Location of the mount corner P1 with respect to the lens vertex with a sharp corner interface. (For design and analysis, use File 3.11 of the CD-ROM).

A lens of diameter $D_G = 2.100$ in. (53.340 mm), aperture A = 2.000 in. (50.800 mm), and surface radius of 20.000 in. (508.000 mm) is interfaced with a 90 deg sharp corner on a mount shoulder. What is the dimension Δx if: (a) the surface is convex, (b) concave. Repeat these calculations as parts (c) and (d) assuming that the corner is cut at 135°.

(a) Per Fig. 3.26(a) and Eq. (3.46), $y_C = y_S = (0.505)(2.000) = 1.010$ in. (25.654 mm). By Eq. (3.47), $\Delta x = 20.000 - (20.000^2 - 1.010^2)^{1/2} = 0.025$ in. (0.648 mm)

(b) By Eq. (3.46),
$$y_S = (0.505)(2.000) = 1.010$$
 in. (25.654 mm). Fig. 3.26(b) applies.
By Eq. (3.47), $y_C = \left(\frac{1.010}{2}\right) + \left(\frac{2.100}{4}\right) = 1.030$ in. (26.162 mm)
By Eq. (3.46), $\Delta x = 20.000^2 - (20,000^2 - 1.030^2)^{1/2} = 0.026$ in. (0.660 mm)

- (c) By Eq. (3.46), $y_S = (0.505)(2.000) = 1.010$ in. (25.654 mm). Fig. 3.27(a) applies. By Eq. (3.47), $y_C = \left(\frac{1.010}{2}\right) + \left(\frac{2.100}{4}\right) = 1.030$ in. (26.162 mm) By Eq. (3.46), $\Delta x = 20.000^2 - (20,000^2 - 1.030^2)^{1/2} = 0.026$ in. (0.660 mm)
- (d) By Eq. (3.46), $y_S = (0.505)(2.000) = 1.010$ in. (25.654 mm). Fig. 3.27(b) applies. By Eq. (3.47), $y_C = \left(\frac{1.010}{2}\right) + \left(\frac{2.100}{4}\right) = 1.030$ in. (26.162 mm) By Eq. (3.46), $\Delta x = 20.000^2 - (20,000^2 - 1.030^2)^{1/2} = 0.026$ in. (0.660 mm)

We observe that cases (b) through (d) give the same result. This is because y_s is the same for these three cases.

Easily made by modern machining technology, the conical interface tends to produce smaller contact stress in the lens for a given preload than the sharp corner interface. This attribute of the conical interface is discussed in Sect. 13.5.2.

The cone half-angle, φ , is determined by the following equation:

$$\varphi = 90 \text{ deg} - \arcsin\left(\frac{y_c}{R}\right).$$
 (3.49)

We again define y_c as the midpoint between the y_s and the rim of the lens per Eq. (3.48). The tolerance for φ in a given design depends primarily on the desired radial width of the conical annulus, or land, on the metal part and the allowable error in axial location of the lens vertex. Usually, the tolerance on φ can be at least ± 1 deg.

Equations (3.46) and (3.48) through (3.53) can be used to establish Δx , the axial distance from P_1 to V, as shown in Fig. 3.28. Example 3.12 illustrates the use of these equations.

$$x_s = y_s \tan \varphi, \tag{3.50}$$

$$x_2 = \frac{R}{\sin \varphi},\tag{3.51}$$

$$x_1 = x_2 - x_s, (3.52)$$

$$\Delta x = R - x_1. \tag{3.53}$$

Example 3.12: Location of the mount corner P_1 with respect to the vertex of a convex-lens surface with a tangential interface. (For design and analysis, use File 3.12 of the CD-ROM.)

A lens of diameter 2.100 in. (53.340 mm) has a convex surface of radius 20.000 in. (508.000 mm). The mechanical interface is tangential. What is Δx if the required aperture is 2.000 in (50.800 mm)?

By Eqs. (3.46) and (3.48),

$$y_{S} = (0.505)(2.000) = 1.010$$
 in. (25.654 mm),
 $y_{C} = \left(\frac{1.010}{2}\right) + \left(\frac{2.100}{4}\right) = 1.030$ in. (26.162 mm),
By Eq. (3.49), $\varphi = 90$ deg - arcsin $\varphi = 90$ deg - arcsin $\left(\frac{1.030}{20.000}\right) = 87.048$ deg,
By Eqs. (3.50 through (3.53),
 $x_{S} = \left(\frac{1.010}{\tan 87.048 \text{ deg}}\right) = 0.052$ in. (1.323 mm),
 $x_{2} = \left(\frac{20.000}{\sin 87.048 \text{ deg}}\right) = 20.027$ in. (508.675 mm),
 $x_{1} = 20.027 - 0.052 = 19.975$ in. (507.365 mm)
 $\Delta x = 20.000 - 19.975 = 0.026$ in. (0.635 mm)

3.8.3 The toroidal interface

A convex toroidal (donut-shaped) interface can be used with either convex or concave surfaces and is particularly useful on concave-lens surfaces, where the tangent interface cannot be used. The reasons for this (which have to do with minimizing contact stress) will be explained further in Sect. 13.5.3. Figure 3.29(a) shows a toroidal (donut-shaped) mechanical surface contacting a convex-spherical-lens surface of radius R at height y_C .

The center of the toroidal surface with cross-sectional radius R_T is at C_T as shown in Fig. 3.29(b). Considering all radii as positive, then:

$$\theta = \arcsin\left(\frac{Y_c}{R}\right),\tag{3.54}$$

$$h = (R \pm R_T) \cos \theta, \qquad (3.55)$$

$$k = (R \pm R_T)\sin\theta, \qquad (3.56)$$

$$x_{1} = h \pm \left[R_{T}^{2} - \left(k - y_{S} \right)^{2} \right]^{1/2}, \qquad (3.57)$$

$$\Delta x = R - x_1 \tag{3.58}$$

In Eqs. (3.55) and (3.56), the positive sign is used for a convex lens surface and the negative sign is used for a concave surface. In Eq. (3.57), the choice of sign is reversed. Example 3.13 demonstrates the use of these equations for a typical design with a convex lens surface.

Figure 3.30 shows a toroidal interface on a concave spherical lens surface. Note that R_T is smaller than R. Equations (3.46), (3.48) and (3.56 through (3.58) are used to find Δx . Example 3.14 shows how to do this.

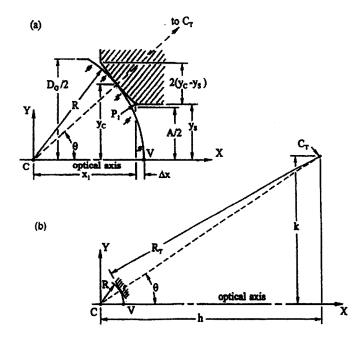


Figure 3.29 Schematics of a toroidal interface on a convex spherical lens surface: (a) detailed view (b) expanded view.

Example 3.13: Location of the mount corner P_1 with respect to a convex lens surface vertex with a toroidal interface. (For design and analysis purposes, use File 3.13 of the CD-ROM.)

A lens of diameter 2.220 in. (56.388 mm) has a convex surface of R = 10.000 in. (254.000 mm), interfacing with a toroidal mechanical surface of $R_T = 100.000$ in. (2540 mm). The aperture is 2.000 in. (50.800 mm). What is Δx ?

By Eqs. (3.46), (3.48), and (3.54) through (3.58) using plus signs in Eqs. (3.55) and (3.56), and minus signs in Eq. (3.57):

$$y_{S} = (0.505)(2.000) = 1.010 \text{ in.} (25.654 \text{ mm}),$$

$$y_{C} = \left(\frac{1.010}{2}\right) + \left(\frac{2.220}{4}\right) = 1.060 \text{ in.} (26.924 \text{ mm}),$$

$$\theta = \arcsin\left(\frac{1.060}{10.000}\right) = 6.085 \text{ deg},$$

$$h = (10.000 + 100.000)(\cos 6.085 \text{ deg}) = 109.380 \text{ in.} (2778.259 \text{ mm}),$$

$$e (10.000 + 100.000)(\sin 6.085 \text{ deg}) = 11.660 \text{ in.} (296.174 \text{ mm}),$$

$$x_{1} = 109.380 - [100.000^{2} - (11.660 - 1.010)^{2}]^{1/2} = 9.949 \text{ in.} (252.705 \text{ mm}),$$

$$\Delta x = 10.000 - 9.949 = 0.051 \text{ in.} (1.295 \text{ mm}).$$

The radial extent of the toroidal land against which the lens can touch in each case can be seen from Figs. 3.28 and 3.29 to be $2(y_C - y_S)$. In Examples 3.12 and 3.13, this annular width is (2)(1.025 - 1.000) = 0.050 in. (1.270 mm). Knowledge of this dimension is useful when defining tolerances for the mechanical parts to prevent degeneration of the interface into line (sharp-corner) contact at the inner or outer edges of this land. Sharp-corner contact rather than toroidal contact would, in those cases, increase the stress for a given level of axial preload.

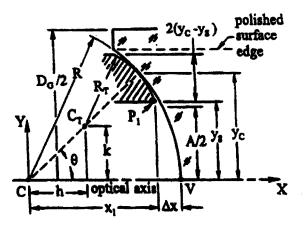


Figure 3.30 Schematic of a toroidal interface on a concave spherical lens surface.

Either of these toroidal surfaces can be cut on a numerically controlled lathe or diamond turning machine with little difficulty. The exact cross-sectional radius produced is not overly critical. Reasonable tolerances might be of the order of ± 25 % of R_T . The basis for this conclusion is twofold: the simple geometry allows such a variation and our considerations of contact stresses for these types of interfaces later in this book allows even greater variations.

Example 3.14: Location of the mount corner P_1 with respect to a concave-lens surface vertex with a toroidal interface. (For design and analysis purposes, use File 3.14 of the CD-ROM).

A lens of diameter 2.220 in. (56.388 mm) has a concave surface of R = 10.000 in. (254.000 mm) interfacing with a toroidal mechanical surface of $R_T = 5.000$ in. (127.000 mm). The aperture is 2.000 in. (50.800 mm). What is Δx ?

By Eqs. (3.46), (3.48), and (3.54) through (3.58) using minus signs in Eqs. (3.55) and (3.56), and plus signs in Eq. (3.57):

 $y_{S} = (0.505)(2.000) = 1.010 \text{ in.} (25.654 \text{ mm}),$ $y_{C} = \left(\frac{1.010}{2}\right) + \left(\frac{2.220}{4}\right) = 1.060 \text{ in.} (26.924 \text{ mm}),$ $\theta = \arcsin\left(\frac{1.060}{10.000}\right) = 6.085 \text{ deg},$ $h = (10.000 - 5.000)(\cos 6.085 \text{ deg}) = 4.972 \text{ in.} (126.284 \text{ mm})$ $k = (10.000 - 5.000)(\sin 6.085 \text{ deg}) = 0.530 \text{ in.} (13.462 \text{ mm})$ $x_{1} = 4.972 + [5.000^{2} - (0.530 - 1.010)^{2}]^{1/2} = 9.949 \text{ in.} (252.702 \text{ mm})$ $\Delta x = 10.000 - 9.949 = 0.051 \text{ in.} (1.295 \text{ mm})$

3.8.4 The spherical interface

Spherical glass-to-metal interfaces on convex and concave surfaces are shown in Fig. 3.31(a) and (b), respectively. The contact height y_c may be considered to be at the midpoint of the land contacting the lens surface. However, for this interface to work properly, intimate contact is needed over the entire area of the land so no particular height of contact is different from any other such height. The defining characteristic of this type interface is this very point, i.e., intimate contact between surfaces whose radii match within a few wavelengths of light.

To produce a mechanical surface of a particular radius within this degree of accuracy requires that the surface be carefully lapped using tools commonly found only in an optical shop. In fact, these surfaces typically are finished by opticians using fine grinding tools matched to the tools used to make the companion optical surface. The configuration of the mechanical part must provide easy access to the surface to be lapped so the lapping tool can move across and beyond the area to be finished.

Because the manufacture and testing of these mechanical parts are expensive, the spherical interface is not frequently used. Its use is generally confined to systems expected to encounter extreme vibration or shock or ones needing intimate contact between the glass and metal for heat transfer reasons.

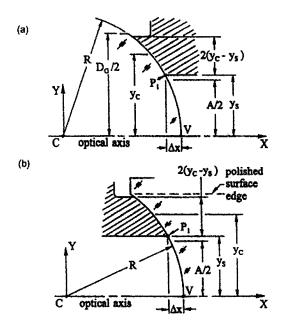


Figure 3.31 Schematics of spherical interfaces on (a) a convex lens surface and (b) a concave lens surface.

3.8.5 Interfaces with bevels on optics

It is standard optical shop practice to lightly bevel all sharp edges of optics. This minimizes the danger of chipping, so such bevels are called protective bevels. Larger bevels, or chamfers, are used to remove unneeded material when weight is critical or packaging constraints are tight, and/or to provide mounting surfaces. Usually all these secondary surfaces are ground with progressively finer abrasives. If the lenses are likely to have to endure severe stress, the bevels and the lens rims may also be given a crude polish by buffing with polishing compound on a cloth or felt-covered tool. These grinding and polishing procedures tend to strengthen the lens material by removing subsurface damage resulting from the grinding operations. Acid etching of the ground surfaces also can be accomplished to achieve a similar result.

Views (a) through (c) of Fig. 3.32 show three lenses with bevels. The plano convex element of view (a) has minimum protective bevels that typically might be specified as "0.5-mm maximum face width at 45 deg "or" 0.4 ± 0.2 mm face symmetric to surfaces."

Each surface of the biconcave lens of view (b) has a wider annular bevel oriented perpendicular to the lens's optical axis. Applying axial preload cannot center this lens; some external means must be utilized. Tight tolerances on perpendicularity must be specified for such bevels if both centers of curvature of the lens surfaces are to be brought to the mount's mechanical axis simultaneously by lateral translation of the lens. Tolerances for this 90-deg angle of \pm 30 arcsec or less are common for precision lenses.

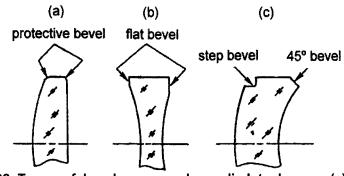


Figure 3.32 Types of bevels commonly applied to lenses. (a) Protective bevels, (b) flat bevels on concave surfaces, and (c) step and 45-deg angled bevels.

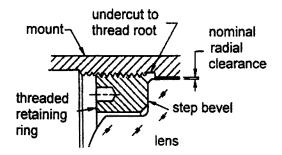


Figure 3.33 Details of a step bevel providing space for a threaded retaining ring.

Figure 3.32(c) shows a meniscus lens with a wide 45-deg bevel on the concave side and a step bevel ground into the rim on the other (convex) side to form a flat surface recessed into the lens. A conventional retainer or a spacer can be brought to bear against the latter surface. The step bevel is shown in more detail in Fig. 3.33. The step bevel is used primarily to provide room for a spacer or retaining ring when another optic must be located close to the surface to be beveled. A 45-deg bevel or radius should be provided on the inner leading edge of the mechanical part so that it does not interfere with a rounded inside corner that usually is created in a step bevel during manufacture.

It is not good practice to apply an axial preload directly against an angled bevel such as that shown at right in Fig. 3.32(c) since that surface may not be precisely located. A toroidal surface-contact interface on the concave side of the lens would be preferable. Protective bevels should be provided at the edges of all bevels.

Cemented doublet lenses are sometimes designed to have different diameters for the crown and flint elements so mounting can be accomplished on one or the other element without interfacing mechanically with both elements. Figures 3.34 and 3.35 are representative of these designs. There are at least two technical advantages to both designs: the weight of one element is reduced and any geometrical wedge in the cantilevered element or in the cement joint will not affect the symmetry of the mounting interface. Those wedges could, of course, affect optical performance.

In Fig. 3.34, the crown element is the larger. The mechanical interface on the convex surface is at a conical shoulder surface in the cell, while that on the concave side is on a precision flat bevel. The retainer preloads the lens axially by applying force to that bevel. The flint lens rim OD is smaller than the ID of the retainer so that rim is clear of the cell wall.

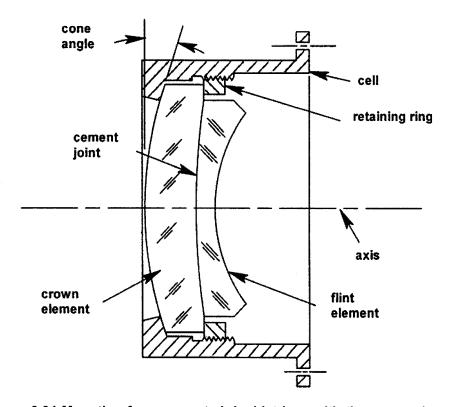


Figure 3.34 Mounting for a cemented doublet lens with the crown element larger in diameter than the flint element.

In Fig. 3.35, the flint element is the larger one. It has a precision flat bevel on the first (left) side and a step bevel on the second (right) side. The flat portion of the latter bevel is ground to be perpendicular to the lens' axis. The retainer applies axial preload. The rim of the flint element is shown as cylindrical. Its OD is slightly smaller than the ID of the cell. In an alternate configuration, the rim of that element could be a fine-ground sphere with a radius equal to $D_G/2$. This would allow the lens to fit closely into the cell ID for centering purposes, yet not jam against that surface before it seats properly against the shoulder if tilted somewhat during insertion.

The radius of the lens rim does not always need to equal $D_G/2$. A longer radius (called a crowned rim) would serve almost as well and does not require as much glass to be removed during edging as for the spherical rim. The maximum tilt angle that can be tolerated during assembly is reduced with a crowned rim, but this is not usually a problem.

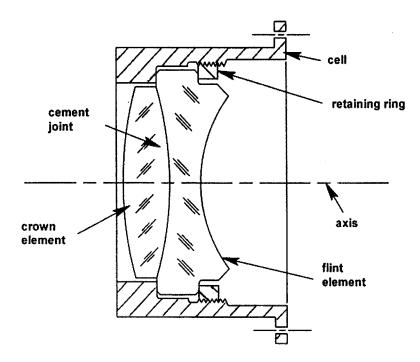


Figure 3.35 Mounting for a cemented doublet lens with the flint element larger in diameter than the crown element.

3.9 Elastomeric Mountings

A deceivingly simple technique for mounting lenses, windows, filters, and mirrors is illustrated schematically in Fig. 3.36. Shown is a typical design for a lens constrained by an annular ring of a resilient elastomeric material (typically epoxy, urethane, or RTV) within a cell. Hopkins¹⁶ reported that Dow Corning RTV732 is an appropriate material for this purpose, while Bayar⁸ indicated that Dow Corning RTV3112 has frequently been used in aerial cameras. General Electric RTV-88 and GE RTV8112 are representative of materials meeting U.S. military specification MIL-S-23586E. EC2216B/A epoxy made by 3M has been used for this purpose and is representative of that class of elastomer. Its outgassing characteristics are sufficiently low for it to be used with great success in many space applications. Characteristics of some elastomers are given in Tables B15a and B15b. Some epoxies are described in Table B14. Unfortunately, values for important properties such as Poisson's ratio and Young's modulus are not generally available for these materials. If possible, these should be measured before a given material is used in a specific application.

One side of the elastomeric ring is usually and intentionally left unconstrained so the material can deform under compression or tension caused by temperature changes. Registry of one optical surface against a machined reference surface in the mount helps maintain alignment established through use of temporary shims or an external fixture prior to adding the elastomer. The detailed view of Fig. 3.36 shows one means for holding the lens in place and constraining the elastomer while it cures. The fixture, which is made of Teflon or a similar plastic or metal coated with a mold release, is removed after curing. The elastomer is typically injected with a hypodermic syringe through radially oriented access holes in the mount until the space around the lens is filled.

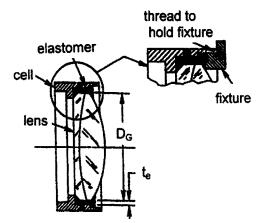


Figure 3.36 Technique for mounting a lens in an annular ring of cured-inplace elastomer. The detail view shows one way to hold the lens in place and constrain the elastomer with a tool while it cures.

If the annular elastomer layer has a particular thickness, t_e , the assembly will, to a firstorder approximation, be athermal in the radial direction. This minimizes stress buildup within the optomechanical components that is caused by differential radial expansion or contraction of the lens, cell, and elastomer under temperature changes. This thickness may be determined by the following equation, commonly referred to as Bayar's equation:⁸

$$t_{e \text{ Bayar}} = \left(\frac{D_G}{2}\right) \frac{(\alpha_M - \alpha_G)}{(\alpha_e - \alpha_M)},$$
(3.59)

where α_G , α_M , and α_e are the CTEs of the lens, mount, and elastomer, respectively, and the dimensions are as shown in Fig. 3.36.

The axial length of the elastomer layer approximately equals the edge thickness of the lens. Because Eq. (3.59) neglects the effects of this length, Poisson's ratio (v_e) , Young's modulus (E_e) , and the shear modulus, it should be considered an approximation. The application of this equation does not make the assembly athermal in the axial direction. When the temperature changes, the same nominal lengths of mounting wall, elastomer, and lens change at different rates are proportional to the applicable CTEs. Some amount of shear is then introduced into the elastomer layer.

Elastomers commonly used in optical applications typically have Poisson's ratios in the range of 0.4300 to 0.4999.^{*} Epoxies lie at the lower end of this range, while RTVs characteristically have the high values. The importance of this property has been revealed by various authors.¹⁷⁻¹⁹ For instance, Genberg¹⁷ provided a graph based on finite element analysis showing the dependence of effective elastomer CTE (α_e^*) relative to the bulk value (α_e) on v_e and joint aspect ratio. This graph is reproduced in Fig. 3.37. The aspect ratio is expressed as the ratio of joint axial length to joint thickness, or L/t_e . Most

^{*} The maximum value for Poisson's ratio is 0.5000.

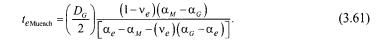
elastomeric ring mounts for lenses have L/t_e values on the order of 250/1, so the pertinent portions of the curves in the figure lie to the right of that value.

Herbert gave the most recent and thorough discussion of elastomeric athermal mountings for optics.²⁰ He considered the stress-strain relationship within the joint and compared several published equations for t_e , including Eq. (3.59). One conclusion reached in that study was that the effective CTE for the elastomer (α_e^*) can be determined by the following equation:

$$a_{e}^{*} = \frac{\alpha_{e}(1 + v_{e})}{(1 - v_{e})}.$$
(3.60)

Herbert also suggested that using α_e^* instead of the bulk value in Bayar's equation gives a better approximation for t_e .

Another equation for t_e that does include the effect of Poisson's ratio is the so-called, "Muench equation." This is Eq. (3.61), where the elastomer's bulk CTE is used:



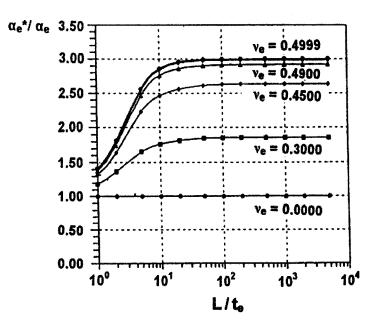


Figure 3.37 Variations of the ratio of an elastomer's effective CTE to it's bulk CTE with Poisson's ratio and with joint aspect ratio. (From Genberg.¹⁸)

Example 3.15 compares the results of using Eqs. (3.59) and (3.61) to find t_e for a typical design application.

Example 3.15: Athermal thickness for an elastomer ring mounting for a lens. (For design and analysis, use File 3.15 of the CD-ROM.) A 2.051 in. (52.095 mm) diameter germanium lens is to be mounted in an annular ring of elastomer in an aluminum cell. The material properties are: $\alpha_G = 3.22 \times 10^{-6}$, $^{\circ}$ F (5.8×10^{-6} , $^{\circ}$ C), $\alpha_M = 12.78 \times 10^{-6}$, $^{\circ}$ F (23.0×10^{-6} , $^{\circ}$ C), $\alpha_e = 137.8 \times 10^{-6}$, $^{\circ}$ F (248×10^{-6} , $^{\circ}$ C), and $v_e = 0.49$. (a) Apply Eqs. (3.59) and (3.60) to obtain a value for the ring thickness $t_{e \text{ Bayar}}$. (b) Use Eq. (3.61) to obtain the ring thickness $t_{e \text{ Muench}}$. (c) Compare. (a) From Eq. (3.60), $\alpha_e^* = (1 + 0.49)(137.8 \times 10^{-6})/(1 - 0.49) = 402.5 \times 10^{-6}$, $^{\circ}$ F From Eq. (3.59), $t_{e \text{ Bayar}} = (2.051/2)(12.78 - 3.22)(10^{-6})/[(402.5 - 12.78)(10^{-6})] = 0.0251$ in. (0.637 mm) (b) From Eq. (3.61), $t_{e \text{ Muench}} = (2.051/2)(12.78 - 3.22)(10^{-6})/[137.8 - 12.78 - (0.49)(137.8 - 3.22)(10^{-6})] = 0.0262$ in. (0.665 mm)

(c) The results from the two calculations are essentially the same (~4% variation).

Miller²¹ described a finite element analysis of three generic elastomerically mounted lens designs involving germanium lenses of the same [2.051 in. (52.095 mm)] diameter and with meniscus, biconvex, and biconcave shapes (see Fig. 3.38). The lenses were all mounted in aluminum cells with a silicone rubber ring. Poisson's ratio was chosen as 0.49. Miller estimated the elastomer layer thickness from Bayar's equation [Eq. (3.59)] to be 0.077 in. (1.956 mm).

In his study, Miller varied t_e from the nominal "athermal" value to 0.005 in. (0.127 mm) and determined (by FEA) the stresses within the lenses and the cells. Figure 3.39 shows the stress distributions at 122°F (50°C) for each lens shape at several values for t_e . The maximum stress levels (darkest ends of the gray scales) are indicated for each case. The contours of the lenses and of the elastomer layers have been emphasized here to make those shapes more visible. Fidelity of reproduction is poor in the figures, but definite changes are apparent. The highest stresses are found, as might be expected, with the thinner elastomer thicknesses. Note that the elastomer bulges outward at high temperatures for the "athermal" thickness cases (three cases shown on the right). This indicates compression. The layer is obviously in tension, as shown by the concave boundaries, at the smaller thicknesses (three cases shown on the left). Apparently the elastomer would fill the available space between the lens rim and the cell ID for some intermediate thicknesse.

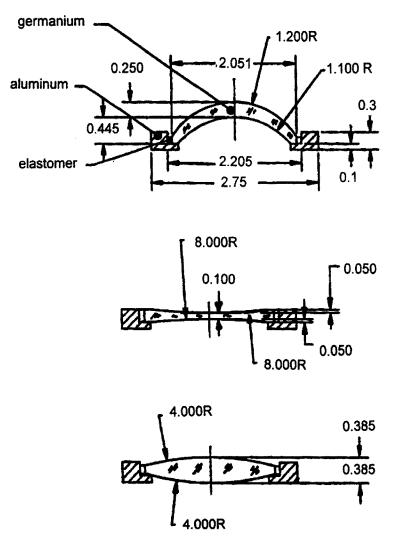


Figure 3.38 Schematics of lens configurations studied by Miller:²¹ (a) meniscus, (b) biconcave, and (c) biconvex.

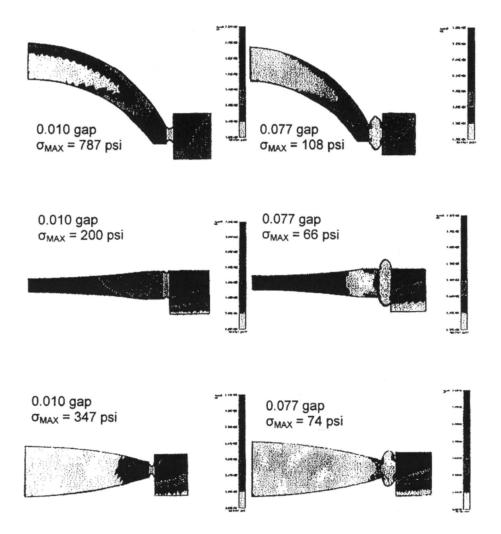


Figure 3.39 Stress magnitudes and distributions for the lenses of Fig. 3.38 at 122°F (50°C) for elastomer layers of 0.077 in. (1.956 mm) (supposedly "athermal") to 0.010 in. (0.254 mm) annular thicknesses. (Adapted from Miller.²¹)

Figure 3.40 shows the variations of maximum stresses with t_e for all three lens types. The most interesting general conclusions to be drawn from this figure are: (a) reducing the thickness by as much as a factor of 2 has little effect on stress, (b) Eq. 3.59 does not give the lowest stress, (c) the stress is nearly independent of lens shape and constant for $t_e > 0.03$ in. (0.762 mm), and (d) the meniscus lens shape is the most sensitive to temperature increases for thin elastomer layer thicknesses. Although Miller does not specifically discuss it, increasing t_e beyond the "athermal" value should tend to decrease temperature-induced stress because of increased flexibility of the joint. From the stress and strain viewpoint, a large tolerance on thickness of the elastomer layer would apparently be allowed.

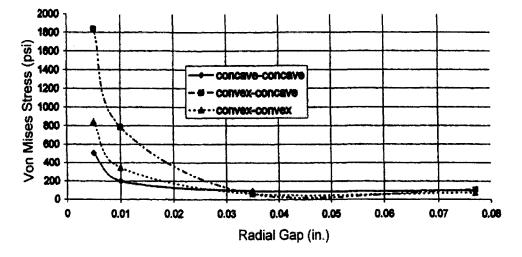


Figure 3.40 Variations of the maximum stress with thickness of the elastomer layer in the lenses of Fig. 3.38 at elevated temperature. (Adapted from Miller.²¹)

The mounted lens analyzed in Example 3.16 has the same design and material parameters as those in Miller's lenses and mountings. The calculations indicate that t_e should be about 0.025 in. (0.635 mm). The reason for the discrepancy between this value and the apparent minimum stress value from Fig. 3.40 [~0.045 in. (~1.143 mm)] has not been identified. The 0.077 in. (1.956 mm) "athermal" thickness value that Miller obtained by applying Eq. (3.59) without correcting the elastomer CTE for its Poisson's ratio is not correct. This fact does not affect the results of his FEA study.

Valente and Richard²² reported an analytical technique for estimating the decentration, δ , of a lens mounted in a ring of elastomer when subjected to radial gravitational loading, i.e., with the lens axis horizontal. Their equation has been extended very slightly here to include more general radial acceleration forces by adding an acceleration factor a_G as follows:

$$\delta = \frac{2a_G W_e}{\left(\pi D_G t_E \left[\left\{\frac{E_e}{\left(1-v_e^2\right)}\right\} + S_e\right]\right)},$$
(3.62)

where W is the lens weight, t_e is the elastomer layer thickness, D_G is the lens diameter, t_E is the lens edge thickness, v_e is the elastomer Poisson's ratio, E_e is the elastomer Young's modulus, and S_e is the elastomer shear modulus given by

$$S_{e} = \frac{E_{e}}{\left[(2)(1+v_{e})\right]}.$$
 (3.63)

The decentrations of modest-sized optics that correspond to static gravity loading are generally quite small, but may grow significantly under shock and vibration loading (see Example 3.16). A resilient material is naturally elastic and will tend to restore the lens to its

unstressed location and orientation when the acceleration force dissipates. Normally, the instrument is not expected to perform to specifications during shock or other brief accelerations. One application in which full performance is expected during lateral acceleration is a target seeker on a guided missile used in combat with an aircraft or another missile. In that case, lateral accelerations can be quite large as the missile is tracking evasive maneuvers of the target. Optical component decentrations then might well degrade performance sufficiently for the seeker to fail to maintain contact with the target.

Example 3.16: Decentration of a lens in a radially athermal elastomeric mounting. (For design and analysis, use File 3.16 of the CD-ROM.)

Consider a BK7 lens with diameter $D_G = 10.000$ in. (254.000 mm), edge thickness t_E 1.000 in. (25.400 mm), and weight W = 7.147 lb (3.242 kg) mounted in a DC3112 elastomeric ring inside a titanium cell. Assume the following properties for the materials: $\alpha_G = 3.94 \times 10^{-6}$ /°F (7.1×10⁻⁶/°C), $\alpha_M = 4.90 \times 10^{-6}$ /°F (8.8×10⁻⁶/°C)

 $\alpha_e = 167 \times 10^{-6/6}$ F (300.6×10⁻⁶/°C), $\upsilon_e = 0.499$ $E_e = 500$ lb/in.² (3.447 MPa)

(a) What should be the elastomer thickness t_e using Eq. (3.44)? (b) How much should the lens decenter under lateral accelerations of 1 and 250 times gravity?

From Eq. (3.63),

$$S_e = \frac{500}{\left[(2)(1+0.499)\right]} = 167 \text{ lb/in.}^2 (1.150 \text{ MPa}).$$

From Eq. (3.61):

$$t_e = \left(\frac{10.000}{2}\right) \frac{(1-0.499)(4.90-3.94)}{[167-4.90-(0.499)(3.94-167)]} = 0.030 \text{ in. } (0.762 \text{ mm}).$$

From Eq. (3.62), for $a_G = 1$:

$$\delta_{1} = \frac{(2)(1.0)(7.147)(0.030)}{\left(\left[\pi\right]\left[10.0\right]\left[1.0\right]\left[\frac{500}{\left(1-0.499^{2}\right)}\right]+167\right)} = 1.6 \times 10^{-5} \text{ in. } (4.1 \times 10^{4} \text{ mm}).$$

From Eq. (3.62), for $a_G = 250$:

 $\delta_{250} = (250)(1.6 \times 10^{-5}) = 0.004$ in. (0.102 mm).

The self-weight deflection seems negligible, but the higher acceleration deflection probably would be significant if full performance is expected under that acceleration.

Elastomer rings between lenses and their mounts are frequently used as seals. See, for example, Fig. 2.20(c). If the elastomer ring is completely encapsulated, as it would be if it were injected to completely fill a closed annular space between a lens rim, mount ID, shoulder, and a retainer, the elastomer will try to expand at elevated temperatures to a volume greater than the space available. The lens may then become radially stressed since most elastomers are virtually incompressible. This problem can be minimized by keeping the amount of elastomer used in any such design as small as possible while still producing the desired seal.

Many successful designs have one side of the elastomer seal exposed so it can expand outward when heated without causing excessive stress in the optic. Examples are shown in Figs. 3.41, 5.1, 5.9, and 5.22(c).

Elastomeric constraint is a good technique for mounting nonsymmetrical optics such as lenses and windows with rectangular apertures since threaded retainers and continuous-ring clamps do not adapt well to noncircularly symmetric applications. Elastomeric constraint also may be appropriate for optical components lacking rotational symmetry of their optical surfaces. See, for example, the plano concave lens shown in Fig. 3.41. One edge of the lens involving a portion of the aperture that is not needed to transmit useful rays to the image has been cut away. Removal of this unneeded material reduces weight and provides clearance needed for other system components. The plano surface should be in contact with a registration shoulder of the mounting plate and centered mechanically. Elastomer is then inserted into the annular gap between the lens rim and the mount ID and cured.

Equation (3.59) [used with Eq. (3.60)] or (3.61) could be used to estimate the elastomer layer thicknesses required in this design to make the configuration athermal in the height or width direction. The appropriate overall linear dimension of the lens (height or width) is substituted for D_G . As pointed out earlier, this thickness need not be tightly toleranced.

The surfaces of a lens mounted in this manner can be curved or aspherical (including wild "potato-chip" types) if specific points on those surfaces can be identified as registration points for alignment purposes and convex mechanical pads (such as segments of spherical surfaces) are provided to contact those points. Without such registration features, the lens must be aligned optically and held by shims or other means until the elastomer has cured.

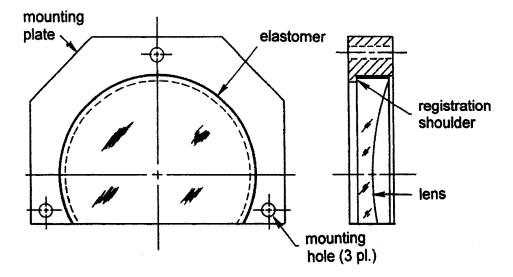


Figure 3.41 A typical technique for mounting a lens with noncircular aperture by potting it into a mount with an elastomeric layer.

3.10 Flexure Mountings for Lenses

In order to achieve optimum image quality, very high performance lenses must be assembled to extremely tight axial (despace), tilt, and decentration tolerances relative to other lenses in an optical assembly or to some other reference within the assembly, such as one or more mechanical surface(s). Alignment must then be fully retained under operational levels of shock, vibration, atmospheric pressure, and temperature variations. Furthermore, misalignments occurring during exposure to survival levels of these environments must be reversible without hysteresis. Mounting techniques that involve mechanical clamping of a lens component or elastomeric encapsulation do not always prevent the relative motion of a lens with respect to the mount to the required degree. It may then be advantageous to attach the lens to symmetrically disposed flexures in the manner depicted schematically in Fig. 3.42. Then, differential expansion of materials caused by uniform temperature changes will not affect tilt or centration. Although they may be similar in appearance, a flexure is not the same as a spring. A flexure is an elastic element that allows small controlled relative motions of components, while a spring provides a controlled force through elastic deformation. Vukobratovich and Richard²³ discuss the use of flexures extensively.

Several concepts for flexure mountings for optics are described next. The first concept is shown in Fig. 3.43. It schematically illustrates a design by Ahmad and Huse²⁴ in which the rim of the lens is bonded with an adhesive (such as epoxy) to the ends of three thin blades that are parts of flexure modules. Those blades are compliant radially, but stiff in all other directions. When the dimensions of the mount (shown here as a simple cell) and lens change with temperature, any mismatch in CTEs causes the flexures to bend slightly. Since this action is symmetrical with respect to the axis, the lens stays centered.

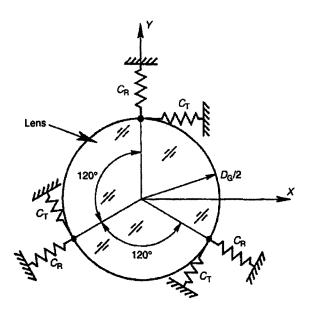


Figure 3.42 Schematic for a "three-point" flexure support for a lens. (Courtesy of D. Vukobratovich, Tucson, AZ.¹¹)

The detailed view in Fig. 3.43 shows a flexure module manufactured separately and attached to the mount with screws so the bonded subassembly comprising the lens and the three flexures can be removed and replaced without damage. Since they are separate, the flexure modules can be made from a material appropriate for the application, such as titanium or beryllium copper, while the mount can be made of a different material, such as stainless steel. The surface on the flexure that is to be adjacent to the lens rim can be shaped as a concave cylinder to approximate the curvature of that rim. Alternatively, localized flats can be ground onto the lens rim to match the shape of a flat. In either case, the thickness of the adhesive layer will then be uniform over the joint. This is desirable for maximum strength of the bond. Not shown in Fig. 3.43, but mentioned in the original authors' description, are dabs of epoxy applied around the edges of the flexure modules after they are screwed in place to secure the modules to the cell in the plane of the figure. In other versions of this concept, mechanical pins are used to lock the modules in place.

A slightly different configuration for the flexure module in which the flexible portion is attached at both ends and the lens is bonded to a pad in the middle of the blade was also suggested by Ahmad and Huse.²⁴ The function of this design is similar to that of the single-ended design.

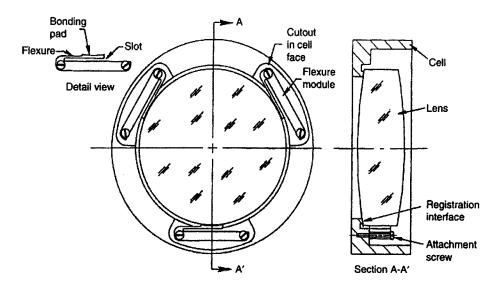


Figure 3.43 A mounting concept in which the rim of a lens is bonded to three removable flexure modules. (Adapted from Ahmad and Huse.²⁴)

Another configuration for a flexure mounting, described by Bacich,²⁵ is shown in Fig. 3.44. Here the flexure blades are formed integrally with the cell, so they cannot be removed or realigned after assembly. The cell material must be chosen in part so the integral flexures function reliably throughout the many temperature cycles inherent in the desired long useful life of the instrument. As in the design shown in Fig. 3.43, these flexures must be accurately machined with a specific and uniform thickness. Precisely curved slots are easily created by the electric discharge machining (EDM) process. In this process a wire of an appropriate diameter is passed through a hole and then raised to an electric potential significantly higher

than the cell so an arc is formed that eats away at the metal while the wire is moved in the desired path; this motion usually is under computer control.

Two versions of the basic mount design are depicted in Fig. 3.44, where in (a) and (b) the rim of the lens is bonded to the flexures in the same general manner as in the design of Fig. 3.43. Figures 3.44(c) and 3.44(d) show the bottom surface of the lens bonded to shelves built into the flexure blades.

Yet another flexure mounting is shown in Fig. 3.45. This design also uses integral flexure blades. They are formed in a box-shaped cell by locally machining the corners from the top and bottom rims of the box in the three regions where the flexure is to be located. The blades then remain attached at both ends. As in the flexure mountings of Ahmad and Huse²⁴ and Bacich,²⁵ their function is to allow dimensional changes between the lens and cell with the temperature without disturbing the centration of the lens.

The description of this mounting configuration by Steel et al.²⁶ includes a discussion of the manner in which the angular subtense φ of the bonded region was determined. Using finite element analysis, 30-deg and 45-deg angles were compared. The larger angle was selected because it resulted in a smaller degree of lens-surface distortion at the extreme operational temperature, it had a higher natural frequency, and it produced less stress in the cell and adhesive (RTV) joint. These attributes of the design were all within the tolerances allowed for the application.

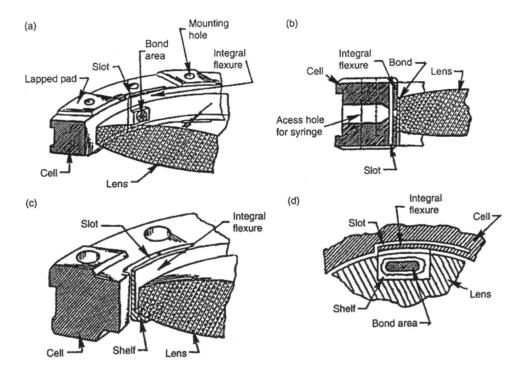


Figure 3.44 Flexure mountings with the lens bonded to integral flexures. In (a) and (b) the bonds are on the lens rim while, in (c) and (d), the bonds are on local areas in the lens' face. (Adapted from Bacich.²⁵)

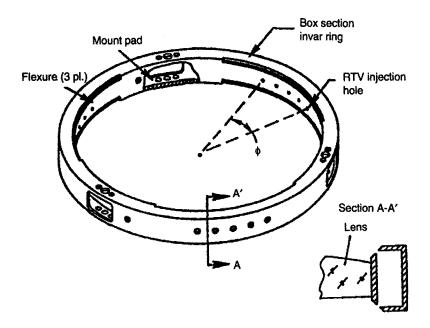


Figure 3.45 A flexure-type lens mounting in which the lens is bonded to three flexure blades machined into a box-section cell. (Adapted from Steel et al.²⁶)

Bruning et al.²⁷ advanced the technology just described in several ways. One technique is illustrated schematically in Fig. 3.46(a). Here, a ring with solid rectangular cross section has three elongated curved slots cut through the ring to create an inner ring that is attached to the outer ring only by way of three narrow flexures. These flexures isolate the inner ring from the outer ring so minute distortions of the latter that can occur when it is mechanically attached to an external structure by screws passing through the indicated recessed mounting holes will not be transferred to the inner ring. A lens (not shown) with an outer diameter substantially the same as the inner diameter of the slots that is mounted on top of the inner ring. View (b) of the same figure shows details of one flexure from view (a). It has two blind holes bored into the flexure blade from top and bottom faces of the rings to weaken that flexure in the direction perpendicular to the figure. The holes also allow the flexure to twist slightly. This feature further isolates the inner ring and lens from mounting disturbances.

Alternative flexure arrangements for the ring mount of Bruning et al.²⁷ are illustrated in Figs. 3.47(a) and 3.47(b). In view (a), the ring is slotted along an arc to provide radial flexibility for a seat protruding inward from the inside surface of the ring at the midpoint of the slot. Three such slot/flexure/seat features at 120-deg intervals provide attachment points for a lens that would be secured with adhesive on the tops of the three pads. In view (b), a more complex slotting arrangement provides dual flexures as indicated. The long slots conform to the general design shown in Fig. 3.46 in that they isolate an inner ring from the outer ring.

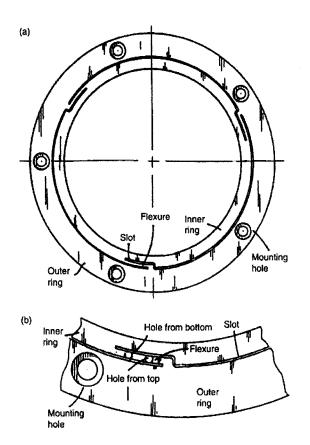


Figure 3.46 A flexure mounting in which a lens (not shown) is bonded to an inner ring that is integral with, but mechanically isolated from, an outer ring by flexures. (Adapted from Bruning et al.²⁷)

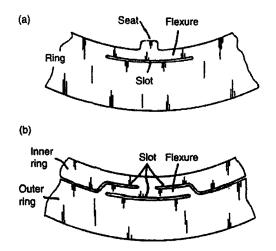


Figure 3.47 Two design concepts for flexures that can be used to support a lens in its mount. (Adapted from Bruning et al.²⁷)

Figure 3.48(a) is a plan view of a lens/mount subassembly incorporating the flexure configuration shown in Fig. 3.47(a). The lens is attached to three seats as shown in the side view in Fig. 3.48(b). Distortions of the ring lens mount by forces exerted by the five screws that attach the ring to an external structure are prevented from reaching the lens by the multiple flexures provided.

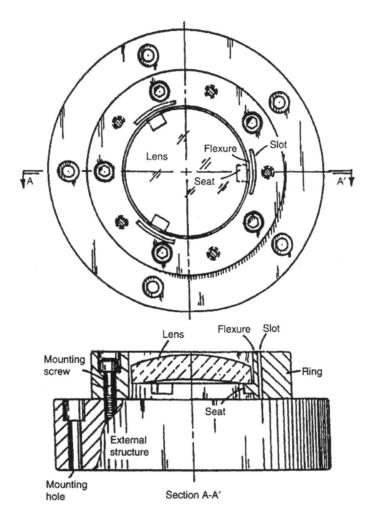


Figure 3.48 Top and side views of a flexure mounting using flexures of the type shown in Fig. 3.47(a). (Adapted from Bruning et al.²⁷)

3.11 Mounting Plastic Components

Injection and compression molding at high temperatures and high pressures are the most common techniques for manufacturing lenses from plastics.^{28,29} Polymethylmethacrylate, polystyrene, polycarbonate, styrene acrylonitrile, polyetherimide, polycyclohexylmethacrylate, or one of the newer materials such as cyclic olefin copolymer are the most commonly used materials.³⁰ Windows and filters also can be fabricated from plastic. Plastic prisms and mirrors are used only in low-performance applications. Diamond-turning

MOUNTING INDIVIDUAL LENSES

techniques are frequently used for short production runs and prototype quantities of plastic optics if made of compatible materials. The resulting products may be configured in the same manner as conventionally manufactured glass optics with cylindrical rims, bevels, and convex or concave surfaces. Figure 3.49 shows a set of molded plastic lenses for a 28-mm (1.10 in.) focal length, *f*/2.8 photographic objective.³¹ Each of these elements is configured for conventional mounting. Plastic lenses can have all these features plus Fresnel or diffractive surfaces, aspherics, or integral mounting features such as flanges, tabs, holes, locating pegs, brackets, or spacers. Plastic lenses can also be configured to nest together, eliminating separate spacers and facilitating centration. Figure 3.50 shows a plastic lens molded into a square configuration. It has flat rectangular tabs extending beyond the optical aperture on two edges to facilitate insertion into a mechanical assembly. The tabs have unequal thicknesses to discourage incorrect installation. All these integral features greatly simplify assembly and reduce the number and complexity of mechanical interfacing parts.

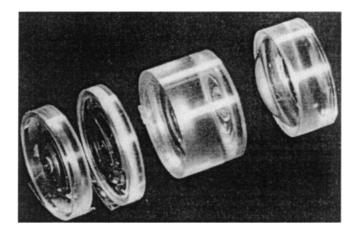


Figure 3.49 Photograph of a four-element plastic objective with elements configured for conventional mounting. (From Lytle.³¹)

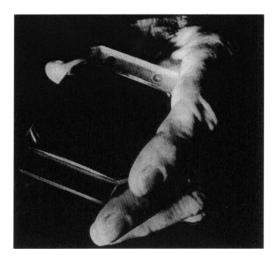


Figure 3.50 A plastic lens with square configuration and integral mounting features. (From Lytle.³¹)

Plastic optics can be assembled by clamping, adhesive or solvent bonding, heat staking, or welding with ultrahigh-frequency sound.²⁸ All these means of assembly are applied well outside the optical apertures. Plastics are not very well suited for making cemented doublets, since optical cements may soften the plastic and destroy the surfaces. The materials used to make the lenses have large CTEs, so temperature changes may cause excessive internal or surface stress because of differential expansion or contraction. Plastic materials tend to absorb moisture from the atmosphere; dimensions and refractive index then change with time. Antireflective coatings can be applied to some plastic materials to enhance transmission.

Figure 3.51 shows section views through several molded plastic lenses that, according to Altman and Lytle,²⁹ demonstrate favorable and unfavorable characteristics. The lens in Fig. 3.51(a) is a conventional glass meniscus lens while Fig. 3.51(b) shows the equivalent plastic lens. The latter has slightly shorter radii because of the lower refractive index of the plastic relative to that of the glass and is thicker to facilitate the flow of the raw material into the mold before heating and compression. The thin lens of Fig. 3.51(d) has a poor design for plastic molding since the material will not flow easily into the central region. The one shown in Fig. 3.51(c) would be better in this regard. Figures 3.51(e) and 3.51(f) show a lens molded with a protruding rim that forms a recess into which a companion lens can fit to make a nested air-spaced doublet. Close dimensioning of the molds to compensate for any shrinkage during curing ensures centration and spacing. Figures 3.51(g) and 3.51(h) show a cemented doublet (lens materials and cement not specified) and a well-proportioned meniscus lens.

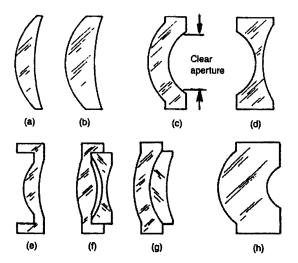


Figure 3.51 Different shaped molded plastic lens elements that have distinct advantages and disadvantages as discussed in the text. (Adapted from Altman and Lytle.²⁹)

Nonspherical surfaces can be molded on plastic optics nearly as easily as spherical ones. The trick is in making the molds to the negative contour of the desired surface. Plummer³² has described the high-order nonrotational polynomial aspherics built into the refracting corrector plate, concave mirror, and eye lens of the viewfinder for the unique single-lens reflex Polaroid SX-70 camera. Each element was made by injection molding.

The molds were made by hand correcting steel surfaces shaped initially by single-point diamond turning. A customized measuring machine capable of at least 0.1-µm (3.8×10^{-6} in.) precision while sliding a 0.8-mm (0.030-in.) diameter sapphire ball lightly over the surface to be measured provided error maps that defined where correction was required and how large the correction should be. This same technique can be applied to make molds for nearly any aspheric surface to be used in other applications.

Baumer et al.³³ described the design of a CMOS^a imaging sensor using an injection molded plastic singlet. Intended for large-scale production, this sensor, lens, and mount are shown in a conceptual section view in Fig. 3.52. The lens is biconvex and has a circular recess in its entrance face for a filter. The lens fits into a barrel that, in turn, fits into a mount holding the focal-plane assembly. No centering adjustments are provided. The required alignment results from tolerancing and careful molding in precision molds. Performance of the f/2.2 system is compatible with a 352×288 -pixel format with 5.6×5.6 -µm pixels.

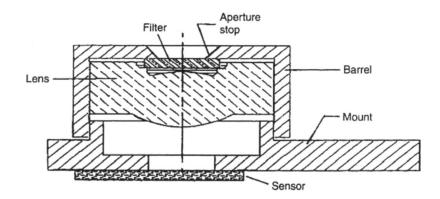


Figure 3.52 A small image-forming optical system featuring injection-molded plastic optical and mechanical parts. (From Baumer et al.³³)

3.12 References

- 1. Yoder, P.R., Jr., *Opto-Mechanical Systems Design*, 3rd. ed., CRC Press, Boca Raton, 2005.
- 2. Smith, W.J., "Optics in practice," Chapter 15 in *Modern Optical Engineering*, 3rd ed., McGraw-Hill, New York, 2000.
- 3. Horne, D.F., *Optical Production Technology*, Adam Hilger Ltd., Bristol, England, 1972.
- 4. Jacobs, D.H., Fundamentals of Optical Engineering, McGraw-Hill, New York, 1943.
- 5. Plummer, W.T., "Precision: how to achieve a little more of it, even after assembly," *Proceedings of the Firs t World Automation Congress (WAC '94)*, Maui, TST Press, Albuquerque, 1994:193.

^a CMOS is a low power consumption <u>complementary metal oxide semiconductor used in camera</u> sensor chips.

- Baldo, A.F., "Machine rlements," Chapter 8.2 in *Marks' St andard Ha ndbook f or Mechanical Engineers*, E.A. Avallone and T. Baumeister III, eds., McGraw-Hill, New York, 1987:8.
- 7. Yoder, P.R., Jr., "Lens mounting techniques," Proceedings of SPIE 389, 1983:2.
- 8. Bayar, M., "Lens barrel optomechanical design principles," Opt. Eng. 20, 1981:181.
- 9. Kowalski, B.J., "A user's guide to designing and mounting lenses and mirrors," *Digest of Papers, OS A Workshop on Opt ical Fabri cation and Testi ng, N orth Falmouth,* 1980:98.
- Vukobratovich, D., "Optomechanical systems design," Chapter 3 in *The Infrared & Electro-Optical Syst ems Handbook*, 4, ERIM, Ann Arbor, and SPIE, Bellingham, 1993.
- 11. See Chapter 8, "The Design of Screws, Fasteners, and Connections," in Shigley, J.E. and Mischke, C.R., *Mechanical Engineering Design*, 5th ed., McGraw-Hill, New York 1989.
- Roark, R.J., Formulas for Stress and Strain, 3rd ed., McGraw-Hill, New York, 1954. See also Young, W.C., Roark's Formulas for Stress & Strain, 6th ed., McGraw-Hill, New York, 1989.
- 13. Hopkins, R.E., "Lens Mounting and Centering" Chapter 2 in Applied Opt ics and Optical Engineering, VIII, Academic Press, New York, 1980.
- 14. Delgado, R.F., and Hallinan, M., "Mounting of Optical Elements," *Opt*. Eng. ,14, 1975:S-11,. (Reprinted in SPIE Milestone Series, 770, 1988:173.)
- Yoder, P.R., Jr., "Location of mechanical features in lens mounts," *Proceedings of* SPIE 2263, 1994:386.
- Hopkins, R. E. "Lens mounting and centering," Chapter 2 in Applied Optics and Optical Engineering, VIII, R. R. Shannon and J. C. Wyant, eds., Academic Press, New York, 1980.
- 17. Genberg, V., "Structural analysis of optics," Chapter 8 in Handbook of Optomechanical Engineering," CRC Press, Boca Raton, 1997.
- Michels, G.J., Genberg, V.L., and Doyle, K.B., "Finite Element Modeling Of Nearly Incompressible Bonds" *Proceedings of SPIE* 4771, 2002: 287.
- 19. Doyle, K.B., Michels, G.J., and Genberg, V.I., "Athermal design of nearly incompressible bonds" *Proceedings of SPIE* **4771**, 2002: 298.
- 20. Herbert, J.J., "Techniques for deriving optimal bondlines for athermal bonded mounts," *Proceedings of SPIE* **6288**, 62880J-1, 2006.
- 21. Miller, K.A., "Nonathermal potting of lenses," Proceedings of SPIE 3786, 1999: 506.
- 22. Valente, T.M. and Richard, R.M., "Interference fit equations for lens cell design using elastomeric mountings," *Opt. Eng.*, 33, 1994: 1223.
- 23. Vukobratovich, D. and Richard, R.M., "Flexure mounts for high-resolution optical elements" *Proceedings of SPIE* **959**, 1988: 18.
- 24. Ahmad, A. and Huse, R.L. (1990). "Mounting for high resolution projection lenses" U.S. Patent No. 4,929,054, issued May 29, 1990.
- 25. Bacich, J.J. (1988). "Precision lens mounting," U.S. P atent No. 4,733, 945, issued March 29, 1988.
- Steele, J.M., Vallimont, J.F., Rice, B.S., and Gonska, G.J., "A compliant optical mount design," *Proceedings of SPIE* 1690, 1992: 387.
- Bruning, J.H., DeWitt, F.A., and Hanford, K.E. (1995). "Decoupled mount for optical Element and stacked annuli assembly," U.S. Patent No. 5,428,482, issued June 27, 1995.

- 28. Welham, W. (1979). "Plastic optical components," Chapter 3 in *Applied Optics and Optical Eng ineering*, VII, R. R. Shannon and J. C. Wyatt, eds., Academic Press, New York.
- 29. Altman, R.M. and Lytle, J.D. (1980). "Optical design techniques for polymer optics," *Proceedings of SPIE* 237:380.
- 30. Lytle, J.D., "Polymeric optics," Chapter 34 in OSA Handbook of Optics, 2nd ed., Vol. II, M. Bass, E. Van Stryland, D. R. Williams, and W.L. Wolfe, eds., McGraw-Hill, Inc., New York.
- 31. Lytle, J.D., "Specifying glass and plastic optics: what's the difference?," *Proceedings of SPIE* 181:93.
- 32. Plummer, W. T., "Unusual optics of the Polaroid SX-70 Land Camera," *Appl. Opt.*, 21, 1982:196.
- 33. Bäumer, S., Shulepova, L., Willemse, J., and Renkema, K., "Integral optical system design of injection molded optics," *Proceedings of SPIE* **5173**, 2003:38.

CHAPTER 4 Multiple-Component Lens Assemblies

The assembly of two or more optical components such as lenses, windows, mirrors, or filters into a common mechanical surround generally involves multiple applications of the basic mounting techniques discussed in Chapter 3. Here, we discuss several technical aspects unique to multiple-component optical assemblies. Topics include the design and fabrication of spacers that are commonly used to separate adjacent components in a mount; assembly techniques (drop in, lathe, poker chip, and modular) that provide varying degrees of control over intercomponent alignment; techniques for sealing and purging completed assemblies; and mechanisms for moving one or more lenses relative to other optics for focus adjustment, focal length variation, magnification change, etc. Numerous examples of hardware designs illustrate the various types of construction. Methods and instrumentation for precision alignment of multiple refracting and reflecting components are discussed in Chapter 12.

4.1 Spacer Design and Manufacture

The axial separation of multiple lenses generally requires the presence of one or more spacers to establish the proper separations of adjacent optical surfaces and to provide a means for registering the lenses for alignment purposes. Alternatively, shoulders machined integrally into the mount may accomplish these functions. For simplicity, we refer here to either as a spacer. Figure 4.1 shows an example. The parameters indicated are those needed to find the length $L_{j,k}$ of the spacer between the contact points P_j and P_k at heights y_j and y_k on the spherical surfaces of absolute radii $|R_j|$ and $|R_k|$ to produce the separation $t_{j,k}$ between the adjacent vertices. The sagittal depths of the surfaces are S_i and S_j , respectively. These are calculated by the following equations and assigned positive signs if contact occurs to the right of the vertex, and negative signs if contact occurs to the left of the vertex. In the figure, S_j is negative while S_k , $t_{j,k}$, and $L_{j,k}$ are positive.

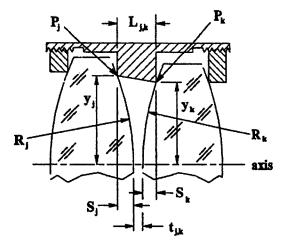


Figure 4.1 Geometry for dimensioning a spacer between two lenses.

$$S_j = |R_j| - (R_j^2 - y_j^2)^{1/2},$$
(4.1)

$$S_k = |R_k| - (R_k^2 - y_k^2)^{1/2}, \qquad (4.2)$$

$$L_{j,k} = t_{J,k} - S_j + S_k, (4.3)$$

Example 4.1: Calculation of a spacer's length. (For design and analysis, use File 4.1 of the CD-ROM).

Assume that two biconvex lenses with radii $R_2 = 30.000$ in. (762.000 mm) and $R_3 = 7.500$ in. (190.500 mm) are to be separated axially by $t_A = 0.0750$ in. (1.905 mm). If the contact heights are $y_2 = 1.500$ in. (38.100 mm) and $y_3 = 1.667$ in. (42.342 mm), what length of spacer is needed?

From Eqs. (4.1) and (4.2): $S_2 = -[30.000 - (30.000^2 - 1.500^2)^{1/2} = -0.0375 \text{ in.} (-0.953 \text{ mm}),$ $S_3 = +[7.500 - (7.500^2 - 1.667^2)^{1/2} = 0.1876 \text{ in.} (4.765 \text{ mm}).$

From Eq. (4.3): $L_{2,3} = 0.0750 + 0.0375 + 0.1876 = 0.3001$ in. (7.622 mm).

As mentioned in Sect. 2.1.3.2, an advantage of surface-contact interfaces is that the lenses can have decentered or tilted rims without affecting the alignment of their optical axes. This is illustrated in Fig. 4.2. Here three lenses with rim errors have been aligned to a common axis using two spacers that contact the spherical surfaces. Centration of the lenses using the spacers and some metrology apparatus that senses alignment errors will minimize the axial distance between the outer surfaces. Hopkins¹ described a technique for assembling and aligning the lenses and spacers in such an assembly as well as one form of error-measuring apparatus to be used to ensure that this is the case. Once aligned, the components are secured to maintain that alignment. Next, the method is summarized.

Figure 4.3 shows the lens assembly of Fig. 4.2 schematically in a mechanical housing with two additional (top and bottom) spacers and the alignment sensing apparatus. The latter is a laser autocollimator based on one described by Brockway and Nord.² The lenses and spacers can each be moved transversely in two orthogonal directions by mechanisms such as orthogonal radial setscrews. The beam from the autocollimator is aligned normal to the upper surface of the slightly wedged glass reference plate forming the base of the housing. The first spacer is centered approximately to the beam and anchored to the plate with clamps or wax. The first lens is then placed on that spacer. Ring patterns will be seen through the evepiece; two result from interference between the reference surface and each spherical surface of the lens, while another comes from interference between the lens surfaces. The lens is moved until the patterns are centered to each other using the translation mechanism. The lens is then clamped or bonded to the spacer with adhesive. The next spacer is added and clamped or bonded to the first lens. The second lens is added and manipulated until the interference rings formed from its surfaces are centered. That lens is then secured in place. The process is repeated for the third spacer and the third lens. The fourth spacer is added and the entire assembly is clamped together axially.

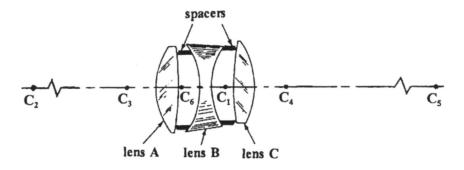


Figure 4.2 A surface-contacted triplet lens that is perfectly centered while neither the lens rims nor spacers are perfect or centered. (Adapted from Hopkins.¹)

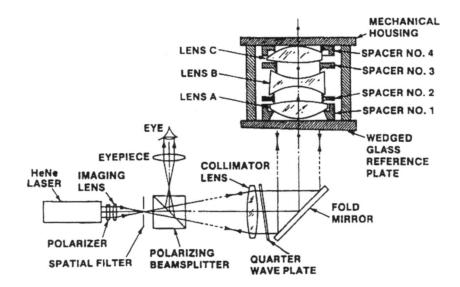


Figure 4.3 A setup for monitoring and adjusting centration of multiple lenses as they are assembled. (Adapted from Hopkins¹ and Brockway and Nord.²)

There are cases where some interference rings will appear too large or too small for accurate centration. In those cases, refocusing of the autocollimator and/or addition of an auxiliary lens to the beam may adjust the apparent sizes sufficiently to facilitate accurate alignment of the surfaces producing the interference.

Since the annular widths of most spacers are small compared with their diameters (see Fig. 4.4 for a typical case), their manufacture can be difficult. If turned on a lathe, they may end up out of round and have nonparallel faces. To minimize these problems, a stress-free manufacturing procedure such as that described by Westort³ should be employed. His technique ensures the correct IDs and ODs, roundness, parallelism of faces, and perpendicularity to the axis of faces for any reasonably proportioned spacer.

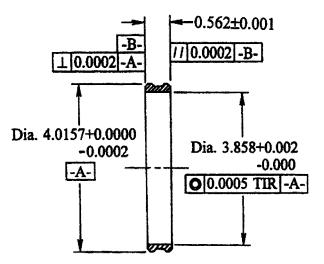


Figure 4.4 A typical lens spacer. Dimensions are in inches. (From Westort.³)

Figure 4.5 illustrates the main steps of this procedure. The material used is a 400-series stainless steel that matches that used in the lens cell, which was chosen to provide a reasonable thermal match to the lenses to be mounted. The spacer is first rough machined from a blank [dashed outline in Fig. 4.5(a)] to near finished dimensions and heat-treated to relieve stresses. The spacer is then potted with a low-temperature melting metal alloy into a fixture [Fig. 4.5(b)] and bored to the final ID. After disassembling by heating gently, multiple spacers are then installed onto a precision arbor and ground to the final OD [Fig. 4.5(c)]. This ensures the concentricity as well as the dimensional correctness of these surfaces. Each spacer is then inserted into a fixture with a precision bore that matches and squares on to the spacer OD [Fig. 4.5(d)]. The top surface is ground flat and its edges burnished. The spacer is then turned over and the top surface is ground flat and to the final spacer thickness. Once the edges of that face are burnished smooth, the spacer is completed.

This procedure results in a spacer with 90-deg sharp-corner interfaces to the lenses. One or both faces could be ground conically with minor modifications of this procedure if the facing operations were done on a precision spindle with the grinding wheel set at the appropriate angle. Toroidal interfaces could also be made on a spindle; the grinding wheel then needs to move in the correct curve. This is quite feasible on a modern computer numerically controlled (CNC) lathe or when using SPDT machining.

Figure 4.6 shows a cell in which the spacer of Fig. 4.4 is intended to fit. The complete assembly is shown in Fig. 4.7. The spacer discussed above is the first on the left. The maximum clearance between the spacer OD and the cell ID is 0.0008 in. (20.3 μ m). Care must be employed during assembly to keep the spacer from binding as it is inserted. The rim of the spacer could be ground spherically or crowned in the manner described for lenses in Section 2.1.3.1 to prevent this potential problem. Each spacer would then have to be ground individually on a spindle instead of in groups as shown in Fig. 4.5(c).

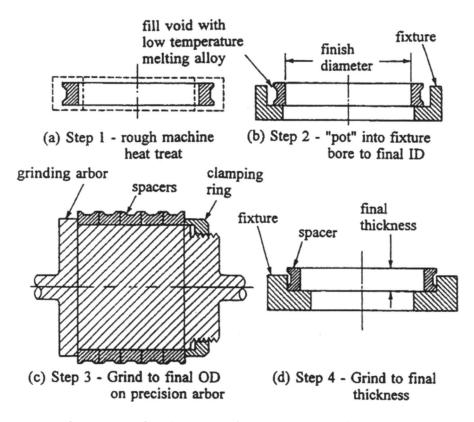


Figure 4.5 Sequence of major steps (described in text) in one method for making precision spacers. (From Westort.³)

The other spacers in the assembly of Fig. 4.7 are worth mentioning. The thickness of the second spacer is shown exaggerated for clarity. Typically, such spacers are die-cut rings of metal, such as stainless steel of a specific thickness as required by the lens design. The minimum thickness probably is in the region of 0.0020 in. (50 μ m). These spacers are flexible enough that they conform to a "best fit" to the adjacent glass surfaces (which usually are quite close in radius) when preloaded by a retainer. Of course, the spacers must be thick enough to ensure that the surfaces do not contact at the axis if the sagittal depth of the convex face exceeds that of the concave face.

The third spacer of Fig. 4.7 serves two purposes. It acts as a metallic slip ring so rotation of the retaining ring during tightening does not drag the lens with it and disturb the rotational alignment around the axis of the last lens. In precision assemblies, lenses often are manually rotated ("clocked" or "phased") during installation so that their residual optical wedges tend to cancel each other and produce the best possible image. This spacer also is long enough to bring the retainer to a conveniently accessible location. Note that two retaining rings are used. The second locks the first to help prevent loosening under vibration. In addition, the threads of both rings should be treated with locking compound.

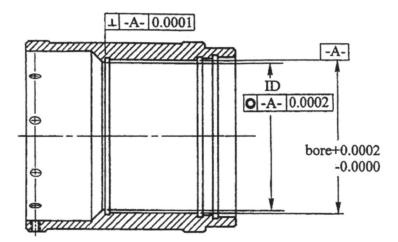


Figure 4.6 Cell in which the spacer shown in Fig. 4.4 is used. Dimensions are in inches. (Adapted from Westort.³)

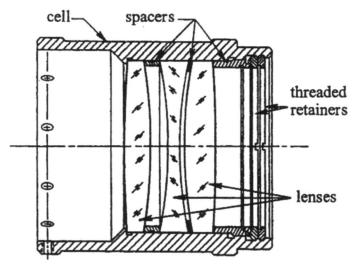


Figure 4.7 High-performance relay lens assembly. (Adapted from Westort.³)

Figure 4.8(a) shows a die-cut plastic spacer shaped with three tabs that can be inserted between two lenses that need a small axial air space. Addis described such spacers made of polyester film as a viable and inexpensive means for separating air-spaced doublets.⁴ The outer ring supports the tabs and lies outside the lens ODs. The tabs protrude between the lens surfaces to the clear apertures of the elements. An advantage pointed out by Addis is that air or nitrogen can easily flow into the space between the lens surfaces when the assembly is purged to remove moisture. A continuous shim would not allow this to happen unless grooves were cut into the spacer and the lens rims.

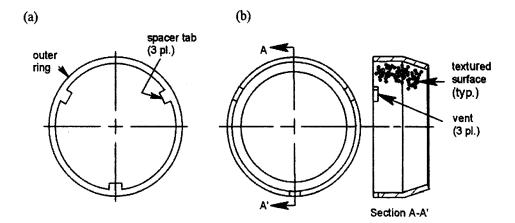


Figure 4.8 (a) A thin plastic spacer with tabs to separate the surfaces. (b) A molded plastic spacer with ventilation grooves. (Adapted from Addis.⁴)

Figure 4.8(b) shows a typical molded plastic spacer with grooves of this type. Contact with the lens occurs most of the way around the assembly to distribute preload. Molded spacers are advantageous from the cost viewpoint in quantity production; they can be made accurately enough for applications in which extreme accuracy is not required. If the spacers are made of black plastic and textured internally, stray light can be attenuated.⁴

Before we leave the subject of lens spacers, we should consider the appropriateness of edge-contacting lens surfaces without any spacer. Figure 4.9 shows a typical configuration. Here the adjacent surfaces are strongly curved and the convex side of the left lens element touches the bevel of the concave surface. Usually when this is encountered, the size of the axial air space is a strong driver of some aberration, so it must be controlled accurately. Either of two techniques can be used to establish the contribution of each element to this air space: the sagittal depth of each surface from the plane of the contact can be measured directly using, for example, a ring spherometer with an appropriately sized ring, or the diameter of the contact (i.e., that of the bevel on the concave surface) can be measured and the sagittal depths of each surface corresponding to that diameter calculated from Eqs. (4.1) and (4.2) using the known radii of curvature of each surface. In either case, the difference between these dimensions is the air space.

The real issue here is not design, but manufacture of the edge-contacted lens assembly. Price⁵ rather humorously stated the usual reaction of optical shop personnel to such designs as follows: "You want to put that sharp bevel on a flint glass, right? Where do you think flint glass got its name? You're right, it chips easily, and especially at the edge of a bevel. Further, if you have to move one lens on the other to align the two, you can scratch the coating on the second lens. What do you get? High scrap losses." These disadvantages must be balanced against the advantages: no spacer is needed and one element *may* self-center.

The logical approach then is to evaluate each design on its own merits, considering the relative radii of the adjacent surfaces, the ability to set the contact at a reasonable diameter, the brittleness and hardness of the glasses, the ratio of radius to diameter at the contact diameter, the sensitivity of the design to air space variations and element tilts, the ability of the design to achieve self-centering, and the sizes of the lenses and their resulting weights.⁵

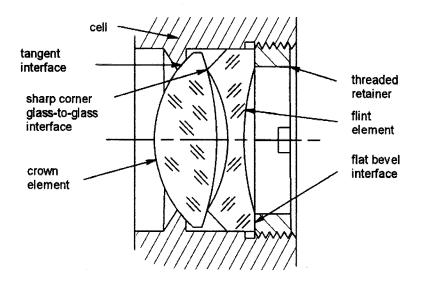


Figure 4.9 Schematic mounting for an edge-contacted doublet in which the crown element is centered between a shoulder and a sharp corner on the flint element. (Adapted from Price.⁵)

4.2 Drop-in Assembly

Designs in which all lenses and the interfacing surfaces of the mount are fabricated to specified dimensions within specified tolerances, assembled without further machining, and without adjustment are called "drop-in" assemblies. Low cost, ease of assembly, and simple maintenance are prime criteria for these designs. Typically, relative apertures are f/4.5 or slower, and performance requirements are not high.

An example is shown in Fig. 4.10. This is a fixed-focus eyepiece for a military telescope.⁶ Both lenses (identical doublets oriented crown to crown) and a spacer fit into the ID of the cell with 0.003-in. (0.075-mm) radial clearance. A threaded retainer holds these components in place. Sharp-corner mechanical interfaces are used throughout. The accuracy of centration depends primarily upon the accuracy of the lens edging and the ability of the axial preload to "squeeze out" differences in edge thickness (i.e., wedge) before the rims of the lenses touch the cell ID. The axial air space between the lenses depends upon the spacer dimensions, which typically are held to design values within 0.010 in. (0.25 mm).

Lens assemblies for many military and commercial applications with modest performance requirements traditionally follow the drop-in design concept. Most involve high-volume production and some are intended for assembly by "pick-and-place" robots. Cost is of prime importance. Careful tolerancing, guided by knowledge of standard optical and mechanical shop practices is essential, since parts are usually selected from stock at random, and few, if any, adjustments at the time of assembly are feasible. Usually, focus is the only adjustment. It is expected that a small percentage of the end items will not meet all performance requirements. Those that fail are discarded, since that is generally more costeffective than troubleshooting and fixing the problem affecting any individual "out-oftolerance" component.

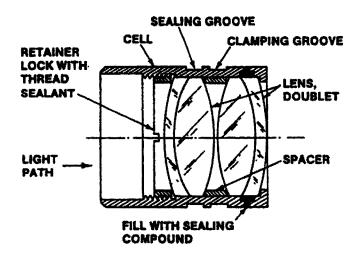


Figure 4.10 Example of a drop-in assembly. (From Yoder.⁶)

4.3 Lathe Assembly

As described in Section 3.13.2 for a single lens element, a "lathe-assembled" lens is one in which the seat in the mount is custom machined on a lathe or diamond-turning machine to fit closely to the measured OD of a specific lens. The axial position of each seat is also usually determined during this machining operation. Spherical or crowned rims may be appropriate if the edge thicknesses are large. This design feature, described in Section 2.1.3.1, is intended to facilitate the assembly of lenses into closely fitting mount IDs without jamming from inadvertent lens tilts.

Figure 4.11 illustrates the measurement and fitting sequence for a two-lens assembly. View (a) shows an air-spaced doublet in a cell. Measurements to be made on the "as manufactured" lenses are indicated by the circled numbers 1 through 5 in Fig. 4.11(b). Actual values can be written in the boxes below the numbers on a copy of this figure to create a record for a given assembly. Lens surface radii from test plate or interferometric measurement also must be made part of the recorded data. The mechanical surfaces designated by the letters A through E are machined to suit this specific set of lens measurements and to position the lenses axially within specified tolerances. Machining of surface D, which provides a tangential interface for lens 1, is an iterative process, with trial insertions of the lens and measurement of its vertex location relative to flange surface B to ensure achievement of the 57.150-mm axial dimension within the specified ± 10 -µm tolerance. The spacer thickness is also machined iteratively, with trial assembly and measurement of overall axial thickness to ensure meeting the design tolerance for this dimension.

Figure 4.12 shows a 24-in. (61-cm) focal length, f/3.5 aerial-camera objective lens designed for lathe assembly.⁷ The titanium barrel is made in two parts so a shutter and iris can be inserted between lenses 5 and 6. The machining of the lens seats to fit the measured lens ODs and to provide proper air spaces begins with the smaller diameter components and

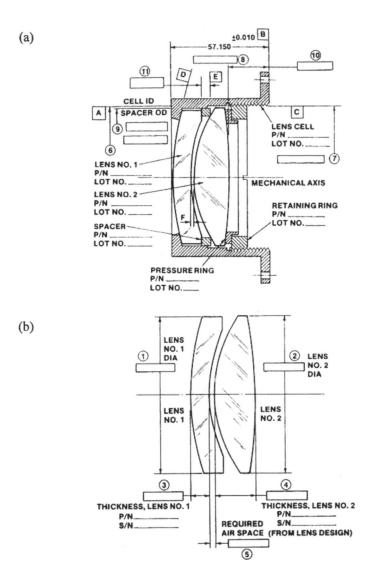


Figure 4.11 An air-spaced doublet assembly made by the lathe-assembly process. (a) Complete assembly, (b) measurements made on lenses. (From Yoder. 6)

progresses toward the larger ones. Each lens is held with its own retaining ring, so no spacers are required. The lenses are fitted into the front and back barrel components in single lathe setups to maximize centration. These optomechanical subassemblies are mechanically piloted together (the pilot diameters are machined in the same setups as the lens seats) so their mechanical and optical axes coincide. An O-ring is used to seal the barrel-to-barrel interface with metal-to-metal contact between the flanges. Tangent surfaces are used in the convex surface interfaces. Flat bevels are ground on the concave surfaces with accurate perpendicularity to the lens' optical axes to ensure proper centration. Because of space constraints between lenses 2 and 3, and 3 and 4, step bevels are ground into the lens rims to provide spaces for the retainers. Injected elastomer rings (not shown) seal

lenses 1, 5, 6, and 7 to the barrel. All internal air spaces are interconnected with ducts (not shown) to facilitate purging with dry nitrogen to minimize condensation of internal moisture at low temperatures.

This technique is also often used in the assembly of lens systems for high-performance optical systems such as those that might be used in military or space systems. Many such systems involve several lenses and some have mirrors. The application for the air spaced doublet shown in Fig. 4.11 in a catadioptric system is described in Section 15.10. Lathe assembly usually starts with the smallest diameter element and progresses to the largest, as in the assembly of Fig. 4.12. Lathe assembly may also be used to advantage (perhaps without the elaborate data retention if not required by the customer) when the assembly is to be subjected to severe vibration and/or shock conditions because the customizing of lens fits limits alignment changes under such exposures.

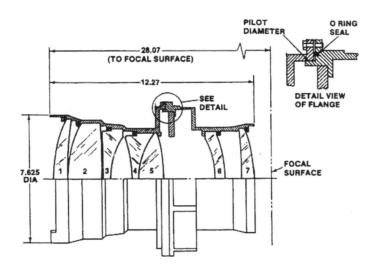


Figure 4.12 Sectional view of a lathe-assembled aerial camera lens assembly. Dimensions are given in inches. (From Bayar.⁷)

4.4 Elastomeric Mountings

The elastomeric mounting technique described in Section 3.9 is quite adaptable for use in multiple-lens assemblies. Making the individual element mountings athermal radially by providing the appropriate radial thickness of elastomer between the mount ID and lens OD allows different glasses to be used in a mount made of a single material without introducing drastic radial forces on those lenses when the temperature changes to extremes. In this section we give an example of such a design.

Figure 4.13 shows an aerial camera objective with a focal length of 66 in. (1.67 m) and a fixed relative aperture of f/8, which was described by Bayar in Ref. 7. Four singlets and one doublet are mounted separately within annular rings of RTV elastomer. For added protection against lens motion within the barrel, each lens is constrained axially against a shoulder by a threaded retainer. The lens barrel is made in two parts that pilot together at a flange. These barrel halves are joined with screws (see detail view) after the lenses are installed.

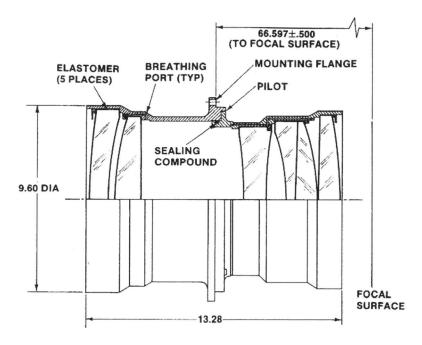


Figure 4.13 An aerial camera lens with elastomerically mounted lenses. Dimensions are given in inches. (From Bayar.⁷)

The front half of the objective is assembled with the barrel mounted with its axis vertical on a rotary table and adjusted until the lens seats and pilot diameter all have minimum mechanical runout as the table and barrel are rotated slowly. The innermost lens is installed and centered to the rotational axis within allowed tolerances using mechanical and/or optical error-detecting means, such as those described in Chapter 12. The retainer is tightened against that lens to preload it against the opposite shoulder. The annular gap between the lens rim and the barrel ID is then filled with RTV compound through several radially directed holes (not shown) by means of a hypodermic syringe. The outermost lens is then inserted, aligned to the rotary axis, clamped in place, and "potted" in place. The same procedure is used to assemble and align the rear half of the objective. Once the sealant has cured, the barrel halves are joined to each other and sealed with RTV sealing compound.

The thickness of each annular layer is customized to make each lens mounting athermal radially using Eq. (3.59). In his paper, Bayar did not indicate that he applied the correction for Poisson's ratio expressed by Eq. (3.60). Since he did not specify the materials used in this assembly, we cannot now determine how close the design came to minimizing the radial stress generated within the lenses at low temperatures. Inasmuch as it was a successful optomechanical design, we can speculate that the compensation for differential expansion was adequate for the application. We observe that, in such a design, the elastomer is completely surrounded by metal and glass, so the RTV has no space into which to expand when the temperature increases. The high CTE of the RTV relative to those of the metal and glass would then be expected to create radial force upon the lens rim. The lens assembly shown in Fig. 4.14 is a five-element, 81.102 in. (2.06 m) focal length, f/10 objective lens developed at the Optical Sciences Center of the University of Arizona, for astrographic use by the U.S. Naval Observatory.⁸ It features individual lens elements and a filter bonded into subcells with annular layers of Dow Corning Type 93-500 elastomer on the order of 0.2-in. (5-mm) thick. The cells are installed into a barrel with interference fits and further constrained axially with threaded retainers. Two spacers are also used. All metal parts are 6Al-4V titanium to provide a reasonable CTE match to the glasses. The overall weight of the assembly is about 44.6 kg (98.3 lb), 21.9 kg (48.4 lb) of which is in the glass. The lens barrel and its component parts are shown in Fig. 4.15.

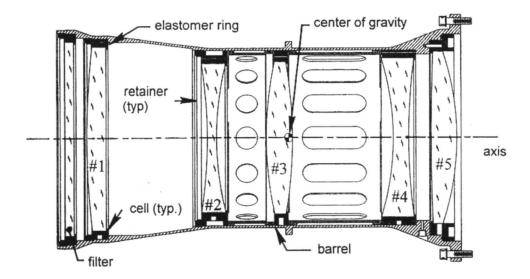


Figure 4.14 Optomechanical configuration of an 81.102 in. (2.06 m) focal length, f/10 astrographic objective. (From Vukobratovich.⁸)

Each cell is made with a shoulder at one side to provide a reference for squaring on the lens, as indicated schematically in Fig. 4.15(a). The construction technique allows the lenses to be accurately centered in their subcells. The elastomer is then added and cured. The optomechanical subassemblies are pressed into place in the barrel so as to rest against axial locating shoulders.

Valente and Richard⁹ described how to determine the radial stress introduced into the lenses by the radial compression created during assembly. Using a finite element analysis technique, those authors verified that essentially the same result would be obtained by their equations and by FEA. Figure 4.15(b) illustrates the model used. Once the radial stress in the glass is known, the stress birefringence can be estimated from the stress optical coefficient of the refractive material and the optical path length in the glass.

As discussed in Section 3.9, Eq. (3.62) can be used to calculate the radial deflection of a lens element used in this assembly that results from elastic deformation of the elastomer under normal gravity and higher accelerations. Using that equation and assuming 0.200-in. (5.080-cm) thick elastomer layers, it can be shown⁹ that the worst-case self-weight decentration of any of the five lenses in this assembly will not exceed 0.0002 in. (5.1 μ m),

which is considerably smaller than the corresponding decentration tolerance of 0.001 in. (25.4 μ m). Acceleration-induced decentration due to compliance of the elastomer layer probably is not a serious problem for any assembly that does not experience extreme lateral accelerations.

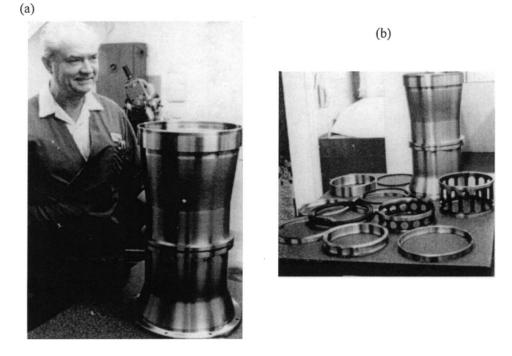


Figure 4.15 Mechanical parts of the astrographic telescope objective: (a) main titanium barrel; (b) barrel, six cells (solid rings), two spacers (slotted rings), and three retainers. (Courtesy of D. Vukobratovich, Tucson, AZ.)

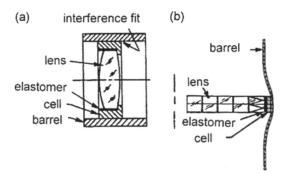


Figure 4.16 (a) Concept for elastomeric mounting, (b) FEA model used to confirm radial stress calculations. (Adapted from Valente and Richard.⁹)

4.5 Poker-Chip Assembly

Optomechanical subassemblies with the lenses mounted and aligned precisely within individual subcells that are inserted in sequence into precisely machined IDs of lens barrels (in the manner of a stack of poker chips) have been described by many authors.^{1,10-15} One such design is shown in Fig. 4.17.¹⁴ The lenses of this low-distortion, telecentric projection lens were individually aligned within their respective stainless steel cells to tolerances as small as 0.0005 in. (12.7 μ m) of decentration, a 0.0001-in. (2.5- μ m) edge thickness full indicator movement due to wedge, and a 0.0001-in. (2.5- μ m) surface edge full indicator movement due to tilt. They then were potted in place with 0.015-in.(0.381-mm) thick annular rings of 3M 2216B/A epoxy adhesive injected through radial holes in the subcells. After curing, the axial thicknesses of the subcells were final machined so that the air spaces between lenses were within design tolerances. The subcells were then inserted into the stainless-steel barrel and secured with retainers. No adjustments were needed.

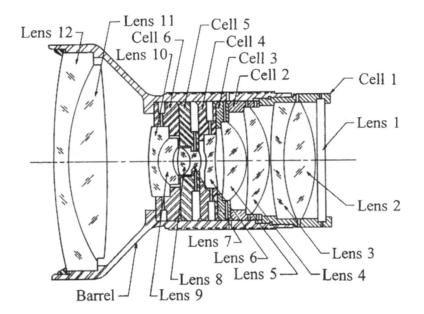


Figure 4.17 A high-performance projection lens assembled by the poker chip technique. (From Fischer.¹⁴)

Vukobratovich¹⁰ discussed the last-described technique for mounting lenses within subcells and illustrated it with examples in which each lens was burnished or epoxied into its own subcell. The cell ODs were machined to the proper OD for insertion along with similarly machined subcells into a common barrel. In another case, the cells were machined to the proper ODs within tight tolerances. Then the lenses were installed, centered to those ODs, and potted in place with rings of epoxy.

The assembly of lenses in the poker-chip configuration allows performance optimization by fine transverse adjustment of one or more lenses during the final stage of assembly. Figure. 4.18 shows an example of such an assembly in which the third element can be transversely adjusted with three radially directed screws to allow modification of its

aberration contributions to compensate for residual aberrations of the optical system. When this technique is employed, the moveable lens must have sufficient sensitivity to the specific aberration that is to be compensated so reasonable movement produces the desired effect. It must however not be too sensitive to this and other aberrations, for this would make the adjustment too critical. The choice of which element to move is usually made by the lens designer. Multiple components are sometimes chosen as "compensators"; each affects one specific aberration more than others. The construction of complex high performance poker-chip assemblies and their adjustment at final assembly are discussed in more detail in Chapter 12.

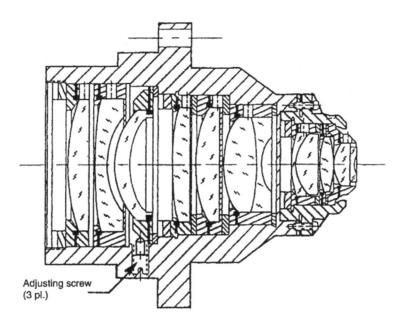


Figure 4.18 Sectional view of a poker chip type lens assembly with one lens element adjustable to allow optical performance to be measured and optimized after final assembly. (From Vukobratovich.¹⁰)

4.6 Assemblies Designed for High-Shock Environments

Figure 4.19 shows the optomechanical configuration of an air spaced objective for a relatively high-performance military telescope intended to withstand a severe shock and vibration environment. The three singlets are edged to the same outside diameter within tight tolerances and fit into a type 316 stainless-steel cell with a nominal 0.005-mm (0.0002-in.) radial clearance. All lenses are inserted from the right side. The first is Schott SF4 glass; it has a plano face that registers against a flat shoulder on the cell. The first spacer is 0.066 ± 0.005 -mm (0.0026 ± 0.0002 -in.) thick and is made of stainless-steel shim stock. The second lens is Schott SK16 glass and the third, Schott SSK4 glass. The second spacer is made of the same steel as the cell and is shaped for tangential contact on the adjacent convex surfaces. The retainer is also made of steel and is machined square to interface with a precision annular flat on the third lens. The threads on the retainer have a Class 2 fit into the cell threads as recommended in Chapter 3. All metal parts are black passivated.

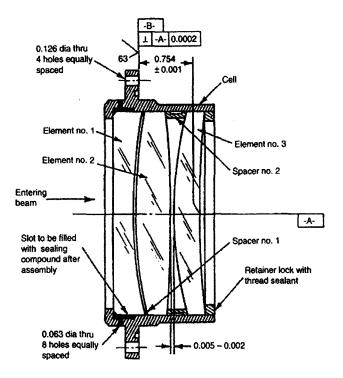


Figure 4.19 A high-performance telescope objective intended for high acceleration loading. (Adapted from Yoder.⁶)

The wedge tolerances on the lenses and spacers are 10 arc-seconds, whereas the maximum edge thickness variation from the annular flat to the first surface on the third element is toleranced at 10 μ m (0.0004 in.). At assembly, two of the lenses are phased by rotation about their axis for maximum symmetry of the on-axis aerial image in combination with the third (fixed) lens.

Figure 4.21 is a sectional view of a collimating lens assembly designed as part of a military flight-motion simulator. This assembly was intended to project an infrared target into a forward-looking infrared (FLIR) system during vibration testing of the latter device. The collimator has two air-spaced lens groups: the front group is a doublet with a diameter of approximately 9 in. (23 cm), while the rear group is a triplet with an average diameter of about 1.5 in. (3.8 cm). The lenses are made from silicon and germanium, so the optical system can operate in the 3 to 5- μ m spectral band. The overall dimensions of the assembly are 24.179 in. (61.41 cm) long and 12.43 in. (31.57 cm) diameter at the largest end (neglecting the larger mounting flange). It weighs about 80 lb (356 kg).

Palmer and Murray¹⁶ indicated that, because of the high cost of the larger lenses, the end user of the assembly specified that they should survive, without damage, a failure on the part of the simulator system that caused severe impact. Rather than designing the entire assembly to withstand the shock, it was designed so that the mechanical supports for those costly lenses would fail at an acceleration of 30 times gravity, and that those components would be constrained in a safe manner so they could be salvaged and reused even if other parts of the system were damaged.

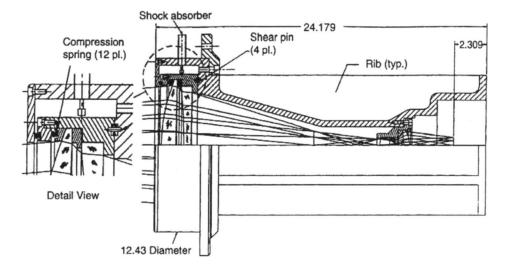


Figure 4.20 Sectional view of a collimating lens assembly designed to withstand high shock loading. Dimensions are in inches. (Courtesy of Janos Technology, Inc. Keene, NH)

It was determined that this severe impact would occur only in a direction transverse to the axis of the assembly. To make the assembly mechanically stiff under bending forces, the main housing was made of 6061-T6 aluminum and configured with a unique cross-section over most of its length. This construction may be seen in Fig. 4.21(a), which is a view of the exterior of the assembly. The lens groups occupy the cylindrical portions of the housing, while the structure between the cylinders has a cross section similar to a "paddle wheel" with six ribs of full external-diameter-supporting conformal wall portions, which enclose the internal light beam that converges from the smaller lenses and expands to fill the apertures of the larger lenses [see Fig. 4.21(b)]. The ribs enhance assembly stiffness while minimizing weight. Grooves machined into the internal surfaces of the walls reduce stray light reflections that could degrade image contrast.

The cell for the larger lenses was designed with a retaining flange that presses multiple axially oriented compression springs against an annular pressure ring contacting the first lens. The preload so introduced presses a spacer against the second lens, which, in turn, registers against a shoulder. The cell is constrained radially in the housing by three axially oriented aluminum shear pins that engage stainless steel inserts pressed into the lens cell and the housing. Without these pins, the cell would be able to slide laterally within clearances provided all around the rim of the cell. At assembly, the pins locate the cell and its lenses radially. The cell is pressed firmly against a shoulder in the housing by additional axially oriented compression springs that bear against the outermost flange.

The three pins are designed to shear under the prescribed shock load, allowing the cell to move. This cell motion is dampened by shock absorbers oriented radially at four points around the periphery of the assembly. Three of these are shown in Fig. 4.21(a) and one is indicated in Fig. 4.20. The shock absorbers are nonlinear; they become stiffer under higher accelerations.

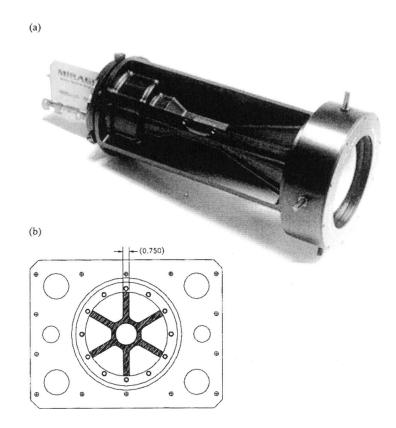


Figure 4.21 (a) Exterior view of the collimator assembly of Fig. 4.20 without its mounting flange. (b) Section view through the center of the assembly. (Courtesy of Janos Technology, Inc. Keene, NH)

4.7 Photographic Objective Lenses

Figure 4.22 shows a classic design for a f/1.9 objective for a 16-mm motion picture camera.¹⁷ Its focal length is 25.4 mm. Three of the four lenses are held in place by threaded retaining rings while the fourth (biconcave shaped) is held by metal spring clips spot welded to the mount. The outermost threaded ring (left end) is provided to hold a filter in place. A large portion of the mount is devoted to the focus mechanism, which can be seen in the section view.

A very fast (f/0.71) fixed-focus, fixed aperture, 64-mm focal length, lens for 16-mm motion picture photography in a medical application is shown in Figure 4.23. The deeply curved lenses are mounted in cells with threaded retainers. Of special interest is the configuration of the cell holding the smaller diameter doublet and the fact that the front surface is contacted by a conical interface and the retainer rests against a deep bevel. In Chapter 3, this author suggests avoiding the latter type interface. The construction of this lens is, of course, commercial. Obviously, the lens was not intended to withstand severe handling or exposure to adverse environments.

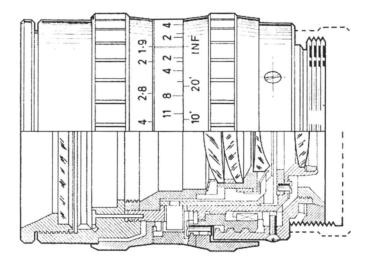


Figure 4.22 Sectional view of a 25.4 mm focal length, f/1.9 motion picture lens assembly. (Adapted from Horne.¹⁷)

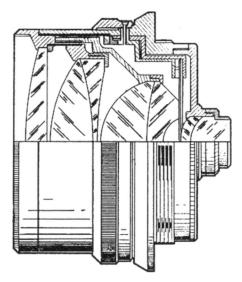


Figure 4.23 Sectional view of a 64-mm focal length, f/0.71 lens for a medical application. (Adapted from Horne.¹⁷)

Figure 4.24 shows a high quality 85-mm focal length, f/1.4 objective of the planar type made by Carl Zeiss for a 35-mm film camera. Featured here are toroidal interfaces on concave lens surfaces (2 places), a glass-to-glass interface between the 2nd and 3rd lens elements, a step bevel, and a differential-thread focus mechanism. Figure 4.25 shows detailed views of these features of the lens. Note that the outlines of elements 3 and 4/5 (cemented doublet) have been emphasized with white lines for clarity. In view (a), the lens interfaces are indicated, and in view (b), the focus mechanism is shown. To focus,

turning the focus ring rotates the intermediate ring, which has coarse and fine threads on its ID and OD, respectively. These threads mesh with corresponding threads on the fixed housing and the lens cell. The lenses move as a group to focus. The principle of operation of the differential thread mechanism is discussed in more detail in Section 4.11.

The aerial camera lens shown earlier in Fig. 4.13 is a 66 in. (1.67 m) focal length, f/8 assembly. Its length is 13.28 in. (33.73 cm) and its entrance aperture diameter is 9.60 in. (24.38 cm). The long focal length leads to increased detail in the photographs as compared to that which could be obtained with standard camera lenses as used by the general public.

A significantly larger aerial camera lens is shown in Fig. 4.26. Its focal length is 72 in. (1.8 m), which is 9% larger than the previously discussed assembly. The relative aperture of the present lens is, however, f/4, so it has 475% greater light gathering power. The optomechanical system of the f/4 lens assembly was designed to perform to specification on film in sunlight with a minus-blue filter at an altitude of at least 100,000 ft (91 km) over a range of temperatures from -65 to 130°F (-54 to 54°C).¹⁰ The film supply/take-up cassette is the black subassembly shown below the woman's right hand in the photograph.

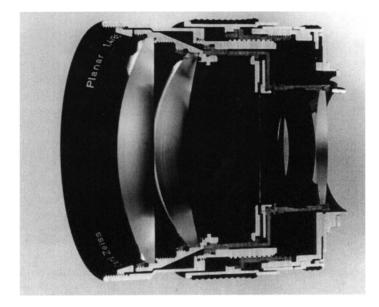


Figure 4.24 Cut-away photograph of a high performance 85-mm focal length, *f*/1.4 Zeiss Planar lens for a 35-mm camera. (Courtesy of Carl Zeiss, Germany.)

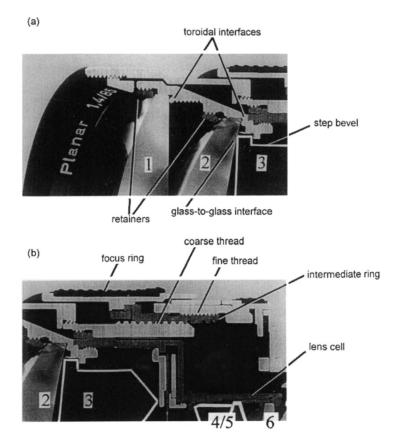


Figure 4.25 Enlarged views of some features of the Zeiss lens of Fig. 4.24.

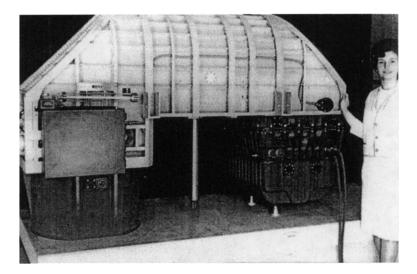


Figure 4.26 Photograph of a 72 in. (1.8 m) focal length, f/4 aerial camera objective with large 18 in. (45.7 cm) aperture. (From Yoder.¹¹)

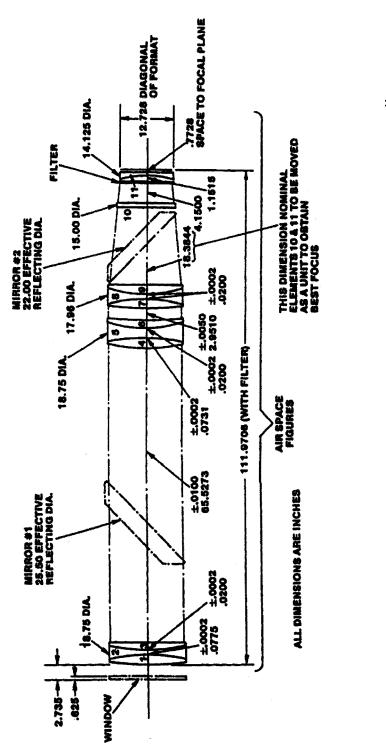
Figure 4.27 is the optical layout of this assembly. It is of the Petzval type and has three air-spaced triplets, two with approximately the same OD of 18.75 in. (47.62 cm) and the third having an OD of 17.96 in. (45.62 cm). The U-shaped configuration provides space for a stabilizing mount (not shown) under the center portion of the assembly and aligns the center of gravity of the optical assembly with the gimbal axes of the mount.

Figure 4.28 shows the tolerances to be met in order for the performance required from this lens system to be achieved. It is not a simple task to mount lenses of these diameters to tight tolerances on centration [typically 0.0005 in. (12.7 µm)] and in a stress-free condition over the specified temperature range. Scott described the mounting for the central triplet subassembly used in this objective.¹⁸ Figure 4.29 shows a sectional view through that subassembly. The positive elements are BaLF6 glass (CTE = 6.7×10^{-6} /°C), while the central negative element is KzFS4 glass (CTE = 4.5×10^{-6} /°C). The cell material was chosen to match as closely as possible the CTE of the BaLF6 glass. This material is A70 titanium (CTE 8.1×10^{-6} /°C), and was machined from a forged cylinder.

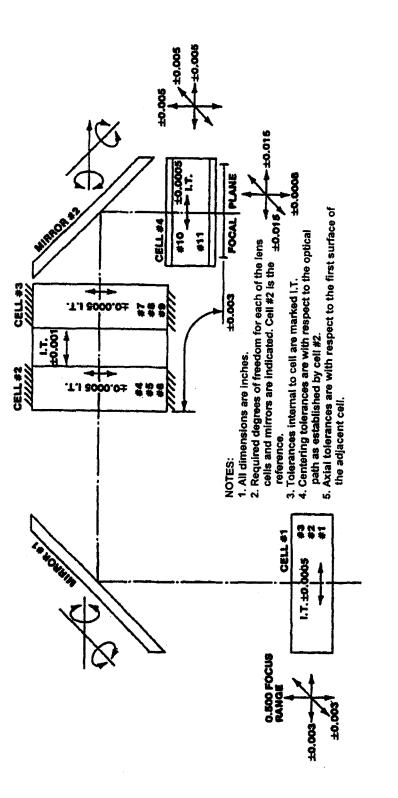
At assembly, radially oriented aluminum plugs, nominally 1.0-in. (25.4-mm) long, were inserted against the rim of the center lens at 120-deg intervals. The caps over the ends of the plugs were bottomed against the cell wall. Mylar shims were placed between the lens rims and the adjacent metal to act as padding at the interfaces. This combination of materials of various dimensions served to athermalize the mount in the radial direction. Aluminum plugs also were used to athermalize the subassembly in the axial direction. These plugs were lapped until the right-hand surface of the third lens was pushed against the hard-mounted flange at that end of the assembly and the caps over the plug ends were bottomed against the outer surface of the left flange retainer. After assembly, the rims of the positive lenses were sealed to the barrel ID with a polysulfide-type sealant (3M EC-801) to maintain a desiccated environment within the assembly after purging with dry nitrogen.¹⁸

The method of mounting the lens cells in the housing had to be rigid, compact, lightweight, temperature-stable, and in the case of the front and rear cells, axially adjustable for focus. In the final design, each large cell was supported from the camera structure on three tangent arms, each of which was approximately 0.13-in. (3.3-mm) thick by 3.0-in. (76.2-mm) wide. In the case of the smaller (rear) lens cell, the arms were 1.5-in. (38.-mm) wide. Their thicknesses were the same as the other arms. One end of each arm was bolted and pinned directly to its cell. In the case of the second and third cells, the fixed ends of the arms were attached to fittings on the structure. In the case of the first and fourth cells, the arms were attached to the structure through adjustable eccentric bolts.

The structure chosen for this assembly was of the semimonocoque type. Here annular aluminum bulkheads were spaced at intervals along the lens axis and tied together by longitudinal stringers in much the same fashion as aircraft fuselages are built. An aluminum skin was attached by rivets to the bulkheads and the stringers to provide structural stiffness. The skin was located internally to these stringers to provide space for thermal insulation and a protective cover on the outside. Figure 4.26 shows the assembly before the insulation and cover were added.









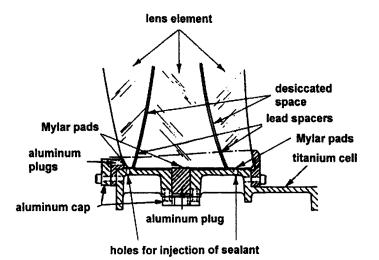


Figure 4.29 Technique for radially athermalizing the large 18 in. (45.7 cm) aperture lenses in the aerial camera objective of Fig. 4.26. (Adapted from

To establish the proper focus of the image at the film plane, four adjustable hardened pads were built into the surface of the last lens cell facing the image. Four fixed pads were provided on the film magazine side of the interface. Iterative photographic tests and adjustments of the lens cell pads during final alignment ensured that the focus was correct.¹⁸

4.8 Modular Construction and Assembly

The assembly, alignment, and maintenance of optical instruments are simplified if groups of related optomechanical components are constructed as prealigned and interchangeable modules. In some cases, the individual modules are considered nonmaintainable, and instrument repairs are made by replacing defective modules, usually without requiring system realignment.

The manufacture of a modular instrument design is somewhat more complex than the equivalent nonmodular version because of the added requirement for interchangeability without compromising performance. This tends to increase production costs, which is at least partially compensated for by significantly reduced assembly and alignment costs. In most cases, mounting surfaces are machined to specific orientations and locations with respect to optical axes and focal planes. Assembly is greatly facilitated by the design and fabrication of optomechanical fixtures specifically intended for alignment of the modules. Vukobratovich reviewed alignment accuracy and structural stiffness typically resulting from modular design and fabrication methods in Ref. 19.

A good example of modular design is illustrated in Fig. 4.30 where we see an exploded view of a military 7×50 binocular. Its optomechanical layout is shown in Fig. 4.31. This

Scott.¹⁸)

instrument has identical prealigned and parfocalized^{*} objectives and eyepieces, as well as left and right housings with prealigned Porro-type erecting prisms. The thin-wall-housing castings are identical until the final machining stage. The housings are connected with a hinge mechanism that allows adjustment of the interpupillary distance.

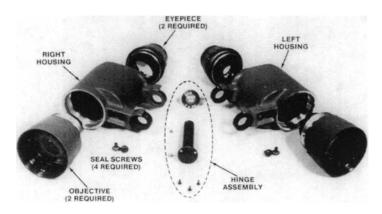


Figure 4.30 Exploded view of the modular military Binocular M19. (From a U.S. Army photograph.)

One objective module is shown in Fig. 4.32. It is an air spaced triplet of telephoto configuration with a focal length of 6.012 in. (152.705 mm) and an aperture of 1.969 in. (50.000 mm). It works at f/3.05. The elements are types 517647 (first two elements) and 689309 glasses. The lens diameters are 2.067 in. (52.502 mm), 1.909 in. (48.489 mm), and 1.457 in. (37008 mm), respectively. Each OD is held to tolerances of +0, -0.001 in. (+0 - 0.25 mm). The objective housing is wrought aluminum. The crown lenses are mounted directly into a recess in the housing with an intermediate tapered spacer. A threaded retainer provides axial preload to these elements through an O-ring from distortion as the retainer is tightened. The flint lens is mounted into a focusable cell that screws into the housing to tune the focal length of the triplet. This cell is sealed to the housing after adjustment with injected elastomeric sealant. The flint element is sealed into its cell with an O-ring compressed axially through a thin pressure ring by a retainer that also serves as a light baffle. The entire objective assembly is sealed to the binocular housing by an O-ring compressed radially during assembly.

All parts of the objective are assembled and sealed in a dry atmosphere before its external mounting interfaces are machined. Centration of the objective to the system's optical axis and focus is accomplished by machining the registration OD and shoulder of the module (surfaces -A- and -B-, respectively, of Fig. 4.32) as well as the mating surfaces in the body to close tolerances using sophisticated optical alignment techniques to position the modules in holding and transfer fixtures. The objective cell (still in its fixture) is precision machined on a numerically controlled lathe that is equipped with a hollow spindle

Parfocalized optical assemblies have preset and nominally equal back focal distances.

and an optical collimator aligned to the spindle axis. The image produced by the objective is observed with an eyepiece or microscope with a crosshair also aligned to the same axis and at the proper distance from the flange.

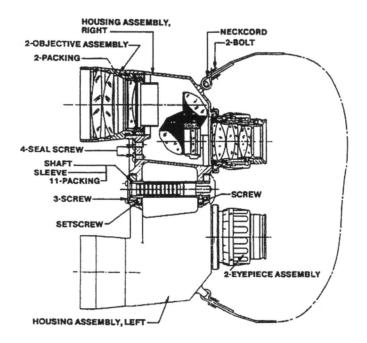


Figure 4.31 Sectional view of the modular Binocular M19 (From a U.S. Army drawing.)

The overall construction of the eyepiece modules for the M19 Binocular is discussed in Section 4.11 in the context of mechanisms for focusing optical assemblies.

Many photographic and video camera lenses, microscope objectives, and telescope eyepieces are essentially optomechanical modules. For example, the interchangeable objectives for most laboratory microscopes are modular in that their mechanical interfaces are aligned optically at assembly to produce a high quality image on axis at a predetermined distance from the mounting flange. In photographic applications, a variety of modular lens assemblies can be interchanged on a single camera body or moved from one camera to another of similar type. These lens modules are parfocalized so their calibrated infinity focal planes automatically coincide with the camera's film plane.

Single-point diamond-turning (SPDT) techniques facilitate the creation of optomechanical modules that have precisely located and contoured optical component mounting interfaces and integral mounting features. An example is shown in Fig. 4.33(a). This is a concave toroidal metal mirror with an integral flange for attaching the module to an instrument without requiring adjustment for alignment. The application is in the dual-channel short wavelength spectrometer (SWS) designed for use with a 60-cm (23.6-in.) aperture, 9-m (29.5-ft) focal length Cassegrain telescope in the European Space Agency's Infrared Space Observatory. This instrument was intended to measure stellar spectra in wavelength bands of 2.5 μ m to 13 μ m and 12 μ m to 45 μ m using dual optical systems.²¹

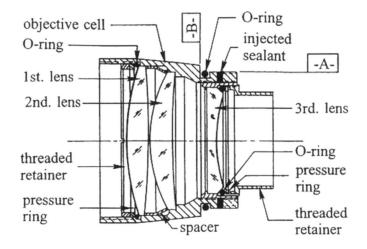


Figure 4.32 Sectional view of the modular objective assembly for the Binocular M19. (Adapted from Trsar et al.²⁰)

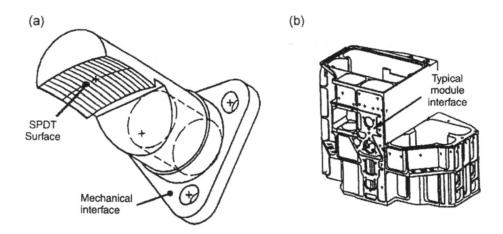


Figure 4.33 Two components of the SWS: (a) a precision toroidal mirror module made by SPDT and (b) the main housing showing cutouts and multiple optical module interfaces. (From Visser and Smorenburg.²¹)

Figure 4.34 illustrates one optical system that is located after the f/15 focal plane of the main telescope. Dichroic beamsplitters feed two functionally identical grating spectrometers outputting to multiple detector arrays. The longer-wavelength channel also feed two tunable Fabry-Perot detectors. Many mirrors used here are aspheric and/or anamorphic to provide the necessary beam sizes at the gratings while fitting within the allowable package space. Several radiant sources are included for calibration purposes.

A front view of the machined main housing for the spectrometer is shown in Fig. 4.33(b). This was machined from one piece of 6082 aluminum alloy to obtain maximum uniformity of CTE. This housing attached with screws to the satellite structure through

one rigid foot and two flexible (i.e., flexure) feet. The various modular mirror subassemblies protruded through openings in the housings and were secured with screws from the outside. Modular construction facilitated assembly and ensured that the component alignment would be retained for a long time. If necessary, replacement of a mirror would be a simple operation. The same aluminum alloy as the housing was used in these mirrors and in the gratings to equalize the system's thermal properties and enhance stability. The mirrors were gold-coated to enhance reflectivity in the IR. The gratings were ruled directly into coated aluminum blanks.

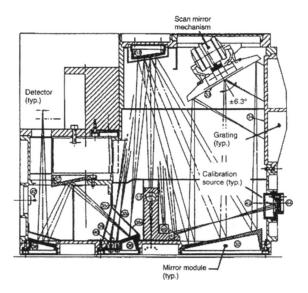


Figure 4.34 One optical system of the modular short wavelength spectrometer (SWS) instrument. (From Visser and Smorenburg.²¹)

4.9 Catoptric and Catadioptric Assemblies

A catadioptric optical system is one that has both refracting (lens) and reflecting (mirror) components. Most are adaptations of classic all-reflecting (catoptric) designs such as the Cassegrain or Gregorian types. Usually the refractive (dioptric) components are added to improve performance or to increase the usable field of view of a reflective system. The catadioptric system usually works at a faster relative aperture, is physically shorter, and has a larger field than the equivalent catoptric version. Although a reflecting system with field lenses added near the focal plane is technically catadioptric, this term is more frequently associated with systems having full-aperture refracting components such as in Schmidt or Maksutov objectives or derivatives of the Schmidt-Cassegrain system.

A representative catoptric instrument is the Infrared Astronomical Satellite (IRAS), which was a Ritchey-Chretien variant of the Cassegrain optical system. Figure 4.35 shows a schematic diagram of the optomechanical system for this 24-in. (60.96-cm) aperture telescope. It operated in the 8 to 120-µm spectral region. All major structural parts and the two mirrors were made of beryllium. This single-material construction resulted in an athermal design, as was necessitated by the fact that the mirrors were made at room

temperature, but used at a cryogenic temperature of about 2K. At this low operational temperature, the system changed dimensions, but remained in focus and gave full performance. The mirrors were supported on the telescope structure by flexures to minimize transfer of forces and moments into the optical surfaces. These flexures are described in Chapter 10.

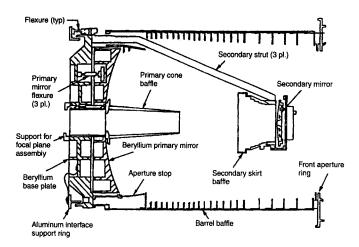


Figure 4.35 Optomechanical configuration of the 24 in. (61 cm) aperture, *f*/1, all-beryllium IRAS Telescope. (Adapted from Schreibman and Young.²²)

Full-aperture lenses located at the entrance apertures of many catadioptric systems are usually called "corrector plates," since they usually have near-zero total optical power and serve primarily to correct aberrations that cannot be corrected by the associated mirrors. They also serve as windows to preserve the integrity of the interior of the instrument.

Frequently, large corrector plates are thin, but large in diameter, so have reduced structural strength. Consequently, they suffer more from gravitational, acceleration, and thermal effects than smaller components. Large diameter threaded retaining rings are difficult to make and install, so flange-type retainers are frequently employed to preload the plates. Sealing by elastomeric formed-in-place gaskets usually is best for the larger correctors. Smaller ones can be sealed with O-rings.

To illustrate a typical catadioptric system, Fig. 4.36 shows one of the 20-in. (50.8-cm) focal length, f/1 Baker-Nunn, "Satrack" cameras developed in the mid-1950s to photograph orbiting satellites. Fig. 4.37 shows two sectional schematic (top and side) views of the assembly. The optical design, created by James G. Baker, was an enhancement of the Schmidt objective. The single corrector plate of the classic Schmidt was replaced here by a full-aperture air-spaced triplet (black profiles).

A good description of this system is found in MIL-HDBK-141 entitled *Optical Design.*²³ A portion of that description follows:

"It will be noted that the aperture of the system is very close to the center of curvature of the primary mirror, but the single corrector plate, which normally is located there, is split up into a color corrected triplet for the purpose of eliminating the small amount of residual axial color in the single Schmidt plate. The four inner surfaces of this system are aspheric.

"It is presumed that, because of the high relative aperture of this system (f/1), the curvature of the Schmidt plate required would be more extreme than usual, leading to more axial color than the designer could tolerate. The splitting up of the single plate into three, with the central glass different from the outside, and the distribution of the Schmidt curvature among four surfaces would tend to alleviate this situation.

"It will be noted from [Fig. 4.37] that the film is transported over a spherically curved gate, which matches the curved focal plane of the image. The curvature in the plane at right angles must necessarily be zero, because of the mechanical impossibility of bending the moving film into a compound curve. Consequently, the field coverage in this direction is limited to only 5 degrees, while in the direction of film travel it reaches the amazing value of 31 degrees. It was found that at the edges of this extreme field the focal surface departs slightly from a spherical shape, so the film runners are not quite circular. The combination of careful design and excellent execution resulted in a system wherein 80 percent of the point energy anywhere in the field is within a circle 0.001 inch in diameter. This instrument was conceived for the purpose of tracking the U.S. Vanguard satellite, and the first instrument arrived just in time to be used for the original Sputniks."

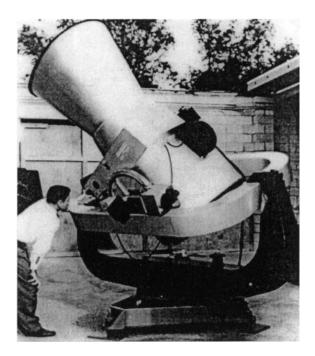
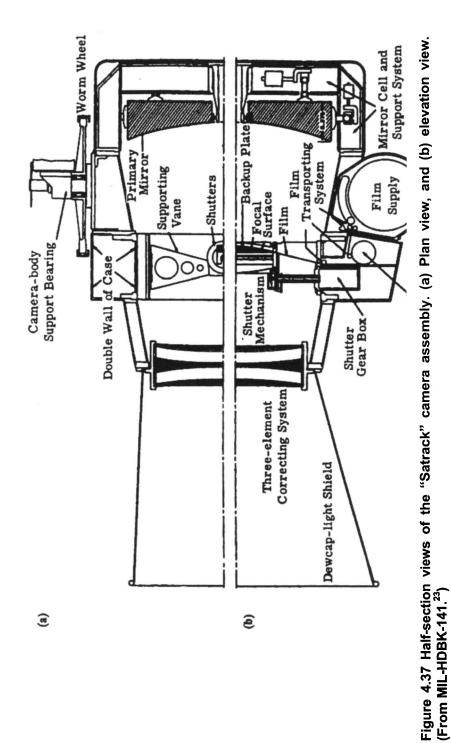


Figure 4.36 One of the first Satrack Cameras during final checkout. (Courtesy of Goodrich Corp., Danbury, CT.)



Although not much optomechanical detail can be derived from Fig. 4.37, one can note that the primary mirror is supported radially and axially on counterbalanced lever mechanisms. Supports of this type are described in Chapter 11. The diameter of the mirror is about 31 in. (79 cm) to prevent vignetting at the edges of the field.

The optics for all the Satrack systems were built by Perkin-Elmer Corporation in Norwalk, Connecticut. Joseph Nunn of South Pasadena, California provided the mechanical design. Boller and Chivens, also in South Pasadena, California did the mechanical fabrication, assembly, and testing. Twelve of these cameras were built and used with great success at various locations around the world. Periodic maintenance kept them operational for many years. The most serious repair to the optics was repolishing of the exposed plano surface of the outer corrector plate, which became stained over time by deposits from visiting birds. The aluminum/silicon monoxide coatings on the primary mirror also needed to be replaced occasionally, and the slightly aspheric film runners (i.e., film platen) needed to be repolished because of wear from the film passage. The cameras remained in operation until after 1980. Some systems are believed to still exist—but not necessarily in their original configuration nor in their full operating condition.

4.10 Assemblies with Plastic Housings and Lenses

Plastic lenses are used in many consumer products such as camera viewfinders and objectives, magnifiers, television projection systems, compact disc players, and cell phone camera lenses. They also are used in some military applications, such as night vision goggles, and head-mounted displays. In hardware requiring the lowest cost and low to moderate performance, mechanical parts also can be made of plastic. Low-cost housings can be molded of plastic using techniques similar to those employed in making optics. Figure 4.38 shows a collet-type housing in which four molded lenses with integral tabs are inserted so the tabs fit into slots in the housing. End caps are slipped over the ends of the housing, thereby compressing the housing around the lens rims. The caps are then secured by ultrasonic bonding. Disassembly of such a unit is not generally considered practical.

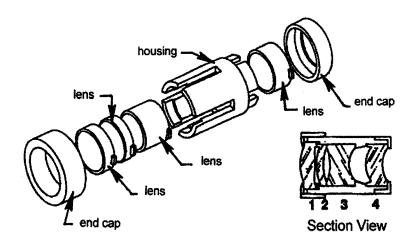


Figure 4.38 Lens assembly featuring all-plastic construction. (Adapted from Lytle.²⁴)

Figure 4.39 illustrates two versions of a mounting arrangement in which lenses are located in separate cylindrical housings. These housings are joined during assembly. In view (a), the joint is shown next to the negative lens. The aperture stop is integral with the second housing. A spacer separates the left two lenses and both positive lenses are secured with retainers. The retainers might be bonded in place with adhesive or heat sealed to the housing. In the design represented by view (b), the joint is between a cylindrical housing and an attached flange. This flange is outside the right positive lens. Here all lenses are mounted in the same housing, thereby achieving better centration since all lens seats are molded into the same part. The aperture stop must be a separate part, so the cost of the bottom design would probably be somewhat higher than that of the top design. The tradeoff is then between cost and centration (i.e., performance).

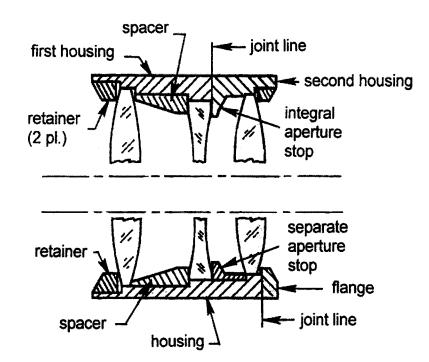


Figure 4.39 Schematics of two all-plastic, three-lens assemblies with identical optics. The upper design (two axially joined housings) should have a lower cost, but the lower one (one housing and a flange) should have better lens centration. (Courtesy of 3M Precision Optics, Inc., Cincinnati, OH.)

Figure 4.40 shows photographs of a series of all-plastic lens assemblies manufactured by U.S. Precision Lens (now 3M Precision Optics, Inc.) for use in rear projection television systems. Focus is achieved in some by turning an inner cell (containing the lenses) inside an outer housing, with axial motion driven by a pin sliding in a helical cam slot in that housing. Other assemblies have two axially adjustable cells with lenses. Each of these cells has a pin that slides in its own cam slot. These two motions allow focus and magnification adjustment.



Figure 4.40 Photographs of several all-plastic lens assemblies. (Courtesy of 3M Precision Optics, Inc., Cincinnati, OH.)

An example of one design for an all-plastic assembly is shown in Fig. 4.41. It is labeled as a Delta 20 design and consists of an air-spaced triplet mounted in a plastic cell that can be moved axially to focus by rotating that cell within the housing. The dimensions of the assembly are 104.5 ± 3.5 -mm $(4.11 \pm 0.14$ -in.) long, including focus motion, and a 117-mm (4.1-in.) diameter, not including the mounting flanges. The weight of the assembly is approximately 660 g (1.5 lb). The device was designed to operate at a fixed relative aperture of f/1.2 with a nominal magnification of $9.3 \times$.



Figure 4.41 Photograph of an all-plastic television projection lens assembly.

The internal construction of the lens cell is similar to that described by Betinsky and Wellham²⁵ for earlier designs (see Fig. 4.42). The lenses are supported within a longitudinally split cell molded as symmetrical halves. This is commonly called a "clamshell mounting." At the time of assembly, the lenses are inserted into recesses in one half of the cell [see Fig. 4.43(a)]. When the halves are joined together, the lenses are captured and held in position by internal features such as pads that touch the lens rims to center the optics and narrow tabs that extend inward radially in front and back of the lens at several locations around its periphery. These tabs bend slightly when the optic is inserted to secure it axially. The plastic components deform slightly when self-tapping screws are inserted through the flanges to clamp the cell halves together. Figure 4.43(b) shows the interior of one cell half with only one lens in place. Some of the radial pads, axial constraint tabs, and stray light suppressing grooves are visible. Light passing through the holes in the flanges for the screws can also be seen in the shadow area.

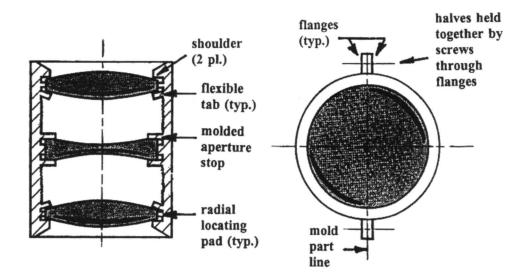


Figure 4.42 Schematic side and end views of the clamshell lens mounting. (Adapted from a publication by U.S. Precision Lens, Inc., Cincinnati, OH.²⁶)

The assembled lens cell fits snugly into the ID of the housing. Two screws pass through helical cams on either side of the housing and thread into holes in the cell wall. Wing nuts on the screws are tightened to clamp the adjustment after focusing. The housing is designed to be attached to the structure of the television system by screws passing through the mounting ears visible in Fig. 4.41.

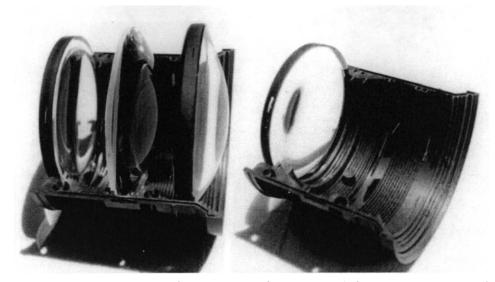


Figure 4.43 Photographs of the interior of the lens cell from the assembly of Fig. 4.41. (a) With all lenses in place and (b) with two or the lenses removed. Lens mounting features and stray light reducing grooves can be seen.

Figure 4.44 is a photograph of a one-piece molded plastic modular assembly designed for use in an automatic coin-changer mechanism. It is made of polymethyl methacrolate (acrylic) and has two lens elements, one of which is aspheric, molded integrally with a mechanical housing that has prealigned mounting interfaces for attaching two detectors. When it is manufactured in large quantities so as to amortize cost of the injection tooling, this type module is inexpensive. Because it needs no adjustments, it is easy to install and virtually maintenance free.

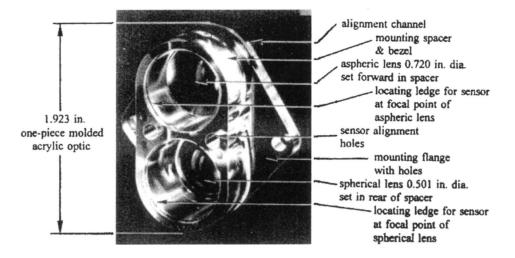


Figure 4.44 One-piece molded acrylic optical module with two integral prefocused lenses, interfaces with detectors, and an integral mounting interface. (Courtesy of 3M Precision Optics, Inc. Cincinnati, OH.)

4.11 Internal Mechanisms

4.11.1 Focus mechanisms

In many optical instruments, internal adjustments are required during normal operation. Examples are focusing a camera or binocular on objects at different distances, changing the focal length of a zoom lens, or adjusting a telescope eyepiece to suit the focus requirements of the observer's eye. Most of these adjustments involve axial motions of certain lenses or groups of lenses within the instrument. A few applications, such as the range compensator of some camera range finders or rectification of converging images of parallel lines in architectural photography, may involve decentration or tilting of lenses.

In some cameras, focus is changed by moving the entire objective lens relative to the film or sensor, while in others it is changed by moving one or more lens components within the objective relative to the rest of the components. The required motions may be small or large, depending on the lens focal length and object distance, but these motions must always be made precisely and with minimum decentration or tilt of the moving parts.

Figure 4.45 illustrates a type of mechanism often used in a camera lens to couple rotation of an external knurled focus ring through a differential thread to change the spacing between lens elements. The differential thread consists of a coarse pitch thread^{*} and a finer one on either end of a cylindrical part, labeled "focus ring." Mating threads are placed on the front cell (A in the figure) and the back cell (B). When the ring is turned, the threads act together to move the lens subassemblies axially as if they were driven by a thread of a pitch much finer than that actually used in the mechanism. Such a fine thread would be harder (read "more expensive") to manufacture and might cause thread damage during assembly if crossed inadvertently.

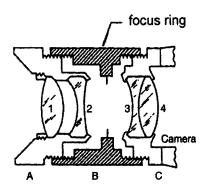


Fig. 4.45 Schematic diagram of a differential thread focus mechanism. (From Jacobs.²⁷)

The "pitch" of a thread is the axial distance between crests. Its reciprocal is the number "N" of threads per unit length, such as "threads per inch" (tpi). The "lead" of a single threaded screw equals its "pitch". That of a "multiple lead" screw is "n" times the "pitch" of that screw, where "n" is the number of parallel threads on the screw.

A few words about the lens system in Fig. 4.45 seem appropriate. Lenses 1 and 2 are mounted in cell A, while lenses 3 and 4 are mounted in cell C. With this type of lens (Tessar), the spacing between elements 1 and 2 is very critical so an accurately machined shoulder maintains this spacing. Lens 1 is held in place by a threaded retaining ring, while all the other elements are burnished into their cells. The airspace between lenses 2 and 3 can be varied to focus the assembly, as just explained. The lenses do not rotate about the axis, so the image does not shift laterally while focusing. Some means, such as a pin sliding in a fixed external slot, would be appropriate for each cell.

Quantitatively, the threads connecting A and B might be 32 threads per inch, while those connecting B and C might be 48 threads per inch. If the threads were made subtractive with rotation of the focus ring, the motion would be equivalent to a single thread with 96 threads per inch. If rapid focusing were desired (in some other lens assembly), these threads could be made additive, giving the effect of a single thread of 19.2 threads per inch. Equations (4.4) and (4.5) express the simple mathematical relationships used to find these results and Example 4.2 shows how to use them:

$$N_{E \text{ FINE}} = \frac{N_1 N_2}{(N_2 - N)_1},$$
(4.4)

$$N_{E \text{ COARSE}} = \frac{N_1 N_2}{(N_2 + N_1)},$$
 (4.5)

where $N_{E \text{ FINE}}$ creates a finer motion by combining threads N_1 and N_2 and $N_E \text{ COARSE}$ creates the opposite effect with the same threads. In the first case, the threads are both right handed while in the second case, they are opposite handed.

Another useful attribute of the differential thread is the increase in resolution of the resulting thread as compared to the finer thread used in the mechanism. This is sometimes called its "gain." Kittell²⁸ determine this as:

$$Gain = \frac{N_2}{(N_1 - N_2)}.$$
 (4.6)

Yet another bit of useful information about the effect of using a differential thread is the distance moved per complete turn of the connecting member (such as the focus ring in Fig. 4.45). Defining this factor as δ_{DIF} , we obtain its magnitude quite simply as

$$\delta_{\rm DIF} = \frac{1}{N_E},\tag{4.7}$$

where N_E is either $N_{E \text{ FINE}}$ or $N_{E \text{ COARSE}}$ as the case may be.²⁸

A previously discussed example of a modern high-performance photographic lens assembly using the differential thread principle is the Carl Zeiss 85-mm f/1.4 Planar lens shown in Fig. 4.25(b). Rotation of the knurled ring (outermost dark-colored part of the barrel) turns an intermediate threaded ring on fine and coarse threads to slide the entire inner subassembly carrying the lenses in the axial direction without rotating. The means for preventing lens rotation is not obvious in the figure.

Kittel²⁸ pointed out some things to watch out for in designing and tolerancing the normal differential thread mechanism. While this mechanism creates a finer motion than would be produced by either thread alone, it reduces the available range of motion considerably in comparison with that available from either thread. Moving from one end to the other within this small range requires a larger rotation of the intermediate member than would be required with just one thread. Finally, the inevitable random errors in the pitch of each thread are additive in the differential mechanism. This effect may make the translation per degree of rotation nonlinear. Backlash might be a problem during reversal of travel if the differential thread mechanism is not spring loaded in one direction. Not withstanding these potential problems, the differential mechanism is very practical for many applications to optical instrumentation where small motions are the norm.

Because they are customarily used to observe objects at long distances, the objectives of most military telescopes, binoculars, and periscopes traditionally have a fixed focus at infinity. The angular calibration of reticle patterns used for weapon-fire control then remains constant since the image distance to that pattern equals the objective focal length. Refocusing the eyepiece(s) would have no effect on reticle calibration since the image of the target and the pattern are axially coincident at the eyepiece's object plane. If the magnification of such an instrument is greater than 3 power, its eyepiece(s) should be individually focusable to suit the user's eye accommodation.

Example 4.2: Differential thread. (For design and analysis, use File 4.2 of the CD-ROM).

Two parts of a camera lens are to have a variable separation for focus adjustment using a fine thread created by a differential mechanism. The threads are chosen as 32 and 48 threads per inch (tpi). (a) What is the effective pitch of the combination? (b) What is the gain of the mechanism? and (c) What is δ_{DIF} for this mechanism?

(a) By Eq. (4.4):

$$N_{\rm E \, FINE} = \frac{(32)(48)}{(48-32)} = 96 \, {\rm tpi} \, (3.78 \, {\rm tpmm})$$

(b) By Eq. (4.5):

$$Gain = \frac{48}{(48-32)} = 3$$

(c) By Eq. (4.6):

$$\delta_{\text{DIF}} = \frac{1}{96} = 0.0104 \text{ in./turn} = 0.265 \text{ mm/turn.}$$

Example 4.3: Required axial motion to focus an eyepiece. (For design and analysis, use File 4.3 of the CD-ROM).

The focus of an eyepiece with focal length $f_E = 1.569$ in. (39.843 mm) is to be changed by ± 4 D. What total axial motion is required?

By Eq. (4.8),
$$\Delta_E = \frac{1.569^2}{39.37} = 0.0625$$
 in. (1.588 mm) for 1 D change,
so ± 4 D = $\pm (4)(0.0625) = \pm 0.250$ in. (6.350 mm).

The total movement is 0.250 + 0.250 = 0.500 in. (12.700 mm).

A focus adjustment (commonly called the "diopter adjustment") of at least ± 4 diopters (D)^{*} is common for military instruments, while a range of at least ± 2 to -3 D is common on consumer equipment. A scale calibrated in 1/2 or 1/4 D increments usually is placed on the eyepiece focus ring for ease of setting. Assuming that the entire eyepiece is moved axially to make this adjustment, the axial displacement Δ_E for a 1 D change in collimation of the beam entering the eye is determined from Eq. (4.8):

$$\Delta_{E} = \frac{f_{E}^{2}}{1000} \text{ (if } f_{E} \text{ is in mm)},$$

$$\Delta_{E} = \frac{f_{E}^{2}}{39.37} \text{ (if } f_{E} \text{ is in in.}),$$
(4.8)

or

where f_E is the eyepiece focal length. Example 4.3 illustrates use of this equation.

The thread defined in this example is very coarse. Additive differential threads (combination of a right-handed thread and a left-handed thread turning simultaneously) or multiple-lead threads (such as a set of four or more individual coarse threads in parallel) can be used to advantage here. Helical cams with cam followers also can be used in such cases. We here concentrate on threaded mechanisms.

An example of an eyepiece with multiple-lead threads is shown in Fig. 4.46. To determine the number of leads, one counts the number of starts at either end of the threaded region. In this thread, there are 6 starts so it has 6 leads. By design, the number of thread crests per inch is 16 so the pitch is 16 tpi (0.630 tpmm). The pitch diameter of each thread in this particular assembly is 1.180 in. (29.972 mm), while the axial length of the thread engagement is about 0.28 in. (7.11 mm). With six threads engaged, much averaging of minor manufacturing errors and thread imperfections takes place so that the motion feels smooth to the user with a minimal amount of lubrication.

Example 4.4 applies the above equations to a practical application.

^{*} One diopter corresponds to a change in focus corresponding to a focal length of 1000 mm or 39.37 in. This is the same as terminology for specifying optical power of eyeglasses.

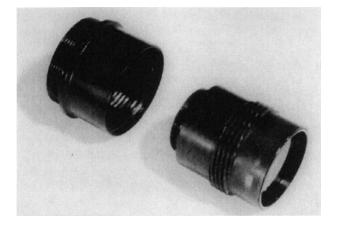


Fig. 4.46 Mating threaded parts for a multiple lead thread on an eyepiece. The six parallel grooves are spaced at 0.0625-in. (1.578-mm) intervals so are equivalent to 16 tpi (0.630 tpmm).

Consumer telescopes and binoculars utilize different means for focusing on objects at different distances. Since there is usually no reticle pattern to keep in focus, either the eyepiece(s) or the objective(s) can be moved for this purpose. The classic design for focusable binoculars, as exemplified by Fig. 4.47, moves both eyepieces simultaneously along their axes as the knurled ring located on the central hinge is rotated. One eyepiece has individual focus capability to allow accommodation errors between right and left eyes to be balanced. The eyepieces in this design slide in and out of holes in the cover plates on the prism housings. It is very difficult to seal the gaps between the eyepieces and these plates adequately. Most commercial instrument designs do not attempt to do so and the interior of the instrument eventually becomes contaminated.

Figure 4.48 shows an individually focusable eyepiece for a low-cost commercial binocular in which the entire internal lens cell rotates on a coarse thread to move axially for diopter adjustment. This rotation may cause the line of sight of the lenses to nutate because of residual optical wedge effects. Eye strain can result over an extended period of use.

Another simple eyepiece is shown in Fig. 4.49. Its focus mechanism is typical of many designs used in military binoculars and telescopes. The basic difference from that in Fig. 4.48 is that the exposed eye lens is sealed and heavy grease is applied to the thread in an attempt to seal the rotary joint. This reduces intrusion of moisture and dust and improves the feel of the motion, but causes serious problems at low temperature when the grease may become very stiff.

Example 4.4: Thread pitch for less than one turn rotation of an eyepiece focus ring to produce a given number of diopters change. (For design and analysis, use File 4.4 of the CD-ROM).

A focus ring is to be turned 240 deg to produce 4 D focus change of an eyepiece with focal length of 1.110 in. (28.1940 mm). (a) What pitch is needed if a standard single thread is used? (b) What would be the pitch of a 6 lead thread? (c) What are the threads per unit length?

From Eq. (4.8):
$$\Delta_{\mathcal{E}} = \frac{1.110^2}{39.37} = 0.0313$$
 in./diopter (0.795 mm/diopter).
(a) Single thread: $p = \left(\frac{360}{240}\right) (4) \left(\frac{0.0313}{1}\right) = 0.1878$ in. (4.7701 mm).
(b) Lead thread: $p = \left(\frac{360}{240}\right) (4) \left(\frac{0.0313}{6}\right) = 0.0313$ in. (0.795 mm).
(c) For (a): threads/unit length $= \frac{1}{0.1878} = 5.325$ tpi (0.210 tpmm).

For (b): threads/unit length $=\frac{1}{0.0313}=31.949$ tpi (1.258 tpmm).

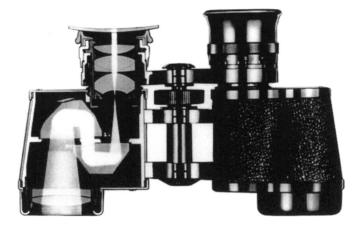


Figure 4.47 A consumer binocular of traditional design with focusable eyepieces. (Courtesy of Carl Zeiss, Inc. Germany.)

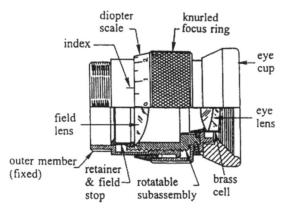


Figure 4.48 A simple focusing eyepiece in which the lens cell rotates on a coarse thread. (Adapted from Horne.¹⁷)

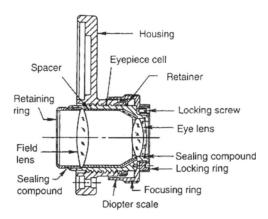


Figure 4.49 Another focusing eyepiece in which the lenses rotate to focus. (From a U.S. Army drawing.)

In the more complex eyepiece shown in Fig. 4.50, the entire internal lens cell slides axially without turning. The latter configuration has the performance advantage of maintaining better line-of-sight centration when refocused and it can be sealed better than any with a rotating cell. This design is used in the military Binocular M19 discussed earlier.

Here the lenses are mounted in a cell (11) that slides axially to focus when the knurled ring (28) is turned on thread (29). The stop pin (34) slides in a slot in the housing (13) to prevent rotation of this cell as the knurled ring is turned. The housing typically interfaces with the optical instrument by means of a threaded clamping ring (not shown) that is slipped over the eyepiece housing (13) before the ring (28) is installed. When engaged with a corresponding thread on the instrument housing, this ring presses against the right side of the flange on housing (13) to hold the eyepiece in place. Since there is no mechanical indexing means provided, care must be exercised with this design to ensure that the eyepiece is rotated about its axis so that the reference mark on housing 13 for the diopter adjustment scale is visible in the normal position of use for the instrument before clamping in place. An O-ring in the groove adjacent to the flange seals the eyepiece to the housing.

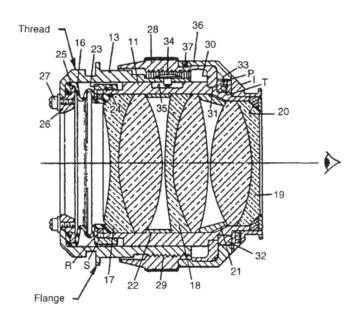


Figure 4.50 Typical construction of a more complex eyepiece for a military telescope. In this design, the lenses translate instead of rotate to focus. (From Quammen et al.²⁹)

Figure 4.51 shows one approach for providing focus in a consumer binocular without involving dynamic seals of the moving parts. Rotation of the focusing ring on the hinge moves an internal lens in each telescope to refocus. Rotation of another knurled ring adjacent to the focus ring biases the position of the moveable lens in one telescope to provide diopter adjustment. All externally exposed lenses are statically sealed to the housings. Rotary seals are provided on the shaft that carries the focusing ring.

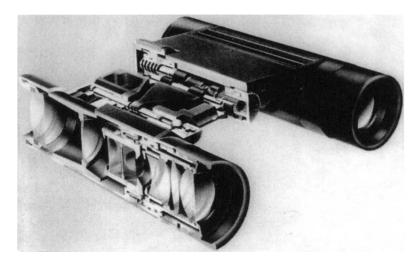


Figure 4.51 A consumer 8×20 pocket binocular with internal focusing mechanisms. (Courtesy of Swarovski Optik K.G., Austria.)

4.11.2 Zoom mechanisms

Zoom lenses for consumer and professional photography and television, as well as military applications are the result of diligent efforts to design mechanisms capable of moving lens groups smoothly, accurately, and quickly along the axis to change the focal length and field of view of the lens from wide-angle to telephoto positions, while maintaining sharp focus. The associated lens designs provide appropriate levels of image quality at reasonably relative apertures over large zoom ranges. Although most of these lenses work in the visible spectral region, infrared zoom telescopes suitable for incorporation into forward-looking infrared (FLIR) systems also have been developed for military and security use. Other zoom systems are described in Chapter 15.

Ashton described the classic design for a zoom lens intended for use with 35-mm motion picture cameras shown in Fig. 4.52.³⁰ This 25 to 250-mm EFL, f/3.6, 10:1 range lens covers a maximum of 45-deg horizontal field and can be focused down to about a 1.2-m (4-ft) object distance. Approximate dimensions of the assembly are a 300-mm (11.8-in.) barrel length and a 150-mm (5.90-in.) diameter. The outermost doublet (left) is fixed; the next air-spaced doublet is movable by an external motor (not shown) to change focus. As in most zoom lenses, the zoom functions are accomplished by moving two groups of lenses. The first group, consisting of a singlet and triplet, moves over a relatively long distance from the telephoto position shown in the figure to a forward position for wide angle use. The second group is a doublet that moves forward from the telephoto position shown to a location where it reverses direction and then moves back to the wide-angle position. The remaining components in the lens system are fixed and serve to bring the image to focus at the film plane.

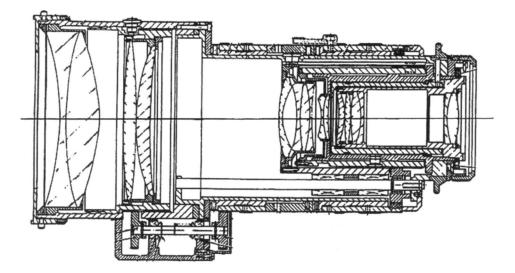


Figure 4.52 Sectional view of a 25- to 250-mm focal length, f/3.6 photographic zoom lens assembly. (Adapted from Ashton.³⁰)

Figure 4.53 is an exploded view of the zoom mechanism. There are three main components: two sleeves (A and B) and a carriage (C). The front zoom group is attached to the front of the carriage. This carriage moves ball bushings in along a rod fitted into the lens housing parallel to the axis. The bushings are laterally spring-loaded to reduce image wander during zooming. A key fixed to the housing rides in a slot in the carriage to prevent rotation of the carriage. The second lens group (doublet) is attached to the front of sleeve B. This sleeve has an external ring gear and a helical cam machined into its outer surface. A cam follower (not shown) attached to the carriage rides in this cam to impart controlled motion to the doublet as the sleeve is rotated by an external motor (not shown). Sleeve A is fixed to the lens housing and carries another cam slot. A cam follower attached to sleeve B engages this slot and moves sleeve B as it rotates. The ring gear on sleeve B is long enough to maintain engagement with the drive gear throughout this axial motion. The movement of the carriage may be seen as the sum of the two cam forms.

Sleeves A and B are a matched subassembly. Raised rings on the outside of sleeve A serve as bearing surfaces against the inner diameter of sleeve B during zoom motion. Both mating surfaces are hard anodized for good wear characteristics; the rings on sleeve A are diamond-turned to fit closely and smoothly within the honed inside surface of sleeve B. Permissible clearance between these bearing surfaces is 7 to 10 μ m (0.0003 to 0.0004 in.). A smaller clearance causes too much torque resistance and poor wear. Larger clearance causes the image to jump and go out of focus when the zoom motion reverses. Cam slots are diamond-turned to reproduce a master contour form. Cam followers are polyurethane and fit the slots without clearance to eliminate backlash. The lenses are mounted into aluminum cells and held by threaded retaining rings. The cells are, in turn, attached to the aluminum sleeves and carriage.

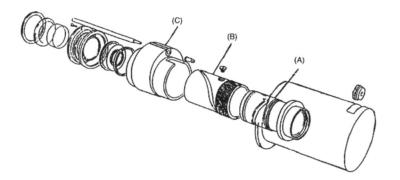


Figure 4.53 Exploded view of the zoom mechanism for the lens of Fig. 4.52. (Adapted from Ashton.³⁰)

A newer concept for designing a zoom system is represented by the 5:1 afocal zoom attachment for a military forward-looking infrared (FLIR) sensor operating in the spectral range from 8 to $12 \mu m$.³¹ The optomechanical design is shown in Fig. 4.54(a) and (b). The first element is fixed, as are the smaller lenses at right. The moveable lenses are designated Groups 1 (air spaced doublet) and 2 (singlet). All of these lenses are made of germanium, as is the second small fixed lens. The other small fixed lens is zinc selenide. There are four aspherics in the design, whose image quality would meet all requirements

over the specified temperature and target distance ranges if the locations of the moveable lens groups could be reoptimized for each combination of operating parameters. The usual technique of driving the lens motions by one or two mechanical cam(s) will not suffice here because there are too many variables.

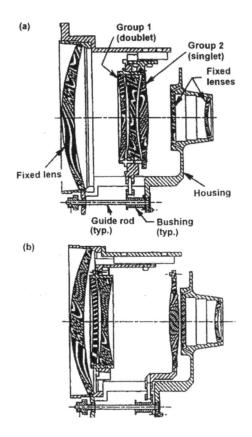


Figure 4.54 Optomechanical layout for an athermalized zoom system, (a) wide angle setting and (b) telephoto setting. (From Fischer et al.³¹)

To athermalize this design, the moveable lens groups are attached to linear bushings that slide on guide rods. Two stepper motors acting through appropriate gear trains drive them independently. See Fig.4.55(a). The motors are controlled during operation by a local microprocessor. The operator commands the magnification to be provided and the target range. The electronics system then refers to a look-up table stored in a built-in erasable programmable read only memory (EPROM) to determine the appropriate settings for the moveable lenses at room temperature. Thermistors attached to the lens housing sense the temperature of the assembly. Signals from these sensors are used by the electronics to select, from a second look-up table stored in the EPROM, the required refinements of the lens settings to correct for temperature effects on system focus. The corrected signals then drive the motors to position the lenses for best imagery at the measured temperature. The lens group motions vary as functions of magnification and temperature as indicated in Fig. 4.55(b). Similar relationships exist for group motion variations as functions of magnification and target range at constant temperature.³¹

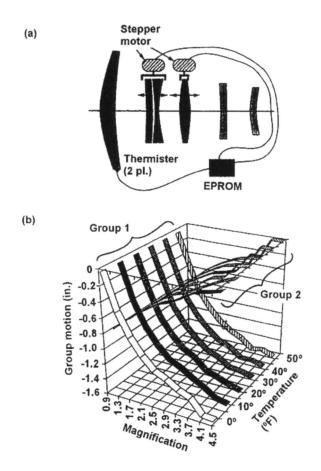


Fig. 4.55 (a) Lens motion control system for the zoom system of Fig. 4.54. (b) Motions of lenses as functions of temperature and magnification for a constant target range. (From Fischer et al.³¹)

4.12 Sealing and Purging Lens Assemblies

Sealing is an important aspect of optical instrument design. The primary purpose is to keep moisture, dust, and other contaminants from entering and depositing on optical surfaces, electronics, or delicate mechanisms. The need for protection from adverse environments depends on the intended use. Military and aerospace optical equipments are subjected to very severe environmental exposures; the optics used in scientific or clinical laboratories and in consumer applications, i.e., interferometers, spectrographs, microscopes, cameras, surveyor's transits, binoculars, laser copiers and printers, and compact disc players, usually experience much more benign environments. Low-cost instruments may have few, if any, provisions for sealing.

Sealing exposed windows and lenses with cured-in-place elastomeric gaskets or Orings (see Fig. 2.20) provides static protection from the environment at normal temperatures and pressures. Seals for hard-vacuum applications can be created with preformed gaskets made of resilient metals such as lead, gold, indium, or gold-plated Inconel.^{32,33} Some examples of window seals are shown in Chapter 5. Nonporous materials are preferred for housings and lens barrels. Castings usually need impregnation with thermosetting plastic resins to seal pores. O-rings made of Viton may be the best for long-term reliability.

Exposed sliding and rotating parts are frequently sealed to fixed members of the instrument with dynamic seals such as O-rings, glands with formed lips (for example, "quad rings"), rolling diaphragms, or flexible bellows made of rubber or metal. See Fig. 2.21. In the eyepiece module for the Binocular M19 shown in Fig. 4.50, a dual-purpose rubber bellows seals the focusing lens cell to a fixed housing at the left (not shown) as well as to the innermost lens at the left of the cell, while the outermost lens at the right of the cell is sealed statically with elastomer. The moving inner subassembly (cell and lenses) slides rather than turns when the focusing ring is turned. A fixed pin riding in a slot in the moving part prevents rotation.²⁹

Many instruments and some subassemblies thereof are flushed with dry gas, such as purified nitrogen or helium, after sealing. A pressure differential above ambient of perhaps 5 lb/in.^2 ($3.4 \times 10^4 \text{ Pa}$) is sometimes generated within the instrument to help prevent intrusion of contaminants. Access through the instrument walls may, in such cases, be provided by a spring-loaded valve that is basically similar in function to those used on automobile tires. Access for flushing nonpressurized instruments can be provided by threaded through holes into which seal screws (typically round or pan head machine screws with O-rings under the head) are inserted after flushing. Applications of seal screws, O-rings, and injected elastomers as seals in a military binocular are shown in Fig. 4.31.

The internal cavities of sealed instruments, such as the housings of an aerial camera lens, should be connected by leakage paths to the main cavity (small bored or cast-in holes inside housing walls, grooves through the edges of lenses, spacers with tabs or vents, etc.) in order for the flushing process to work properly. An example is discussed in Section 4.4 and shown in Fig. 4.13.

Removal of air, moisture, and/or products of outgassing from these ancillary cavities is facilitated if the instrument is evacuated and backfilled two or three times with the dry gas. Baking the instrument at a slightly elevated temperature for several hours also tends to vaporize moisture and expedite stabilized outgassing of volatile materials. To prevent potentially harmful pressure changes that are due to temperature changes, sealed instruments that do not have sturdy walls can be allowed to "breathe" through desiccators and dust filters.¹¹ An example of the latter construction is discussed in Section 15.11.

4.13 References

- 1. Hopkins, R.E., "Some thoughts on lens mounting," Opt. Eng. 15, 1976:428.
- 2. Brockway, E.M., and Nord, D.D., "Lens axial alignment method and apparatus," U.S. *Patent No. 3,507,597*, issued April 21, 1970.
- 3. Westort, K.S., "Design and fabrication of high-performance relay lenses," *Proceedings* of SPIE **518**, 1984:21.
- 4. Addis, E.C., "Value engineering additives in optical sighting devices," *Proceedings of SPIE* **389**, 1983:36.
- 5. Price, W.H., "Resolving optical design/manufacturing hang-ups," *Proceedings of SPIE* 237, 1980: 466.
- 6. Yoder, P. R., Jr., "Lens mounting techniques," Proceedings of SPIE 389, 1983:2.

- 7. Bayar, M., "Lens barrel optomechanical design principles," Opt. Eng. 20, 1981:181.
- 8. Vukobratovich, D., "Design and construction of an astrometric astrograph," *Proceedings of SPIE* **1752**, 1992, 245.
- 9. Valente, T.M. and Richard, R.M., "Interference fit equations for lens cell design using elastomeric lens mountings," *Opt. Eng.*, 33, 1994: 1223.
- Vukobratovich, D., Valente, T.M., Shannon, R.R., Hooker, R. and Sumner, R.E., "Optomechanical Systems Design," Chapt. 3 in *The Infrared & E lectro-Optical Systems Handbook*, Vol. 4, ERIM, Ann Arbor and SPIE Press, Bellingham, WA, 1993.
- 11. Yoder, P.R., Jr., Opto-Mechanical Systems Design, 3rd ed., CRC Press, Boca Raton, 2005.
- 12. Hopkins, R.E., "Lens Mounting and Centering," Chapt. 2 in *Applied Opt ics and Optical Engineering*, Vol VIII, Academic Press, New York, 1980.
- Carnell, K.H., Kidger, M.J., Overill, M.J., Reader, A.J., Reavell, F.C., Welford, W.T., and Wynne, C.G., "Some experiments on precision lens centering and mounting," *Optica Acta*, 21, 1974: 615. (Reprinted in *Proceedings of SPIE* 770, 1988: 207.
- 14. Fischer, R.E., "Case study of elastomeric lens mounts," *Proceedings of SPIE* **1533**, 27, 1991.
- 15. Bacich, J.J., "Precision lens mounting," U.S. Patent No. 4,733,945, issued March 29, 1988.
- 16. Palmer, T.A. and Murray, D.A., personal communication, 2001.
- 17. Horne, D.F., Optical Production Technology, Adam Hilger, England, 1972.
- 18. Scott, R.M., "Optical Engineering," Appl. Opt., 1, 1962:387.
- 19. Vukobratovich, D., "Modular optical alignment," Proceedings of SPIE 376, 1999:427.
- 20. Trsar, W.J., Benjamin, R.J., and Casper, J.F., "Production engineering and implementation of a modular military binocular," *Opt. Eng.* 20, 1981:201.
- 21. Visser, H. and Smorenburg, C., "All reflective spectrometer design for Infrared Space Observatory" *Proceedings of SPIE* **1113**, 1989:65.
- 22. Schreibman, M. and Young, P. "Design of Infrared Astronomical Satellite (IRAS) primary mirror mounts," *Proceedings of SPIE* **250**, 1980:50.
- 23. MIL-HDBK-141, *Optical Desi gn*, Section 19.5.1, Defense Supply Agency, Washington, D.C., 1962.
- 24. Lytle, J.D., "Polymeric optics," Chap. 34 in OSA Handbook of Optics Vol. II, Optical Society of America, Washington, 1995:34.1.
- Betinsky, E.I. and Welham, B.H., "Optical design and evaluation of large asphericalsurface plastic lenses," *Proceedings of SPIE* 193, 1979:78.
- 26. U.S. Precision Lens, Inc., *The Handbook of Plastic Optics*, 2nd. Ed., Cincinnati, OH, 1983.
- 27. Jacobs, D.H., Fundamentals of Optical Engineering, McGraw-Hill, New York, 1943.
- 28. Kittell, D., Precision Mechanics, SPIE Short Course, 1989.
- 29. Quammen, M.L., Cassidy, P.J., Jordan, F.J., and Yoder, P.R., Jr., "Telescope eyepiece assembly with static and dynamic bellows-type seal," U.S. Pat ent No. 3,246,5 63, issued April 19, 1966.
- 30. Ashton, A., "Zoom lens systems," Proceedings of SPIE 163, 1979:92.
- 31. Fischer, R.E., and Kampe, T.U., "Actively controlled 5:1 afocal zoom attachment for common module FLIR," *Proceedings of SPIE*, 1690, 1992:137.
- 32. Manuccia, T.J., Peele, J.R., and Geosling, C.E., "High temperature ultrahigh vacuum infrared window seal," *Rev. Sci. Instr.* 52, 1981, 1857.
- Kampe, T.U., Johnson, C.W., Healy, D.B., and Oschmann, J.M., "Optomechanical design considerations in the development of the DDLT laser diode collimator," *Proceedings of SPIE* 1044, 1989:46

CHAPTER 5 Mounting Optical Windows, Filters, Shells, and Domes

The optical components considered in this chapter do not form images. They are intended either to act as a transparent barrier between the outside environment and the interior of the instrument or, in the case of a filter, to modify the spectral characteristics of the transmitted (or reflected) beam. Typically they have the form of plane parallel plates or meniscusshaped elements (shells and domes). Special cases are conformal ones, i.e., ones whose contours approximate that of the surrounding skin or structural envelope. Candidate materials for all these optics include optical glasses, fused silica, optical crystals, and plastics. With the exception of filters, they have as their prime purposes the exclusion of dirt, moisture, and other contaminants and/or supporting a pressure differential between interior and exterior atmospheres. Critical aspects of mountings for such components include mechanically or thermally induced surface distortions and stresses, as well as sealing. Since most filters are plane parallel plates, their mountings are usually the same as those for flat windows. The location within the optical system of a window or a filter is important because the tolerances on defects such as surface deformation, wavefront tilt, and homogeneity of the refractive index are more stringent near pupils than near images. Tolerances on component cleanliness, material inclusions, and surface blemishes (scratches and digs) are tighter near images than near pupils. In this chapter, we discuss typical mountings for various configurations of the optics of interest.

5.1 Simple Window Mountings

Figure 5.1 shows schematically a typical mounting design for a small circular-aperture window used to seal the interior of an optical system from the outside world. This window is a 20.000-mm (0.787-in.) diameter by 4.000-mm (0.157-in.) thick disk of filter glass. It is intended to be used in the f/10 beam of a military telescope reticle-projection subsystem. The illumination source is located outside the main instrument cavity to facilitate maintenance and to reduce heat input to the other telescope optics. The optical performance requirements are low: surfaces need to be flat only to 10 waves peak-to-valley (p-v) of visible light, and parallel to 30 arcmin.

The window is sealed into its stainless steel (303 CRES) cell with RTV sealing compound. After room temperature curing, this material secures the window in place and forms the seal. Note that the glass is positioned axially against a flat annular shoulder inside the cell and that the sealant fills the annular groove created by undercutting the shoulder, as well as the small annular clearance between the window rim and the cell ID. In this design, the nominal radial gap between glass and metal is 0.050 ± 0.005 in. $(1.270 \pm 0.127 \text{ mm})$. Uniformity of the encapsulating adhesive layer's radial thickness can easily be achieved by temporarily inserting shims between the glass and metal before the adhesive is inserted. The voids left by the shims after removal would be backfilled with sealant. In the application, very little atmospheric pressure differential would be expected, so excessive shear stress would not be applied to the sealant joint.¹

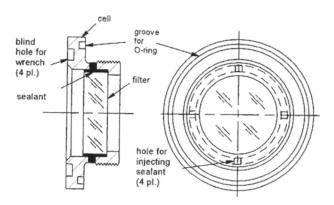


Figure 5.1 Low optical performance glass window constrained by an elastomerically formed-in-place seal.

An alternative configuration for the subassembly of Fig. 5.1, which has only slightly less reliability because of the increased chance of pinholes in the seal, does not have the sealant injection holes. The sealant would then be inserted in either of two ways; (1) applied carefully to the window rim and/or the cell's inside diameter before inserting it into the window, centering it, and registering it against the shoulder; and (2) applied with a hypodermic syringe to the gap between the window and cell with the centered window pressed securely against the shoulder.² With either design, excess sealant should be cleaned from the window and cell surfaces before it cures.* The suitability of the seal can be inferred by pressure proof testing. The external thread on the cell is mated to a threaded hole in the instrument housing. An O-ring would be used between the cell's flange and the housing to seal the interface.

The subassembly shown schematically in Fig. 5.2 has a BK7 window of 50.800-mm (2.000-in.) diameter and 8.800-mm (0.346-in.) thickness into a stainless steel (416 CRES) cell and secured with a threaded retainer that also is made of stainless steel. The subassembly is sealed with a RTV sealant injected into access holes through the cell wall. This window is intended to be used as an environmental seal in front of the objective of a 10-power telescope. The light beam transmitted through this window is collimated and nearly fills the clear aperture at all times, so the critical optical specifications are the transmitted wavefront error (0.25 wave p-v of spherical power and 0.05 wave p-v irregularity for green light) and wedge angle (30 arcsec maximum).

The cell is provided with an annular groove for an O-ring that is used to seal the cell to the instrument housing at the next level of assembly. The dimensions of the groove are shown in the detail view. Note that the mounting holes are outside this seal. Screws used to attach the subassembly would thread into blind holes in the instrument housing. The telescope is to be pressurized to ~5 lb/in.² (~ 3.45×10^4 Pa) with dry nitrogen after assembly. Since the retainer is on the inside, this pressure differential presses the window against the shoulder. Once again, pressure testing is advised to confirm the integrity of the seals.

^{*} Polyurethane foam swabs with pointed or rounded ends can be used for this clean up.

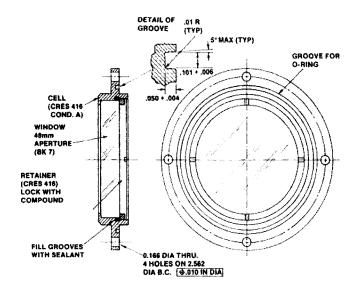


Figure 5.2 Higher optical performance glass window constrained by a threaded retaining ring and sealed with elastomer. Dimensions are in inches.

A vacuum-tight window assembly developed for cryogenic laboratory applications in a double-walled dewar was described by Haycock et al.³ It is illustrated schematically in Fig. 5.3. The window was germanium and had a racetrack-shaped aperture of about 5.25 by 1.30 in. (133.3 by 33.0 mm). Since a hermetic seal was required, a gasket of indium wire was compressed by a spring-loaded piston onto the gap between the heavily beveled rim of the window and the interior edge of the cell as shown in Fig. 5.4. The authors indicated that the deflection of the spring plate provided a total preload [on the order of 530 lb (2350 N)] to hold the window in place at all temperatures between 77 K and 373 K and to create a peak compressive stress in the indium of about 1200 lb/in.² (8.27 MPa).

The spring plate was slit radially around its inner boundary to distribute its force evenly around the edge of the window. Titanium was used for the spring because of its low CTE, high Young's modulus, and high yield strength. This plate functions in the same manner as the circular flange discussed as a lens constraint in Section 3.7. The window frame was Nilo 42 (Ni₄₂Fe₅₈), which approximated the CTE of the germanium. The piston was aluminum for ease of fabrication to its unusual shape.

One important design parameter was the width at the narrower end (bottom) of the triangular gap into which the indium was pressed. A small dimension was needed to maintain the required pressure within the seal, but if too small, it would be difficult to assemble the seal with complete packing of the indium into the volume. It was found that a dimension of 0.010 in. (0.254 mm) was satisfactory for this application. Another critical dimension was the gap on either side of the piston under spring load. A value of 0.001 in. (25 μ m) was found to be appropriate to minimize extrusion of the indium at higher temperatures so the seal would remain intact over time periods of 1 week. Testing of the assembly at cryogenic temperature indicated that it was leak proof to the accuracy of the test apparatus (10⁻¹⁰ std atm cc/s) during and after repeated cycling (> 200 cycles) throughout the temperature range of 293 K to 77 K.

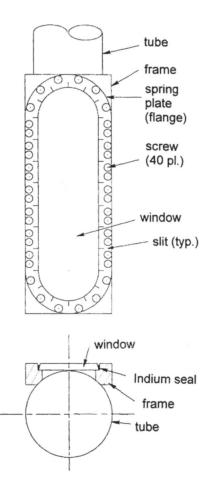


Figure 5.3 Plan and end views of a cryogenic window assembly with pressure-loaded indium seal. (Adapted from Haycock et al. 3)

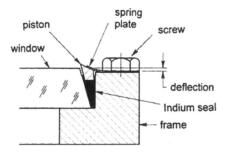


Figure 5.4 Detail view of the pressure-loaded indium seal for the window assembly of Fig. 5.3. (Adapted from Haycock et al.³)

5.2 Mounting "Special" Windows

In this "special" category included are windows for military electro-optical sensors [such as forward-looking infrared (FLIR) systems, low-light-level television (LLLTV) systems, laser range finder/target designator systems], for aerial and space based reconnaissance and mapping cameras, and for optical systems in deep submergence vehicles. We do not consider windows used in high-energy laser systems because an adequate explanation of their complex design and their unique mounting problems would be too lengthy. Readers interested in this type of window are referred to the voluminous literature on laser-induced effects on optical materials, including papers by Holmes and Avizonis,⁴ Loomis,⁵ Klein,⁶⁻⁸ Palmer,⁹ Weidler,¹⁰ and the published reports from the annual symposium on laser-induced damage in optical materials (commonly known as the Boulder Damage Conference). While the latter do not often directly address the problems of mounting windows, they do cover the subject of material damage and thermal effects.

Most airborne electro-optical sensors and cameras are located within environmentally controlled equipment bays in the aircraft fuselage or in a wing-mounted pod. Typically, an optical window is provided to seal the bay or pod and to provide aerodynamic continuity of the enclosure. Its quality must be high and long lasting in spite of exposure to adverse environments. Single- and double-glazed configurations are used as dictated by thermal considerations. Both types are discussed here.

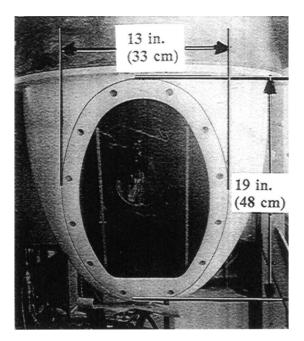


Figure 5.5 An elliptically shaped laminated glass window used in a low-lightlevel television system mounted in an aircraft pod. (Courtesy of Goodrich Corp., Danbury, CT.)

Figure 5.5 shows a window subassembly for a typical LLLTV camera utilizing light in the spectral region of 0.45 to 0.90 µm. Two plane parallel plates of crown glass are laminated together to form a single 19-mm (0.75-in.) thick glazing with an elliptical aperture measuring approximately 25 by 38 cm (9.8 by 15.0 in.). It is mounted in a cast and machined aluminum frame. The frame interfaces with the curved surface of the camera pod and is held in place by twelve screws as shown in the figure. The internal construction of this subassembly is shown schematically in the exploded view in Fig. 5.6. The wires connect to an electrically conductive coating applied to one glass plate before lamination. This coating provides heat over the entire aperture for anti-icing and defogging purposes during a military mission. It also attenuates electromagnetic radiation. The exposed window is susceptible to erosion from impacts of particulate matter, rain, and/or ice, therefore it is designed so the optic can be replaced if it is damaged. The assembled window is sealed to allow no more than 0.1 lb/in.2 (6900 Pa) of air leakage each minute when the pod is pressurized at about 7.5 lb/in.² (5.2×10⁴ Pa) above ambient sea level. The design is also capable of withstanding, without damage, a proof pressure differential of 11 lb/in.² (7.6×10⁴ Pa) in either direction. Both exposed surfaces of the window are broadband antireflection coated.

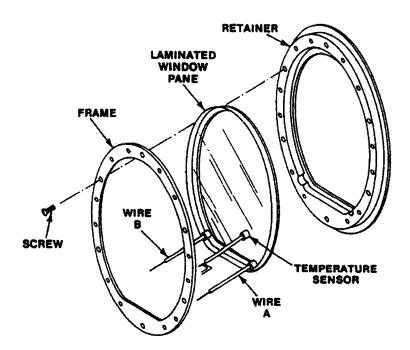


Figure 5.6 Exploded view of the window subassembly shown in Fig. 5.5. (Courtesy of Goodrich Corp., Danbury, CT.)

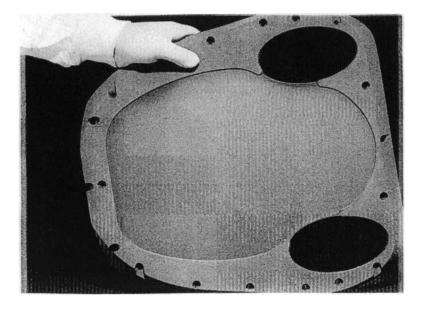


Figure 5.7 A multiaperture window subassembly. The larger element is IRtransmitting zinc sulfide while the smaller ones are BK7 optical glass. (Courtesy of Goodrich Corp., Danbury, CT.)

The multiaperture window assembly shown in Fig. 5.7 is designed for use in an airborne military application involving a FLIR sensor operating in the spectral region 8 to 12 μ m and a laser range finder/target designator system operating at 1.06 μ m. The larger window is used by the FLIR system and is made of a single plate of chemical vapor-deposited (CVD) zinc sulfide (ZnS) approximately 1.6-cm (0.63-in.) thick. Its nominal aperture is 30 by 43 cm (11.8 by 16.9 in.). The smaller windows are similar to each other and have elliptical apertures nominally 9 by 17 cm (3.5 by 6.7 in.). They are used by the laser transmitter and receiver systems and are made of BK7 glass 1.60-cm (0.63-in.) thick. All surfaces are appropriately antireflection coated for maximum transmission at the specified wavelengths at a 47-deg \pm 5-deg angle of incidence. Robinson et al.¹¹ indicated that the coatings also resist erosion caused by rain at a rate of ~1 in. (~2.5 cm) per hour with an impact velocity approaching 500 MPH (224 m/sec) for at least 20 min. The specifications for transmitted wavefront quality are 0.1 wave p-v at 10.6 μ m over any 2.5-cm (1-in.) diameter instantaneous aperture for the FLIR window and 0.2 wave p-v power, 0.1 wave irregularity at 0.63 μ m over the full apertures for the laser windows.

The CVD ZnS used in this design is not an easy material with which to work. Fortunately, it transmits in the visible adequately enough for the optician to identify the volume within an oversized raw material blank where the element should be located in order to avoid the worst inclusions and bubbles. The mechanical strengths of the ZnS and BK7 glazings are maximized by controlled removal of material using progressively finer abrasives during grinding, as described by Stoll et al.,¹² to ensure that all subsurface damage caused by previous operations has been removed. This process is called "controlled grinding" and specifies removal of material with each grade of abrasive to a depth of three

times the prior grit diameter. The optical wedge is brought within specified limits during this grinding process. The edges of all windows are control ground and cloth polished, primarily to maximize strength. This multistep grinding followed by polishing minimizes the risk of breakage from forces imposed by the mounting, shock and vibration, or temperature changes. All three glazings are bonded with adhesive into a lightweight frame made of 6061-T651 aluminum plate, and anodized after machining to the complex contours shown in Fig. 5.7. The bonded assembly is attached to the aircraft pod by screws through several recessed holes around the edge of the frame. The mating surfaces of the frame and pod as well as the hole pattern must match closely in order not to deform the optics or disturb the seals during integration and/or exposure to adverse environments.

5.3 Conformal Windows

Figure 5.8 illustrates schematically a segmented window subassembly typical of those used with panoramic aerial cameras designed to photograph from horizon to horizon transverse to the flight path of a military aircraft. The size of the window required for use with such a camera is determined primarily by the size of the lens's entrance pupil and the camera's instantaneous field of view. The exterior envelope of the window shown here conforms generally with that of the skin on the surrounding structure. Hence, this type of window is called a *conformal* one.

One example of the type of window shown in Fig. 5.8 has dual-glazing construction with fused silica glazings outside and BK7 glass glazings inside (see Fig. 5.9). Because the aircraft flies very fast, a very critical design problem here is thermal. At high velocities, boundary-layer effects heat the outer window glazing; that material acts as a blackbody with an emissivity of about 0.9 so it would normally radiate heat into the camera and its surrounding equipment. To combat this deleterious effect, the inside surfaces of the outer glazings are coated with a thin layer of gold having low emissivity and high visible-light transmission. All other window surfaces are conventionally antireflection coated to maximize transmission in the sensitivity spectral region of the film or detector.

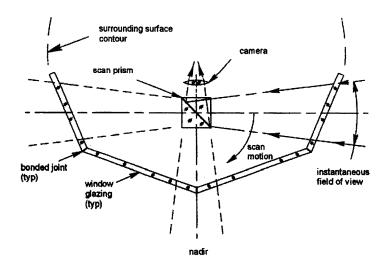


Figure 5.8 Schematic showing how a segmented conformal window can be used with a panoramic aerial camera to photograph from horizon to horizon.

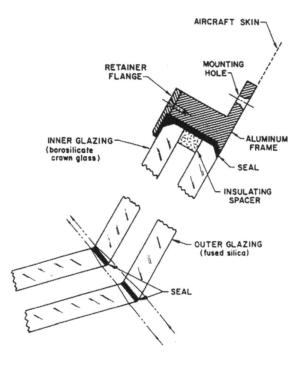


Figure 5.9 Schematic of a double-glazed segmented window of the type shown in Fig. 5.8. Refraction through the tilted glazings minimizes obscuration at the bonded joints without affecting field coverage. (From Yoder.¹)

The square glazings in the center of this assembly have dimensions of approximately 12.6 by 13.0 in. (32 by 33 cm) and are 0.40-in. (1.0-cm) thick. The side glazings are smaller in one dimension, and all of the glazings are separated by a few millimeters. Conditioned air from the aircraft is circulated through the space between the glazings before and during flight to help stabilize the temperature within the camera region. The adjacent edges of the individual elements in each of the inner and outer glazings of this segmented window are beveled and polished. These edges are bonded with a semiflexible adhesive. The glazings are sealed into recesses machined into the aluminum frame with a RTV sealant and secured by a metal retainer flange. The contours of the subassembly and its mounting-hole pattern are made to match those of the aircraft interface by reference to special match tooling and fixtures.

Other important applications of conformal windows are in electro-optical sensors built into vehicle surfaces and in guidance systems for missiles. Figure 5.10 shows two possible installations of the first type. View (a) illustrates a cylindrical window installed into the leading edge of an aircraft wing while view (b) shows a toroidal meniscus window installed into the curved skin of a missile. These windows typically are meniscus in shape. Aberration correcting optical components may be needed in the optical system following the windows to compensate for obliquity of the beam paths as they refract through the windows.¹³ Mounting these windows would pose no problems not addressed in our earlier considerations of lens mountings.

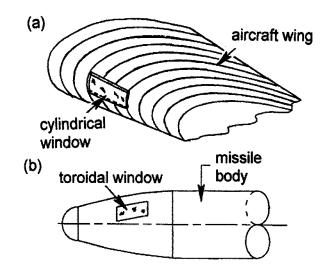


Figure 5.10 Conformal window configurations: (a) cylindrical window installed in an aircraft wing, (b) toroidal window installed in conical skin of a missile.

For applications to the nose of a missile, windows comprising a series of triangular flat plates represent an early attempt to shape the window so it conforms to the natural shape of the skin. The aerodynamic aspects of this approach are very poor. Surface heating at high velocity, especially at the point, can destroy the windows. The molybdenum (TZM) tip shown at the apex of the flat-plate window in Fig. 5.11(a) provides some protection, but is not a complete solution. Joining the segments would be problematic because adhesives do not survive well at very high temperatures.¹⁴ Hemispherical domes were used on early models, but domes shaped as half-ellipsoids, as suggested by Fig. 5.11(b), were found to be better approximations to the missile's skin contour. They allow a higher velocity of flight and reduce drag; therefore increasing the useful range to target.

The image quality obtained by an optical sensor looking through a hemispherical dome is generally unchanged with the scan angle if the entrance pupil of the sensor is located at the center of the spherical surfaces. However, a similar situation involving an ellipsoidal dome can give poor imagery. The reason is suggested in Fig. 5.12(a) and (b). Looking straight ahead, the ellipsoidal element is symmetrically disposed about the axis of the sensor. Axially symmetric correctors could be used in the sensor's optical system to optimize imagery. However, looking obliquely, as in view (b), refraction is unsymmetrical, so performance is compromised; astigmatism is the worst defect, with spherical aberration and coma following close behind.¹⁵ Fig. 5.12(c) shows one approach for restoring good image quality. The two corrector elements are fixed while the sensor optics, including the detector and its window, are mounted in a two-axis gimbal system so it can scan the line of sight.^{16,17}

Techniques for mounting ellipsoidal domes into missiles are generally complex because of the aerodynamic aspects and the high temperatures involved. The subject is discussed briefly in Section 5.6. Manufacture of hemispherical and ellipsoidal domes is addressed by Harris.¹⁸

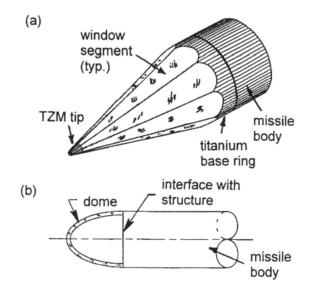


Figure 5.11 Conformal windows for missile applications: (a) triangular flat plates (adapted from Fraser and Hemingway¹⁴), and (b) ellipsoidal dome.

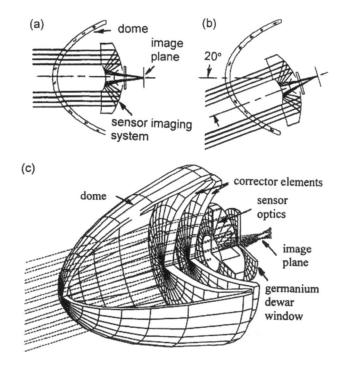


Figure 5.12 Sensor configuration using an ellipsoidal dome: scan angle at (a) at 0 deg, (b) -10 deg. (Adapted from Knapp et al.¹⁶ and from Trotta.¹⁷)

5.4 Windows Subject to Pressure Differential

5.4.1 Survival

When a plane-parallel circular window is subjected to a pressure differential of ΔP applied uniformly over an unsupported aperture A_W , the minimum value for its thickness t_W to provide a safety factor of f_S over the material's fracture strength S_F is as given by Eq. (5.1):

$$t_{W} = \left[0.5A_{W}\right] \left[\frac{\left(K_{W}f_{S}\Delta P\right)}{S_{F}}\right]^{1/2},$$
(5.1)

where: K_W = a support condition constant = 1.25 (if unclamped) or 0.75 (if clamped).

Figure 5.13 shows the geometry defining these two conditions. The unclamped condition applies approximately if the plate is supported in a ring of elastomer as described in Section 3.10. Maximum stress occurs at the center of this window. The other condition applies if it is constrained by a threaded ring or flange. Maximum stress then occurs at the edge of the clamped area. The customary (conservative) value for f_S is 4. Typical values for S_F at room temperature for some commonly used infrared-window materials as given by Harris¹⁸ are listed in Table B16. Example 5.1 illustrates this type of calculation.

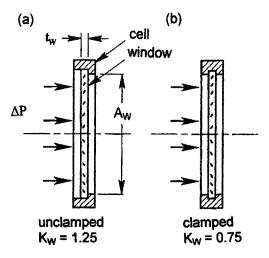


Figure 5.13 Schematics of: (a) an unclamped circular window and (b) a clamped circular window. (Adapted from Harris.¹⁸)

Dunn and Stachiw¹⁹ investigated the thickness-to-unsupported diameter ratio t_W/A_W for the relatively thick plane parallel plate and conical-rim windows typically used in deep submergence vehicles and that experience large pressure differentials. The material they considered was Rohm and Haas grade B plexiglas (polymethylmethacrylate). The varied parameters included diameter, thickness, pressure differential, mounting flange configuration, and in the case of the conical windows, they included the cone angle (from

30 deg to 150 deg). During testing, the pressure was increased at a rate of 600 to 700 lb/in.² (4.13 to 4.83 MPa) per minute to failure. The cold-flow displacement (extrusion) of the window material into the lower-pressure space was also measured. The strength of conical windows was found to increase nonlinearly with the included cone angle, the greatest improvements coming at the lower end of the angular range. Flat windows and 90 deg conical windows of the same (t_W/A_W) ratio were found to fail at approximately the same critical load. Typically, a 1.00-in. (2.50-cm) aperture 90-deg window with $t_W/A_W = 0.5$ failed at about 16,000 lb/in.² (1.10 MPa). The authors concluded from their study that the failure pressure-differential scales as the t_W/A_W ratio.

Two typical high-pressure window-design configurations are shown in Fig. 5.14. The 90-deg conical window of view (a) is supported around its entire rim, and its inner surface is flush with the smaller end of the conical mounting surface. The retainer compresses the neoprene gasket enough to constrain the window at low pressure differentials. The flat window in view (b) is sealed with an O-ring midway along its rim and it is kept from falling out at a zero pressure differential by a retaining ring. The rims of both windows are coated with vacuum grease prior to assembly. Dunn and Stachiw did not specify complete details

Example 5.1: Thickness required of in a circular plane parallel plate window to safely withstand a given pressure differential. (For design and analysis, use Files 5.1 of the CD-ROM).

A sapphire window with unsupported aperture of 14 cm (5.51 in.) is subjected to a pressure differential of 10 atm (1.013 MPa or 147.0 lb/in.²). Assuming a safety factor f_s of 4, (a) what should be the window thickness if potted into its mount with a ring of RTV (i.e., unclamped)? And (b) what should be the thickness if the window is clamped?

From Table B16, the minimum S_F for sapphire is approximately 300 MPa (45,511 lb/in.²).

(a) Unclamped: by Eq. (5.1):

$$t_{W} = \left[(0.5)(14) \right] \frac{\left[(1.25)(4)(1.013) \right]^{1/2}}{300} = 0.909 \text{ cm}(0.358 \text{ in.}).$$

(b) Clamped: by Eq. (5.1):

$$t_{W} = \left[(0.5)(14) \right] \left[\frac{(0.75)(4)(1.013)}{300} \right]^{\sqrt{2}} = 0.704 \text{ cm} (0.277 \text{ in.}).$$

Note, by examination of Eq. (5.1):

 $\frac{t_{W \text{ CLAMPED}}}{t_{W \text{ UNCLAMPED}}} = \left(\frac{0.75}{1.25}\right)^{1/2} = 0.775.$

Therefore, the answer to (b) is (0.775)(0.909) = 0.704 cm.

for these operational designs, but one can infer that the tolerances reported for the experimental versions might also apply. Accordingly, conical angles would be held to \pm 30arcmin, minor diameters of conical windows to \pm 0.001 in. (\pm 25 µm), and the surface finish of window rims and mating metal surfaces to 32 rms. A radial clearance of 0.005 to 0.010 in. (0.13 to 0.25 mm) should typically be provided around flat windows.

These parametric study results typically indicate that for $A_W = 4.0$ in. (10.2 cm), the windows should be ~2.0-in. (~5.1-cm) thick. Failure would be anticipated at about 4000 lb/in.² (27 MPa). The window material would be expected to extrude through the aperture of the mount by about 0.5 in. (1.3 cm) at the time of failure. The authors wisely advised proof-testing all windows intended for man-rated applications. Logically, the test level should exceed any anticipated exposure during operation. In addition, some representative samples should be tested to failure so as to verify the safety factor of the design.

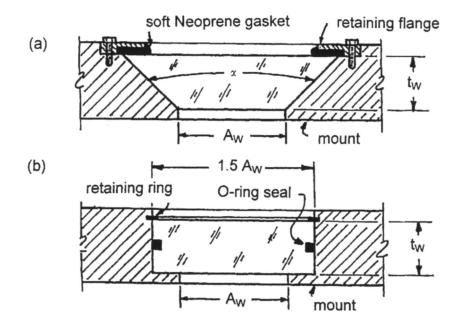


Figure 5.14 Typical plane-parallel plate polymethylmethacrylate window configurations for high-pressure application in deep submergence vehicles: (a) 90-deg conical rim and (b) cylindrical rim. (From Dunn and Stachiw.¹⁹)

The ability of a thin shell or dome to withstand a pressure differential was addressed by Harris.¹⁸ The pertinent geometry is as shown in Fig. 5.15. The thickness of the optic is t_W , its spherical radius R_W , its diameter D_G , and its included angle 2 θ . It is either simply supported or clamped, as in the case of the plane-parallel plate window. The stress S_W generated in the optic by a uniform pressure applied to the outside surface is compressive if the window is simply supported and tensile if it is clamped. We only need consider the latter case because glass fails at a lower stress level under tension. The following equation attributed by Harris¹⁸ to Pickles and Field²⁰ applies:

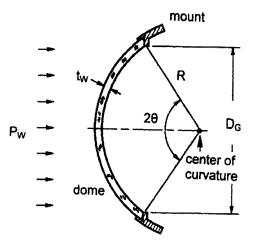


Figure 5.15 Geometry of a dome with simply supported base subjected to a pressure differential. (Adapted from Harris.¹⁸)

$$S_{W} = \left[\frac{\left(R_{W}\right)\left(\Delta P\right)}{2t_{W}}\right] \left\{\cos\theta \left[1.6 + 2.44\sin\theta \left(\frac{R_{W}}{t_{W}}\right)^{1/2}\right] - 1\right\}.$$
(5.2)

We see how to solve a typical design problem with this equation in Example 5.2.

The ability of a window to survive imposed stress is determined largely by the condition, i.e., presence of scratches, digs, and subsurface cracks in its surfaces. Optics that have been improperly manufactured (i.e., possesses residual internal stress), mishandled, or exposed to impacts from dust, sand, rain, hail, etc., are much more likely to fail than ones with pristine surfaces. Larger windows have a higher probability of having defects than smaller ones simply because they have greater areas. Hence, these larger windows may fail at a lower stress level than smaller ones with equivalent defects. Or, they may fail more quickly—at a given stress level—than a smaller one. This is because failure usually results from growth of defects over time that was originally too small to be a problem. The rate of this growth depends, in part, on the relative humidity of the surrounding atmosphere. Higher moisture content leads to higher propagation velocity. Statistical methods can be used to predict the probability of failure under given stress if sufficient information is known about that particular optic. Many authors have described these methods. Notable are Vukobratovich²¹, Fuller et al.,²² Harris,¹⁸ and Pepi.²³

An example of the use of these methods is the design and verification of that design for a dual-glazing BK7 window intended for use with high performance photographic equipment aboard a commercial aircraft as described by Fuller et al.²² Figure 5.16 shows a partial cross-section view of the window and its mounting frame. U.S. Federal Air regulations required that 95% confidence of 99% survival probability of the window after at least 10,000 hours (417 days) under full operating conditions. Pepi²³ described, in detail, the design created to meet these requirements. He also described the complex program of analysis and testing that supported the design. It was shown that the design was fail-safe in that, if the outer glazing suffered a catastrophic failure, the inner glazing would survive and maintain the full required pressure differential for at least 8 hours after the outer glazing failure.

Example 5.2: Thickness required of a dome to safely withstand a given pressure differential. (For design and analysis, use File 5.2 of the CD-ROM).

A ZnS dome with exterior radius R = 50.000 mm (1.968 in.) is to be subjected to a uniform compressive pressure differential ΔP of 1.42 MPa (205.95 lb/in.²). The angle θ is 30 deg, so assuming a safety factor of 4, how large must the dome thickness be?

From Table B16, the fracture strength S_F is 100 MPa (14,503 lb/in.²). The allowable S_W is then 100/4 = 25 MPa (3626 lb/in.²). Knowing that Eq. (5.2) is to be solved by iteration, we assume an initial value for t_W and solve for S_W . This repeats until the stress equals the maximum allowed value. Linear interpolation is used between trials.

For
$$t_W = 5.00$$
 mm:

$$S_W = \left[\frac{(50.0)(1.42)}{(2)(5.000)}\right] \left(\cos 30 \deg\left\{1.6 + \left[(2.44)(\sin 30 \deg\left(\frac{50.0}{5.000}\right)^{1/2}\right] - 1\right\}\right) = 26.460 \text{MPa}.$$

For
$$t_W = 5.10 \text{ mm}$$
:

$$S_W = \left[\frac{(50.0)(1.42)}{(2)(5.100)}\right] \left(\cos 30 \text{ deg}\left\{1.6 + \left[(2.44)(\sin 30 \text{ deg})\left(\frac{50.0}{5.100}\right)^{1/2}\right] - 1\right\}\right) = 25.712 \text{ MPa.}$$

Next,
$$t_{W} = 5.00 + \left[\frac{(26.460 - 25.000)(5.00 - 5.100)}{(26.460 - 25.712)} \right] = 5.195 \text{ mm},$$

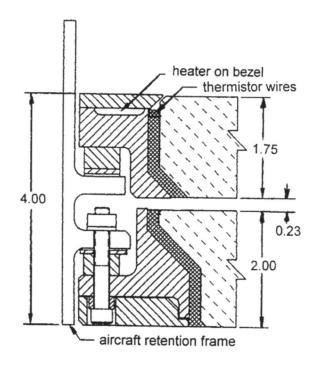
$$S_{W} = \left[\frac{(50.0)(1.42)}{(2)(5.195)}\right] \left(\cos 30 \deg\left\{1.6 + \left[(2.44)(\sin 30 \deg)\left(\frac{50.0}{5.195}\right)^{V^2}\right] - 1\right\}\right) = 25.034 \text{ MPa.}$$

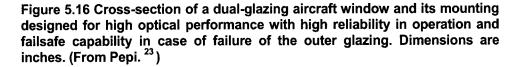
Next,
$$t_{W} = 5.195 + \left[\frac{(25.034 - 25.000)(5.100 - 5.195)}{(25.034 - 25.712)}\right] = 5.200 \text{ mm.}$$

 $S_{W} = \left[\frac{(50.0)(1.42)}{(2)(5.200)}\right] \left(\cos 30 \text{ deg}\left\{1.6 + \left[(2.44)(\sin 30 \text{ deg})\left(\frac{50.0}{5.200}\right)^{1/2}\right] - 1\right\}\right) = 24.999 \text{ MPa.}$

Hence, 5.200 mm is the smallest t_W that meets the stress limitation.

Because the methods applied by Pepi and others lead to the assignment of a reasonable tolerance for stress applied to optics by their mechanical surround that we need in order to decide if a given optomechanical design is acceptable, we will defer detailed consideration thereof until Section 13.2.





5.4.2 Optical effects

The following approximate equation for the optical path difference (OPD) introduced by a rim mounted circular flat plate window deflected by a pressure differential ΔP_W was given by Sparks and Cottis²⁴ per Vukobratovich.²⁵

OPD=0.00889
$$(n-1)\frac{\Delta P_{W}^{2}A_{W}^{4}}{E_{G}^{2}t_{W}^{5}},$$
 (5.3)

where the window aperture is A_W , thickness is t_W , Young's modulus E_G , and index n.

Roark,²⁶ gave a relationship for the deflection Δx of a circular plane-parallel plate at its center where it experiences a pressure differential ΔP_W over its aperture. When rearranged slightly, we obtain Eq. (5.4), which can be applied to an optical window of this same shape:

$$\Delta x = 0.0117 \left(1 - v^2 \right) \frac{\Delta P_w A_w^4}{\left(E_G t_w^3 \right)},$$
(5.4)

where *m* is the reciprocal of Poisson's ratio, E_G is the Young's modulus for the glass, and *P* is the total force applied to the window's area. All other terms were defined earlier. The Δx value can then be compared to the tolerance on surface deflection from the optical system design to see if it is acceptable. Example 5.3 demonstrates the use of Eqs (5.3) and (5.4).

Example 5.3: Deflection and optical performance of a window under a pressure differential. (For design and analysis, use File 5.3 of the CD-ROM.)

(a) Calculate the thickness required of a plane parallel circular N-BK7 window under a pressure differential ΔP of 0.5 atm applied uniformly over its 3.000 in. (76.2 mm) aperture in order to limit the central deflection to 1.0 wave of light at $\lambda =$ 0.633 µm (2.492×10⁻⁵ in.) wavelength. The window is clamped outside its aperture. (b) What OPD is created by this deflection at this same wavelength?

(a) From Table B1:

$$v_G = 0.206$$
 and $E_G = 1.19 \times 10^7$ lb/in.² (8.2×10⁴ MPa).

From Fig. 1.7:

$$n = 1.51509 @ 0.633 \ \mu m$$
,

1.0 wave @ 0.633
$$\mu$$
m= $\frac{(0.633 \times 10^{-3})}{25.4} = 2.492 \times 10^{-5}$ in.

$$\Delta P = 0.5 \text{ atm} = (0.5)(14.7) = 7.35 \text{ lb/in.}^2 (0.051 \text{ MPa}).$$

Rearranging Eq. (5.4):

$$t_{W}^{3} = \frac{(0.0117)(7.35)(3.00^{4})(1-0.206^{2})}{\left[(1.19\times10^{7})(2.492\times10^{-5})\right]},$$

$$W = 0.282$$
 in. (7.163 mm).

t

(b) From Eq. (5.3):

OPD =
$$\frac{(0.00889)(0.5151)(7.35)^2(3.000)^6}{\left[(1.19 \times 10^7)^2(0.282)^5\right]} = 7.014 \times 10^{-10}$$
 in.

At $\lambda = 2.492 \times 10^{-5}$ in.:

OPD =
$$\frac{7.014 \times 10^{-10} \text{ in.}}{2.492 \times 10^{-5}} = 2.8 \times 10^{-5} \text{ wave.}$$

5.5 Filter Mountings

Glass and better-quality plastic absorption filters are employed extensively in such applications as cameras, photometers, and chemical analysis equipment. Glass interference filters used singly or in combination with absorption (blocking) filters are convenient means for isolating specific narrow transmission bands in systems such as those using lasers. They generally require temperature control. Gelatin filters are noted for their low cost and wide variety, but inhomogeneities of optical quality, thickness variations, and surface figure errors, as well as their low mechanical strength and poor durability, limit their use to rather low-performance applications. They are generally used in protected environments. If the gelatin is sandwiched between transparent plates of a more durable material such as glass, physical strength and durability can be greatly improved. Glass filters do not suffer these limitations.

Many applications for optical filters require only that the component be supported approximately centered in and roughly aligned normal to the transmitted light beam. Cell mountings such as the snap-ring, elastomeric, and retaining-ring designs described in Chapter 3 are frequently employed to hold filters in other instruments. A simple mounting for a series of glass disks held in a filter wheel by snap rings, is illustrated in Fig. 5.17. This wheel is driven by a Geneva mechanism that rotates the wheel in 4 steps and holds it in the chosen position until turned to the next position.

Heat-absorbing filters for projectors and other high-temperature applications are typically restrained by spring clips because they allow thermal expansion. Interference filters require precise angular orientation to the beam so strict attention must be paid to that aspect of their mounting design.

Some thin filters are cemented to a refracting substrate (i.e., a window) that provides mechanical rigidity to the subassembly. An example is shown in Fig. 5.18. In this case, a 1.20-mm (0.05-in.) thick sheet of red filter glass is cemented with conventional optical cement to a crown glass window of 7.50-mm (0.30-in.) axial thickness. The 88-mm (3.46-in.) diameter subassembly serves two purposes: it transmits properly in the spectral region characteristic of the filter and it is sufficiently stiff to function as a sealing window.

Another laminated filter is shown in Fig. 5.19. This is a composite filter consisting of a mosaic of narrow bandpass interference filter elements cemented between two 290-mm (11.4-in.) diameter crown glass plates. The nominal thicknesses of the plates and all filters are 6 mm (0.24 in.), and all have the same thickness within 0.1 mm (0.004 in.). Rather than controlling the wedge angles of the filter elements to an extremely tight tolerance, they are made to a "reasonable" wedge tolerance and oriented variously at assembly to minimize average deviation. This is permissible since the filter is intended for a nonimage-forming application. The outside diameter of the filter mosaic is made somewhat smaller than the outside diameters of the windows so that an annular "guard ring" made up of crown glass segments can be cemented between the windows to protect the edges of the interference filter coatings from the environment. The outside diameter of the assembly is edged after cementing.

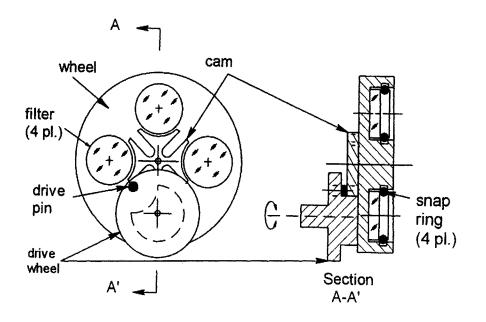


Fig. 5.17 Schematic views of a simple filter wheel with four glass filters held in place with snap rings and driven by a Geneva mechanism.

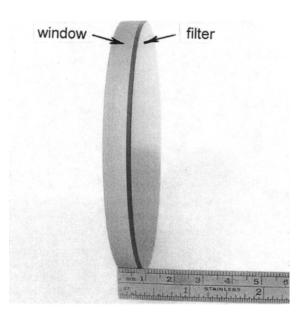


Fig. 5.18 A laminated filter and pressure window.

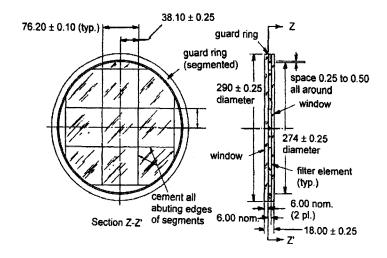


Figure 5.19 Composite filter design comprising a laminated and heated mosaic of interference filter elements. (Courtesy of Goodrich Corp., Danbury, CT.)

Because a narrow-bandpass filter is temperature sensitive, the coating is designed to operate at a temperature of 45°C (113°F), which is above the expected operational ambient. A strip heater is built into the mount to heat the filter from its rim. A thermistor is mounted on one window surface outside the clear aperture; it is used to drive a temperature control electrical circuit elsewhere in the instrument. The filter is designed to have a spectral passband with a nominal full-width-to-half-maximum of 30 Å centered at a specific near-infrared laser wavelength. Out-of-passband radiation is blocked with separate absorption cut-on and cut-off filters of conventional design elsewhere in the system.

The cemented filter assembly is mounted in an aluminum cell and clamped around its edge by a retaining flange secured to the cell with several screws. Two O-rings and a flat gasket seal the assembly. Figure 5.20 shows a sectional view through the mount. The assembly is not intended to be exposed to a significant pressure differential. The cell is insulated thermally from the body of the optical instrument of which it is a part with G10 fiberglass-epoxy resin material.

5.6 Mounting Shells and Domes

Meniscus-shaped windows are usually called "shells" or "domes." They are commonly used on electro-optical sensors requiring access to large fields of view by scanning a line of sight over a large conical space and in wide-field astronomical telescope objectives such as the Bouwers, Maksutov, or Gabor types (see Kingslake²⁷). They also are frequently used as protective windows for underwater vehicles. Hyperhemispheres are domes that extend beyond a 180-deg angular extent. An example is shown in Fig. 5.21. The outside diameter of this optic is 127 mm (5 in.), the dome thickness is 5 mm (0.2 in.), and the angular aperture is approximately 210 deg. This dome is made of crown glass; many other domes are made of infrared-transmitting materials such as fused silica, germanium, zinc sulfide, zinc selenide, silicon, magnesium fluoride, sapphire, spinel, and CVD diamond. With the exception of fused silica and diamond, which are relatively durable, these materials are

susceptible to erosion and damage caused by impacts with water drops, ice, dust, and sand in the atmosphere, especially if moving at a high relative velocity.

Domes are typically mounted on instrument housings by potting them with elastomers or clamping them through soft gaskets with ring-shaped flanges. Figure 5.22 illustrates three of these techniques. Hard mounting these optics against metal mechanical interfaces and constraining them by metal retainers is generally not attempted.

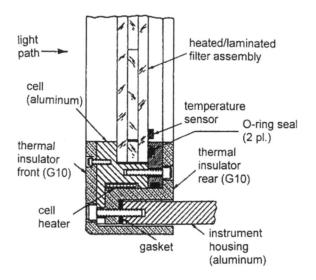


Figure 5.20 Schematic sectional view of the mount for the filter subassembly shown in Fig. 5.19. (Courtesy of Goodrich Corp., Danbury, CT.)

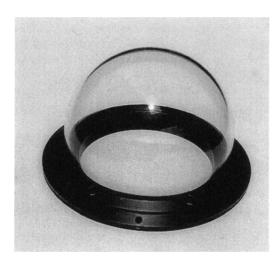


Figure 5.21 A crown glass hyperhemispheric dome potted with elastomer into a metal flange. Its dimensions are given in the text.

Harris¹⁸ discusses methods for designing a dome that would be resistant to the aerodynamic pressure load that would be experienced if the dome were at the nose of a missile or airborne sensor in high-velocity flight. He also explains why those domes reach high temperatures in flight.

In order to solve the latter problem, some very sophisticated designs have evolved. Figure 5.23 from Sunne,³⁰ and Sunne et al.,^{31,32} shows two configurations for ceramic domes (typically sapphire) attached to the cylindrical 6Al-4V titanium housing of an airto-air missile by brazing. In view (a), the base of the dome is butt brazed with Incusil-ABA (an alloy of ~27.25% copper, ~12.5% indium, ~1.25% titanium, and the remaining percentage silver. This alloy melts at ~700°C)^b to a flat surface on an intermediate transition cylindrical ring made of 99% Niobium alloyed with 1% zirconium. This material has a CTE of 4 to 4.5×10^{-6} °C, closely matching sapphire. The crystal's *c*-axis is approximately normal to the dome base. Four locating tabs are machined into the left end of the cylindrical ring to locate the dome during brazing. The right end of the cylindrical ring is inserted into the machined bore of the titanium missile's nose with an interference fit accomplished by heating the titanium and cooling the Niobium for assembly. Upon returning to ambient temperature the metals seize. The joint is then brazed with Gapasil-9 (an alloy of ~82% silver, ~9% palladium, and ~9% gallium that melts at ~930°C). Brazing the two joints is accomplished in a vacuum of $\leq 8 \times 10^{-5}$ torr and in two steps because the melting temperatures of the materials are so different. The metal-to-metal joint is brazed first and the metal-to-ceramic joint is brazed second. The missile housing extends axially beyond the second braze joint to provide aerodynamic continuity of the external surface of the missile. In Fig. 5.23(a), a polysulfide seal is shown filling the gap between the dome and the nose. The transition ring in this design is thin enough to be slightly compliant radially. This allows differential expansion and contraction between the sapphire and titanium (which have significantly different CTEs of $\sim 5.3 \times 10^{-6}$ /°C and 8.8×10^{-6} c, respectively) as the temperature changes and minimizes the chance for breakage of the dome.

In Fig. 5.23(b), an improved brazing method is depicted. Here, the cylindrical flexure ring is integral with the titanium missile housing thereby eliminating the need for the separate precisely machined transition cylinder and its interference fit into the bore ID of the missile nose. It is thin and radially compliant for the reason explained above. Two braze joints are used between the dome base and the flat end of the cylindrical ring. A flat washer of thickness 0.008 in. (0.20 mm) made of a 99% niobium 1% zirconium alloy is brazed to the dome base with Incusil-ABA alloy, while the other side of the washer is brazed to the flat end of the transition ring with Incusil-15 alloy. This material has essentially the same composition as Incusil-ABA, but without titanium. The melting points of the two braze materials are practically the same at approximately 700°C. A titanium aerodynamic shield is brazed to a shoulder on the nose using the Incusil-15 alloy. All three joints are brazed in vacuum at the same time, thereby facilitating manufacture as compared to the design of view (a).

^D Incusil and Gapasil are registered trademarks of WESGO, Inc., San Carlos, CA (www.wesgometals.com).

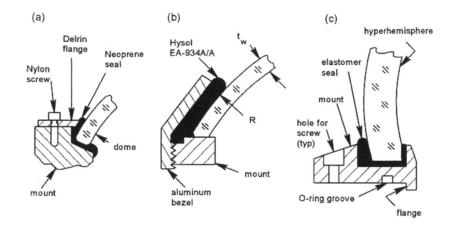


Figure 5.22 Three configurations for shell/dome mountings. (a) A shell clamped mechanically through a soft gasket by a flange. (Adapted from Vukobratovich.²⁸) (b) A shell constrained by an internal threaded retainer. (Adapted from Speare and Belloli.²⁹) (c) A hyperhemisphere potted with elastomer into its mount.

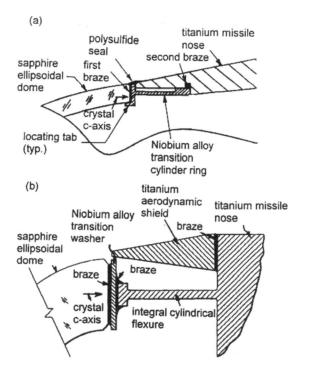


Figure 5.23 Schematic diagrams of mountings for ellipsoidal domes with the optic brazed to the metal mounting. (a) With a separate transition cylinder ring (adapted from Sunne et al.³⁰) and (b) with an integral cylinder ring. (Adapted from Sunne et al.³¹)

Both of the above-described designs with brazed domes have yielded durable dometo-missile joints in production. They are proof tested at a pressure differential of 90 lb/in.² to verify strength and joint integrity. Sunne³² indicated that other ceramics, such as ALON, have been brazed successfully using the techniques described above.

5.7 References

- 1. Yoder, P.R., Jr., "Nonimage forming optical components," *Proceedings of SPIE* 531, 1985: 206.
- 2. Yoder, P.R., Jr., Opto-Mechanical Systems Design, 3rd ed., CRC Press, Boca Raton, 2005.
- Haycock, R.H., Tritchew, S., and Jennison, P., "A compact indium seal for cryogenic optical windows." *Proceedings of SPIE* 1340, 165, 1990.
- 4. Holmes, D.A. and Avizonis, P.V. (1976). "Approximate optical system model," *Appl. Opt.* 15, 1976:1075.
- Loomis, J.S., "Optical quality of laser windows," Proceedings 4th Conference on Infrared Laser Window Materials, Air Force Material Labs, Wright Patterson AFB, 1976.
- 6. Klein, C.A. "Thermally induced optical distortion in high energy laser systems", *Opt. Eng.* 18, 1979:591.
- 7. Klein, C.A. "Mirrors and windows in power optics," *Proceedings of SPIE* **216**, 1980:204.
- Klein, C. A. "Optical distortion coefficients of laser windows—one more time," *Proceedings of SPIE* 1047, 1989:58.
- Palmer, J.R., "Thermal shock: catastrophic damage to transmissive optical components in high power CW and pulsed laser environments," *Proceedings of SPIE* 1047, 1989:87.
- Weidler, D.E., "Large exit windows for high power beam directors," *Proceedings of* SPIE 1047, 1989:153.
- 11. Robinson, B., Eastman, D.R., Bacevic, J., Jr., "Infrared window manufacturing technology," *Proceedings of SPIE* 430, 1983:302.
- 12. Stoll, R., Forman, P.F., and Edleman, J., The effect of different grinding procedures on the strength of scratched and unscratched fused silica," *Proceedings of Symposium on the Strength of Glass and Ways to Improve It*," Union Scientifique Continentale du Verre, Charleroi, Belgium, 1961:1.
- Marushin, P.H., Sasian, J.M., Lin, J.E., Greivenkamp, J.E., Lerner, S.A., Robinson, B., and Askinazi, J., "Demonstration of a conformal window imaging system: design, fabrication, and testing," *Proceedings of SPIE* 4375, 2001:154.
- 14. Fraser, B.S. and Hemingway, A., "High performance faceted domes for tactical and strategic missiles," *Proceedings of SPIE* 2286, 1994:485.
- 15. Shannon, R.R., Mills, J.P., Trotta, P.A., and Durvasula, L.N., "Conformal optics technology enables window shapes that conform to an application, not to optical limitations," *Photonics Spectra*, 35, 2001:86.
- 16. Knapp, D.J. Mills, J.P., Trotta, P.A., and Smith, C.B., "Conformal optics risk reduction demonstration," *Proceedings of SPIE* **4375**, 2001:146.
- Trotta, P.A., "Precision conformal optics technology program," *Proceedings of SPIE* 4375, 2001:96.
- 18. Harris, D.C., Materials for In frared Windows and D omes, Prop erties and Performance, SPIE Press, Bellingham, WA, 1999.

- 19. Dunn, G. and Stachiw, J., "Acrylic windows for underwater structures," *Proceedings* of SPIE 7, 1966: D-XX-1.
- 20. Pickles, C.S.J. and Field, J.E., "The dependence of the strength of zinc sulfide on temperature and environment," J. Mater. Sci., 29, 1994:1115.
- 21. Vukobratovich, D. "Optomechanical system design," Chapter 3 in *The Infrared & Electro-Optical Syst ems H andbook*, Vol. 4, ERIM, Ann Arbor and SPIE Press, Bellingham.
- Fuller, E.R., Freiman, S.W., Quinn, J.B., Quinn, G.D., and Carter, W.C., "Fracture mechanics approach to the design of glass aircraft windows: a case study," *Proceedings of SPIE* 2286, 1994:419.
- 23. Pepi, J.W., "Failsafe design of an all BK-7 glass aircraft window," *Proceedings of* SPIE 2286, 1994:431.
- 24. Sparks M. and Cottis, M., "Pressure-induced optical distortion in laser windows," J. *Appl. Phys.*, 44, 1973:787.
- 25. Vukobratovich, D., "Principles of optomechanical design," Chapter 5 in *Applied Optics and Optical Engineering*, XI, R.R. Shannon and J.C. Wyant, Eds., Academic Press, New York, 1992.
- 26. Roark, R.J., Formulas for Stress and Strain, 3rd. ed., McGraw-Hill, New York, 1954.
- 27. Kingslake, R., Lens Design Fundamentals, Academic Press, New York, 1978:311.
- 28. Vukobratovich, D., "Introduction to optomechanical design," SPIE S hort Course Notes SC114, 2003.
- 29. Speare, J. and Belioni, A., "Structural mechanics of a mortar launched IR dome," *Proceedings of SPIE* **450**, 1983:182.
- 30. Sunne, W.L., Nagy, P.A., and Liquori, E., "Vehicle having a ceramic radome affixed thereto by a compliant metallic 'T-fixture' element," U.S. Patent 5,941,479, 1999.
- 31. Sunne, W., Ohanian, O., Liguori, E., Kevershan, M., Samonte, J., and Dolan, J., "Vehicle having a ceramic dome affixed thereto by a compliant metallic transmission element," U.S. Patent No. 5,884,864, 1999.
- 32. Sunne, W., "Dome attachment with brazing for increased aperture and strength," *Proceedings of SPIE* **5078**, 2003:121.

CHAPTER 6 Prism Design

Many types of prisms have been designed for use in various optical instrument applications. Most have unique shapes as demanded by the geometry of the ray paths, reflection and refraction requirements, and compatibility with manufacture, weight reduction considerations, and provisions for mounting. Before we consider how to mount these prisms, we should understand how they are designed. Our first topics in this chapter are the functions of prisms, geometric relationships that govern those functions, refractive effects, total internal reflection, and the construction and use of tunnel diagrams. We then see how to determine aperture requirements and reference analytical means for calculating third-order aberration contributions from prisms. The chapter closes with design information for 30 types of individual prisms and prism combinations frequently encountered in optical instrument design.

6.1 Principal Functions

The principal functions or uses of prisms (and of some mirrors) are as follows:¹

- To bend (deviate) light around corners,
- To fold an optical system into a given shape or package size,
- To provide proper image orientation,
- To displace the optical axis,
- To adjust optical path length,
- To divide or combine beams by intensity or aperture sharing at a pupil,
- To divide or combine images at an image plane,
- To dynamically scan a beam,
- To disperse light spectrally, and
- To modify the aberration balance of the system of which they are a part.

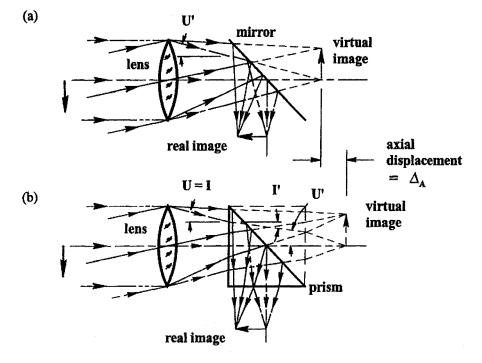
Some prisms accomplish more than one function simultaneously.

6.2 Geometric Considerations

6.2.1 Refraction and reflection

The laws of refraction and reflection of light govern the passage of rays through prisms and mirrors. In Fig. 6.1, we see a comparison of ray paths from an object passing through a lens and a reflector (mirror) en route to the image. In Fig. 6.1(a), the reflector is a flat mirror while in Fig. 6.1(b); it is a right-angle prism where reflection occurs at an internal surface. The most significant differences are the ray deviations that occur at the prism's refracting surfaces and the axial displacement of the image caused by the replacement of air by glass in part of the path. Refraction in the prism, of course, follows Snell's law, which may be written as:

$$n_j \sin I_j = n'_j \sin I'_j, \tag{6.1}$$



where n_j and n'_j are the refractive indices in object and image spaces of surface "j" and I_j and I'_j are the ray angles of incidence and refraction, respectively.

Figure 6.1 90-deg deviations by reflection of rays, (a) at a 45-deg mirror and (b) in a right-angle prism. In (b), angles U, U', I, and I' pertain to the first surface of the prism.

Reflection follows the familiar relationship

$$I'_j = I_j, \tag{6.2}$$

where I_j and I'_j are the angle's incidence and reflection at surface "j". The angles in these equations are measured with respect to the surface normal at the point of incidence of the ray on the surface. A change in algebraic sign of the ray angle occurs upon reflection. The algebraic signs of the angles are not shown in either of these equations.

The entrance and exit faces of most prisms are oriented perpendicular to the optical axis of the optical system. This promotes symmetry and reduces aberrations for noncollimated beams passing through the prism. Notable exceptions are the Dove prism, the double-Dove prism, wedge prisms, and prisms used to disperse light in monochromators and spectrographs.

A prism with faces normal to the optical axis refracts rays exactly as would a plane parallel plate oriented normal to the axis. The geometrical path length, t_A , through the prism measured along the axis is the same as the thickness of the plate. Any reflections occurring inside the prism do not affect this behavior. The axial displacement, Δ_A (see Fig. 6.1), of an image formed by rays passing through the prism is given by PRISM DESIGN

$$\Delta_A = t_A \left[1 - \frac{\tan U'}{\tan U} \right] = \left(\frac{t_A}{n} \right) \left[n - \left(\frac{\cos U}{\cos U'} \right) \right].$$
(6.3)

For small angles, this equation reduces to the paraxial version:

$$\Delta_A = \frac{(n-1)t_A}{n}.\tag{6.4}$$

Example 6.1: Image axial displacement due to insertion of a prism. (For design and analysis, use File 6.1 of the CD-ROM).

Assume that a converging lens images a distant object with an f/4 beam. How much does the image move axially (a) exactly and (b) paraxially, when a right-angle prism made of FN11 glass with thickness $t_A = 38.100 \text{ mm} (1.500 \text{ in.})$ is inserted in the beam?

(a) The marginal ray angle for this f/4 beam is

$$\sin U' = \frac{0.5}{(\text{f-number})} = \frac{0.5}{4} = 0.1250$$

hence, U' = 7.1808 deg.

From Table B2, the refractive index for FN11 glass is 1.621. Since the entrance face of the prism is normal to the axis, I = U' so, by Eq. (6.1)

$$\sin I' = \frac{\sin 7.1808 \deg}{1.621} = 0.07711 \text{ and } I' = 4.4226 \deg.$$

By Eq. (6.3), the image moves by

Δ

$$\Delta_A = \left(\frac{38.100}{1.621}\right) \left[1.621 - \left(\frac{\cos 7.1808 \deg}{\cos 4.4226 \deg}\right)\right] = 14.711 \,\mathrm{mm}\left(0.579 \,\mathrm{in.}\right)$$

(b) By Eq. (6.4), the paraxial approximation of this displacement is

$$A_{A} = \frac{(1.621 - 1)(38.1)}{1.621} = 14.596 \text{ mm}(0.579 \text{ in.}).$$

The reflection within a prism folds the light path. In Fig. 6.1(b), the object (an arrow, not shown) is imaged by the lens through the prism as the indicated virtual image. After reflection, the real image is located as shown. If the page were to be folded along the line representing the reflecting surface, the real image and the solid-line rays would coincide exactly with the virtual image and the dashed-line rays. A diagram showing both the original prism (ABC) and the folded counterpart (ABC') is called a "tunnel diagram" (see Fig. 6.2). The rays a-a' and b-b' represent actual reflected paths, while rays a-a'' and b-b'' appear to pass directly through the folded prism with proper refraction, but without the reflection. Successive folds of the page represent multiple reflections. This type of diagram, which can be drawn for any prism, is particularly helpful when designing an optical instrument using prisms since it simplifies the estimation of required apertures and hence the sizes of those prisms.

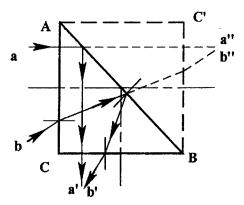


Figure 6.2 Illustration of a tunnel diagram for a right-angle prism.

To illustrate the use of a tunnel diagram, let us consider the telescope optical system of Fig. 6.3. This could be a spotting telescope or one side of a binocular. The Porro prisms serve to erect the image as indicated by the "arrow crossed with a drumstick" symbols at various locations in the figure. Figure 6.4(a) shows the front portion of the same system with the Porro prisms represented by tunnel diagrams. The diagonal lines indicate folds in the light path. We designate all the prism apertures as "A"; the axial path length of each prism is then 2A. In Fig. 6.4(b) the prism path lengths are shown as 2A/n; these are the thicknesses of air optically equivalent to the physical paths through the prisms. The airequivalent thickness is sometimes called the "reduced thickness." In a reduced thickness diagram, the marginal rays converging to the axial image point can be drawn as straight lines (i.e., without refraction). The ray heights at each prism surface (including the reflecting surfaces) are paraxial approximations of the true values that would be obtained by trigonometric ray tracing. Paraxially, angles in radians replace the sines of the angles. In most applications, this degree of approximation is adequate. For example, an angle of 7 deg is 0.12217 radians and its sine is 0.12187. The differences between these values are not significant for prism design purposes.

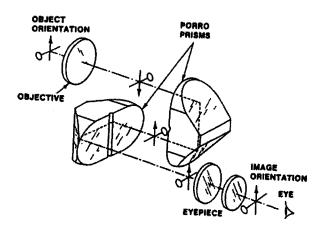


Figure 6.3 Optical system of a typical telescope with a Porro prism erecting system. (From Yoder.¹)

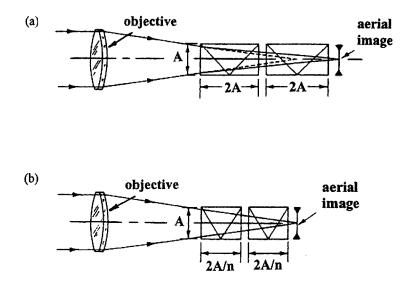


Figure 6.4 Lens and Porro prisms from Fig. 6.3 with prisms shown (a) by conventional tunnel diagrams and (b) by tunnel diagrams with reduced (air equivalent) thicknesses. (Adapted from Smith.²)

Warren Smith used tunnel diagrams to illustrate the determination of the minimum Porro prism apertures required for use in a typical prism erecting telescope.² With a diagram similar to Fig. 6.4(b), he noted that the proportion of face width A_i to reduced thickness was A_i : $(2A_i/n_i)$ or $n_i/2$. He then redrew the diagram in the form shown in Fig. 6.5 to facilitate calculating the minimum value for A_1 and A_2 . The dashed lines drawn from the top front prism corners to the opposite vertices both have slopes, m, equaling one-half the ratio just derived or $n_i/4$. These lines are loci of the corners of a family of prisms with the proper proportions. The intersections of these two dashed lines with the outermost full-field ray (frequently called the "upper rim ray" or URR) locate the corners of the two Porro prisms. Note that the air spaces between optical components must be known for this procedure to succeed.

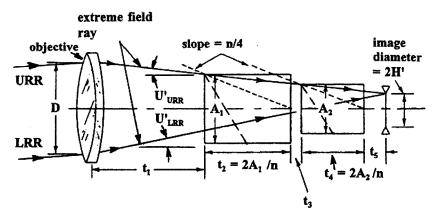


Figure 6.5 Determination of minimum prism apertures from geometric proportions and the outermost unvignetted upper full-field (rim) ray (URR). The corresponding lower field (rim) ray (LRR) is used to determine the ability of the prisms to function by total internal reflection (TIR).

It is easy to see from Fig. 6.5 that the slope of the URR is

$$\tan U'_{\rm URR} = \frac{\left[\left(\frac{D}{2}\right) - H'\right]}{\rm EFL_{OBJ}},\tag{6.5}$$

and that the semiaperture of the second prism is $A_2/2 = H' + (t_4 + t_5)(\tan U')$. This semiaperture also is given by the expression $A_2/2 = (m)(t_4) = (n_i)(t_4)/4$. Equating these expressions for A_2 , we find that the thickness and aperture of the second prism are

$$t_{4} = \frac{\left[(t_{5}) (\tan U'_{\text{URR}}) + H' \right]}{\left(\frac{n_{i}}{4} - \tan U'_{\text{URR}} \right)},$$

$$A_{2} = \frac{(n_{i}) (t_{4})}{2}.$$
(6.7)

We can write expressions for the axial thickness and aperture of the first Porro prism as

$$t_{2} = \frac{\left[\left(t_{3} + t_{4} + t_{5}\right)\left(\tan U_{\text{URR}}'\right) + H'\right]}{\left(\frac{n_{i}}{4}\right) - \tan U_{\text{URR}}'},$$
(6.8)

$$A_{1} = \frac{(n_{i})(t_{2})}{2}.$$
 (6.9)

Example 6.2 illustrates the use of this technique.

The apertures derived by these calculations should be confirmed by more precise techniques, such as trigonometric ray tracing—especially if a specific amount of vignetting is needed for off-axis aberration control. To allow for protective bevels and dimensional tolerances, we probably need to increase the apertures of both prisms by small amounts, such as a few percent of their apertures.

The same general technique can be applied to other types of prisms and prism assemblies to determine the required apertures when used in converging or diverging beams. Space limitations preclude inclusions of these considerations here. In all cases, once the prism apertures are known, we are next concerned about the areas on the prism refracting and reflecting surfaces actually used by the beams. We call these "beam prints." The approach described in Section 8.4 for defining mirror aperture requirements can easily be adapted to prisms. The chief difference between the two types of reflectors in such an analysis is the use of the refracted ray angles for all interior ray paths.

Example 6.2: Calculation of prism size. (For design and analysis, use File 6.2 of the CD-ROM).

Find the minimum apertures A_1 and A_2 of both prisms in a system as in Fig. 6.5 if EFL_{OBJ} is 177.800 mm (7.000 in.), objective aperture is 50.000 mm (1.968 in.), image diameter is 15.875 mm (0.625 in.), t₃ is 3.175 mm (0.125 in.), t₅ is 12.7 mm (0.500 in.), and prism index is 1.500.

By Eq. (6.5):

$$\tan U'_{URR} = \frac{\left[\left(\frac{50.000}{2}\right) \times \left(\frac{15.875}{2}\right)\right]}{177.800} = 0.09596$$

so $U'_{URR} = 5.481 \text{ deg.}$

By Eq. (6.6):
$$t_4 = \frac{\left[(12.700)(0.09596) + \left(\frac{15.875}{2}\right) \right]}{\left[\frac{1.500}{4} \right] - 0.09596} = 32.801 \text{ mm}(1.291 \text{ in.}).$$

By Eq. (6.7):

$$A_2 = \frac{(1.500)(32.813)}{2} = 24.601 \text{ mm} (0.968 \text{ in.}).$$

45.185 mm (1.779 in.).

By Eq. (6.8),

$$\int (3.175 + 32.801 + 12.700) (0.09596)$$

1.500

By Eq. (6.9):

9).
$$A_{\rm l} = \frac{(1.5)(45.185)}{2} = 33.889 \text{ mm (1.334 in.)}.$$

6.2.2 Total internal reflection

A special case of refraction can occur when a ray is incident upon an interface where *n* is greater than n' as, for example, at the hypotenuse surface (surface 2) inside a right-angle prism. In the last section, we assumed that all rays would reflect, as indeed they would if the surface had a reflective coating such as silver or aluminum. If that surface is uncoated, however, Snell's law [Eq. (6.1)] says that for small angles of incidence and low values of prism index, a ray can refract through that surface into the surrounding air, (see ray a-a' in Fig. 6.6). This ray is vignetted and does not contribute to the image formed below the prism. If we increase the ray angle I_2 , the angle I'_2 also increases. For some value of I_2 , I'_2 can reach 90 deg. Then sin I'_2 is unity. Since this sine cannot exceed unity, we find that for

still larger values of I_2 , the ray reflects internally just as if the surface were silvered. The particular value of I_2 corresponding to $I'_2 = 90$ deg is called the "critical angle," I_C . This angle is calculated from the equation:

$$\sin I_C = \frac{n_2'}{n_2}.$$
 (6.10)

Usually, the medium beyond surface 2 is air, so n'_2 is unity and sin $I_C = 1/n_2$.

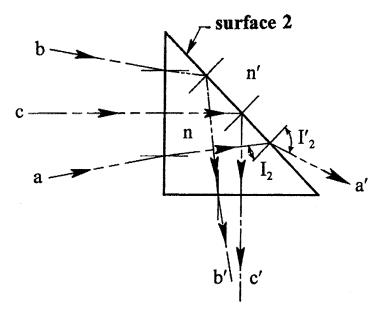


Figure 6.6 Ray paths through an unsilvered right angle prism of low refractive index. Ray *a*-*a*' is at an angle of incidence l_2 smaller than l_c so it "leaks" through the surface, while l_2 for each of rays *b*-*b*' and *c*-*c*' exceeds l_c so they "totally reflect" internally.

We can take advantage of total internal reflection (TIR) in prisms by choosing a refractive index high enough that all rays we want to reflect exceed I_C at the surface in question. Then the reflections take place without photometric loss, and reflective coatings are not needed on that surface. It is important to note that TIR occurs only at clean surfaces, so special care must be taken not to let the surface become contaminated with condensed water, fingerprints, or other foreign matter that can change the refractive index outside that surface.

In Fig. 6.5, we note that the LRR has a smaller incident angle with respect to the diagonal reflecting surface inside both prisms than any other ray lying between the LRR and the URR. It therefore is more likely than those other rays not to totally reflect if the glass index is not high enough to cause TIR. This ray is then the one that determines the size of the unvignetted field of view of the optical system. Example 6.3 shows how this unvignetted field can be determined for a given telescope system design using Porro prisms.

Example 6.3: Unvignetted field of view for TIR in a prism erecting telescope. (For design and analysis, use File 6.3 of the CD-ROM.)

Assume that the prisms of Example 6.2 are not silvered and are made of F2 glass with a refractive index of 1.620. Let EFL_{OBJ} be 177.8 mm (7.000 in.) and the lens's aperture D be 50.000 mm (1.968 mm).

What field of view can the prisms transmit without vignetting caused by loss of TIR?

From Eq. (6.10):

$$\sin I_C = \frac{1}{1.620} = 0.61728,$$

so $I_C = 38.1181$ deg.

From the geometry of Fig. 6.6, I' for the LRR inside the prism at its entrance face is $(45 \text{ deg} - I_C) = 6.8819 \text{ deg}.$

From Eq. (6.1):

$$\sin I_{\text{LRR}} = (1.620)(\sin 6.8819 \text{ deg}) = 0.19411,$$

so $I_{LRR} = 11.1930$ deg.

This ray angle equals the slope of the LRR passing from the bottom of the lens aperture to the top of the image. Hence, $U'_{LRR} = 11.1930 \text{ deg and tan } I_{LRR} = 0.19788.$

Modifying Eq. (6.5) to apply to the LRR (by changing the minus sign to a plus sign in the numerator), we get:

$$\tan U'_{LRR} = \frac{\left\lfloor \left(\frac{D}{2}\right) + H' \right\rfloor}{EFL_{OBJ}} = \frac{\left\lfloor \left(\frac{1.968}{2}\right) + H' \right\rfloor}{7.000} = 0.19788.$$

Solving for the image height, we get

$$H' = (0.19788)(7.000) - \left(\frac{1.968}{2}\right) = 0.4012$$
 in. (10.1905 mm).

The principal ray (PR) of an optical system is defined as the ray that passes through the center of the aperture stop at the semi-field angle and intercepts the image plane at the height H'. Here, the aperture stop is at the objective. The following relationship applies:

tan
$$U'_{PR} = \frac{H'}{EFL_{OBJ}} = \frac{0.4012}{7.000} = 0.0573$$
 and $U'_{PR} = 3.280 \text{ deg}$.

The total system unvignetted field of view is then ± 3.280 deg or 6.560 deg.

6.3 Aberration Contributions of Prisms

As mentioned earlier, prisms usually are designed so their entrance and exit faces are perpendicular to the optical axis of the transmitted beam. If that beam is collimated, no aberrations are introduced. Aberrations do result if the beam is not collimated. In a converging or diverging beam, a prism introduces longitudinal aberrations (spherical, chromatic, and astigmatic) as well as transverse aberrations (coma, distortion, and lateral chromatic). Smith provided exact and third-order equations for calculating the aberration contributions of a plane parallel plate or the equivalent prism.² Ray-tracing programs give the aberration contributions of prisms in a given design on a surface-by-surface basis.

6.4 Typical Prism Configurations

Chapter 13 of MIL-HDBK-141 Optical Design³ gives generic dimensions, axial path lengths, and tunnel diagrams for many types of common prisms. Most of these designs were described earlier in ORDM 2-1, *Design of Fire Control Optics*,⁴ a two-volume treatise on telescope design written by Frankford Arsenal's long-time chief lens designer, Otto K. Kaspereit, and published by the U.S. Army in 1953. Since copies of the latter book are hard to find and both, like later references,^{2,3,5} do not always include all the information we need to design mounts for the prisms, we include here design data and/or functional descriptions for 33 types of prisms, some of which were not included in any of these references. Included are orthographic projections, prism dimensions, axial path length, and in many cases, isometric views, tunnel diagrams, approximate prism volume, and bonding area information (discussed in Sect. 7.5). The following parameter definitions apply:

A	= prism face width
<i>B</i> , <i>C</i> , <i>D</i> , etc.,	= other linear dimensions
a, b, c, etc.,	= widths of typical bevels
δ, θ, φ, etc.,	= angular dimensions
t_A	= axial path length
V	= prism volume (neglecting small bevels)
ρ	= glass density
W	= prism weight
a_G	= acceleration factor measured as "times gravity"
Q_{MIN}	= minimum bond area for adhesive strength J and safety factor f_S
Q_{MAX}	= maximum circular (C), or racetrack (RT) bond area achievable on the
	prism mounting surface

Also included in the figure captions for most designs are numerical results for each parameter assuming that A is 1.500 in. (38.100 mm), the glass is BK7 with n = 1.5170, $\rho = 0.0907$ lb/in.³ (2.510 g/cm³), $a_G = 15$, J = 2000 lb/in.² (13.79 MPa), and $f_S = 4$. The pertinent reference to a file in the CD-ROM for design and analysis also is given. Minor discrepancies between values indicated in the figure captions and those given in the CD-

^{*} As discussed in Section 14.6, stresses induced by temperature changes may set upper limits on bond sizes. Those limits prevail in design of the bond.

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ROM examples result from differences in the number of significant figures used in the calculations.

6.4.1 Right-angle prism

Figure 6.1(b) shows the function of this prism in its most common role as a means for deviating a beam by 90 deg whereas Fig. 6.2 shows its tunnel diagram. Figure 6.7 shows three views of the prism with a bonded interface on one of its triangular sides.

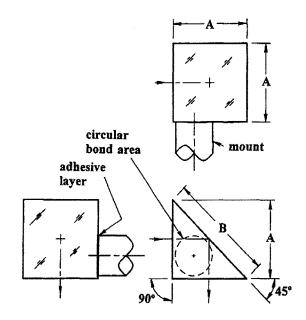


Figure 6.7 Right angle prism. (For design and analysis, use File 6.4 of the CD-ROM) $t_A = A = 1.500$ in. (38.100 mm); B = 1.414A = 2.121 in. (53.881 mm); $V = 0.500A^3 = 1.688$ in.³ (27.661 mm³); $W = V\rho = 0.154$ lb (0.070 kg); $Q_{MIN} = V\rho a_G f_S / J = 0.0046$ in² (2.968 mm²); $Q_{MAX C} = 0.230A^2 = 0.517$ in.² (333.870 mm²).

6.4.2 Beamsplitter (or beamcombiner) cube prism

Two right-angle prisms cemented together at their hypotenuse surfaces with a partially reflective coating at the interface form a cube-shaped beamsplitter or beamcombiner. This type of prism is shown in Fig. 6.8. If this prism (or any multiple-component prism) is to be bonded to a mechanical mounting, the adhesive joint should be limited to one component; the bond would then not bridge the cemented joint. This is because the two glass surfaces may not be accurately coplanar and the strength of the bond may be degraded by differences in adhesive thickness. If the adjacent surfaces are reground after cementing, bonding across the joint may be acceptable.

Most of the design equations for the beamsplitter cube apply also to a monolithic cube such as might be used as a rotating prism in a high-speed camera. With a solid cube, the bond area, Q, can be as large as $Q_{MAX} = 0.785 \text{A}^2$.

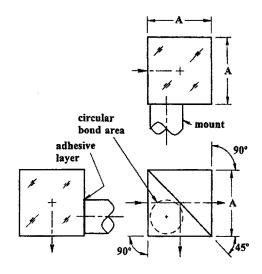


Figure 6.8 Beamsplitter cube prism. (For design and analysis, use File 6.5 of the CD-ROM.) $t_A = A = 1.500$ in. (38.100 mm); $V = A^3 = 3.375$ in.³ (55.306 cm³); $W = V\rho = 0.307$ lb (0.139 kg); $Q_{MIN} = V\rho a_G f_S / J = 0.0092$ in.² (5.935 mm²); $Q_{MAX C} = 0.230A^2 = 0.517$ in.² (333.870 mm²).

6.4.3 Amici prism

The Amici prism (see Fig. 6.9) is essentially a right-angle prism with its hypotenuse configured as a 90-deg "roof" so a transmitted beam makes two reflections instead of just one. A right-handed image is produced. The prism can be used in such a manner that the transmitted beam is split by the dihedral edge between the roof surfaces or (with a larger prism for constant beam size) so the beam hits the roof surfaces in sequence. These possibilities are illustrated in Figs. 6.10(a) and 6.10(b), respectively. In the former case, the dihedral edge must be accurately 90 deg (i.e., within a very few arcseconds) in order to not produce a noticeable double image. This makes the smaller component's cost higher because of the added labor, fixturing, and testing required to correct the roof angle. The prisms of Fig. 6.10 are shown to be of equal size so the beam in view (b) cannot be larger than A/2. The beam axis is displaced laterally by A/2 in this case. In view (a), the beam can be almost as large as A, and is centered to the roof edge.

6.4.4 Porro prism

A right-angle prism arranged so the beam enters and exits the hypotenuse surface, as shown in Fig. 6.11(a), is called a "Porro prism." Ray a-a' travels parallel to the axis while rays b-b'and c-c' enter at different field angles. Note that rays a-a' and b-b' turn around and exit parallel to the entering rays; this shows that the prism is retrodirective in the plane of refraction. Path c-c' represents a field ray entering near the edge of the prism. It intercepts the hypotenuse A-C internally and hence has three reflections, producing an inverted image. Such a ray is called a "ghost" ray since it does not contribute useful information to the main image. It does add stray light and thus should be eliminated. The groove cut into the center

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of the hypotenuse does just that, so it is a usual feature of the Porro design. The tunnel diagram of Fig. 6.11(b) shows all these rays and the groove. The prism design is given in Fig. 6.12.

Another useful feature of this (and several other types of prisms) is that it produces the same deviation (180 deg for the Porro) even if it is rotated about an axis perpendicular to the plane of refraction. Such a prism is said to be a "constant deviation" prism. Note that the Porro prism produces constant deviation only in this one plane.

6.4.5 Porro erecting system

Two Porro prisms oriented at a right angle and connected together, as shown in Fig. 6.3, constitute a Porro erecting system. The axis is displaced laterally in each direction by 2A plus the width of the bevel on the apex of each prism. This system is most frequently used in binoculars and telescopes to erect the image. A design in which the prisms are cemented together is shown in Fig. 6.13. An air space between the prisms does not alter their function. This is not a constant deviation configuration.

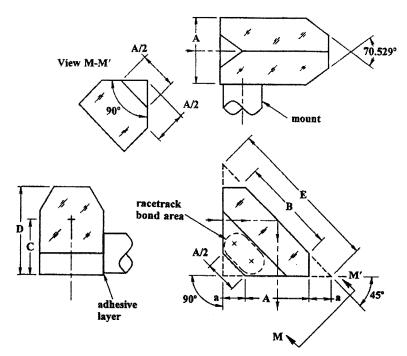


Figure 6.9 Amici prism. (For design and analysis, use File 6.6 of the CD-ROM.) $t_A = 2.707A = 4.061$ in. (103.140 mm); a = 0.354A = 0.530 in. (13.472 mm); B = 1.414A = 2.121 in. (53.881 mm); C = 0.854A = 1.280 in. (32.522 mm); D = 1.354A = 2.031 in. (51.587 mm); E = 2.415A = 3.621 in. (91.981 mm); $V = 0.888A^3 = 2.997$ in.³ (49.118 cm³); $W = V\rho = 0.273$ lb (0.124 kg); $Q_{MIN} = V\rho a_G f_S / J = 0.008$ in.² (5.264 mm²); $Q_{MAX C} = 0.164A^2 = 0.369$ in.² (238.064 mm²); $Q_{MAX RT} = 0.306A^2 = 0.689$ in.² (444.338 mm²).

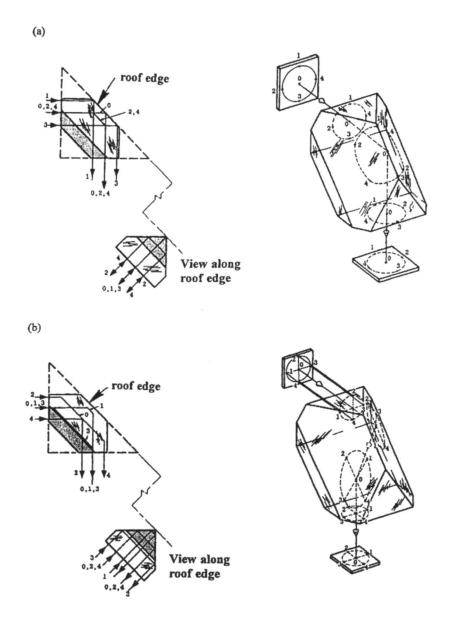


Figure 6.10 The Amici prism used (a) symmetrically as a split-beam reflector and (b) off-center as a full beam reflector. (From MIL-HDBK-141. 3)

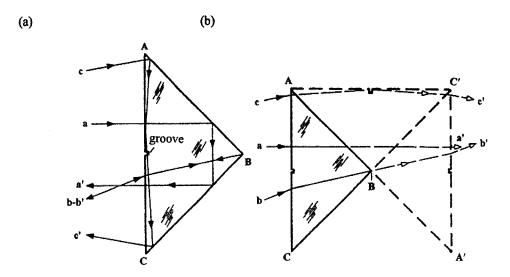


Figure 6.11 (a) Typical ray paths through a Porro prism. (b) Its tunnel diagram.

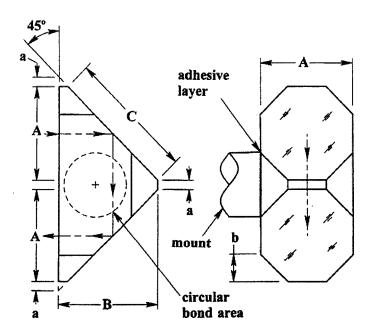


Figure 6.12 Porro prism. (For design and analysis, use File 6.7 of the CD-ROM). $t_A = 2.3A = 3.450$ in. (87.630 mm); a = 0.1A [assumed] = 0.150 in. (3.810 mm); b = 0.293A = 0.439 in. (11.163 mm); B = 1.1A = 1.650 in. (41.910 mm); C = 1.414A = 2.121 in. (53.881 mm); $V = 1.286A^3 = 4.340$ in.³ (71.124 cm³); $W = V\rho = 0.395$ lb (0.179 kg); $Q_{MIN} = V\rho a_G f_S / J = 0.012$ in.² (7.644 mm²); $Q_{MAX C} = 0.608A^2 = 1.368$ in.² (882.579 mm²).

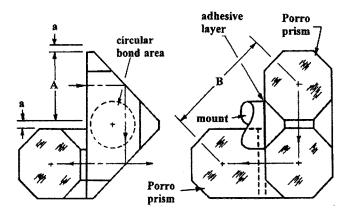


Figure 6.13 The Porro erecting system. (For design and analysis, use File 6.8 of the CD-ROM.) $t_A = 4.6A = 6.900$ in. (175.260 mm); a = 0.1A [assumed] = 0.150 in. (3.810 mm); B = 1.556A = 2.334 in. (59.284 mm); $V = 2.573A^3 = 8.634$ in.³ (142.303 cm³); $W = V\rho = 0.790$ lb (0.358 kg); $Q_{MIN} = V\rho a_G f_S / J = 0.024$ in.² (15.295 mm²); $Q_{MAX c} = 0.459A^2 = 1.033$ in.² (666.289 mm²).

6.4.6 Abbe version of the Porro prism

Ernst Abbe modified the design of the Porro prism by rotating one half of the prism about the optic axis by 90 deg with respect to the other half. Figure 6.14 illustrates this prism and provides its design equations in the caption. For a given aperture A, this prism is slightly larger than the standard version because it includes larger bevels. The presence of these bevels and their sizes are design options.

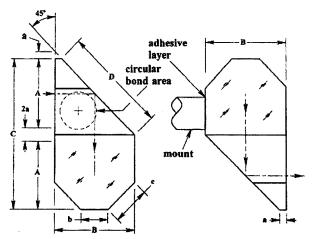


Figure 6.14 Abbe version of the Porro prism. (For design and analysis, use File 6.9 of the CD-ROM.) $t_A = 2.400A = 3.600$ in. (91.440 mm); a = 0.1A[assumed] = 0.150 in. (3.810 mm); b = 0.414A = 0.621 in. (15.773 mm); B =1.200A = 1.800 in. (45.720 mm); C = 2.200A = 3.300 in. (83.820 mm); D =1.556A = 2.334 in. (59.284 mm); $V = 1.832A^3 = 6.183$ in³ (101.321 cm³); W = $V\rho = 0.561$ lb (0.254 kg); $Q_{MIN} = V\rho a_G f_S / J = 0.017$ in.² (10.967 mm²); $Q_{MAX C} =$ 0.388 $A^2 = 0.873$ in.² (563.225 mm²).

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6.4.7 Abbe erecting system

The combination of two Abbe prisms cemented together with entrance and exit faces on opposite sides of the assembly creates an erecting prism subassembly that functions like a Porro erecting system. It usually is made by cementing two right-angle prisms side by side, but facing in opposite directions, on the hypotenuse of a Porro prism (see Fig. 6.15). This construction is somewhat less expensive than making two Abbe prisms and cementing them together. It also is much easier to mount since the Porro prism presents a larger surface for bonding.

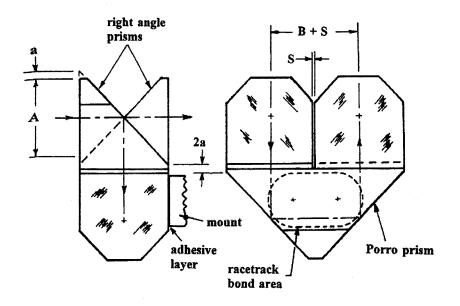


Figure 6.15 The Abbe erecting system. (For design and analysis, use File 6.10 of the CD-ROM.) $t_A = 4.450A = 6.675$ in. (169.545 mm); a = 0.1A [assumed] = 0.150 in. (3.810 mm); S = 0.050A [assumed] = 0.075 in. (1.905 mm); B = 2.250A = 3.375 in. (85.725 mm); $V = 3.808A^3 = 12.852$ in.³ (210.606 cm³); $W = V\rho = 1.169$ lb (0.530 kg); $Q_{\text{MIN}} = V\rho a_G f_S / J = 0.035$ in.² (22.636 mm²); $Q_{\text{MAX } C} = 0.459A^2 = 1.033$ in.² (666.289 mm²) ; $Q_{\text{MAX } RT} = 0.841A^2 = 1.892$ in.² (1220.643 mm²).

6.4.8 Rhomboid prism

Although usually made of one piece of glass, the rhomboid prism shown in Fig. 6.16 is functionally the integration of two right-angle prisms with their reflecting surfaces parallel and a plane parallel plate of axial thickness variable from zero to some value (B in the figure). It produces a particular lateral displacement of the optical axis. The prism is insensitive to tilt in the plane of reflection so it provides constant deviation in that plane.

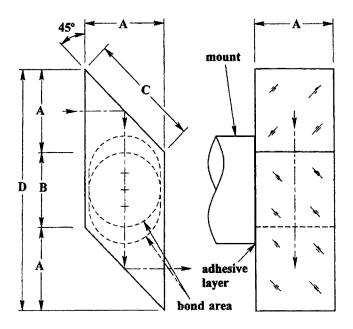


Figure 6.16 Rhomboid prism. (For design and analysis, use File 6.11 of the CD-ROM.) *B* [assumed] = 0.500 in. (12.700 mm); $t_A = 2A + B = 3.500$ in. (88.900 mm); C = 1.414A = 2.121 in. (53.881 mm); D = 2A + B = 3.500 in. (88.900 mm); $V = A^2$ (A + B) = 4.500 in.³ (73.742 cm³); $W = V\rho = 0.408$ lb (0.185 kg); $Q_{MIN} = V\rho a_G f_S / J = 0.012$ in.² (7.900 mm²); For B = 0, $Q_{MAX C} = 0.393A^2 = 0.884$ in.² (570.483 mm²) $Q_{MAX RT} = 0.686A^2 = 1.543$ in. (995.804 mm²); For B > 0.414A, $Q_{MAX C} = 0.785A^2 = 1.767$ in.² (1140.094 mm²); $Q_{MAX RT} = 0.578A^2 + 0.500AB = 1.676$ in.² (1081.401 mm²).

6.4.9 Dove prism

The Dove prism is a right-angle prism with the top section removed and the optical axis oriented parallel to the hypotenuse face as shown in Fig. 6.17. This single-reflection prism inverts the image only in the plane of refraction. It is most commonly used to rotate the image by turning the prism about its optical axis. The image then rotates at twice the speed of the prism. Because of the oblique incidence of the axis at the entrance and exit faces, the prism can be used only in a collimated beam. Alternative versions can have faces tilted at other angles, but here we limit consideration to the 45-deg incidence case because it is the most common.

The prism dimensions depend upon the prism's refractive index because of deviation of the optical axis at the tilted faces. Table 6.1 shows how the dimensions of a typical Dove prism vary with changes in refractive index. These values were all obtained with the equations given in the caption of Fig. 6.17.

Index (n)	1.5170	1.6170	1.7215	1.8052
A (mm)	38.100	38.100	38.100	38.100
B (mm)	177.156	163.154	152.541	145.959
C (mm)	173.346	159.344	148.731	142.148
D (mm)	93.336	79.334	68.721	62.138
E (mm)	56.576	56.576	56.576	56.576
t _A (mm)	141.590	128.283	118.303	112.171

Table 6.1 variations in Dove prism dimensions with refractive index.

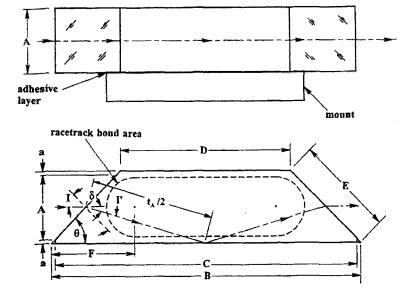


Figure 6.17 Dove prism. (For design and analysis, use File 6.12 of the CD-ROM.) n = 1.5170; $\theta = 45 \text{ deg}$; $l = 90 \text{ deg} - \theta = 45 \text{ deg}$; $l' = \arcsin [(\sin l)/n] = 27.783 \text{ deg}$; $\delta = l - l' = 17.217 \text{ deg}$; a = 0.050A [assumed] = 0.075 in. (1.905 mm); $t_A = (A + 2a)/\sin \delta = 5.574$ in. (14.580 mm); $B = (A + 2a)[(1/\tan \delta) + (1/\tan \theta)] = 6.975$ in. (177.156 mm); C = B - 2a = 6.825 in. (173.355 mm); D = B - 2(A + 2a) = 3.675 in. (93.345 mm); $E = (A + a)/\cos \theta = 2.227$ in. (56.566 mm); $F = (A + 2a)/[2\tan (\theta/2)] = 1.992$ in. (50.590 mm); $V = (A)(B)(A + 2a) - (A)(A + 2a)^2 - Aa^2 = 13.170$ in.³ (215.819 cm²); $W = V\rho = 1.198$ lb (0.542 kg); $Q_{MIN} = V\rho a_G f_S/J = 0.036$ in.² (23.226 mm²); $Q_{MAX C} = \pi[(A + a)/2]^2 = 2.138$ in.² (1379.511 mm²); $Q_{MAX RT} = Q_{MAX C} + (A + 2a)(B - 2F) = 7.074$ in.² (4563.678 mm²).

6.4.10 Double Dove prism

This prism consists of two Dove prisms, each of aperture A/2 by A, attached at their hypotenuse faces. Figure 6.18 shows the configuration. It is commonly used as an image rotator. The prisms can be air spaced by a small distance and held mechanically. TIR then will occur. They also can be cemented together. In that case, a reflective coating such as aluminum or silver is placed on one prism face before the prisms are cemented to keep the light from passing through the interface. For a given aperture, A, the double Dove prism is one-half the length of the corresponding standard Dove prism. To minimize obscuration of

the beam, the leading and trailing edges of both prisms are given only minimal protective bevels.

As shown in the end view of Fig. 6.18, the shape of a circular beam entering a double Dove prism at left is converted into a pair of "D-shaped" beams with curved edges adjacent as it exits at right. If vignetting is to be avoided, the apertures of subsequent optics must be large enough to accept the divided beam of diameter 1.414*A*. The modulation transfer function (MTF) of the optical system in which the prism is used is somewhat degraded by the divided aperture. The 45-deg angles of the prisms must be very accurate in order to minimize image doubling.

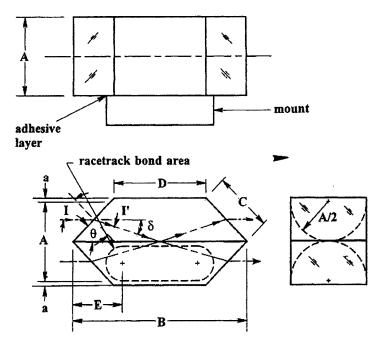


Figure 6.18 Double Dove prism. (For design and analysis, use File 6.13 of the CD-ROM.) n = 1.5170; $\theta = 45 \text{ deg}$; $i = 90 \text{ deg} - \theta = 45 \text{ deg}$; $l' \arcsin [(\sin l)/n] = 27.783 \text{ deg}$; $\delta = l - l' = 17.217 \text{ deg}$; a = 0.05A = 0.075 in. (1.905 mm); $t_A = [(A/2) + a]/\sin \delta) = 2.787 \text{ in}$. (70.795 mm); $B = (A + 2a)[(1/\tan \delta) + (1/\tan \theta)]/2 = 3.487 \text{ in}$. (88.578 mm); $C = [(A/2) + a]/\cos \theta = 1.167 \text{ in}$. (29.635 mm); D = B - (A + 2a) = 1.837 in (46.668 mm); $E = [(A/2) + a]/[2 \tan (\theta/2)] = 0.996 \text{ in}$. (25.295 mm); $V = (A)(B)(A + 2a) - (2A)[(A/2) + a]^2 = 6.589 \text{ in.}^3 (107.979 \text{ mm}^3)$; $W = V\rho = 0.598 \text{ }/B (0.271 \text{ kg})$; $Q_{\text{MIN}} = V\rho a_G f_S/J = 0.018 \text{ in.}^2 (11.548 \text{ mm}^2)$; $Q_{\text{MAX } C} = (\pi/4)[(A/2) + a]^2 = 0.535 \text{ in.}^2 (344.878 \text{ mm}2)$; $Q_{\text{MAX } RT} = Q_{\text{MAX } C} + [(A/2) + a](B - 2E) = 1.768 \text{ in.}^2 (1140.919 \text{ mm}^2)$.

A cemented "cube-shaped" version of the double Dove prism (with reflecting coating at the interface) is sometimes used as a means for scanning the line of sight (LOS) of an optical system in the plane of refraction. In such a prism, the faces with dimensions "C" shown in Fig. 6.18 are extended so the faces marked "D" are reduced to almost zero. The prism is rotated about an axis normal to that plane (parallel to the hypotenuse faces) and

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passing through the prism's geometric center. When it is located in front of a camera, periscope, or other optical instrument with a collimated beam entering from the object, such a prism can scan the system's LOS at well over 180 deg in object space. A single Dove prism also can be used to scan the LOS. A hardware example of this type is shown in Fig. 7.15. It has a reduced deviation range as compared to the double Dove assembly (see Fig. 7.16).

6.4.11 Reversion, Abbe Type A, and Abbe Type B prisms

A two-component (cemented) image rotator prism is shown in Fig. 6.19. It has three reflections and is called a reversion prism. Functionally, it differs from the Dove and double Dove prisms in that it can be used in converging or diverging beams. The central reflecting face, of dimension "C" in the figure, must have a reflective coating to prevent refraction through it. This surface usually is then covered by a protective coating such as electroplated copper and paint like a "back-surface" mirror, (see Section 8.2).

The Abbe Type A and Type B prisms (see Figs. 6.20 and 6.21) serve the same function as the reversion prism, but they have the central reflecting surface changed into a roof so they can invert the image in the direction transverse to the dihedral edge. With an even number of reflections, they can be used as an erecting assembly, but not as an image rotator. They differ in construction and the number of elements cemented together. The Type A prism has two elements while the Type B prism comprises three elements.

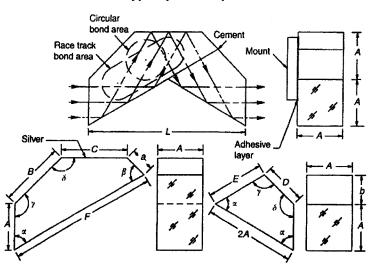


Figure 6.19 Reversion prism. (For design and analysis, use File 6.14 of the CD-ROM.) α = 30 deg; β = 60 deg; γ = 45 deg; δ = 135 deg; a = 0.707A = 1.060 in. (26.937 mm); b = 0.577A = 0.865 in. (21.984 mm); c = 0.500A = 0.750 in. (19.050 mm); t_A = 5.196A = 7.794 in. (197.968 mm); B = 1.414A = 2.121 in. (53.881 mm); C = 1.464A = 2.196 in. (55.778 mm); D = 0.867A = 1.300 in. (33.020 mm); E = 1.268A = 1.902 in. (48.311 mm); L = 3.464A = 5.196 in. (131.978 mm); V = 4.196 A^3 = 14.161 in.³ (232.065 cm³); W = $V\rho$ = 1.289 lb (0.585 kg); $Q_{\text{MIN}} = V\rho a_G f_S / J$ = 0.039 in.² (25.161 mm²); $Q_{\text{MAX} c}$ = 1.108 A^2 = 2.493 in.² (1608.384 mm²); $Q_{\text{MAX RT}}$ = 2.037 A^2 = 4.583 in.² (2956.930 mm²).

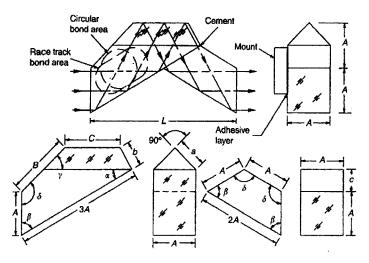


Figure 6.20 Abbe Type A prism. (For design and analysis, use File 6.15 of the CD-ROM.) α = 30 deg; β = 60 deg; γ = 45 deg; δ = 135 deg; a = 0.707*A* = 1.060 in. (26.937 mm); *b* = 0.577*A* = 0.866 in. (21.996 mm); *c* = 0.500*A* = 0.750 in. (19.050 mm); *t_A* = 5.196*A* = 7.794 in. (197.968 mm); *B* = 1.414*A* = 2.121 in. (53.873 mm); *C* = 1.309*A* = 1.963 in. (49.860 mm); *L* = 3.464*A* = 5.196 in. (131.978 mm); *V* = 3.719*A*³ = 12.552 in.³ (205.684 cm³); *W* = *V* ρ = 1.138 lb (0.516 kg); *Q*_{MIN} = *V* $\rho a_G f_S / J$ = 0.034 in.² (21.935 mm²); *Q*_{MAX C} = 0.802*A*² = 1.805 in.² (1164.514 mm²); *Q*_{MAX RT} = 1.116*A*² = 2.512 in.² (1620.642 mm²).

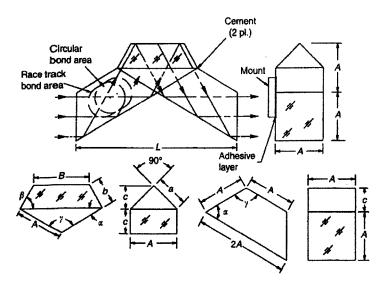


Figure 6.21 Abbe Type B prism. (For design and analysis, use File 6.16 of the CD-ROM.) α = 30 deg; β = 60 deg; γ = 45 deg; δ = 135 deg; a = 0.707A = 1.060 in. (26.937 mm); b = 0.577A = 0.865 in. (21.984 mm); c = 0.500A = 0.750 in. (19.050 mm); t_A = 5.196A = 7.794 in. (197.968 mm); B = 1.155A = 1.733 in. (44.018 mm); L = 3.464A = 5.196 in. (131.978 mm); V = 3.849A³ = 12.991 in.³ (212.880 cm³); W = $V\rho$ = 1.182 lb (0.536 kg); Q_{MIN} = $V\rho a_G f_S / J$ = 0.035 in.² (22.880 mm²); $Q_{\text{MAX } c}$ = 0.589A² = 1.325 in.² (854.998 mm²); $Q_{\text{MAX } RT}$ = 1.039A² = 2.338 in.² (1508.223 mm²).

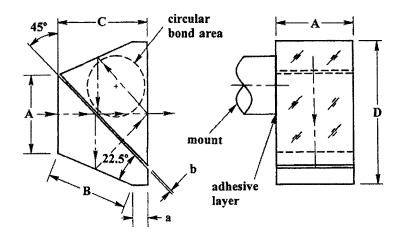


Figure 6.22 Pechan prism. (For design and analysis, use File 6.17 of the CD-ROM.) a = 0.207A = 0.310 in. (7.887 mm); b = 0.004 in. (0.100 mm) [assumed]; B = 1.082A = 1.623 in. (41.224 mm); C = 1.207A = 1.810 in. (45.987 mm); D = 1.707A = 2.560 in. (65.037 mm); $t_A = 4.621A = 6.931$ in. (176.060 mm); $V = 1.801A^3 = 6.078$ in.³ (99.601 cm³); $W = V\rho = 0.551$ lb (0.250 kg); $Q_{MIN} = V\rho a_G f_S/J = 0.017$ in.² (10.968 mm²); $Q_{MAX C} = 0.599A^2 = 1.348$ in.² (869.514 mm²).

6.4.12 Pechan prism

The Pechan prism has an odd number (5) of reflections and is frequently used as a compact image rotator in place of the Dove or double-Dove prisms because it can be used in convergent or divergent beams. The design is shown in Fig. 6.22. The optical axis of the nominal design is displaced very slightly owing to the small central air space, but it is not deviated. The two outer reflecting surfaces must have reflective coatings and protective overcoat and/or paint, while the internal reflections occur by TIR so those surfaces are not coated.

The two prisms are usually held mechanically or bonded to a common mounting plate to create a narrow air space (dimension b in the figure) between them. A gap on the order of 0.1 mm (0.004 in.) is typical. Thin shims of the proper thickness can be placed near the edges of these reflecting surfaces in a clamped mounting. The edges of the air space should be covered by a narrow ribbon of sealant such as RTV to prevent entry of moisture or dust.

6.4.13 Penta prism

The penta prism turns the axis by exactly 90 deg without turning the image over in either meridian. It thus provides constant deviation in the plane of refraction. In the plane normal to the reflection plane, it acts as a mirror by deviating the beam axis by twice the prism tilt. This prism is most frequently used in optical range finders, surveying equipment, optical alignment systems, and metrology equipment where accuracy of the 90-deg deviation is essential. The reflecting surfaces must have reflective coatings and act as second surface mirrors. The design is defined in Fig. 6.23.

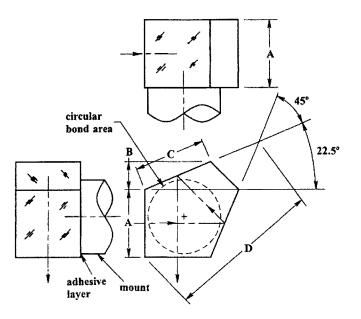


Figure 6.23 Penta prism. (For design and analysis, use File 6.18 of the CD-ROM.) $t_A = 3.414A = 5.121$ in. (130.073 mm); B = 0.414A = 0.621 in. (15.773 mm); C = 1.082A = 1.623 in. (41.224 mm); D = 2.414A = 3.621 in. (91.973 mm); $V = 1.500A^3 = 5.062$ in.³ (82.951 cm³); $W = V\rho = 0.459$ lb (0.208 kg); $Q_{MIN} = V\rho a_G f_S/J = 0.014$ in.² (9.032 mm²); $Q_{MAX C} = 1.129A^2 = 2.540$ in.² (1638.868 mm²).

6.4.14 Roof penta prism

If either reflecting surface of a penta prism is a 90-deg roof, the prism inverts the image in the direction normal to the plane of reflection (see Fig. 6.24). It still deviates the LOS by 90 deg. For a given aperture and glass, the roof penta is about 17% larger and 19% heavier than the penta. The roof angle must be accurate within a few arcseconds to prevent image doubling.

6.4.15 Amici/penta erecting system

An Amici prism combined with a penta prism provides two reflections in each direction perpendicular to the axis so it can be used as an erecting system. Usually the prisms are cemented together as illustrated in Fig. 6.25(a). This design has been used in some binoculars. A functionally similar erecting system can be obtained by combining a right-angle prism with a roof penta prism [see Fig. 6.25(b)]. For a particular aperture A, the indicated height dimensions differ by about 4%. This system has primarily been used in an experimental compact military binocular.⁶ This prism is illustrated in Fig. 6.26. Its 90-deg roof angle can be tested more easily than the corresponding angle of a roof penta prism because it is accessible at normal incidence before the two prisms are cemented together. TIR occurs at the roof surfaces, but the other surfaces must have reflective coatings.

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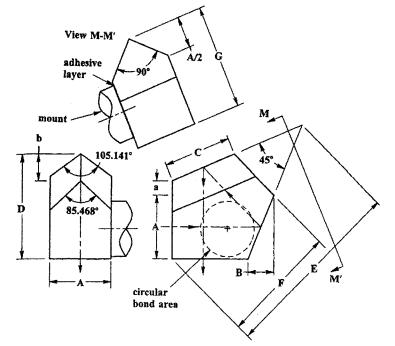


Figure 6.24 Roof penta prism. (For design and analysis, use File 6.19 of the CD-ROM.) $t_A = 4.223A = 6.334$ in. (160.896 mm); a = 0.237A = 0.355 in. (9.030 mm); b = 0.383A = 0.574 in. (14.592 mm); B = 0.414A = 0.621 in. (15.773 mm); C = 1.082A = 1.623 in. (41.224 mm); D = 1.651A = 2.476 in. (62.903 mm); E = 2.986A = 4.479 in. (113.767 mm); F = 1.874A = 2.811 in. (71.399 mm); G = 1.621A = 2.431 in. (61.760 mm); $V = 1.795A^3 = 6.058$ in.³ (99.275 cm³; $W = V\rho = 0.552$ lb (0.250 kg); $Q_{MIN} = V\rho a_G f_S/J = 0.017$ in.² (10.670 mm²); $Q_{MAX c} = 0.824A^2 = 1.854$ in.² (1196.126 mm²).

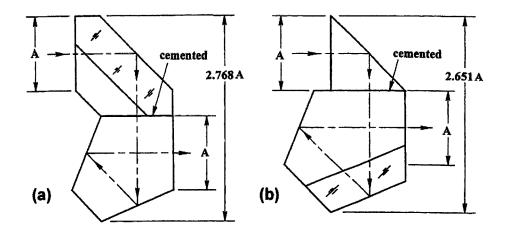


Figure 6.25 Other erecting prism assemblies: (a) Amici/penta system (b) Right angle/roof penta system.

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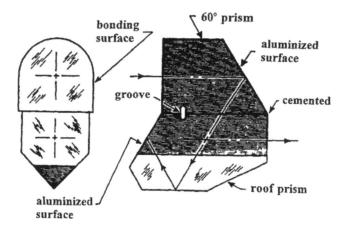


Figure 6.26 A compact erecting system assembly used in an experimental military binocular. (From Yoder.⁶)

6.4.16 Delta prism

Figure 6.27 shows the path of an axial ray through this triangular prism. TIR occurs in sequence at the exit and entrance faces. The intermediate face must be silvered or aluminized to make it reflect. With the proper choice of index of refraction, apex angle, and prism height, the internal path can be made symmetrical about the vertical axis of the prism; the exiting axial ray then is collinear with the entering axial ray. With an odd number of reflections (3), the delta prism can be used as an image rotator. Because it has tilted entrance and exit faces it can be used only in a collimated beam. For a given aperture, the overall size of the delta prism rotator is smaller than the Dove prism.⁷ It has fewer reflections and a shorter t_4 so should have better light transmission than the latter prism.

The design of the delta prism starts with a choice of the index of refraction, n. A value for θ (one-half the apex angle) is then assumed. The angle of incidence, I_1 , at the first surface equals θ . We vary n and θ until the same value for I'_1 is obtained by

$$I_1' = \arcsin\left(\frac{\sin I_1}{n}\right),\tag{6.11}$$

and

$$I_1' = 4\theta - 90 \deg.$$
 (6.12)

We then calculate I_2 from

$$I_2 = 2\theta - I'_1.$$
 (6.13)

This value is compared to I_C from Eq. (6.10) to see if TIR occurs, i.e., $I_2 > I_C$ at the second surface. If so, we continue with the design. If not, we choose a glass with a higher index. For practical reasons, the chosen index must correspond to an available glass type. Typically, TIR will occur for an index >1.7. Figure 6.28 shows the nearly linear variation of θ with n_d for five Schott glasses, four of which are included in Table B1.

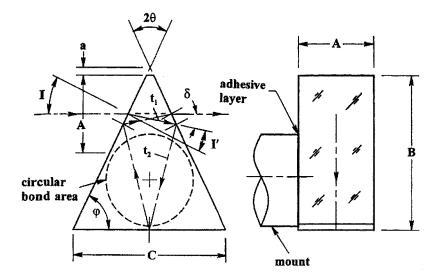


Figure 6.27 Delta prism. (For design and analysis, use File 6.20 of the CD-ROM.) Iterate *n* and θ as explained in the text until Eqs. (6.11 and (6.12) give identical values for I'_1 . Check for TIR. If no TIR, chose higher index glass and repeat. For example, assuming *n* = 1.85025 and θ = 25.916 deg; φ = 90 deg – θ = 64.084 deg; $I_1 = \theta$ = 25.916 deg; I'_1 = arcsin [(sin I)/n] = 13.663 deg; $\delta = I_1 - I'_1$ = 12.253 deg; I_c at surface 2 = arcsin (1/*n*) = 32.715 deg; $I_2 = \delta + \theta = 38.168$ deg; [$I_2 > I_c$, therefore, TIR occurs]; *a* = 0.1*A* = 0.150 in. (3.810 mm); *B* = {(*A* + 2*a*)(sin (180 deg – 4 θ)/[(2)(cos θ)(sin θ)]] – *a* = 2.225 in. (56.508 mm); *C* = 2(*B* + *A*) tan θ = 3.620 in. (91.958 mm); t_1 = [(*A* / 2) + *a*](sin 2 θ)/{[cos θ][sin (90 deg – 2 θ + *I'*)]} = 1.001 in. (25.420 mm); t_2 = [*B* – (*A*/2) – *a* – t_1 sin δ]/cos θ = 1.237 in. (31.413 mm); t_A = 2(t_1 + t_2) = 4.475 in. (113.658 mm); *V* = *A* [(*B* + *a*)² – *a*²] tan θ = 4.094 in.³ (67.087 mm³); *W* = *V* ρ = 0.371 lb (0.168 kg); Q_{MIN} = *V* $\rho a_g f_s/J$ = 0.011 in.² (7.211 mm²); $Q_{MAX} = \pi$ [($C^2/4$) tan²($\varphi/2$)] = 4.131 in.² (2665.156 mm²).

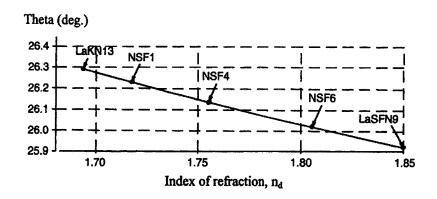


Figure 6.28 Variation of apex half-angle θ with refractive index for TIR in a delta prism.

The numerical values for the delta prism example given in the caption for Fig. 6.27 assume the use of Schott LaSFN9 glass with index of 1.85025 and apply to a ray coincident with the axis. In order for all desired field rays to reflect internally, the extreme lower rim ray must have an angle U_{LRR} with respect to the axis defined by the following equations:

$$I_1' = 2\theta - I_C, \tag{6.14}$$

$$I_1 = \arcsin(n \sin I_1'), \tag{6.15}$$

and

$$U_{\rm LRR} = I_{\rm I} - \theta. \tag{6.16}$$

For the design of Fig. 6.27, $I_C = 32.715 \text{ deg}$, $I'_1 = 19.116 \text{ deg}$, $I_1 = 37.295 \text{ deg}$ and $U_{LRR} = 11.379 \text{ deg}$. Any ray entering the prism at a smaller angle with respect to the axis than this latter value will totally reflect (see Fig. 6.29).

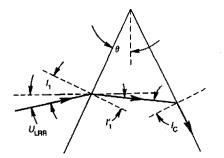


Figure 6.29 Geometry allowing determination of the limiting field ray angle ULRR for TIR in a delta prism.

6.4.17 Schmidt roof prism

The Schmidt roof prism will invert and revert an image, so it is usually used as an erecting system in telescopes. It also deviates the axis by 45 deg, which allows an eyepiece axis orientation to be tilted upward by this angle with respect to the objective axis in applications where the LOS is horizontal. The entrance and exit faces are normal to the axis so it can be used in a converging beam (Fig. 6.30 applies). The prism's refractive index must be high enough for TIR to occur in sequence at the exit and entrance faces.

If a roof is added to the delta prism, an image-erecting system with coaxial input and output optical axes will result. This prism would resemble the Schmidt prism, but the entrance and exit faces would be tilted with respect to the axis, so the roof delta prism must be used in a collimated beam.

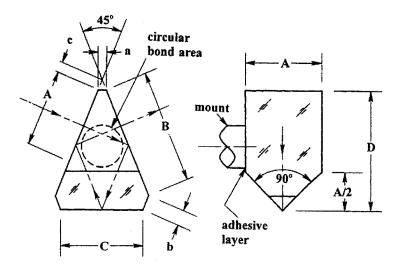


Figure 6.30 Schmidt roof prism. (For design and analysis, use File 6.21 of the CD-ROM.) a = 0.1A = 0.150 in. (3.810 mm); b = 0.185A = 0.277 in. (7.041 mm); c = 0.131A = 0.196 in. (4.980 mm); B = 1.468A = 2.202 in. (55.942 mm); C = 1.082A = 1.624 in. (41.239 mm); D = 1.527A = 2.291 in. (58.194 mm); $t_A = 3.045A = 4.568$ in. (116.022 mm); $V = 0.863A^3 = 2.913$ in.³ (47.729 mm³); $W = V\rho = 0.265$ lb (0.120 kg); $Q_{MIN} = V\rho a_G f_S/J = 0.008$ in.² (5.161 mm²); $Q_{MAX c} = 0.318A^2 = 0.715$ in.² (461.612 mm²).

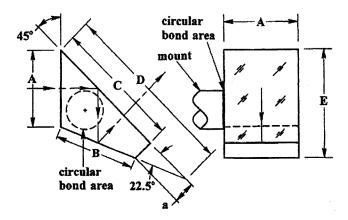


Figure 6.31 45-deg Bauernfeind prism. (For design and analysis, use File 6.22 of the CD-ROM.) $\alpha = 22.5 \text{ deg}$; $\beta = 45 \text{ deg}$; $\delta = 45 \text{ deg}$; a = 0.293A = 0.439 in. (11.163 mm); B = 1.082A = 1.623 in (41.224 mm); C = 1.707A = 2.561 in. (65.040 mm); D = 2.414A = 3.621 in. (91.981 mm); E = 1.414A = 2.121 in. (53.873 mm); $t_A = 1.707A = 2.561$ in. (65.040 mm); $V = 0.750A^3 = 2.531$ in.³ (41.480 cm³); W = Vp = 0.229 lb (0104 kg); $Q_{\text{MIN}} = Vpa_G f_S / J = 0.007$ in.² (4.458 mm²); $Q_{\text{MAX C}} = 0.331A^2 = 0.745$ in.² (480.483 mm²).

6.4.18 The 45-deg Bauernfeind prism

This Bauernfeind prism provides a 45-deg deviation of the axis using two internal reflections. The first reflection is by TIR while the second takes place at a coated reflecting surface. The smaller element of the Pechan prism is of this type. Figure 6.31 shows the design. A 60-deg deviation version of this prism has also been used in many applications.

The combination of a Schmidt prism with a 45-deg Bauernfeind prism forms a popular erecting system for binoculars because of its compact design. It sometimes is called the Schmidt-Pechan roof prism.

6.4.19 Frankford Arsenal prisms nos. 1 and 2

A set of seven special purpose prisms described at the U.S. Army's Frankford Arsenal for use in military telescopes with various requirements for beam deviation to accommodate specific applications were described by Otto Kaspereit.⁴ Rather that being named for their inventors, as was the prior custom, these prisms were named for their place of origin. We show, in Figs. 6.32 and 6.33, the first two of these prisms. Both invert the image in both directions so they can be used with an objective and eyepiece to provide an erect image to the observer. They can be thought of as variations of the Amici prism. F.A. Prism No. 1 deviates the LOS by 115 deg and F.A. prism No. 2 deviates it by 60 deg. By comparison, the Amici prism deviates it by 90 deg. A common use for these three prisms is at the lower fold location of a vertical-offset periscope. The choice as to which to use is based on the LOS angle from the horizontal at the eyepiece. The observer can look upward by 30 deg with F.A. prism No. 2, straight ahead with the Amici prism, or downward by 30 deg with F.A. prism No.1.

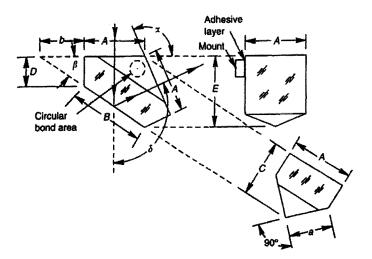


Figure 6.32 Frankford Arsenal prism No. 1. (For design and analysis, use File. 6.23 of the CD-ROM.) $\alpha = \delta = 115 \text{ deg}$; $\beta = 32.5 \text{ deg}$; a = 0.707A = 1.061 in. (26.941mm); b = 0.732A = 1.098 in. (27.889 mm); B = 1.186A = 1.779 in. (45.187 mm); C = 0.931A = 1.396 in. (35.471 mm); D = 0.461A = 0.691 in. (17.564 mm); E = 1.104A = 1.656 in. (42.062 mm); $t_A = 1.570A = 2.355 \text{ in.}$ (59.817 mm); $Q_{MAX \ C} = 0.119A^2 = 0.268 \text{ in.}^2$ (172.875 mm²). (Adapted from Kaspereit.⁴)

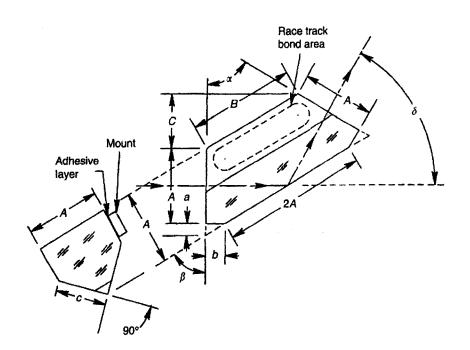


Figure 6.33 Frankford Arsenal prism No. 2. (For design and analysis, use File 6.24 of the CD-ROM.) $\alpha = \beta = \delta = 60 \text{ deg}$; a = 0.155A = 0.232 in. (5.893 mm); b = 0.268A = 0.402 in. (10.211 mm); c = 0.707A = 1.061 in. (26.949 mm); B = 1.464A = 2.196 in. (55.778 mm); C = 0.732A = 1.098 in. (27.889 mm); $t_A = 2.269\text{A} = 3.403 \text{ in}$. (86.436 mm); $Q_{\text{MAX }RT} = 0.776\text{A}^2 = 1.746 \text{ in}$.² (1126.449 mm²). (Adapted from Kaspereit.⁴)

6.4.20 Leman prism

The Leman prism, shown in Fig. 6.34, is most commonly used in binoculars. This is because it provides a large offset of the axis, thus increasing the separation of the instrument's objectives and increasing stereoscopic depth perception as compared to other binocular designs (see Fig. 6.35). At maximum interpupillary distance (IPD), which is typically 72 mm, each prism offsets the axis by 3A so the maximum objective axis separation is 6A + 72mm. Note that the prism face width A in each of these instruments would approximately equal the objective aperture. For observers with smaller IPDs (it can be < 52 mm), the stereoscopic advantage of the Leman prism design is reduced. In binoculars with in-line roof prisms (see Fig. 4.50), the objective and eye piece separations are equal and no depth perception enhancement occurs.

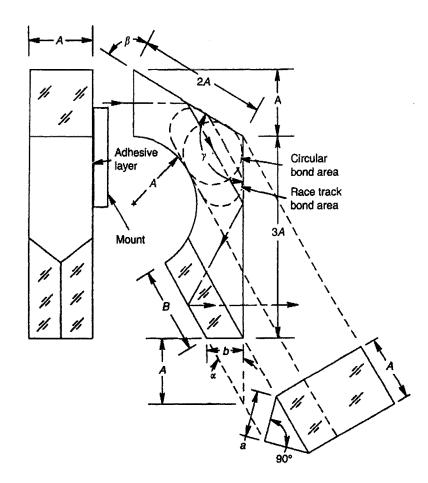


Figure 6.34 The Leman prism. (For design and analysis, use File 6.25 of the CD-ROM). α = 30 deg; β = 60 deg; γ = 120 deg; a = 0.707A = 1.061 in. (26.949 mm); b = 0.577A = 0.866 in. (21.966 mm); B = 1.310A = 1.965 in. (49.911 mm); C = 0.732A = 1.098 in. (27.889 mm); t_A = 5.196A = 7.794 in. (197.968 mm); $Q_{MAX C}$ = 0.676 A^2 = 1.522 in.² (981.829 mm²); $Q_{MAX RT}$ = 0.977 A^2 = 2.198 in.² (1418.393 mm²).

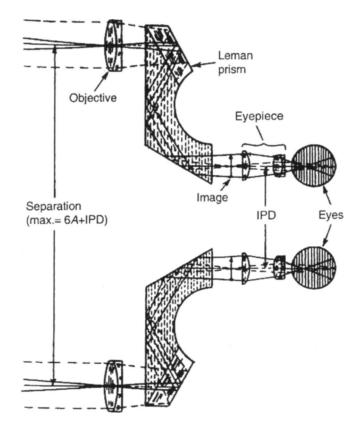


Figure 6.35 Application of the Leman prism to a binocular. (Adapted from Kaspereit.¹⁰)

6.4.21 Internally reflecting axicon prism

With conical surfaces as their active optical surfaces, axicons are frequently used to change a small circular laser beam into an annular beam with a larger outside diameter. The version shown in Fig. 6.36 has a coated reflecting surface to return the beam to and through the conical surface. Because of its rotational symmetry, this axicon is made with a circular cross-section and usually is elastomerically secured in a tubular mount. The apex is sharp or carries a very small protective bevel. A centrally perforated flat mirror at 45 deg can provide a convenient way to separate the coaxial beams if it is located in front of this prism.

An in-line refracting version of this axicon with identical conical surfaces at either end has been used to accomplish the same function, but without the reversal of beam direction. It is twice as long and is more expensive to fabricate because of the additional conical surface.

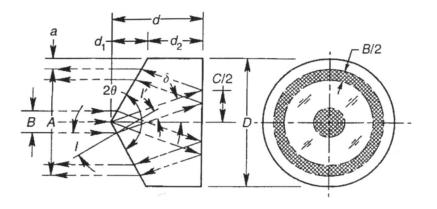


Figure 6.36 Internally reflecting axicon prism. (For design and analysis, use File 6.26 of the CD-ROM.) A = annulus OD = 1.500 in. (38.100 mm); B = input beam OD [assumed] = 0.118 in. (3.000 mm); annulus width = B/2 = 0.059 in. (1.500 mm); a = 0.100A = 0.150 in. (3.810 mm); θ = 60 deg; I_1 = 90 deg – θ = 30 deg; I_1 = arcsin (sin I_1/n) = 19.247 deg; $\delta = I_1 - I_1 = 10.753$ deg; $d = (A/4)[(1/tan <math>\theta) + (1/tan \delta)] = 2.191$ in. (55.642 mm); $d_1 = [(A/2) + a]/tan \theta = 0.520$ in. (13.198 mm); $d_2 = d - d_1 = 1.671$ in. (42.444 mm); $C = (2d)tan \delta = 0.832$ in. (21.139 mm); D = A + 2a = 1.800 in. (45.720 mm); $t_A = A/(2sin \delta) = 4.019$ in. (102.079 mm); $V = (0.785d_2 + 0.262d_1)A^2 = 3.258$ in.³ (53.385 mm³); $W = V\rho = 0.296$ lb (0.134 kg).

6.4.22 Cube corner prism

A corner cut symmetrically and diagonally from a solid glass cube creates a prism in the geometrical form of a tetrahedron (four-sided polyhedron). It has been referred to as a cube corner, corner cube, or tetrahedral prism. Light entering the diagonal face reflects internally from the other three faces and exits through the diagonal face. TIR usually occurs at each internal surface for commonly used refractive index values. The return beam contains six segments, one from each of the pie-shaped areas within the circular aperture shown in Fig. 6.36. If the three dihedral angles between the adjacent reflecting surfaces are exactly 90 deg, the prism is retrodirective, even if the prism is significantly tilted with respect to the input beam. The retrodirective nature of this prism is used to advantage in such applications as laser tracking of cooperative targets on Earth or in space.

The generic cube corner prism of Fig. 6.37 has a triangular form with sharp dihedral edges. If one or more of these dihedral angles differs from 90 deg by an error ε , the deviation differs from 180 deg by as much as 3.26 ε and the reflected beams diverge.⁸ This fact actually is an advantage in some applications in that it spreads the return composite beam to facilitate capture by receivers that are not coaxial with the transmitted beam.

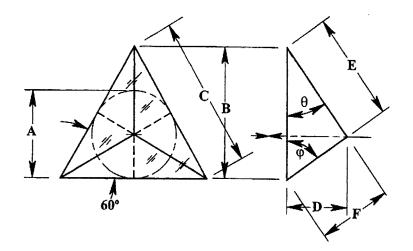


Figure 6.37 Cube corner prism. (For design and analysis, use File 6.27 of the CD-ROM.) θ = 35.264 deg; φ = 54.736 deg; A = aperture = 1.500 in. (38.100 mm); B = [(A/2)/sin 30 deg] + (A/2) = 1.500A = 2.250 in. (57.150 mm); C = 2Btan 30 deg = 1.732A = 2.598 in. (65.989 mm); D = 0.707A = 1.060 in (26.937 mm); E = 1.225A = 1.837 in. (46.672 mm); F = 0.866A = 1.299 in. (32.995 mm); t_A = 2D = 1.414A = 2.121 in. (53.873 mm).

Usually the rim of the cube corner prism is ground to a circular shape circumscribing the aperture (the dashed line). Figure 6.38 shows an example. This is one of the 426 fused silica prisms used on the Laser Geodynamic Satellite (LAGEOS) launched by NASA in 1976 to provide scientists with extremely accurate measurements of movements of the Earth's crust as a possible aid to understanding earthquakes, continental movement, and polar motion. The dihedral angles of the prisms were each 1.25 arcsec greater than 90 deg. A laser beam transmitted to the satellite was returned with sufficient divergence to reach a receiver telescope even though the satellite moved significantly during the beam's roundtrip transit time.

Another possible cube-corner prism configuration has the rim cut to a hexagonal shape circumscribing the prism's circular clear aperture. This allows several of the prisms to be tightly grouped together to form a mosaic of closely packed retrodirective prisms, thereby increasing the effective aperture of the group. Mirror versions of the cube-corner prism are frequently used when operation outside the transmission range of normal refracting materials is needed, (see Sect. 9.3). This so-called "hollow cube corner" has a reduced weight for a given aperture.⁹

6.4.23 An ocular prism for a coincidence rangefinder

In order to facilitate explanation of the design of the next prism type, we first describe its application in a split-field coincidence optical rangefinder. See Fig. 6.39. Light beams from a target enter the rangefinder through windows (not shown) at either end of the instrument. These beams are folded toward the center of the instrument by penta prisms. They pass through two objective lenses, which form two images of the target. The ocular prism

assembly combines these images, which are both then viewed by the observer through an eyepiece. Because the target is at a finite distance, the beams enter the rangefinder at slightly different angles so the images do not coincide in the image plane. The compensator shown in one beam is adjusted by the operator to tilt that beam slightly so the images come together and match. They are then said to be coincident.

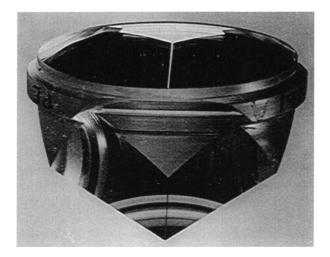


Figure 6.38 Photograph of a precision fused silica cube corner prism with a circular aperture of 1.500 in. (3.810 cm). (Courtesy of Goodrich Corporation, Danbury, CT.)

The optics within the rectangle of the figure are all mounted on a single stiff structural member called an "optical bar," which holds all those optics in alignment. This component must be made of a material with a low CTE so it does not change dimensions or shape significantly when the temperature changes. Although the penta prisms are usually mounted on this bar, it is not essential for them to be attached there because they produce constant deviations in the plane of the figure. Various types of compensators are used to bring the target images into coincidence. The one shown in Fig. 6.39 is an optical wedge that slides along the optical axis (described in Section 6.3.28). The motion of the wedge required to bring the images together is mathematically related to the target range so a nonlinear calibrated scale attached to the wedge allows range information to be obtained.

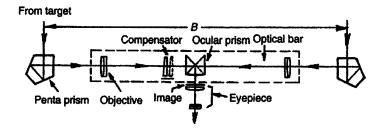


Figure 6.39 Optical schematic of a typical coincidence type optical rangefinder.

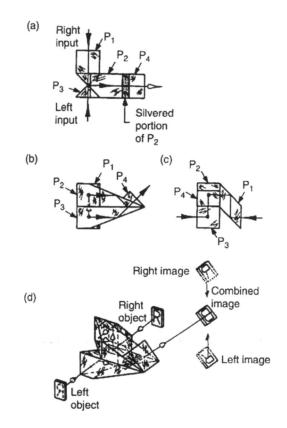


Figure 6.40 An ocular prism designed at Zeiss for a coincidence-type optical rangefinder. (a) Top view, (b) side view, (c) end view, and (d) isometric view. (Adapted from MIL-HDBK-141. 3)

The ocular prism shown in Fig. 6.40 was designed at Carl Zeiss. It comprises four prism elements cemented together. The refracting angles of P_2 , P_3 , and P_4 are all 22.5 deg. The eyepiece axis is inclined upward by 45 deg. The beam from the right objective enters the rhomboid prism P_1 and, after five reflections in P_1 and P_2 , passes through P_4 and focuses at the image plane. The final reflection in this path is at a silvered area on the bottom surface of P_2 . This beam forms the top half of the combined image. The beam from the left objective reflects twice in P_3 and passes through P_4 to the image plane. This beam misses the reflecting area on P_2 and forms the bottom half of the combined image. This rangefinder configuration is called a split-field coincidence type because the image seen by the observer is divided vertically into two parts, coming from different portions of the optical system. When the images are coincident, the observer can read the target range from the scale.

6.4.24 Biocular prism system

The prism system shown in Fig. 6.41 can be used in telescopes and microscopes when both eyes are to observe the same image presented by the objective. It does not provide stereoscopic vision; hence is called "biocular." From Fig. 6.41(a), it can be seen to consist of four prisms: a right-angle prism, P_1 , cemented to a rhomboid prism, P_2 , with a partially reflective coating at the diagonal interface; an optical path equalizing block, P_3 ; and a second rhomboid prism, P_4 . The observer's interpupillary distance is designated as "IPD." By rotating the prisms in opposite directions about the input axis [see view (b)], the IPD is changed to suit the individual using the instrument. The image orientation is not changed by this adjustment.

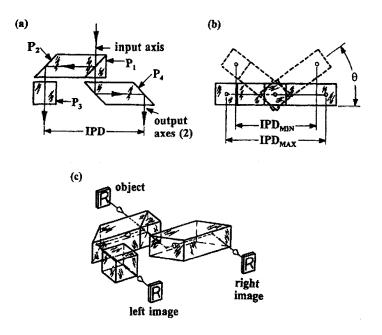


Figure 6.41 Biocular prism system. (a) Top view, (b) end view, and (c) isometric view. IPD is interpupillary distance. Stereoscopic observation of a target is not provided.

6.4.25 Dispersing prisms

Prisms are commonly used to disperse polychromatic light beams into their constituent colors in instruments such as spectrometers and monochromators. The index of refraction, n, of an optical material varies with wavelength, so the deviation of any ray transmitted through a prism at other than normal incidence to the prism's entrance and exit surfaces will depend upon n_{λ} , the angle of incidence at the entrance face, and the prism's apex angle, θ .

Figure 6.42 illustrates two typical dispersing prisms. In each case, a single ray of "white" light is incident at I_1 . Inside each prism, this ray splits into a spectrum of various

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colored rays. For clarity, the angles between rays are exaggerated in the figures. After refraction at the exit faces, rays of blue, yellow, and red wavelengths emerge with different deviation angles, δ_{λ} . The blue ray is deviated the most because $n_{\text{BLUE}} > n_{\text{RED}}$. If the emerging rays are imaged onto a film or a screen by a lens, a multiplicity of images of different colors will be formed at slightly different lateral locations. While the colors here are referred to as blue, yellow, and red, it should be understood that the phenomenon of dispersion applies to all wavelengths, so we really mean the shorter, intermediate, and longer wavelength radiation under consideration in any given application. In the design shown in Fig. 6.42(b), the deviation is unchanged for small rotations of the prism about an axis perpendicular to the plane of refraction; hence the name "constant deviation."

If a single ray or a collimated beam of light of wavelength λ passes symmetrically through a prism so that $I_1 = I_2$ and $I'_1 = I_2$, the deviation of the prism for that wavelength is a minimum and $\delta_{\text{MIN}} = 2I_1 - \theta$. This condition is the basis of one means for experimental measurement of the index of refraction of a transparent medium in which the minimum deviation angle, δ_{MIN} , of a prism made of that material is measured by successive approximations and the following equation is applied:

$$n_{\text{PRISM}} = \frac{\sin\left[\frac{(\theta+\delta)}{2}\right]}{\sin\left(\frac{\theta}{2}\right)}.$$
(6.17)

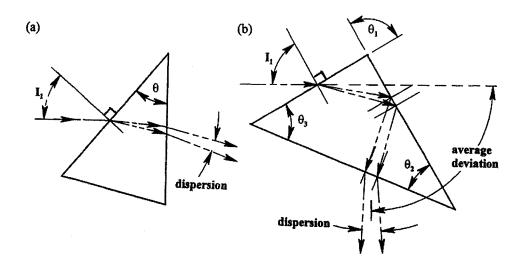


Figure 6.42 Dispersion of a white light ray by (a) a simple prism and (b) a constant-deviation prism involving TIR.

If we want any two of the various colored rays to emerge from the prism parallel to each other, we must use a combination of at least two prisms made of different glasses. Usually, these prisms are cemented together. Such a prism is called an "achromatic prism." Figure 6.43 shows one configuration for an achromatic prism. All such prisms can be designed by choosing refractive indices and the first prism's apex angle, then repeatedly applying Snell's law to find the appropriate incident angle and second prism apex angle that gives the desired deviation for a chosen wavelength and the desired dispersion for two other wavelengths that bracket the chosen one. The angle between the exiting rays with the shortest and longest wavelengths is called the "primary chromatic aberration"; here it should be essentially zero. The angle between either of these extreme wavelength rays and that with an intermediate wavelength is called the "secondary chromatic aberration" of the prism.

To illustrate a typical design procedure, in a two-element prism of the type shown in Fig. 6.43, we might specify that a yellow ray should enter the first prism at I_1 , which should be equal to the value for the minimum deviation condition if that prism were immersed in air. The blue and red rays would then be dispersed. We would assume a value for θ'_1 , calculate $I'_1 = I_2 = \theta_1/2$, and obtain I_1 from Snell's law (Eq. 6.1). We would then add the second prism and redetermine I'_2 . The following equation could then be used to find θ_2 :

$$\cot \tan \theta_2 = \tan I_2' - \left\{ \frac{\Delta n_2}{\left[2\Delta n_1 \sin\left(\frac{\theta_1}{2}\right) \cos I_2' \right]} \right\}.$$
(6.18)

Other than determining the prism glasses and angles required to produce the desired chromatic effect (see Example 6.4), first-order design of a dispersing prism requires calculation of only the required apertures. Usually we assume a collimated input beam and make the apertures of the prism large enough to not vignette any of the dispersed beams. There are so many dispersing prism types that space here does not allow a comprehensive listing of the pertinent equations for computing these apertures. The techniques discussed here for a representative prism type can serve as guidelines for establishing these equations. This task is left to the ingenuity of the reader.

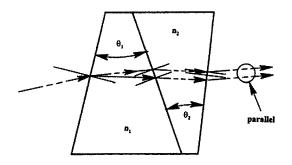


Figure 6.43 A typical achromatic dispersing prism.

Example 6.4: Dispersion through a single prism. (For design and analysis, use File 6.28 of the CD-ROM.)

A BK7 prism with apex angle, θ , of 30 deg disperses a white light collimated beam generally as shown in Fig. 6.42(a). Let the incident angle be $I_1 = 15$ deg. (a) Applying Eq. (6.1) [Snell's law], what are the angular separations between the exiting blue (F), yellow (d), and red (C) beams? (b) If focused by an aberration-free 105 mm focal length lens onto a screen, what are the linear separations of the blue, yellow, and red images at the screen?

(a)wavelength	(µm)	0.486 (F)	0.588 (d)	0.656 (C)	
apex angle, θ	(deg)	30	30	30	
I_1	(deg)	15	15	15	
$\sin I_1$		0.25882	0.25882	0.25882	
n_{λ}		1.52238	1.51680	1.51432	
$\sin I_1$		0.17001	0.17063	0.17091	
I'_1	(deg)	9.7884	9.8247	9.8410	
$I_2 = I'_1 - \theta$	(deg)	20.2116	20.1753	20.1590	
$\sin I_2$		0.34549	0.34489	0.34463	
$\sin I_2$		0.52597	0.52313	0.52188	
I_2	(deg)	31.7332	31.5427	31.4581	
$\delta = I_1 - I'_2 - \theta$	(deg)	16.7332	16.5427	16.4581	
The angular separations between the: blue and yellow beams = $16.7332 \text{ deg} - 16.5427 \text{ deg} = 0.1905 \text{ deg}$, yellow and red beams = $16.5427 \text{ deg} - 16.4581 \text{ deg} = 0.0846 \text{ deg}$, red and blue beams = $16.7332 \text{ deg} - 16.4581 \text{ deg} = 0.2751 \text{ deg}$.					
(b)The image separations = EFL tan (Δ angle):					
blue to yellow			$= 105 \tan 0.1905 \deg = 0.3491 \operatorname{mm} (0.0137 \operatorname{in.}),$		
yellow to red			$= 105 \tan 0.0846 \deg = 0.1550 \operatorname{mm} (0.0061 \operatorname{in.}),$		
red to b	olue	= 105 ta	an $0.2751 \deg = 0.2751 \deg = 0.2251 \deg$	5041 mm (0.0198 in.).	
1					

6.4.26 Thin wedge prisms

Prisms with small apex angles and (usually) axial thicknesses that are small compared with the component apertures are called "optical wedges." One such wedge is shown in Fig. 6.44. Since the apex angle is small, we can assume that the angle expressed in radians equals its sine, and, rewriting Eq. (6.17), we obtain the following simple equation for the wedge deviation:

$$\delta = (n-1)\theta. \tag{6.19}$$

Differentiating this equation, we obtain the following expression for the angular dispersion, i.e., chromatic aberration, of the wedge:

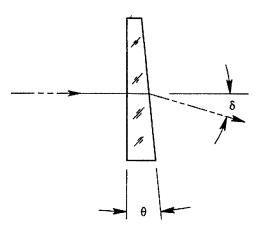


Figure 6.44 A typical thin wedge.

$$d\delta_{\lambda} = dn_{\lambda}\theta. \tag{6.20}$$

The angles in these equations are radians. For small angles, sufficient accuracy is usually obtained using angles in arcsec, arcmin, or degrees.

A wedge so designed is one of minimum deviation. A common arrangement in optical instruments has the incident beam normal to the entrance face. Then $I_2 = \theta$, $I'_2 = \arcsin(n \sin I_2)$, and $\delta = I'_2 - \theta$. If not otherwise specified, we would assume n to apply to the center wavelength of the spectral bandwidth of interest. See Example 6.5.

Example 6.5: Calculate deviation of an optical wedge. (For design and analysis, use File 6.29 of the CD-ROM).

Assume a thin wedge has an apex angle of 1.9458 deg. (a) What is its deviation if the glass index is 1.51680? (b) What is its chromatic aberration for wavelengths corresponding to indices of 1.51432 and 1.52238?

(a) By Eq. (6.19), $\delta = (1.51680 - 1)(1.9458) = 1.0056$ deg. (b) By Eq. (6.20), $d\delta = (1.52238 - 1.51432)(1.9458) = 0.0157$ deg.

6.4.27 Risley wedge system

Two identical optical wedges arranged in series and rotated equally in opposite directions about the optical axis form an adjustable wedge. They are used in collimated beams to provide variable pointing of laser beams, to angularly align the axis of one portion of an optical system to that of another portion of that system, as the means for measuring distance in some optical range finders, etc. They frequently are referred to as Risley wedges. Another name for the system is "diasporometer."

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The action of a Risley wedge system will be understood from Fig. 6.45. Usually the wedges are circular in shape; here their apertures are shown as small and large rectangles for clarity. In views (a) and (c), the wedges are shown in their two positions for maximum deviation. The apexes are adjacent and $\delta_{\text{SYSTEM}} = \pm 2\delta$, where δ is the deviation of one wedge. If the wedges are turned from either maximum deviation position in opposite directions by β [see view (d)], the deviation becomes $\delta_{\text{SYSTEM}} = \pm 2\delta \cos \beta$ and the change in deviation from the maximum achievable value is $2\delta(1 - 2\cos \beta)$. If we continue to turn the wedges until $\beta = 90$ deg, we obtain the condition shown in view (b) where the apexes are opposite, the system acts as a plane parallel plate, and the deviation is zero.

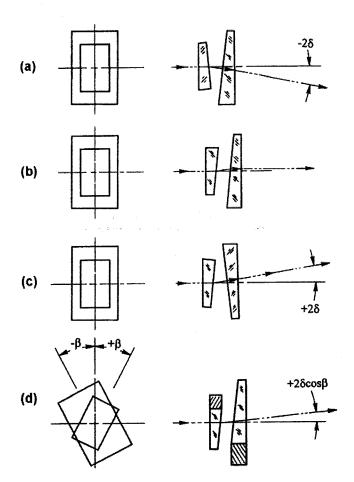


Figure 6.45 Function of a Risley wedge prism system, (a) bases down, (b) bases opposed, (c) bases up, (d) wedges counter rotated by $\pm \beta$.

Since counter rotation of the wedges in a Risley wedge system provides variable deviation in one axis, a second such system, usually identical to the first, is sometimes added to provide independent variation in both orthogonal axes. The deviations from the two systems add vectorially in a rectangular coordinate system. Another arrangement has a single Risley wedge system mounted so both wedges can be rotated together about the optical axis as well as counter rotated with respect to each other. This provides a variation of deviation in a polar coordinate system.

6.4.28 Sliding wedge

A wedge prism located in a converging beam will deviate the beam so the image is displaced laterally by an amount proportional to the wedge deviation and the distance from the wedge to the image plane. See Fig. 6.46 for a schematic of the device. If the prism is moved axially by $D_2 - D_1$, the image displacement varies from $D_1\delta$ to $D_2\delta$. This device was most frequently used in military optical range finders before the advent of the laser range finder. The principle can be used in other more contemporary applications in which an image needs to be variably displaced laterally by a small distance. If used with a long focal length lens, the wedge should be achromatic.

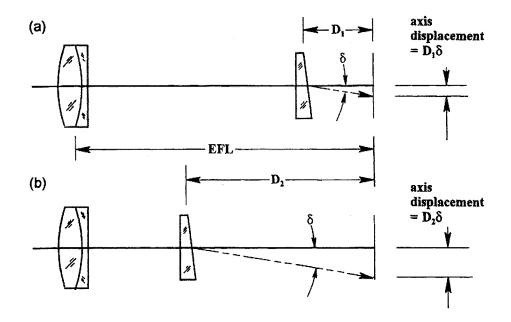


Figure 6.46 A sliding wedge beam deviating system.

6.4.29 Focus adjusting wedge system

Two identical optical wedges arranged with their bases opposite and mechanized so each can be translated laterally by equal amounts relative to the optical axis provide a variable optical path through glass. Figure 6.47 shows the device's principle of operation. At all settings, the two wedges act as a plane-parallel plate. If located in a convergent beam, this system allows the image distance to be varied and can be used to bring images of objects at different distances into focus at a fixed image plane. This type of focus-adjusting system is sometimes used in large-aperture aerial cameras and telescopes such as those found in tracking missiles or spacecraft launch vehicles, where target ranges change rapidly after launch and the large, heavy image forming optics are cannot be moved rapidly or precisely to maintain focus. To a first-order approximation, $t_i = t_0 \pm \Delta y_1 \tan \theta$. The focus variation is $\pm 2t_i [(n-1)/n]$. Here t_0 is the axial thickness of each wedge at its center.

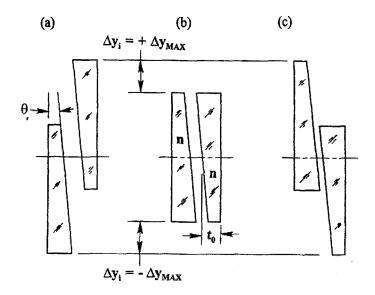


Figure 6.47 A focus adjusting wedge system. (a) Minimum path, (b) nominal path, (c) maximum path.

Figure 6.48 shows the optical schematic for a typical camera application featuring a focus adjusting wedge system. The changes in glass path as the wedges are moved may cause the aberration balance of the optical system to change. This would limit the focus adjustment range in high-performance applications.

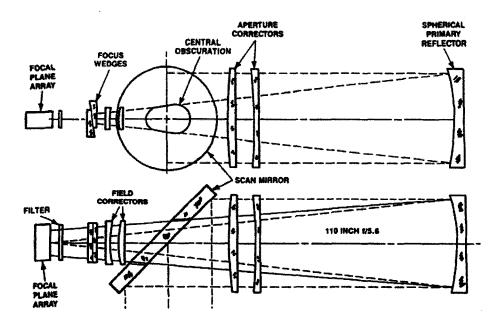


Figure 6.48 Top and side views of the optical system for a 110-in. (2.79-m) focal length, f/5.6 aerial camera lens with a wedge focus adjustment. (From Ulmes.¹¹)

6.4.30 Anamorphic prism systems

If a refracting prism is used at other than minimum deviation, it changes the width of a transmitted collimated beam in the plane of refraction [see Fig. 6.49(a)]. The beam width in the orthogonal direction is not changed so anamorphic magnification results. Beam angular deviation and chromatic aberration are introduced. Both of these defects can be eliminated if two identical prisms are arranged in opposition, as shown in Fig. 6.49(b). Lateral displacement of the axis then occurs, but the angular deviation and chromatic aberration are zero. The beam compression depends upon the prism apex angles, the refractive indices, and the orientations of the two prisms relative to the input axis. The configuration of Fig. 6.49(b) is a unity power telescope in the meridian perpendicular to the figure since the width and degree of the beam collimation are unchanged while it is passing through the optics in that meridian.

Two-prism anamorphic telescopes are attributed by Kingslake¹² to Brewster in about 1835 as a replacement for the cylindrical lenses then used for the purpose. They are commonly used today to change diode-laser beam size and angular divergence differentially in orthogonal directions. The telescope shown in Fig. 6.50(a) has achromatic prisms to allow a broad spectral range to be covered.¹³ Anamorphic telescopes with many cascaded prisms to produce higher magnification have been described.^{14,15} An extreme example with 10 prisms is shown in Fig. 6.50(b). This configuration is reported to be optimal for single material achromatic expanders of moderate to large magnifications.¹⁵ An anamorphic telescope example consisting of only one prism is shown in Fig. 6.50(c).¹⁶ It has three active faces, one of which functions by TIR. The entrance and exit faces can be oriented at Brewster's angle, so the surface reflection (Fresnel) losses are eliminated for polarized beams.

Single material (fused silica) anamorphic prism assemblies have been used quite successfully to convert rectangular Excimer-laser beams into more suitable square ones for materials processing and surgical applications.¹⁷

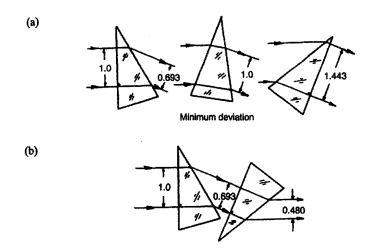


Figure 6.49 Function of anamorphic prisms. (a) Individual prisms at various incident angles. (b) An anamorphic telescope. (Adapted from Kingslake.¹²)

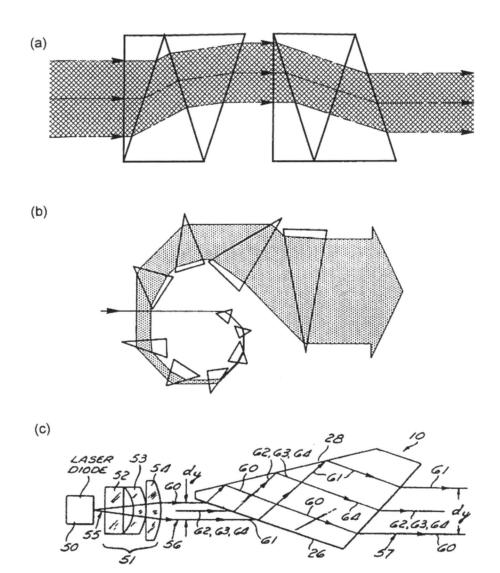


Figure 6.50 Three anamorphic prism telescope configurations. (a) Achromatic prism assembly (adapted from Lohmann and Stork¹³), (b) cascaded assembly (adapted from Trebino¹⁵), (c) single prism telescope (from Forkner.¹⁶)

6.5 References

- 1. Yoder, P.R., Jr., *Opto-Mechanical Systems Design*, 3rd ed., CRC Press, Boca Raton, 2005.
- 2. Smith, W.J., *Modern Optical Engineering*, 3rd ed., McGraw-Hill, New York, 2000.
- 3. MIL-HDBK-141, Optical Design, U.S. Defense Supply Agency, Washington, 1962.
- 4. Kaspereit, O.K., Design of Fire Control Optics, ORDM 2-1, Vol. I, U.S. Army, 1953.

- 5. Smith, W.J., Sect. 2 in *Handbook of Optics*, Optical Society of America, Washington, 1978.
- 6. Yoder, P.R., Jr., "Two new lightweight military binoculars," J. Opt. Soc. Am. 50, 1960:49.
- 7. Durie, D.S.L., "A compact derotator design," Opt. Eng. 13, 1974:19.
- 8. Yoder, P.R., Jr., "Study of light deviation errors in triple mirrors and tetrahedral prisms," J. Opt. Soc. Am. 48, 1958:496.
- 9. PLX, Inc. sales literature, Hard-Mounted Hollow Retroreflector, PLX, Deer Park, NY.
- 10. Kaspereit, O.K., *Designing of Opt ical Syst ems f or Tel escopes*, Ordnance Technical Notes No. 14, U.S. Army, Washington, 1933.
- 11. Ulmes, J.J., "Design of a catadioptric lens for long-range oblique aerial reconnaissance," *Proceedings of SPIE* **1113**, 1989:116.
- 12. Kingslake, R., Optical System Design, Academic Press, Orlando, 1983.
- 13. Lohmann, A.W., and Stork, W., "Modified Brewster telescopes," Appl. Opt. 28, 1989:1318.
- 14. Trebino, R., "Achromatic N-prism beam expanders: optimal configurations," *Appl. Opt.* 24, 1985:1130.
- 15. Trebino, R., Barker, C.E., and Siegman, A.E., "Achromatic N-prism beam expanders: optimal configurations II," *Proceedings of SPIE* **540**, 1985:104.
- 16. Forkner, J.F., "Anamorphic prism for beam shaping," U.S. Patent No. 4,623,225, 1986.
- 17. Yoder, P.R., Jr., "Optical engineering of an excimer laser ophthalmic surgery system," *Proceedings of SPIE* **1442**, 1990:162.

CHAPTER 7 Techniques for Mounting Prisms

In this chapter, we consider several techniques for mounting individual prisms in optical instruments by kinematic, semikinematic, and nonkinematic clamping as well as bonding them to mechanical structures. Techniques for mounting larger prisms on flexures are also described. Although most of the discussions deal with glass prisms interfacing with metal mountings, the designs would generally be applicable to prisms made of optical crystals and to attaching prisms to nonmetallic cells, brackets, and housings. Numerous examples are included to illustrate the use of important design principles.

7.1 Kinematic Mountings

Three positional degrees of freedom (DOFs), i.e., translations, and three orientational DOFs, i.e., tilts, of a prism must be established during assembly and then carefully controlled to tolerances that are dependent upon the optic's location and function within the optical system. Within each class of DOFs, movements are orthogonal to each other and generally correspond to the defined axes of the optical system. In a true kinematic mounting, all six DOFs are uniquely controlled by six constraints at the prism's interfaces with its mechanical surround while six forces hold the prism against those constraints. If possible, the optical material should always be placed in compression by the mounting forces at all temperatures. If more than six constraints and/or forces are applied, the mounting is overconstrained, i.e., nonkinematic. Distortions of the optical surfaces and stress buildup within the prism might then result.

Figure 2.17 illustrates an idealized kinematic mounting for a cube-shaped prism. It is repeated here as Fig. 7.1. The sketch at left indicates how the required six constraints might be provided for a simple cube-shaped prism. A series of six balls of equal diameter are attached to three mutually orthogonal flat surfaces. If the body is held in point contact with all six balls, it will be uniquely or kinematically constrained. Three points in the X-Z plane define a plane on which the lower face of the body rests; these points prevent translation in the Y direction and rotation about the X and Z axes. Two points in the Y-Z plane prevent translation along the X axis and rotation about the Y axis. The single point in the X-Y plane controls the last degree of freedom (translation along the Z axis). Six forces, each perpendicular to one of the flat surfaces and directed through the center of one ball would hold the prism in place. Alternatively, a single force exerted against the outermost corner of the prism and directed toward the origin also could hold the body against all six balls. Ideally, this force should pass through the center of gravity of the body. Components of this single force would then serve as the aforementioned six individual forces. If this applied force or the six individual forces are light and gravity is ignored, the contact surfaces are not deformed and we have a kinematic design.

The minimum preload force P in pounds, to be exerted against the prism in order to prevent its lifting off the constraints under acceleration a_G , may be calculated with the help of Eq. (3.1), which is repeated here for convenience:

$$P = Wa_G, \tag{3.1}$$

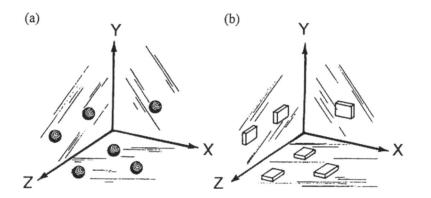


Figure 7.1 Location and orientation defining reference interfaces for a cubeshaped prism: (a) Kinematic mounting with point contacts on six balls; (b) semikinematic mounting with small area contacts. (Adapted from Smith.¹)

where W is the weight of the body in pounds. Note once again that, if the prism weight is expressed in kilograms, this equation must include an additional multiplicative factor of 9.8066 to provide the force in newtons (N).

Because the contacts on the balls have infinitesimal areas, the stresses (force per unit area) introduced into the prism at those contacts are large—perhaps damaging. The prism surfaces would be expected to deflect at the contacts; those deflections may be large enough to degrade optical performance.

7.2 Semikinematic Mountings

View (b) of Fig. 7.1 illustrates one way of reducing this kinematic mount concept to a more reasonable design. The point contacts are replaced by small-area square contacts on pads. The multiple pads in the X-Z and Y-Z planes are machined carefully so that they are very accurately coplanar. The reference surfaces on all the pads must have the proper angular relationship (exactly at 90 deg to each other) so the area contacts with a perfect cube prism do not degenerate into lines. Intimate contacts between the prism faces and the pads over the pad areas result. This is called a semikinematic mounting.

Contact within the clear apertures of optically active surfaces implies obscuration as well as the possibility of surface distortion. Hence, such contact should be avoided. Since reflecting surfaces are much more sensitive to deformation than refracting ones, they are especially critical. Note that TIR surfaces must not touch anything that will frustrate the refractive index mismatch that causes the internal reflection to take place. If the need for periodic cleaning of a TIR surface is anticipated, the design should provide the necessary convenient access.

Figure 7.2(a), similar in concept to Fig. 2.1, schematically shows a semikinematic mounting for a cube-shaped beamsplitter prism similar to one described by Lipshutz.² Here six springs at the points labeled " K_i " preload the cemented prism against six directly

opposite coplanar (i.e., lapped) raised pads indicated as " K_{∞} ." Although several contacts occur on refracting surfaces, they are located outside the optically used clear apertures, thereby avoiding obscurations and minimizing the effects of surface distortions within those apertures. The top view shows that, in the Z direction, the spring and constraint area are displaced from the prism centerline so the interfaces are completely on one half of the cemented prism. This avoids difficulties should the ground faces not be coplanar after cementing. The structure supporting all the fixed points and the springs here is assumed rigid.

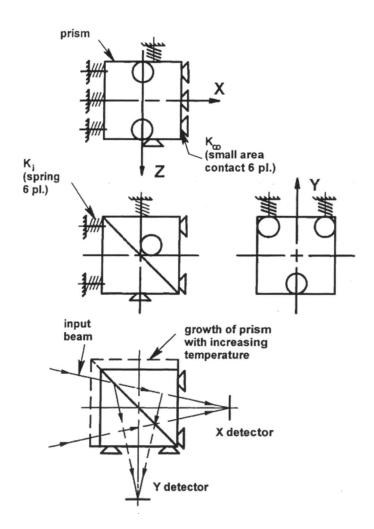


Figure 7.2 (a) Three views of a semikinematic mount for a cube-shaped beamsplitter prism. (b) Schematic of a typical optical function showing the effect of temperature rise. (Adapted from Lipshutz.²)

As shown in Fig. 7.2(b), this beamsplitter is used to divide a beam converging toward an image plane; each beam then forms an image on a separate detector. In order for these images to maintain their proper alignments relative to each other, to the detectors, and to the structure of the optical instrument when the temperature changes, the prism must not translate in the X-Y plane of the figure or rotate about any of the three orthogonal axes. Translation of the prism in the Z direction has no optical effect here, but is controlled. Once aligned, the springs ensure that the prism always presses against the six pads. The dashed outlines in the figures indicate how the prism will expand if the temperature increases. Registry of each prism surface against the pads does not change and the light paths to the detectors do not deviate. This is also true if the temperature decreases.

The preload force, P_i (in pounds), to be exerted by one spring on the prism with the mounting just described may be calculated with the aid of Eq. (7.1). This equation is a minor modification of Eq. (3.1).

$$P_i = \frac{Wa_G}{N},\tag{7.1}$$

where N is the number of springs active in the direction of the preload force. Note that if the prism weight is expressed in kilograms, Eq. (7.1) must once again include the additional multiplicative factor of 9.8066 to give the preload in newtons (N). Friction and moments at the contacts are ignored in the equation. Example 7.1 considers a typical case.

Example 7.1: Clamping force needed to hold a beamsplitter cube prism semikinematically. (For design and analysis, use File 7.1 of the CD-ROM).

A beamsplitter cube weighing 0.518 lb (0.235 kg) is constrained as indicated in Fig. 7.2 and is to withstand accelerations of $a_G = 25$ in any direction. What force should be provided by each spring?

We apply Eq. (7.1) to each case:

Force per spring on the 3-contact face = $\frac{(0.518)(25)}{3}$ = 4.317 lb (19.203 N).
Force per spring on the 2-contact face $=\frac{(0.518)(25)}{2} = 6.475$ lb (28.802 N).
Force per spring on the 1-contact face $=\frac{(0.518)(25)}{1} = 12.950$ lb (57.604 N).

When the prism design is other than a cube, the semikinematic mounting design can be more complex. For instance, it may be difficult or perhaps impossible to apply forces directly opposite support pads. Figure 7.3, adapted from Durie,³ shows two such cases. View (a) shows a right angle prism semikinematically registered against constraints at two of its square refracting faces and one triangular ground face. Three coplanar pads on the baseplate provide constraints in the Y direction while three strategically oriented locating pins pressed into the base plate add three more constraints (X and Z translations and rotation about the Y axis). Ideally, all pads and pins contact the prism outside its optically active apertures (not indicated). In Fig. 7.3(b), the same prism is shown in side view. Note that the required perforations (i.e., apertures) in the plate are not shown in Views (a) and (b). The preload forces F_1 and F_2 are oriented perpendicular to the hypotenuse face and touch the prism near the longer edges of the hypotenuse. F_1 is aimed symmetrically between the nearest pad (b) and the nearest pin (d) while F_2 is aimed symmetrically between pads (a) and (c) and pin (e). Horizontal force F_X holds the prism against pin (f) while vertical components of F_1 and F_2 hold the prism against the three coplanar pads and the remaining two pins. Although it is not optimum in terms of bending tendencies (i.e., moments) because the forces are not directed toward the pads, this arrangement is adequate because the prism is stiff.

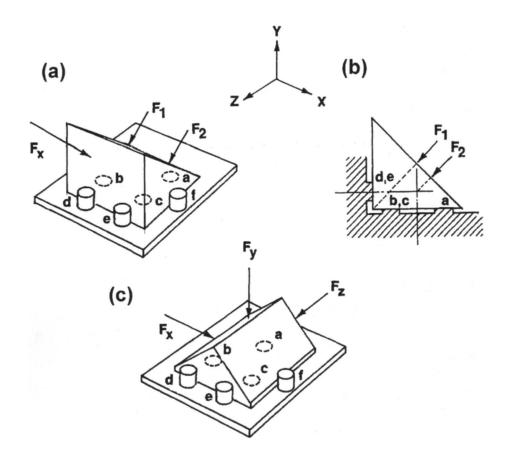


Figure 7.3 Schematics of semikinematic mounts for (a) and (b) a right-angle prism referenced to two refracting faces and one ground face, and (c) a Porro prism referenced to its hypotenuse face, one ground face, and one bevel. (Adapted from Durie.³)

In Fig. 7.3(c), the hypotenuse face of a Porro prism is positioned against three coplanar raised pads on a base plate, while one ground face touches two locating pins and one

beveled edge touches a third pin. A force, F_Z , directed parallel to and slightly above the base plate holds the prism against two pins (d) and (e), while force F_X , also just above the plate, holds it against the third pin (f). A third force, F_Y , holds the prism against the three pads (a), (b), and (c). This force acts against the dihedral edge of the prism at its center. The prism is stiff enough that surface distortion is minimal. Example 7.2 shows how to determine the forces needed in a particular case.

Example 7.2: Clamping force needed to hold a Porro prism semikinematically. (For design and analysis, use File 7.2 of the CD-ROM.)

A Porro prism is constrained as indicated in Fig. 7.3(c). It is made of N-SF8 glass and has an aperture A of 2.875 cm (1.132 in.). The mounting is to withstand accelerations of $a_G = 10$ in any direction. What should be each of the total preloads P_X , P_Y , and P_Z ? Ignore friction.

From Fig. 6.12, the prism volume V is $1.286A^3 = 30.560 \text{ cm}^3$. From Table B1, the glass density ρ is 2.90 g/cm³. Hence, W = (30.560)(2.90) = 88.624 g = 0.089 kg.

From Eq. (7.1): $P_X = P_Y = P_Z = (0.089)(9.8066)(10) = 8.691$ N (1.954 lb).

Figure 7.4 shows a right angle prism referenced to one triangular ground face and to three locating pins in a semikinematic design. The pins contact the refracting faces of the prism outside the clear apertures. The prism is pressed downward against three coplanar raised pads on the baseplate. The clamping plate presses the prism through a resilient (elastomeric) pad under tensile force exerted by three long screws. A leaf spring anchored at both ends (a "straddling" spring) presses the prism against all three locating pins. Other spring types could, of course, be used for the latter purpose. An attractive feature of this mounting is that it can easily be configured so the circular clear apertures of optically active surfaces are not obscured and are not likely to be distorted by the imposed forces.

Assuming the three screws are stiff, the resilient pad provides the spring force (preload) necessary to hold the prism in place under shock and vibration. We can design the subassembly only if the elastic characteristic of the pad material is known. The pertinent material property is the "spring constant," C_P , defined as the load P_i that must be applied normal to the pad surface to produce a unit deflection Δy :

$$C_p = \frac{P_i}{\Delta y}.$$
(7.3)

Most resilient materials have a limited elastic range, tend to creep over time, and typically take a permanent set under a sustained high compressive load, i.e., one greater than that for which the material acts elastically. For these reasons, these materials might be considered unreliable for use in the manner suggested here. However, since they are sometimes used, we consider a typical type of material as it might be used here.

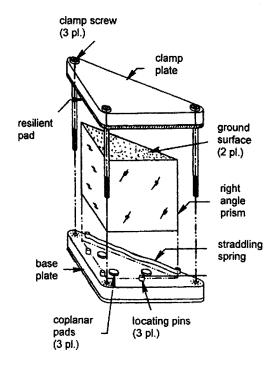


Figure 7.4 A semikinematic mounting for a right angle prism with preload provided by a compressed elastomeric pad. (Adapted from Vukobratovich.⁴)

Sorbothane, a viscoelastic, thermoset, polyether base polyurethane behaves as indicated in Fig. 7.5 for durometers ranging from 30 to 75 and three deflections as percentages of the pad thickness. The material depicted is commonly used in vibration isolators and behaves elastically if the change in thickness is between 10% and 25% of the total thickness.⁵ Obviously, a softer material deflects more per unit load. Since we should design the interface for the maximum applied force (which would occur at maximum acceleration), we might well choose the 20% curve of Fig. 7.5 so deflections under conditions of lesser severity would lie well within the linear range of the material. The manufacturer's literature suggests that deflection, Δy , is related to load, P_i , (in the USC system) as follows:

$$\Delta y = \frac{(0.15)(P_i)(t_p)}{(C_s)(A_p)(1+2\gamma^2)},$$
(7.4)

where: P_i = required force per pad (in N or lb),

 t_P = uncompressed pad thickness (in mm or in.),

$$\gamma = \frac{D_P}{4t_P}$$
, a "shape factor" for a square or circular pad, (7.5)

 D_P = pad width or diameter (in mm or in.),

- $A_P = D_P^2 \text{ for a square pad (in mm² or in.²)},$ (7.6a)
 - $=\pi \frac{D_P^2}{4} \text{ for a circular pad (in mm² or in.²)},$ (7.6b)

 C_s = compressive stress in the pad per Fig. 7.5,

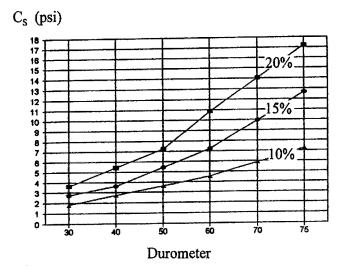


Figure 7.5 Compressive stress (C_S) vs. durometer for a viscoelastic material with deflections of various percentages of pad thickness. (Courtesy of Sorbothane, Inc., Kent, OH.)

Example 7.3 illustrates a typical design for a resilient interface. Similar calculations apply to other resilient materials *if* their elastic properties are available and their force vs. deflection characteristics are appropriate over a reasonable range of deflections.

Another semikinematic mounting is illustrated schematically in Fig. 7.6. Here, a penta prism is pressed against three circular coplanar pads on a base plate. Three cantilevered springs provide the necessary preload directly through the prism against the pads. This provides three constraints: one translation and two tilts. The remaining translations and one tilt are constrained by three locating pins with a straddling leaf spring to provide preload of the prism against the pins.

Equations (3.42) and (3.43) are used to design the cantilevered springs for this application and to check the bending stress in each spring clip just as they were used in Sec. 3.8 in designing similar constraints for lenses. For convenience of the reader, they are repeated here with the deflection defined here as Δy instead of Δx to fit the coordinate system of Fig. 7.6(b).

$$\Delta y = \frac{\left(1 - v_M^2\right) \left(4PL^3\right)}{E_M b t^3 N},$$
(3.42)

$$S_B = \frac{6PL}{\left(bt^2N\right)},\tag{3.43}$$

where E_M is Young's modulus and v_M is Poisson's ratio for the clip material, P is the total preload, L is the free (cantilevered) length of the clip, b and t the width and thickness of the clip, and N is the number of clips employed. As in the case of a lens mounting with clips, the bending stress S_B should not exceed 50% of the yield stress of the clip material.

Example 7.3: Penta prism clamped through a circular resilient pad. (For design and analysis, use File 7.3 of the CD-ROM.)

A SF6 penta prism with aperture A = 2.000 in. (50.800 mm) is to be clamped against three pads on a base plate by a rigid plate in a manner similar to that shown in Fig. 7.4. A circular pad with a thickness of 0.375 in. (9.525 mm) made of a material characterized by Fig. 7.5 is placed between the clamping plate and the prism. (a) Using 30-durometer material, what size pad is required if a 20% deflection is to occur at a maximum acceleration of 10 times gravity? (b) What is C_P for the pad?

(a) From Fig. 6.23, we find that the penta prism volume is $1.5A^3$ and from Table B1, we determine that the density of SF6 glass is 0.186 lb/in.³ (5.18 g/cm³). Hence, the prism weight = $(1.5)(2.000^3)(0.186) = 2.232$ lb. (1.012 kg).

From Eq. (7.1), the preload required is $P_i = \frac{(2.232)(10)}{1} = 22.320$ lb (99.284 N).

From Fig. 7.5, the compressive stress, C_s , in the pad at 30 durometer and 20% deflection is approximately 3.7 lb/in.² (2.55×10⁴ Pa).

From Eq. (7.5),
$$\gamma = \frac{D_p}{4t_p} = \frac{D_p}{(4)(0.375)} = 0.6667D_p$$

From Eq. (7.6b), $A_p = \frac{\pi D_p^2}{4} = 0.785 D_p^2$.

From Eq. (7.4),
$$\Delta y = \frac{(0.15)(22.320)(0.375)}{(3.7)(0.7854D_P^2)[1+(2)(0.6667D_P)]}$$

This deflection also is to be 20% of 0.375 in. or 0.075 in. Equating and applying algebra, we obtain the quadratic equation: $0.194D_P^4 + 0.218D_P^2 - 1.256 = 0$.

Solving for D_P^2 and taking the square root, we obtain $D_P = 1.430$ in. (3.632 cm).

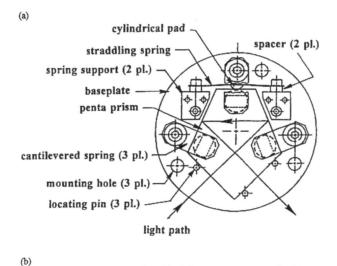
We need to see if the pad will fit onto the side of the prism. From Fig. 6.23, the largest circular area that can be inscribed within the pentagonal face of the prism is found to be $Q_{\text{MAX C}} = 1.13\text{A}^2 = (1.13)(2.000)^2 = 4.520 \text{ in.}^2$. From this we derive

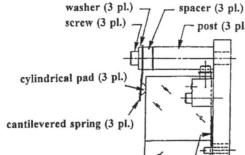
$$D_{PMAX} = \left[\frac{(4)(4.520)}{\pi}\right]^{1/2} = 2.399 \text{ in.}(6.094 \text{ cm}).$$

This shows that the pad will easily fit on the prism face.

(b) The pad's spring constant is $C_P = \frac{22.320}{0.075} = 297.6 \text{ lb/in.}(521.2 \text{ N/cm}).$

A smaller pad would be used if it were a stiffer material (such as 70 durometer) or if the thickness t_P were reduced.





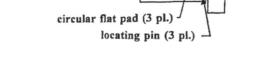


Figure 7.6 Semikinematic mounting for a penta prism with cantilevered and straddling spring constraints: (a) plan view, (b) elevation view.

Another potentially useful equation pertinent to cantilevered spring design is the angle at which the end of the cantilevered portion of the deflected spring is bent relative to the fixed portion of the spring. This equation (adapted from Roark⁶) is

$$\varphi = \frac{\left(1 - v_M^2\right) \left(6L^2 P\right)}{\left(E_M b t^3 N\right)}.$$
(7.7)

post (3 pl.)

Example 7.4 demonstrates the use of these equations.

Example 7.4: Design of a cantilevered spring clip mounting for a prism. (For design and analysis, use File 7.4 of the CD-ROM.)

Assume the following dimensions and material properties for a prism mounting per Fig. 7.6(b) using clips of BeCu: W = 0.267 lb (0.121 kg), $a_G = 12$, N = 3, $S_Y = 155,000$ lb/in.² (1.069×10³ MPa), $E_M = 18.5 \times 10^6$ lb/in.² (1.27×10⁵ Pa), $v_M = 0.35$, L = 0.375 in. (9.525 mm), b = 0.250 in. (6.350 mm), and t = 0.020 in. (0.508 mm). (a) What should be each clip deflection? (b) What bending stress results in that clip? (c) What is its safety factor? (d) Through what angle is each clip bent?

(a) By Eq. (3.1),
$$P = (0.267)(12) = 3.204$$
 lb (14.252 N).
From Eq. (3.42):

$$\Delta y = \frac{(1 - 0.35^2)(4)(0.375^3)(3.204)}{\left[(18.5 \times 10^6)(0.250)(0.020^3)(3)\right]} = 0.0053$$
 in. (0.135 mm).

(b) From Eq. (3.43):

(c)

$$S_{B} = \frac{(6)(0.3/5)(1.068)}{\left[(0.250)(0.020^{2})\right]} = 24,030 \text{ lb/in.}^{2}(165.7 \text{ MPa})$$

$$f_s = \frac{155,000}{24,030} = 6.45$$

(d)
$$\varphi = \frac{(1 - 0.35^2)(6)(0.375^2)(1.068)}{\left[(18.5 \times 10^6)(0.250)(0.020^3)\right]} = 0.021 \, \text{rad} = 1.22 \, \text{deg} \; .$$

The result for Δy in this example is smaller than might be desired for accurate measurement and f_S is larger than necessary. We could decrease t to make a more reasonable design. To do this most easily, we derive an equation for the "optimum" cantilevered spring-clip thickness by setting Eq. (3.43) equal to S_Y/f_S for the chosen spring material. We obtain the following equation:

$$t = \left[\frac{6P_i L f_s}{(bS_\gamma)}\right]^{1/2}.$$
(7.8)

All parameters used in this equation have been defined earlier. Example 7.5 uses this equation to determine the optimum clip thickness for the same design as considered in Example 7.4.

The deflection of the straddling spring shown in Fig. 7.6(a) to preload the prism against the three locating pins with a total preload P can be determined using Eq. (7.9).

Example 7.5: Optimize the cantilevered spring clip thickness using Eq. (7.8) for the same design as considered in Example 7.4. (For design and analysis, use File 7.5 of the CD-ROM.)

Assume the following dimensions and material properties for a prism mounting per Fig. 7.6(b) using clips of BeCu: W = 0.267 lb (0.121 kg), $a_G = 12$, N = 3, $S_Y = 155,000$ lb/in.² (1.069×10³ MPa), $E_M = 18.5 \times 10^6$ lb/in.² (1.27×10⁵ Pa), $v_M = 0.350$, L = 0.375 in. (9.525 mm), b = 0.250 in. (6.350 mm). (a) What should be the clip thickness for $f_S = 2.0$? (b) What is the clip deflection? (c) Is this deflection reasonable? (d) Confirm the bending stress safety factor.

(a) From Eq. (7.8):
$$t = \left\{ \frac{(6)(0.375) \left[\frac{(0.267)(12)(2.0)}{3} \right]}{(155,000)(0.250)} \right\}^{1/2} = 0.011 \text{ in. } (0.279 \text{ mm}).$$

(b) By Eq. (3.1), *P* = (0.267)(12) = 3.204 lb (14.252 N) From Eq. (3.42):

$$\Delta y = \frac{(1 - 0.350^2)(4)(3.204)(0.375^3)}{\left[(18.5 \times 10^6)(0.250)(0.011^3)(3)\right]} = 0.032 \text{ in.} (0.813 \text{ mm}).$$

(c) Assuming that the device used to measure clip deflection can resolve 0.0005 in. (0.0127 mm), the ratio of deflection/resolution = 64. The text recommends that this ratio should be >10 so the deflection is acceptable.

(d) From Eq. (3.43):
$$S_B = \frac{(6)(3.204)(0.375)}{\left[(0.250)(0.011^2)(3)\right]} = 79,438 \text{ lb/in.}^2(547.725 \text{ MPa})$$

Note: S_Y is given as 155,000 lb/in.² so the safety factor is 1.95. It probably should be 2.0, but inasmuch as the material properties and dimensions are surely not known to this degree of accuracy, the agreement is quite reasonable.

$$\Delta x = \frac{(0.0625)(1 - v_M^2)(P)(L^3)}{\left[(E_M)(b)(l^3)\right]}.$$
(7.9)

The bending stress in this spring is given by:

$$S_{B} = \frac{(0.75)(P_{i})(L)}{\left[(b)(t^{2})\right]}.$$
(7.10)

Example 7.6 shows how to use these equations in creating a mounting design as shown in Fig. 7.6(a).

Example 7.6: Design a straddling spring mounting for a prism. (For design and analysis, use File 7.6 of the CD-ROM.)

Consider the prism mounting of Fig. 7.6(a). The dimensions and material properties are as follows: W = 0.267 lb (0.121 kg), $a_G = 12$, N = 1 (BeCu), $S_Y = 155,000$ lb/in.² (1.069×10³ MPa), $E_M = 18.5 \times 10^6$ lb/in.² (1.27×10⁵ Pa), $v_M = 0.35$, L = 1.040 in. (26.416 mm), b = 0.250 in. (6.350 mm), and t = 0.0115 in. (0.292 mm). (a) What should be the spring deflection? (b) What bending stress results in that clip? (c) What is its safety factor?

(a) Required preload =
$$\frac{(0.267)(12)}{1}$$
 = 3.204 lb (14.252 N).

From Eq. (7.9):

$$\Delta x = \frac{(0.0625)(1-0.35^2)(3.204)(1.040^3)}{\left[(18.5 \times 10^6)(0.25)(0.0115^3)\right]} = 0.0281 \text{ in. } (0.714 \text{ mm}).$$

(b) From Eq. (7.10):

$$S_{B} = \frac{(0.75)(3.204)(1.040^{3})}{\left[(0.25)(0.0115^{2})\right]} = 75,590 \text{ lb/in.}^{2}(521.2 \text{ MPa}).$$

(c)
$$f_s = \frac{155,000}{75,590} = 2.05.$$

These results are acceptable.

7.3 The Use of Pads on Cantilevered and Straddling Springs

The use of cylindrical pads between the springs and the prism surfaces in the design of Figs. 7.6(a) and 7.6(b) ensures that line contact will occur in a predictable manner at each interface. In the absence of a pad, a deflected cantilevered spring could touch the prism at the edge of its protective bevel as shown in Fig. 7.7(a). This is highly undesirable since that edge is sharp so the prism is vulnerable to damage from the force exerted by the spring. An alternative interface, again without a pad, is illustrated in Fig. 7.7(b). Here the deflected spring nominally lies flat against the top prism surface by virtue of the wedge-shaped washers placed above and below the spring on the post. The angles of the wedges are set by Eq. (7.7). While this is good from the viewpoint of stress imparted to the prism if the spring is in close contact with an appreciable area on the prism, a potential problem may exist with the latter design. A minor manufacturing error or unfavorable tolerance buildup could cause the end of the spring to touch the prism surface (if the angle is too shallow). Either of these conditions could lead to prism damage or local deformation from a concentration of stress.

Another version of the area interface of Fig. 7.7(b) is shown in Fig. 7.8. Here a flat surface on a wedged pad attached to the end of the spring provides the angular adjustment needed to bring the pad into close contact with the prism surface. Once again, the design is

susceptible to angular errors causing it deteriorate into sharp corner contact at the inner or outer edge of the pad, resulting in undue stress.

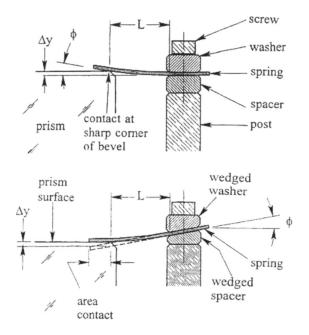


Figure 7.7 Two configurations of the cantilevered spring-to-prism interface: (a) Spring touching prism bevel, (b) spring lying flat on the prism.

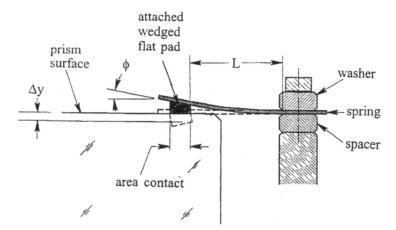


Figure 7.8 Configuration of a cantilevered spring-to-prism interface with a flat pad pressing against a flat surface of the prism.

Line contact at a rounded portion of the spring occurs if the end of the spring is bent to a convex cylindrical shape as indicated in Fig. 7.9(a). Because it may be difficult to form the spring into a smooth cylinder of a particular radius, a better interface results if a pad is machined integrally into the spring as indicated in Fig. 7.9(b). Note that a machined cylindrical pad can also be attached to the spring by screws, welding, or adhesive to achieve the same function. In either case, stress introduced at the interface between the convex cylinder and the flat prism surface can be estimated and adjusted to an acceptable level by careful design. We explain how to do this in Sect. 13.4.

Figure 7.10 shows how a straddling spring can be substituted for the three cantilevered spring clips of Fig. 7.6 to hold a prism in place. A flat pad is shown in View (a) as a means for distributing the force over a small area on the prism. This would be acceptable if we could be sure that the spring action is symmetrical and that the pad lies flat on the prism. Dimensional errors or tolerance buildup could tilt the pad at the wrong angle and cause stress concentration at a pad edge. Curving the pad as shown in Fig. 7.10(b) eliminates this possibility. Additional springs can be used if one is not sufficient to provide the required preload or to reduce stress at each interface.

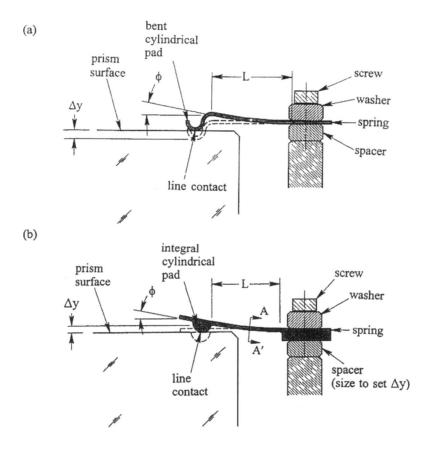


Figure 7.9 Configurations of a cantilevered spring-to-prism interface with. (a) The spring bent into a convex cylindrical curve and (b) an integrally-machined cylindrical pad.

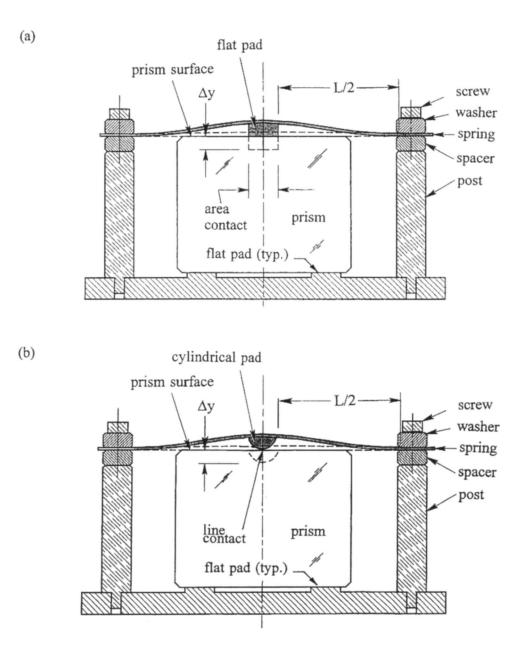


Figure 7.10 Straddling spring constraints for a prism: (a) With a flat pad, (b) with a cylindrical pad.

To further illustrate the use of straddling springs to constrain prisms, consider the hardware design for Porro prism mountings in a commercial binocular as shown in Fig. 7.11. The springs press against the apexes of the prisms and hold them against reference surfaces provided in the housings. One end of each spring is secured with a screw, while the other end simply slips into a slot machined into the housing wall. No pads are used on the springs.

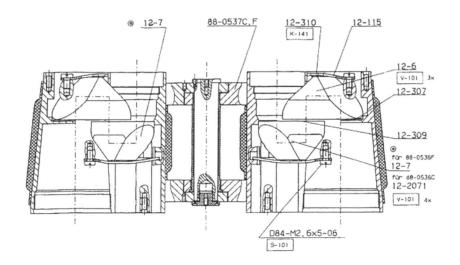


Figure 7.11 Porro prism erecting system mounting in a contemporary commercial binocular. (Courtesy of Swarovski Optik KG, Absam/Tyrol, Austria.)

Yet another version of the straddling spring prism mounting is illustrated in Fig. 7.12. Here an Amici prism is held against flat pads in a small military telescope housing by a spring with bent ends. The screw pressing against the center of the spring forces its ends against the prism. It is very important that the screw not protrude far enough into the housing for the spring to touch the roof edge of the prism. A careful design would specify the correct screw length, while explicit instructions in the manufacturing procedures to check the actual clearance would help avoid damage at assembly or during exposure to shock or vibration, especially at extreme temperatures. Constraint perpendicular to the plane of the figure is provided by resilient pads attached to the inside surfaces of triangular-shaped covers that are attached with screws to both sides of the cast and machined housing.

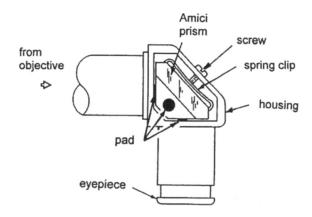


Figure 7.12 Schematic of a small military elbow telescope with an Amici prism spring loaded against reference pads. (Adapted from Yoder.⁷)

The author has examined a telescope with this type of mounting and found a fracture in the prism at one spring-to-prism sharp corner interface. This is believed to have resulted from excess stress that developed during use of the instrument. While it is simple, this type mounting is not recommended for future applications. An improvement that might make the design acceptable would be to add cylindrical pads at each end of the spring and shape that spring so the pads touch the ends of the prism farther from the roof edge.

The designs of the cylindrical pads for use on cantilevered or straddling springs should include consideration of the angular extent of the curved surfaces. Figure 7.13 shows the pertinent geometrical relationships. R_{CYL} is the pad's radius, d_P is the width of the pad, and α is one-half the angular extent of the cylindrical surface measured at its center of curvature. In Fig. 7.13(a), the pad is symmetrically oriented with respect to the prism surface before the spring is bent. In Fig. 7.13(b), the spring is bent to exert preload P_i , tilting the pad by the angle φ per Eq. (7.7). If the worst-case α is greater than φ , no sharp edge contact occurs. Once α is determined, we calculate the minimum value for d_P from this equation:

$$d_{p \text{ MIN}} = 2R_{\text{CYL}} \sin \alpha. \tag{7.11}$$

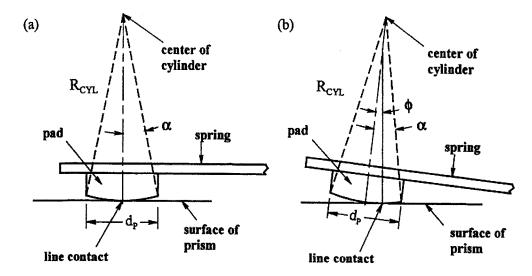
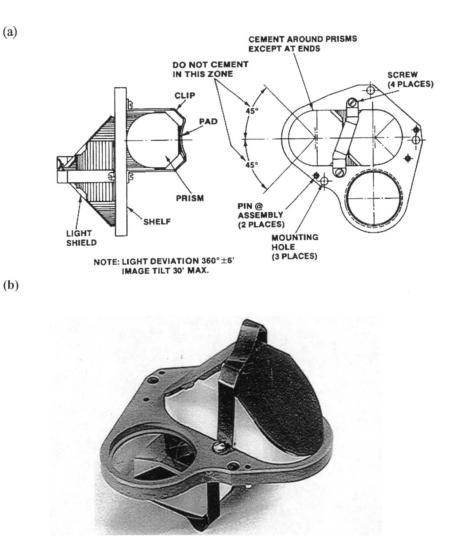
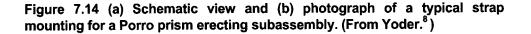


Figure 7.13 Geometric relationships applied in the design of cylindrical pads for springs. The same cross-sectional geometry applies to spherical pads.

7.4 Mechanically Clamped Nonkinematic Mountings

Springs or straps are frequently used to hold prisms in place against extended flat interfaces in optical instruments. These techniques are not kinematic. One is the Porro prism erecting prism assembly shown schematically in Fig. 7.15. This is typical of prism mountings in many military and consumer binoculars and telescopes.⁸ Spring straps (typically made of spring steel) hold each prism against a machined surface in a perforated aluminum mounting shelf that is in turn fastened with screws and locating pins to the instrument housing. The straps are a variation of the straddling spring discussed earlier.





Area contacts occur over annular areas on the racetrack-shaped hypotenuse faces of the prisms, while recesses in the opposite sides of the shelf provide lateral constraints. An elastomeric-type adhesive or cement, as it is called in the figure (rubber cement has been used in some early designs), is used to keep the prism from sliding within the necessary clearances around the recessed faces. In this design, the prisms are made of high-index (flint) glass so TIR occurs at the four reflecting surfaces. Thin aluminum light shields help prevent stray light from reaching the image plane. These shields have bent tabs that touch the edges of the prism faces to space the shields a short distance from the reflecting surfaces. Prisms with silvered or aluminized reflecting surfaces do not need these shields.

Figure 7.15 shows the scanning head assembly from a military periscope. It uses a single prism that resembles a Dove prism with angles between faces of 35 deg, 35 deg, and 110 deg. The prism can be tilted about the horizontal transverse axis to scan the periscope's line of sight in elevation from the zenith to about 20 deg below the horizon. The prism is held in place in its cast aluminum mounting by four spring clips, each attached with two screws to the mount adjacent to the entrance and exit faces of the prism. The edges of the reflecting surface (hypotenuse) of the prism rest on narrow lands machined and lapped into the casting. The lands do not extend into the optical aperture. The prism faces protrude slightly [about 0.5 mm (0.02 in.)] from the mount so that preload is obtained when the clips are flush against the mount. Once centered, the prism cannot slide parallel to the long edge of its hypotenuse because of the convergence of the clamping forces. The vector sum of these forces is nominally perpendicular to the mounting surface and the tangential force components cancel each other. Lateral motion is constrained by friction and limited by a close fit in the mount.

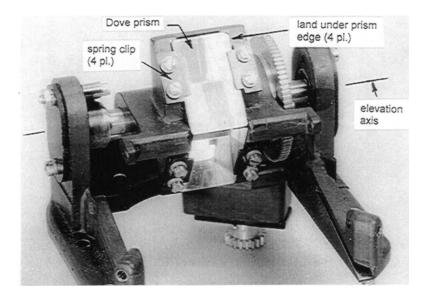


Figure 7.15 A mechanically-clamped Dove prism used in the elevation scanning head subassembly of a military periscope. (From Yoder.⁹ Copyright Taylor and Francis Group, LLC, a division of Informa plc. Reprinted with permission.)

Figure 7.16(a) illustrates the scanning function of the prism shown in Fig. 7.15. This motion is limited optically by vignetting the refracted beam at its top or bottom edges. Mechanical stops are usually built into the instrument to limit physical motion so that the vignetting at the end-point is acceptable for the application. View (b) shows the increased scanning range obtained with a double Dove prism.

The most popular types of derotation prisms are the Dove, double-Dove, Pechan, and delta. In order to function successfully, all these prisms must be mounted securely yet be capable of adjustment at the time of assembly to minimize image motion during operation. A design for one type of adjustable derotation prism mounting is discussed next.

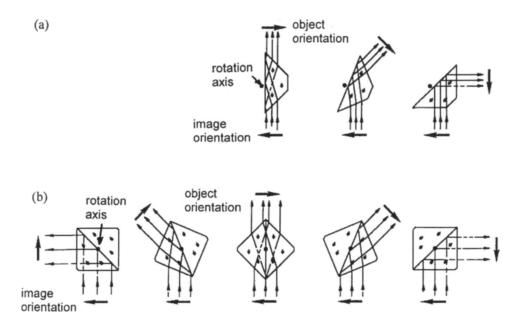


Figure 7.16 (a) Typical elevation scanning function of a Dove prism. (b) Scanning with a double Dove prism. Note the increased scan range in (b).

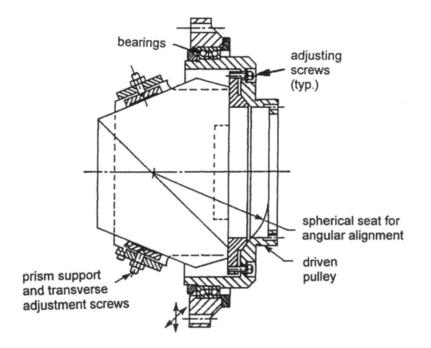


Figure 7.17 A Pechan prism derotation assembly. (From Delgado.¹⁰)

In Fig. 7.17, we see a sectional view of a representative Pechan prism mounting.¹⁰ If it is used in a collimated beam, it requires only angular adjustment of the optical axis relative to the rotation axis. Here it was to be used in an uncollimated beam, so both angular and lateral adjustments were needed. Bearing wobble would cause angular errors. To minimize this in the design considered here, class-5 angular-contact bearings, mounted back to back, were oriented with factory-identified high spots matched and then preloaded. Runout over 180-deg motion was measured as about 0.0003 in. (7.6 μ m). The bearing axis was adjusted laterally by fine-thread screws (not shown) that permitted centration with respect to the optical system axis to better than 0.0005 in. (12.7 μ m). The prism was adjusted laterally within the bearing housing in the plane of refraction by sliding it against a flat vertical reference surface with fine-thread screws pressing against the reflecting surfaces through pressure pads. A spherical seat with its center of rotation at the intersection of the hypotenuse face with the optic axis (to minimize axis cross-coupling) was provided for angular adjustment. This movement was controlled by the adjustment screws indicated.

In each of the last four prism design examples—the Porro, the Amici, the Dove-type, and the Pechan—are loaded against machined surfaces on the mounts. Since it is virtually impossible for the metal surfaces to be as flat as the interfacing glass surfaces, contact will occur first on the three highest points on the mount. Usually these points are not directly opposite the springs that hold the prism in place. Moments are then applied to the glass, and surface deformations may occur. If the spring loading is large, the metal or the glass may bend enough for more point contacts to form. We then have a condition of uncontrolled overconstraint. Since the prisms are stiff and the instruments into which these designs have been incorporated have demonstrated a long service life in relatively adverse environments, we conclude that these problems are usually tolerable. Adding small-area pads that can be lapped flat and coplanar, and located opposite or nearly opposite the force points such as those used in the designs shown in Figs. 7.2, 7.3, and 7.6 reduces the likelihood of prism distortion from the applied constraints.

A potential problem occurs if, under shock or vibration, the prism loses contact with some of its interfacing positional references (i.e., the lapped pads). When the driving force dissipates, the prism may land in a new orientation. It remains there until it is disturbed again. This action introduces uncertainty into the location and/or orientation of the optical component, which may affect performance. If the preloads derived from the springs are large enough that the optic always maintains contact with the references, even at extreme temperatures, this problem should not exist.

7.5 Bonded Prism Mountings

7.5.1 General considerations

Many prisms are mounted by bonding their ground faces to mechanical pads using epoxy or similar adhesives. Contact areas large enough to render strong joints can usually be provided in designs with minimum complexity. The mechanical strength of a carefully designed and manufactured bond is adequate to withstand the severe shock and vibration as well as other adverse environmental conditions characteristic of military and aerospace applications. This technique is also used in many less rigorous applications because of its inherent simplicity and reliability.

The critical aspects of the design are the characteristics and age of the adhesive (it must be used within its specified shelf life), the thickness and area of the adhesive layer, the cleanliness of the surfaces to be bonded, the dissimilarity of coefficients of expansion of the materials, the environmental conditions to be encountered, and the care with which the parts are assembled. Several typical adhesives used for this purpose are listed in Table B14. While the adhesive manufacturer's recommendations should be consulted, experimental verification of the adequacy of the design, the materials to be used, the method of application, and curing temperature and duration are advisable in critical applications.

Guidelines for determining the appropriate bonding area have appeared in the literature.¹¹ In general, the minimum area of the bond, Q_{MIN} , is determined by

$$Q_{MN} = \frac{Wa_G f_S}{J},\tag{7.9}$$

where W is the prism's weight, a_G is the maximum expected acceleration factor, f_S is a safety factor, and J is the adhesive's shear or tensile strength (usually approximately equal).

The safety factor should be at least 2 and possibly as large as 4 or 5 to allow for some unplanned, nonoptimum conditions, such as inadequate cleaning during processing. To simplify the interface design task, most of the prism designs considered in Chapter 6 include equations for calculating the minimum circular or racetrack-shaped bonding area needed and the maximum bonding area achievable ($Q_{MAX C}$ or $Q_{MAX RT}$, as appropriate) on the bonding face of that type of prism.

For maximum glass-to-metal bonding strength, the adhesive layer should have a particular thickness. For example, if 3M EC2216-B/A epoxy is used, experience indicates that a thickness of 0.075 to 0.125 mm (0.003 to 0.005 in.) is best. Some adhesive manufacturers recommend thicknesses as large as 0.4 mm (0.016 in.) for their products. A thin bond is stiffer than a thick one because its effective Young's modulus E^* depends upon the ratio of bond lateral size, i.e., diameter, to bond thickness t_e . According to Genberg, E^* can be ten to several hundred times larger than the bulk E, depending upon the material's Poisson's ratio.¹²

One means for achieving a bond of a particular thickness is to place small spacers of the specified thickness at three locations in the interface between the glass and the metal. If possible, these should be arranged in a triangular pattern and lie outside the bonded area. The glass, mount, and spacers must be held together to ensure contact and to prevent lateral and/or rotational motion throughout curing. A fixture is usually designed and used for this purpose. Adhesive should not be allowed to get onto the spacers. Short pieces of wire or fishing line of appropriate diameter, or a flat metal or plastic shim of appropriate thickness have been used successfully as spacers. Another way to obtain an adhesive layer of a specific thickness is to mix a few percent by volume of small glass beads into the adhesive before applying it to the surfaces to be bonded. When the surfaces are pressed together, the beads separate the glass and metal surfaces. Such beads can be purchased with closely controlled diameters.^a The glass beads have essentially no effect on bond strength.

Adhesives and metals typically have CTEs larger than those of glasses, so differential dimension changes can be large. In addition, adhesives usually shrink by a few percent of their lateral dimensions during curing. Smaller bonds create smaller shear stresses in the bonded region due to these effects. Many mounting designs have the required bonding area subdivided into three (or more) smaller areas. These are preferably arranged in a triangular pattern. This reduces the differential dimension changes and increases stability.

7.5.2 Examples of bonded prisms

A roof penta prism bonded on a circular raised pad on an aluminum mounting bracket is shown in Fig. 7.18. It is intended for use within a military periscope with its plane of reflection nominally vertical, but subject to shock and vibration in any direction. Example 7.7 analyzes this mounting based on certain assumptions regarding its design.

Example 7.7: Design of a bonded mounting for a roof penta prism. (For design and analysis purposes, use File 7.7 of the CD-ROM.)

The face width A of the prism of Fig. 7.18 is 1.102 in. (2.800 cm) and it is made of BK7 glass with density 0.091 lb/in.³ (2.511 g/cm³). Assume it is bonded with 3M 2216B/A epoxy for which J = 2500 lb/in.² (17.24 MPa). How large must its circular bond be if $a_G = 250$ and $f_S = 4$?

From Fig. 6.24, the prism weight is $(1.795)(1.102^3)(0.091) = 0.219$ lb (0.099 kg).

From Eq. (7.9), $Q_{MIN} = \frac{(0.219)(250)(4)}{2500} = 0.088 \text{ in.}^2 (0.565 \text{ cm}^2).$

The minimum bond diameter if a single circular area is

$$(2)\left(\frac{0.088}{\pi}\right)^{V^2} = 0.335$$
 in. (8.502 mm).

From the figure, the bond appears to be much larger than this minimum size.

^a See, for example, size-certified products of Duke Scientific Corp.(www.dukescientific.com).

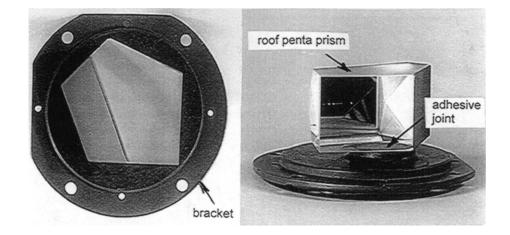


Figure 7.18 Photographs of a typical roof penta prism bonded to a metal bracket. (From Yoder.⁸)

A Pechan prism with three adhesive bonds distributed in a triangular pattern is illustrated schematically in Fig. 7.19. Note that the bonds are on the larger prism only. This is because the ground surfaces of the bases of the two prisms may not be coplanar after assembly. They may be skewed or may have a step in height. Either defect could make the adhesive thickness different on the two prisms and reduce the bond's strength. To prevent adhesive from being drawn into the narrow air space between the prisms by capillary action, the bonds should be kept clear of the adjacent edges of the prisms.

In general, fillets of excess adhesive at the edges of glass-to-metal bonds should be avoided. This is because low-temperature shrinkage along the hypotenuse of a fillet at low temperature is greater than that along the glass or metal surfaces. See Fig. 7.20. Such shrinkage has been known to fracture the glass in some cases.

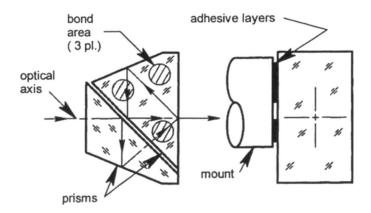


Figure 7.19 A triangular bond distribution on one prism of a Pechan prism subassembly.

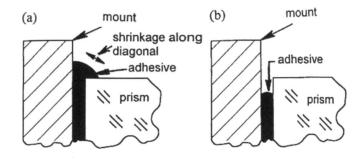


Figure 7.20 (a) An adhesive bond with an undesirable fillet of excess epoxy at the edge of the joint. (b) A more desirable configuration without the fillet.

Figure 7.21 shows a cube-shaped fused silica prism (beamsplitter) with A = 35 mm (1.375 in.) that was bonded with epoxy over its entire square base to a titanium mount. The prism base had been ground flat after cementing to remove any step. When the unit was cooled to -30° C, the shrinkages of the metal and of the adhesive relative to the prism each applied shear forces to the glass and the prism fractured. In Section 14.6, we give an equation for the stress within such a bond and an analysis of this mounting design.

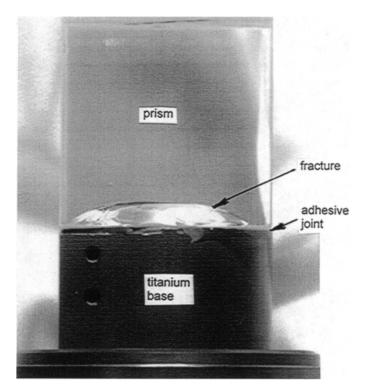


Figure 7.21 Photograph of a fused silica prism bonded to a titanium base with epoxy. It fractured at low temperature because of differential contraction.

A design feature that increases the reliability of glass-to-metal bonds is to specify that the bonding surface on the glass component be fine ground and that the grinding operations consist of multiple steps with progressively finer abrasives. The depth of material removed in each step is at least three times the prior grit size. This process removes subsurface damage from each prior step. Commonly referred to as "controlled grinding," this process (explained in more detail in Section 13.2) produces a surface essentially free of invisible cracks and significantly increases the material's tensile strength at the ground surface.¹³ Experience has shown that bonding on polished glass surfaces may not be as successful as bonding on fine-ground surfaces.

Figure 7.22 shows a Porro prism mounting design that is known to have withstood shocks of approximately 1200 times gravity without damage. This subassembly is the subject of Example 7.8.

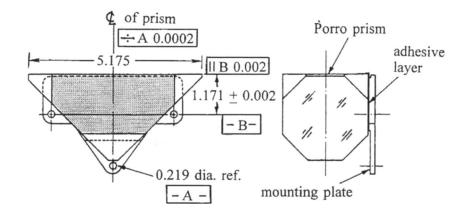


Figure 7.22 Mounting configuration for a Porro prism bonded on one side in cantilevered fashion. Dimensions are in inches. The bonding area is shown shaded. (From Yoder.⁸)

7.5.3 Double-sided prism support techniques

Because, during shipment or use, the mounting surface of any bonded prism subassembly can be oriented in any direction with respect to gravity or other imposed external forces, the cantilevered prism mounting may not be adequate under extreme conditions. Additional support may be desired. This leads us to the following variety of mounting arrangements involving double-sided support.

Some designs for bonding prisms utilize multiple adhesive joints between the prism and structure. In the configuration of Fig. 7.23, increased bonding area and support from both sides are provided to a right-angle prism by bonding it to the ends of two metal stub shafts. Through the use of precision fixturing, these shafts are made collinear. They rest in and are firmly clamped by two precision-machined pillow blocks of conventional split design. Ease of adjustment of the prism's rotational alignment about the transverse axis is a key feature of this design. Limited lateral adjustments along that axis can also be made. It is necessary that the mounting faces of the prism be ground parallel during manufacture.

Example 7.8: Acceleration capability of a large Porro prism assembly bonded in cantilevered fashion. (For design and analysis, use File 7.8 of the CD-ROM.)

The Porro prism of Fig. 7.22 is made of SK16 glass and is bonded with 3M EC2216-B/A epoxy to a 416 stainless steel bracket. The actual bond area Q is 5.6 in.² (36.129 cm²) and the prism weight W is 2.20 lb (0.998 kg). (a) What acceleration a_G would the assembly be expected to withstand with a safety factor f_S of 2? Assume the bonding strength J of the cured joint is 2500 lb/in.². (b) What is the safety factor if shocked at $a_G = 1200$?

(a) Rewriting and applying Eq. (7.9) we obtain:

$$a_G = J \frac{Q}{W f_S} = \frac{(2500)(5.6)}{(2.20)(2)} = 3182$$

(b) The prism should withstand acceleration of 1200 times gravity in any direction with a safety factor of 2.7.

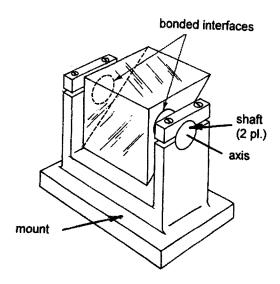


Figure 7.23 Schematic of a right angle prism supported on both sides in a U-shaped mount (Adapted from Durie.³)

Furthermore, the bearing surfaces for the shafts in the bonding fixture and in the instrument must be extremely straight and coaxial. Otherwise, forces exerted during clamping at assembly or during exposure to vibration, shock, or extreme temperatures could strain the bonds and perhaps cause damage.

Problems with differential expansion between glass and metal that may occur at extreme temperatures in the design of Fig. 7.23 can be avoided by building flexibility into one support arm of the double-sided prism mounting. An example of a design with such a feature is shown in Fig. 7.24. Units made to an earlier design without this flexure were

damaged at low temperatures when the aluminum mounting contracted more than the prism, causing the arms to pivot about the bottom edge of the prism and pull away from the prism at the tops of the bonds. Allowing one arm to bend slightly prevented such damage.¹⁴ Holes were provided in each support arm to allow epoxy to be injected into the spaces between the prism and mounting surfaces after the prism was aligned. These holes are designated as "P" in the figure.

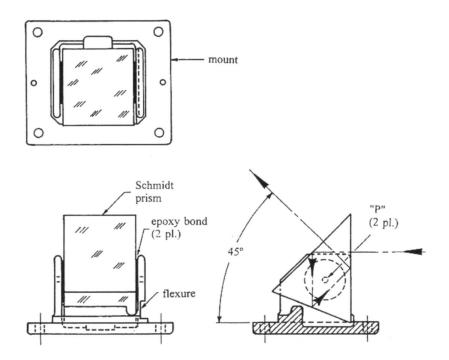


Figure 7.24 A Schmidt prism bonded on both sides to a U-shaped mount. (From Willey.¹⁴)

According to Beckmann,¹⁵ a potential problem in any prism mount design in which epoxy is injected through access holes into a bond cavity is that the "plug" of adhesive in the hole can shrink significantly at low temperatures and perhaps pull the adjacent glass sufficiently to distort or even fracture it. Minimizing the length of the hole and hence of the adhesive plug would help to avoid this problem. Counterboring the injection holes from the outside to reduce this length and to allow excess adhesive to be removed after injection also might help.

Two versions of another design concept with support to a cube-shaped prism provided from two sides by arms forming a U-shaped mount are shown schematically in Fig. 7.25. In view (a), the crown glass prism is bonded to the ends of two cylindrical stainless steel plugs passing through clearance holes in the arms. The mount and both plugs are made of stainless steel to reduce the mismatch of glass and metal thermal expansion coefficients. The prism is supported in a fixture and aligned with respect to the mount prior to bonding and throughout the bond-curing process.

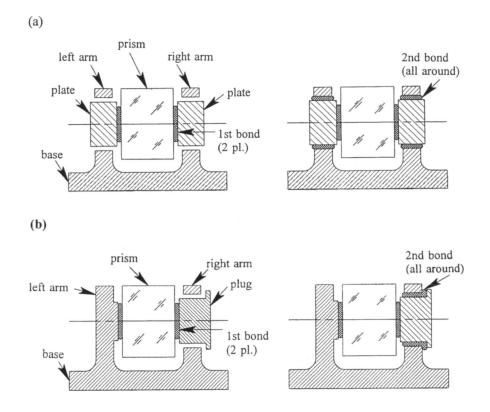


Figure 7.25 Two concepts for double-sided bonding of a prism to a U-shaped mount. [(a) Adapted from Beckmann.¹⁵]

After the first two bonds have cured, the plugs are epoxied to the arms as indicated in view (b). With this approach, tolerances on location and tilt of the surfaces to be bonded are relaxed since the plugs align themselves to the prism in the clearance holes before they are bonded to the arms. It should be noted that metal-to-metal bonds are much more forgiving with respect to alignment and bond thickness variations than in the case of glass-to-metal bonds.

In Fig. 7.25(c), the prism is aligned to the mount and then is bonded to a raised pad on one support arm (left). The metal plug shown protruding through, but not attached to the second (right) arm, is also bonded to the side of the prism. After these bonds have cured, the plug is bonded to the right arm to provide the required dual support [see view (d)]. Once again, tolerances on the bonding surface locations and orientations can be relatively loose. The beauty of the ideas presented here in connection with constraining optical components by sequential bonding is that precise fitting of parts is not required, yet the alignment established optically or by fixturing prior to bonding is retained after the bonds have cured.

A different optomechanical design involving bonding of prisms on both sides is illustrated in Fig. 7.26. Here a Schmidt-Pechan roof prism subassembly is inserted into a close-fitting seat molded in the filled-plastic housing of a commercial binocular. The prism subassembly is then provisionally secured in place with dabs of UV-curing adhesive

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applied through openings in the housing walls.^{16,17} After proper alignment is confirmed, the prisms are secured by adding several beads of polyurethane adhesive through the same wall openings. The slight resiliencies of the housing and the adhesive accommodate the differential thermal expansion characteristics of the adjacent materials. With precision-molded structural members and built-in reference surfaces, adjustments are not required. Figure 7.27 shows some details of the internal configuration of such a prism mounting.

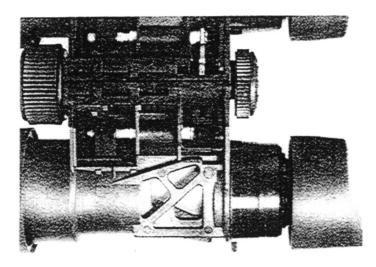


Figure 7.26 A Schmidt-Pechan erecting prism subassembly mounted in a plastic binocular housing. (From Seil.¹⁶)

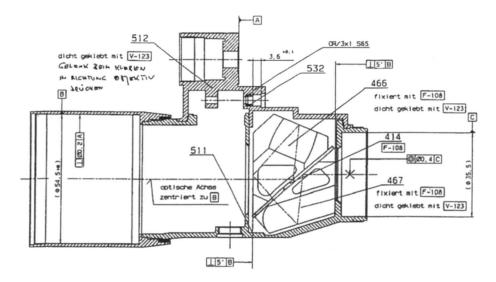


Figure 7.27 Drawing of a roof prism mounted in the manner of Fig. 7.15. (From Seil.¹⁷)

Figure 7.28 is a photograph of an assembly comprising a Porro prism erecting system and a rhomboid prism mounted by the same general technique just described for constraining roof prisms. In this design, one Porro prism is attached with adhesive to its plastic bracket. This bracket then slides on two parallel metal rods to provide axial movement of the prism relative to the second Porro prism for adjusting focus of an optical instrument. The adhesive beads are more clearly shown in Fig. 7.29. Minimization of the number of components and ease of assembly are prime features of this design. Customer experience and acceptance of products made by this technique have demonstrated the durability and adequacy of the optomechanical performance achievable with this type of assembly.

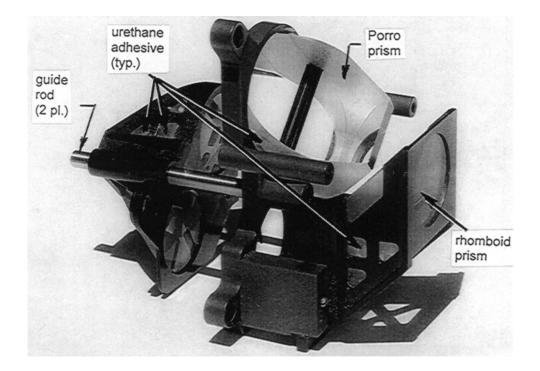


Figure 7.28 Photograph of a Porro prism image-erecting system and a rhomboid prism mounted by adhesive bonding to plastic structural members for use as an optomechanical subassembly in a commercial telescope. (From Seil.¹⁷)

Additional examples of bonded mountings for prisms are given in the context of a discussion of an articulated telescope intended for use in directing fire from the main weapon on an armored vehicle in Section 15.13.

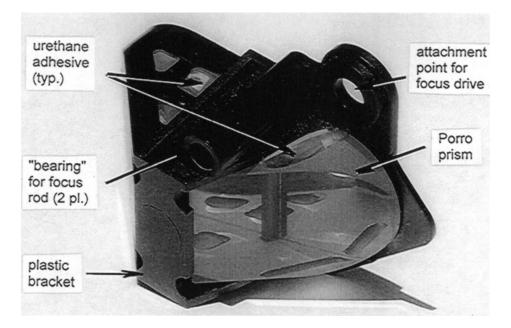


Figure 7.29 Close-up photograph of the movable Porro prism from the subassembly of Fig. 7.17. The prism is mounted by adhesive bonding to a plastic structural bracket. (From Seil.¹⁷)

7.6 Flexure Mountings for Prisms

Some prisms (particularly large ones or ones with critical positioning requirements) are mounted with flexures. A generic example is shown in Fig. 7.30. Three compound flexures are bonded with adhesive directly to the prism base and attached by threaded joints to a base plate (not shown). To reduce strain from differential expansion between the prism material and the base plate resulting from temperature changes, all three flexures are designed to bend in several directions; however, they are very stiff axially. Flexure No. 1 locates the prism horizontally at a fixed point. It has a "universal joint" at its top to allow for angular misalignment at the bonded joint. The second flexure constrains rotation about the fixed point (first flexure), but allows relative expansion along a line connecting the first and second flexures because its universal joints at the top and at the bottom allow it to deform to an "S" shape as relative dimensions change. The third flexure has a universal joint at the top and a single flexure at the bottom; it supports its share of the prism's weight and prevents rotation about the line connecting the other two flexures. This third flexure does not constrain the prism transversely. All three flexures have torsional compliance. Small differences in the lengths of the flexures and/or the parallelism of their top surfaces are accommodated through compliance of the three top universal joints. Because of the flexures, the prism remains fixed in space without being distorted or disturbing the structure to which it is attached, even in the presence of significant temperature changes and differential expansion of the prism and the mounting structure.⁹

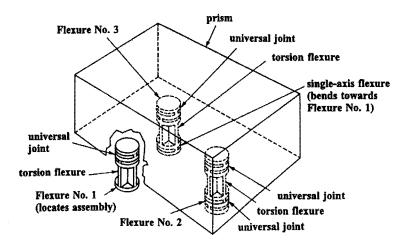


Figure 7.30 Conceptual sketch of a flexure mounting for a large prism.

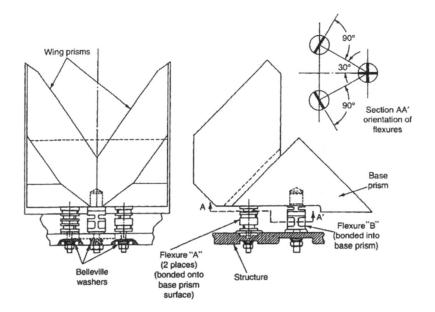


Figure 7.31 Optomechanical configuration of a large prism assembly mounted on three flexure posts. (Courtesy of ASML Lithography, Wilton, CT.)

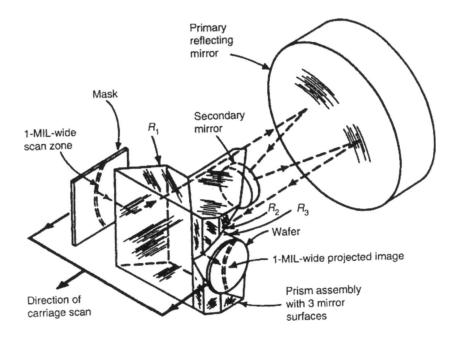


Figure 7.32 Schematic diagram of a microlithography mask projection system using the prism assembly of Fig. 7.31. (Courtesy of ASML Lithography, Wilton, CT.)

7.7 References

- 1. Smith, W.J., Modern Optical Engineering, 3rd. ed., McGraw-Hill, New York, 2000.
- 2. Lipshutz, M.L., "Optomechanical considerations for optical beam splitters," *Appl. Opt.* 7, 1968:2326.
- 3. Durie, D.S.L., "Stability of optical mounts," Machine Des. 40, 1968:184.
- 4. Vukobratovich, D., "Optomechanical Systems Design," Chapt. 3 in *The Infrared & Electro-Optical Systems Handbook*, Vol. 4, ERIM, Ann Arbor and SPIE, Bellingham, 1993.
- 5. Sorbothane, Inc., Engineering Design Guide, Kent, OH.
- 6. Roark, R.J., *Formulas for Stress and Strain*, 3rd ed., McGraw-Hill, New York, 1954. See also Young, W.C., *Roark's Formulas for Stress & Strain*, 6th ed., McGraw-Hill, New York, 1989.
- 7. Yoder, P.R., Jr., "Optical Mounts: Lenses, Windows, Small Mirrors, and Prisms," Chapt. 6 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, 1997.
- 8. Yoder, P.R., Jr., "Non-image-forming optical components," *Proceedings of SPIE* 531, 1985:206.
- 9. Yoder, P.R., Jr., Opto-Mechanical Systems Design, 3rd. ed., CRC Press, Boca Raton, 2005.
- 10. Delgado, R.F., "The multidiscipline demands of a high performance dual channel projector," *Proceedings of SPIE* 389, 1983:75.

- 11. Yoder, P.R., Jr., "Design guidelines for bonding prisms to mounts," *Proceedings of SPIE* **1013**, 1988:112.
- 12. Genberg, V.L., "Thermal and Thermoelastic Analysis of Optics," Chapt. 9 in *Handbook of Optomechanical Engineering*, CR Press, Boca Raton, 1997.
- 13. Stoll, R., Forman, P.F., and Edleman, J. "The effect of different grinding procedures on the strength of scratched and unscratched fused silica," *Proceedings of Symposium on the St rength of Gl ass and Ways t o Improve It*, Union Scientifique Continentale du Verre, Charleroi, Belgium, 1, 1961.
- 14. Willey, R., private communication, 1991.
- 15. Beckmann, L.H.J.F., private communication, 1990.
- 16. Seil, K., "Progress in binocular design," Proceedings of SPIE 1533, 1991:48.
- 17. Seil, K., private communication, 1997.

CHAPTER 8 Mirror Design

As in the case of prisms, a clear understanding of key aspects of mirror design is necessary before we consider the various techniques available for mounting those mirrors. This chapter deals largely with the geometric configurations of different types of mirrors, their functions, and the reasons why they are designed as they are. Because mirror size is a prime driver of design and of material choice, we consider sizes ranging from small ones with diameters of a few millimeters to about 0.5 m (1.6 ft) to large, astronomical-telescope sized ones with diameters as large as about 8.4 m (27.6 ft). How the intended method of manufacture influences design is considered, as appropriate, throughout the chapter. We begin by listing the applications of mirrors, illustrating the uses of mirrors to control image orientation, and defining the relative advantages of first- and second-surface mirror types. How to approximate the minimum aperture dimensions for a tilted reflecting surface located in a collimated or noncollimated beam is considered next. We then describe various substrate configurations that might be employed to minimize mirror weight and/or selfweight deflection. Modern technology for thin facesheet adaptive mirrors is summarized briefly. Selected designs for metallic mirrors are considered. The chapter closes with a few observations about the design and use of pellicles.

8.1 General Considerations

Small mirrors usually have solid substrates shaped as right circular cylinders or rectangular parallelepipeds. They typically have flat, spherical, cylindrical, aspherical, or toroidal optical surfaces. Curved surfaces can be convex or concave. Usually the second, or back, surface of a small mirror is flat, but some are shaped to make the profile into a meniscus. The thickness of the substrate is traditionally chosen as 1/5th or 1/6th the largest face dimension. Thinner or thicker substrates are used as the application allows or demands. Nonmetallic substrates are typically borosilicate crown glass, fused silica, or one of the low-expansion materials (such as ULE or Zerodur). Metallic mirrors usually are made of aluminum unless some special requirement of the application leads to the choice of beryllium, copper, molybdenum, silicon, a composite material (such as graphite epoxy or silicon carbide), or a metal matrix material (such as SXA).

Most of the mirrors used in optical instruments are of the first-surface reflecting type and have thin-metallic-film coatings (such as aluminum, silver, or gold), which have protective dielectric coatings (typically magnesium fluoride or silicon monoxide). Second-surface mirrors have a reflective coating on the mirror's back; the first surface then acts as a refracting surface. The refracting surface typically carries an antireflection coating such as magnesium fluoride to reduce the effects of ghost images from that surface. A special mirror is the plate beamsplitter, which has a partially reflective coating on one surface to redirect some of the incident light and to transmit most of the rest.

Flat mirrors, used singly or as combinations of two or more, serve useful purposes in optical instruments, but do not contribute optical power and, hence, cannot form images by themselves. The principal uses of these mirrors are as follows:

- To bend (deviate) light around corners,
- To fold an optical system into a given shape or package size,

- To provide proper image orientation,
- To displace the optical axis laterally,
- To divide or combine beams by intensity or aperture sharing (at a pupil),
- To divide or combine images by reflection at an image plane,
- To dynamically scan a beam, and
- To disperse light spectrally (as with gratings).

Most of these functions are the same as mentioned in Chapter 6 for prisms. Curved mirrors can do some of these things, but their most common applications involve image formation, as in reflecting telescopes.

8.2 Image Orientation

Reflection from a single mirror results in a reversed image. Figure 8.1 shows this reversal for an arrow-shaped object A-B. Think of this as a view looking parallel to the mirror surface. If the observer's eye is at the indicated position and looks directly at the object (dotted lines), the point B appears on the right. It appears on the left in the reflected image A'-B' (dashed lines). This would be called a "left handed" or "reverted" image as defined in Section 1.1. If the object were a word, it would be harder to read than the object word. Note that the portion of the mirror actually used to form this image extends only from P to P'. If the object or the eye moves, a different portion of the mirror surface will be used. Observation of a bigger object requires a bigger mirror.

With multiple mirrors, the orientation of the image becomes more complex. The image is reverted with each reflection. An odd number of reflections create a left-handed image while an even number creates a right-handed image. In optical systems where an erect and unreverted^{*} image is needed, such as a terrestrial telescope used for bird watching, careful consideration must be given to the number of reflections occurring in each meridian. If the planes of reflection of multiple mirrors are not oriented orthogonally, the image may appear rotated about the axis. An image rotator/derotator (such as the Pechan prism shown in Fig. 6.22) might be needed to correct this potential orientation problem.

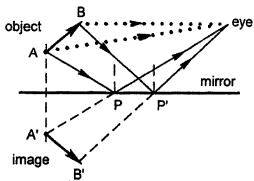


Figure 8.1 Reflection of an arrow-shaped object by a flat mirror as seen by an observer at the point "eye." The image is reversed relative to the object as seen directly.

^{*} A "reverted" image is inverted in the horizontal direction.

Each reflection at oblique incidence deviates a ray by some angle, which we will call δ_i . With reflections in the same plane from multiple mirrors, deviations add algebraically. This is shown in Fig. 8.2 for two mirrors. The total deviation is $\delta_1 + \delta_2$.

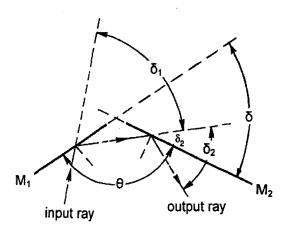


Figure 8.2 Deviations of a light ray upon intersecting two flat mirrors oriented at an angle θ to each other. The total deviation is the sum of δ_1 and δ_2 .

This principle is applied in the layouts of two periscopes in Fig. 8.3. In view (a), mirrors M_1 and M_2 are parallel and inclined to the X axis by 45-deg angles. Since the mirror normals at the reflecting surfaces are opposed, we know that the deviations have opposite signs. Hence, $\delta = \delta_1 + \delta_2 = 0$ and the output ray is parallel to the input ray. The ray path between the mirrors is vertical (Y axis) so the X separation of the points of incidence on the mirrors is zero.

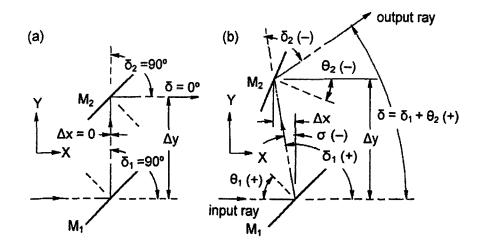


Figure 8.3 Deviations of a light ray upon intersecting two flat mirrors oriented at an angle θ to each other. The total deviation is the sum of δ_1 and δ_2 .

In view (b) of Fig. 8.3, we see the more general case of a periscope in which the intermediate ray between the two mirrors is traveling at an angle φ to the Y axis and the output ray travels at some angle with respect to the input ray direction. Now we have both X and Y separations (called Δx and Δy) of the points of incidence. Once again, the total deviation is the sum of the individual deviations of the two mirrors; the second deviation taken as negative for the reason stated above. Signs assigned to other angles are as noted in the figure. It is important to note that the periscope arrangement of mirrors (or prisms) presents an erect image to the observer viewing back along the direction of the output beam.

To design such a periscope, one typically would start with desired vertical and horizontal offsets Δx and Δy plus a desired deviation δ . Equations that can be used to determine the other parameters are as follows:

$$\tan \sigma = \frac{\Delta x}{\Delta y},\tag{8.1}$$

$$\theta_1 = \frac{(\sigma + 90 \deg)}{2}, \qquad (8.2)$$

$$\theta_2 = \frac{\left(\delta - \sigma - 90 \, \deg\right)}{2},\tag{8.3}$$

$$\delta_1 = 180 \deg - 2\theta, \tag{8.4}$$

$$\delta_2 = \delta + \sigma - 90 \deg. \tag{8.5}$$

The use of these equations in a periscope layout is illustrated in Example 8.1.

The layout of multiple mirror or prism systems in three-dimensional space, and especially ones involving out-of-plane reflections of the axis, is much more complex than what we have just described. For such a task, one might resort to surface-by-surface ray tracing in a lens design program or a vector analysis technique such as that described by Hopkins.¹ The packaging of systems with many reflections yields well to such techniques. The process of laying out such a convoluted optical path is sometimes called "optical plumbing."

A vital aspect of the layout of multiple mirror systems is to determine the orientation of intermediate and final images. A simple technique used by many engineers is to sketch the system in isometric form and visualize the changes that take place at each mirror when an object configured as a "pencil" is bounced from the reflecting surfaces. Figure 8.4 illustrates this. View (a) applies in the meridian of reflection while view (b) shows how the change occurs in both meridians. In three dimensions, we use a "pencil crossed with a drumstick" object. The "drumstick" is not reversed in this case.

Example 8.1: Geometric layout of a 2-mirror periscope. (For design and analysis, use File 8.1 of the CD-ROM.)

A large 2-mirror periscope is to be designed for use as an alignment verification tool for use within a nuclear reactor. The vertical offset is 120.000 in. (3.048 m) and the optical axis intercepts with the mirrors are to be displaced horizontally by -20.000 in. (-0.508 m). The beam deviation is to be +30.000 deg. The sign conventions of Fig. 8.3 apply. (a) What mirror tilts are appropriate? (b) What are the beam deviations at each mirror?

(a) From Eq. (8.1):
$$\sigma = \arctan\left(\frac{-20.000}{120.000}\right) = -9.462 \text{ deg.}$$

From Eq. (8.2): $\theta_1 = \left(\frac{-9.462 \text{ deg} + 90 \text{ deg}}{2}\right) = 40.269 \text{ deg.}$
From Eq. (8.3): $\theta_2 = 30.000 \text{ deg} - (-9.462 \text{ deg}) - 90 \text{ deg} = -25.269 \text{ deg.}$
(b) From Eq. (8.4): $\delta_1 = 180 \text{ deg} - (2)(40.269 \text{ deg}) = 99.462 \text{ deg.}$
From Eq. (8.5): $\delta_2 = 30.000 \text{ deg} + (-9.461 \text{ deg}) - 90 \text{ deg} = -69.462 \text{ deg.}$
Check: $\delta_1 + \delta_2 = 99.462 \text{ deg} + (-69.462 \text{ deg}) = 30.000 \text{ deg.}$

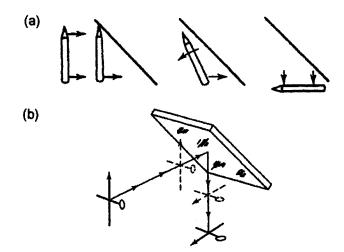


Figure 8.4 (a) Visualization of in-plane changes in image orientation upon reflection at a tilted flat mirror through use of a bouncing "pencil." (b) Twodimensional image orientation changes visualized with an object configured as an "arrow crossed by a drumstick." (Adapted from Smith.²)

The inversion that naturally occurs at an objective lens or relay lens can be included [see Fig. 8.5(a)]. In this figure, the object is to be projected by a lens onto a screen at S. The center of the image is to be located at distances Δx and Δy from the lens and the image orientation is to be as indicated. One of the many possible mirror systems that could be designed for this purpose is shown in Fig. 8.5(b). The orientations of images if they were formed at key points within the system are shown.

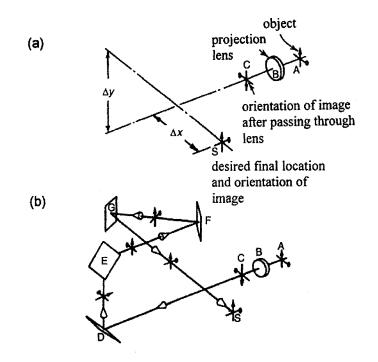


Figure 8.5 (a) Representation of a typical mirror system design problem in which an object at A is to be imaged at a particular location on a screen S with a particular orientation. (b) One of many possible mirror arrangements that could be designed for this purpose. (Adapted from Smith.²)

8.3 First- and Second-Surface Mirrors

Most mirrors used in optical instruments have light reflective coatings made of metallic and/or nonmetallic thin films on their first optically polished surfaces. These are quite logically called "first-surface" mirrors. The metals commonly used as coatings are aluminum, silver, and gold because of their high reflectivities in the UV, visible, and/or IR spectral regions. Protective coatings such as silicon monoxide or magnesium fluoride are placed over metallic coatings to increase their durability. Nonmetallic films consist of single layers or multilayer stacks of dielectric films. These stacks are combinations of materials with high and low indices of refraction. Dielectric reflecting films function over narrower spectral bands than the metals, but have very high reflectivity at specific wavelengths. They are especially helpful in monochromatic systems such as those using laser radiation. Dielectric stacks or dielectric overcoats modify the state of polarization of

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the reflected beam when the beam angle of incidence differs from zero. Figures 8.6 and 8.7 show typical reflectance vs. wavelength curves for different first-surface reflective coatings at normal and/or 45-deg incidence.

Figure 8.8(a) shows reflectance vs. wavelength for a typical multilayer dielectric film, while Fig. 8.8(b) shows reflectance vs. wavelength for a second-surface coating of silver. The latter type of reflective coating is applied to the back surface of a mirror or prism. This can be an advantage from a durability viewpoint because the film then is protected from the outside environment and physical damage that is due to handling or use. The back of the thin-film coating typically is given a protective coating such as electroplated copper plus enamel for this purpose.

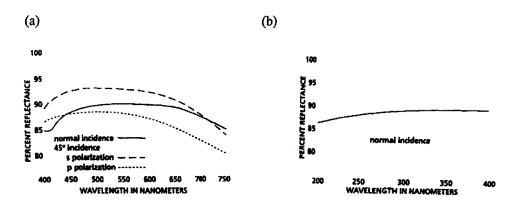


Figure 8.6 Reflectance vs. wavelength for first-surface metallic coatings of (a) protected aluminum and (b) UV enhanced aluminum.

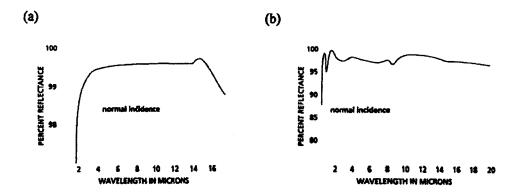


Figure 8.7 Reflectance vs. wavelength for first-surface thin films of (a) protected gold and (b) protected silver.

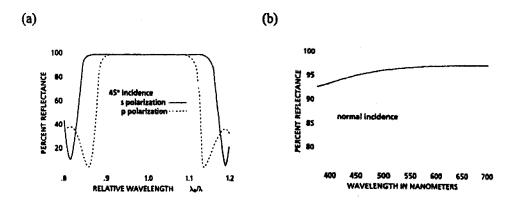


Figure 8.8 Reflectance vs. wavelength for (a) a first-surface multilayer dielectric thin film and (b) a second-surface thin film of silver.

8.4 Ghost Image Formation with Second-Surface Mirrors

Figure 8.9 illustrates a concave second-surface (or Mangin) mirror and its function in forming a normal image of a distant object. This type mirror offers distinct advantages from an optical design viewpoint as compared to the corresponding first surface version because it has more design variables (a glass thickness, a refractive index, and one more radius) to be used for aberration correction. However, it does have one drawback. Because the light to be reflected by any second-surface mirror must pass through a refracting surface to get to the reflecting surface, a ghost image is formed by the first surface. This ghost image of the object is superimposed upon the normal image as stray light and tends to reduce the contrast of the latter image. The axial separation of the two images can be increased or decreased by careful choice of radii and mirror thickness.

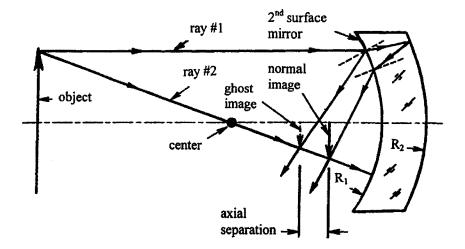


Figure 8.9 Ghost image formation from the first surface of a second-surface mirror with concentric spherical surfaces. (Adapted from Kaspereit.³)

The intensity of the ghost image is calculated from the index of refraction of the substrate using Fresnel's equations.^{2,4} At normal incidence, the reflectance, R_{λ} , of an uncoated interface between two materials with refractive indices n_1 and n_2 is:

$$R_{\lambda} = \left(\frac{(n_2 - n_1)^2}{(n_2 + n_1)^2}\right).$$
 (8.6)

The transmittance of this interface, also at normal incidence, is given by:

$$T_{\lambda} = (1 - R_{\lambda}). \tag{8.7}$$

Example 8.1 illustrates an application of these equations.

To reduce the intensity of beams reflected from uncoated surfaces, we apply a thin-film coating or a stack of coatings [called an "antireflection" (A/R) coating] to that surface. The fundamental purpose of the simplest case (a single-layer coating) is to cause destructive between a first beam reflecting from the air-film interface and a second beam reflecting

Example 8.2: Intensity of the ghost reflection from an uncoated first surface of a second-surface mirror. (For design and analysis, use File 8.2 of the CD-COM.)

A second-surface silvered mirror is made of BaK2 optical glass with $n_{\lambda} = 1.542$ in green light ($\lambda = 0.5461 \ \mu m$). The index of air is assumed as 1.000. What is the intensity of the ghost image relative to that of the normal image if the first surface is uncoated? Assume an incident beam with an intensity of unity and neglect absorption.

Applying Eq. (8.6): $R_1 = \frac{(1.52 - 1.000)^2}{(1.542 + 1.000)^2} = 0.045.$

Multiplying this factor by the intensity of the incident beam (1.0), we obtain the intensity of the ghost beam, I_G as 0.045.

Applying Eq. (8.7): $T_1 = 1 - 0.045 = 0.955$.

From Fig. 8.3(b), the green-light reflectance R_2 of the silvered surface is ~0.97.

The light beam reflected from the mirror's second surface passes twice through the front surface. Hence, its intensity upon exiting the mirror is $(0.955)^2(0.97) = 0.885$ times the intensity of the incident beam (1.0) or 0.885.

The intensity of the ghost image relative to that of the image reflected from the mirror's second surface is then $\frac{0.045}{0.885} = 0.051$ or 5.1%.

from the film-glass interface. This interference occurs when those beams are exactly 180 deg between a first beam reflecting from the air-film interface and a second beam reflecting from the film-glass interface. This interference occurs when those beams are exactly 180 deg or $\lambda/2$ out of phase. Since the second beam passes through the film twice, the desired phase shift results if the optical thickness (*n* times λ) of the film is $\lambda/4$. The combined intensity of the two reflected beams is then zero because their amplitudes subtract.^{2,4} The amplitude of a reflected beam is $(R_{\lambda})^{1/2}$.

Note that complete destructive interference occurs only at a specific wavelength λ and then only if the beam amplitudes are equal. The latter condition occurs if the following equation is satisfied:

$$n_2 = (n_1 n_2)^{1/2}, (8.8)$$

where n_2 = index of the thin film at wavelength λ , n_1 = index of the surrounding medium (typically air with n = 1), and n_3 = index of refraction of the glass at wavelength λ .

If a thin-film material with exactly the right index to A/R coat a given type of glass is not available, imperfect cancellation of the two reflected beams occurs. We calculate the resultant surface reflectance R_s as:

$$R_{s} = \left(R_{1,2}^{1/2} - R_{2,3}^{1/2}\right)^{2}, \qquad (8.9)$$

where: $R_{1,2}$ is the reflectance of the air/film interface and $R_{2,3}$ is the reflectance of the film/glass interface. Example 8.3 quantifies the advantage of one such coating.

Figure 8.10 shows the formation of a ghost reflection from a flat second-surface mirror of thickness *t* oriented at an angle of 45 deg to the axis of a lens. With the mirror at 45 deg in air, the ghost is displaced axially relative to the normal image by $d_A = (2t/n) + d_A$. The ghost is also displaced laterally by $d_L = 2t / [(2)(2n^2 - 1)]^{1/2}$. Once again, superimposition of the ghost image onto the second-surface reflected image tends to reduce the contrast of the latter image. By A/R coating the first surface, the ghost can be reduced in intensity, as quantified earlier. Fresnel's equation Eq. (8.6) must be modified to accommodate the oblique incidence of the beam at the ghost-forming surface.^{2,4}

High-efficiency, multilayer A/R coatings can be designed to have zero reflectivity at a specific wavelength or reduced variations of reflectivity with wavelength. Figure 8.11 shows plots of reflectance vs. wavelength for a single-layer (MgF₂) coating, a "broad band" multilayer coating with low reflectivity over the entire visible spectrum, and two multilayer coatings with zero reflectivity at $\lambda = 550$ nm. All these coatings are applied to crown glass. Coatings V1 and V2 are called "V-coats" because of their characteristic downward-pointing triangular shapes.

Example 8.3: Relative intensity of the ghost reflection from a second-surface mirror with an A/R coating on the first surface. (For design and analysis, use File 8.3 of the CD-ROM.)

A single-film A/R coating of MgF₂ with $n_2 = 1.380$ is applied to the first surface of the Mangin mirror from Example 8.2. Its optical thickness in green light is $\lambda/4 = 0.5461/4 = 0.136 \mu m$. (a) What is the intensity of the ghost image from the coated surface relative to the intensity of the main image? (b) What conclusion can you draw about the efficiency of this coating?

(a) From Eq. (8.8), the film index n_2 should be $[(1.000)(1.542)]^{1/2} = 1.242$ for a perfect antireflection function. This is not the case here, so the coating is imperfect.

From Eq. (8.6): $R_{1,2}$ of the air/film interface is $\frac{(1.380 - 1.000)^2}{(1.380 + 1.000)^2} = 0.0255.$ $R_{2,3}$ of the film/glass interface is $\frac{(1.542 - 1.380)^2}{(1.542 + 1.380)^2} = 0.0031.$

From Eq. (8.9): the intensity of the ghost image is $R_S = [(0.0255)^{1/2} - (0.0031)^{1/2}]^2 = 0.0108.$

From Eq. (8.7): $T_s = 1 - 0.0108 = 0.9892$.

The intensity of the main image reflected from the mirror's second surface is

 $(0.9892)^2(0.97) = 0.9492.$

The relative intensity of the ghost image is then $\frac{0.0108}{0.9492} = 0.0114 = 1.1\%$.

(b) Even though the thin-film index is not optimum for the given substrate material, it does reduce the relative intensity of the ghost image to about one-fifth that of an uncoated surface (which is 5.1% per Example 8.2).

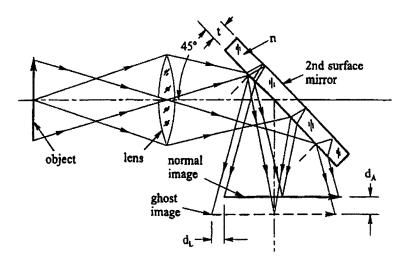


Figure 8.10 Formation of a ghost image at a second-surface mirror inclined 45 deg to the axis. (Adapted from Kaspereit.³)

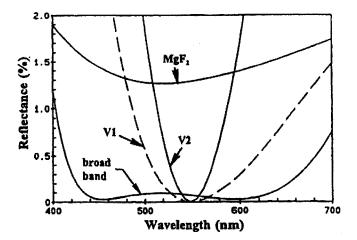


Figure 8.11 Spectral variations of reflectance for several multilayer A/R coatings as identified in the text.

An obvious difference between first- and second-surface mirrors is that a transparent substrate is needed for the latter, but not for the former. Tables B8 and B9 list the mechanical properties and "figures of merit" for the common nonmetallic and metallicmirror substrate materials. Of these, only fused silica has good refractive properties. Hence, second-surface mirrors must be made of that material, one of the optical glasses (see Tables B1 and B2); a crystal (see Tables B4 through B7); or, if the performance requirements are low, perhaps an optical plastic (see Table B3). As mentioned earlier, a related advantage of the second-surface mirror is that an additional surface radius and asphericity, an axial thickness, and an index are available for controlling aberrations as well as for controlling the locations of a ghost image. The second-surface mirror configuration is most frequently used in Mangin-type primary or secondary mirrors in catadioptric systems for photographic and moderate-sized astronomical telescope applications. Second-surface designs obviously do not work in mirrors with contoured or pocketed back surfaces or in built-up mirror substrates.

8.5 Approximation of Mirror Aperture

The size of a mirror is set primarily by the size and shape of the optical beam as it intercepts the reflecting surface, plus any allowances that need to be made for mounting provisions, misalignment, and/or beam motion during operation. The so-called "beamprint" can be approximated from a scaled layout of the optical system showing extreme rays of the light beam in at least two orthogonal meridians. This method is time-consuming to use and may be inaccurate because of compounded drafting errors. Modern computer-aided design methods have alleviated both these problems, especially with software that interfaces ray-tracing capability with the creation of drawings or graphic renditions to any scale and in any perspective. In spite of these advances, some of us rely upon hand calculations, at least at early stages of system design. We include here a set of equations, adapted from Schubert,⁵ that allow minimum elliptical beamprint dimensions for a circular beam intersecting a tilted flat mirror. The geometry is shown in Fig. 8.12. The ellipse is assumed to be centered on the mirror in the minor-axis direction.

$$W = D + 2L\tan\alpha, \tag{8.10}$$

$$E = \frac{W \cos \alpha}{\left[2\sin\left(\theta - \alpha\right)\right]},\tag{8.11}$$

$$F = \frac{W \cos \alpha}{\left[2\sin(\theta + \alpha)\right]},\tag{8.12}$$

$$A = E + F, \tag{8.13}$$

$$G = \left(\frac{A}{2}\right) - F,\tag{8.14}$$

$$G = \frac{AW}{\left(A^2 - 4G^2\right)^{1/2}},$$
(8.15)

where W is the width of the beamprint at the mirror/axis intercept, D is the beam diameter at a reference plane perpendicular to the axis and located at an axial distance L from the mirror/axis intercept, α is the beam divergence angle of the extreme off-axis reflected ray (assuming a symmetrical beam), E is the distance from the top edge of the beamprint to the mirror/axis intercept, θ is the mirror surface tilt relative to the axis (equal to 90 deg minus the tilt of the mirror normal, F is the distance from the bottom edge of the beamprint to the mirror/axis intercept, A is the major axis of the beamprint; G is the offset of the beamprint center from the mirror/axis intercept, and B is the minor axis of the beamprint.

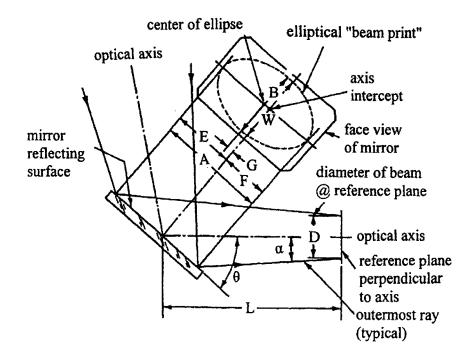


Figure 8.12 Geometric relationships used to estimate the beamprint of a rotationally symmetric beam on a tilted flat mirror. (Adapted from Schubert.⁵)

These equations apply regardless of the direction in which the beam is propagating as long as the reference plane is located where D is smaller than W. For a collimated beam propagating parallel to the axis, α and G are zero and the above equations reduce to the symmetrical case where:

$$B = W = D, \tag{8.15a}$$

$$E = F = \frac{D}{\left(2\sin\theta\right)},\tag{8.11a}$$

$$A = \frac{D}{\sin \theta}.$$
 (8.13a)

The use of these equations is demonstrated in Example 8.4.

The dimensions of the reflecting surface should be increased somewhat from those calculated with Eqs. (8.10) through (8.15) and (8.11a), (8.13a), and (8.15a) to allow for the factors mentioned earlier (mechanical mounting clearances, beam motion, etc.) and for reasonable manufacturing tolerances on all dimensions. All mirrors should have protective bevels; some should be heavily chamfered to remove unneeded material. Very small mirrors might need to have large relative thicknesses to provide material for mounting.

Example 8.4: Beam footprint on a tilted mirror. (For design and analysis, use File 8.4 of the CD-ROM.)

Calculate the beamprint dimensions for a circular beam of D = 25.400 mm (1.000 in.)incident at axial distance L = 50.000 mm (1.968 in.) on a flat mirror tilted at an angle θ = 30 deg to the axis. Assume the beam divergence α to be axisymmetric and (a) 0.5 deg and (b) zero (i.e., collimated).

From Eq. (8.10): $W = 25.400 + (2)(50.000)(\tan 0.5 \text{ deg}) = 26.273 \text{ mm} (1.034 \text{ in}).$ From Eq. (8.11): $E = \frac{(26.237)(\cos 0.5 \text{ deg})}{[2\sin (30 \text{ deg} - 0.5 \text{ deg})]} = 26.677 \text{ mm} (1.050 \text{ in}.).$ From Eq. (8.12): $F = \frac{(26.237)(\cos 0.5 \text{ deg})}{[2\sin (30 \text{ deg} + 0.5 \text{ deg})]} = 25.881 \text{ mm} (1.019 \text{ in}.).$ From Eq. (8.13): A = 26.677 + 25.881 = 52.558 mm (2.069 in.).From Eq. (8.14): $G = \left(\frac{52.558}{2}\right) - 25.881 = 0.398 \text{ mm} (0.016 \text{ in}.).$ From Eq. (8.15): $B = \frac{(52.558)(26.273)}{[52.558^2 - (4)(0.397^2)]^{1/2}} = 26.276 \text{ mm} (1.034 \text{ in}.).$

(b) From Eq. (8.15a):
$$B = W = D = 25.400 \text{ mm} (1.000 \text{ in.}).$$

From Eq. (8.11a): $E = F = \frac{25.4}{(2)(\sin 30 \text{ deg})} = 25.400 \text{ mm} (1.000 \text{ in.}).$
From Eq. (8.13a): $A = \frac{25.4}{\sin 30 \text{ deg}} = 50.800 \text{ mm} (2.000 \text{ in.}).$

As might be expected, the elliptical beamprint in part (a) is slightly decentered upward with respect to the axis in the plane of reflection, but is symmetrical to the axis in the orthogonal plane. In part (b), the beamprint is symmetrical in both directions.

8.6 Weight Reduction Techniques

In large mirrors and even in some small and most modest-sized mirrors, weight minimization can prove advantageous or, in some cases, absolutely necessary. Given a chosen substrate material, a reduction in mirror weight from that of the regular solid can be made only by changing the configuration. The usual ways of doing this are to remove unneeded material from a solid substrate or to combine separate pieces to create a built-up structure with a lot of empty spaces inside. No matter what technique is used to minimize mirror weight, the product must be of high quality and capable of economical fabrication and testing. Rodkevich and Robachevskaya⁶ correctly stated the fundamental requirements for precision mirrors and lightweight versions of them.⁶ These statements are paraphrased here as follows:

- 1. The mirror material must be highly immune to outside mechanical and temperature influences; it must be isotropic and possess stable properties and dimensions.
- 2. The mirror material must accept a high-quality polished surface and a coating having the required reflection coefficient.
- 3. The mirror construction must be capable of being shaped to a specified optical surface contour and must retain this shape under operating conditions.
- 4. Lightweight mirrors must have a lower mass than those made to traditional designs while maintaining adequate stiffness and homogeneity of properties.
- 5. Similar techniques should be used for fabricating conventional and lightweight mirrors.
- 6. If possible, mounting and load relief during testing and use should employ conventional techniques and should not increase the mass of the mirror and/or the complexity of the mechanisms involved.

These idealized principles would serve as useful guidelines for the design of a mirror of any size. Material selection, fabrication methods, dimensional stability, and configuration design are key to meeting these guidelines. Tables B8a and B8b list material properties such as the coefficient of thermal expansion, thermal conductivity, and Young's modulus, which relate to inherent behavior under changing environmental conditions. Dimensional stability and homogeneity of properties differ from one material type to another. Further information on these topics may be found in other publications, such as those by Englehaupt⁷ and Paquin.⁸ Comparisons of mechanical and thermal material figures of merit especially pertinent to mirror design are given in Table B9. Table B10a lists the characteristics of different types of aluminum alloys that are candidates for making metallic mirrors. Typical fabrication methods, surface finishes, and coatings for mirrors made of various materials are listed in Table B11. Mounting methods for smaller mirrors are considered in Chapter 9 of this book, while those typically used to mount metallic and larger mirrors are discussed in Chapters 9 and 10, respectively. The following sections deal with configurations for reducing the weight of mirrors (lightweighting).

8.6.1 Contoured-back configurations

We start by considering various techniques for reducing mirror weight with solid mirrors that have circular apertures and flat backs. First-surface reflection takes place on an optical surface that is flat, concave, or convex. In general, the following discussion would apply also to rectangular or nonsymmetrical designs. Thinning the baseline substrate clearly reduces weight, but also reduces stiffness and increases self-weight deflection. Hence, that technique can be used only within limits. The simplest means for lightweighting solid frontsurface mirrors is to contour the back surface (R_2). Figure 8.13 illustrates this approach for a series of six concave mirrors made of the same material and having the same diameter D_G and the same spherical radius of curvature of the reflecting surface (R_1). Fabrication complexity generally increases from left to right in views (a) through (f). Figure 8.13(g) shows a double-concave version that is *not* lightweighted. It is included here because it is a viable candidate configuration for some applications and has been used successfully in the past.

Mirror-shape variations are discussed in turn and how to calculate its volume is shown, which when multiplied by the appropriate density, gives the mirror weight. Typical examples are discussed in which the mirror diameters, the radius of curvature of the reflecting surfaces, the material types, and the axial thicknesses are the same, but their R_2

surfaces are variously contoured. This allows direct comparison of relative weights of the mirrors.

Figure 8.13(a) shows a concave mirror with flat back that will serve as the baseline for the comparison. Its axial and edge thicknesses are t_A and t_E , respectively. The sagittal depth, S_1 , is given by Eq. (8.16) while the mirror volume is given by Eq. (8.17):

$$S_{1} = R_{1} - \left[R_{1}^{2} - \left(\frac{D_{G}}{2} \right)^{2} \right]^{1/2}, \qquad (8.16)$$

$$t_E = t_A + S_1, (8.17)$$

$$V_{\text{BASELINE}} = \pi R_2^2 t_E - \left(\frac{\pi}{3}\right) \left(S_1^2\right) \left(3R_1 - S_1\right).$$
(8.18)

In Example 8.5, we calculate the mirror's volume and weight assuming it (and all others in this group) to be made of Corning ULE.

Example 8.5: Baseline solid flat-back concave mirror. (For design and analysis, use File 8.5 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(a) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). It has a radius of curvature R_1 of 72.000 in. (1828.800 mm). Calculate (a) the volume and (b) the weight of the mirror.

(a) By Eq. (8.16):

$$S_1 = 72.000 - \left[72.000^2 - \left(\frac{18.000}{2}\right)^2 \right]^{1/2} = 0.565 \text{ in. (14.351 mm)}.$$

By Eq. (8.17): $t_E = 3.000 + 0.565 = 3.565$ in. (90.551 mm). By Eq. (8.18):

$$V_{\text{BASELINE}} = (\pi) (9.000^2) (3.565) - \left(\frac{\pi}{3}\right) (0.565^2) [(3)(72.000) - 0.565]$$

 $= 907.180 - 72.018 = 835.162 \text{ in.}^3 (13,685.853 \text{ cm}^3).$

(b) From Table B8a: ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³). $W_{\text{BASELINE}} = V\rho = (835.162)(0.0797) = 66.562$ lb (30.192 kg).

Note: this weight is the baseline for comparison in the following seven examples.

The simplest back-surface contour is tapered (conical), as indicated in Fig. 8.13(b). Thickness varies linearly between some selected radius, r_1 , and the rim. Equation (8.19) defines the axial extent of the tapered region and Eq. (8.20) gives the mirror volume.

$$t_1 = t_A + S_1 - t_E, (8.19)$$

$$V_{\text{TAPERED}} = V_{\text{BASELINE}} - \left(\pi \frac{t_1}{2}\right) (r_2^2 - r_1^2).$$
 (8.20)

Example (8.6) applies these equations to the tapered back mirror.

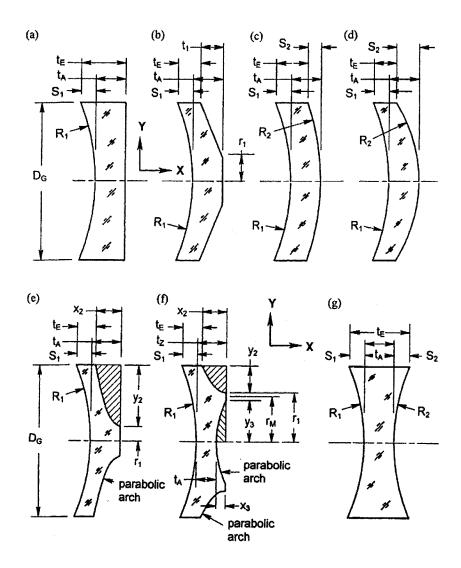


Figure 8.13 Examples of concave mirrors with weight reduced by contouring the rear surface: (a) baseline with flat rear surface, (b) tapered (conical) rear surface, (c) concentric spherical front and rear surfaces with $R_2 = R_1 + t_A$, (d) spherical rear surface with $R_2 < R_1$, (e) single-arch configuration, (f) double-arch configuration, (g) double-concave configuration (not lightweighted, for comparison).

Example 8.6: Solid tapered-back concave mirror. (For design and analysis, use File 8.6 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(b) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). It has a radius of curvature R_1 of 72.000 in. (1828.800 mm) and a conical back starting at an inner radius $r_1 = 1.500$ in. (38.100 mm) and tapering to an edge thickness, t_E , of 0.500 in. (12.700 mm). (a) What is the mirror weight and (b) how does that weight compare with the baseline mirror weight of 66.562 lb (30.192 kg)? Note: the baseline volume is 835.162 in.³ (13,685.853 cm³).

(a) From Eq. (8.16):

$$S_1 = 72.000 - \left[72.000^2 - \left(\frac{18.000}{2}\right)^2\right]^{1/2} = 0.565 \text{ in. } (14.351 \text{ mm}).$$

From Eq. (8.19): $t_1 = t_A + S_1 - t_E = 3.000 + 0.565 - 0.500 = 3.065$ in. (77.851 mm). From Eq. (8.13):

$$V_{\text{TAPERED}} = 835.162 - \left[(\pi) \left(\frac{3.065}{2} \right) (9.000^2 - 1.500^2) \right]$$

= 835.162 - 379.141 = 456.021 in.³ (7472.843 cm³).

From Table B8a: ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³). Mirror weight = (V_{TAPERED})(ρ_{ULE}) = (456.02)(0.0797) = 36.345 lb. (16.4860 kg).

(b) The mirror's relative weight is (36.345/66.562)(100) = 55% of the baseline mirror weight.

à

In Fig. 8.13(c) and 8.13(d), we see meniscus-shaped mirrors with concentric radii and $R_2 < R_1$, respectively. The first case has a uniform thickness over the aperture, so only a modest reduction in weight from the plano-concave case [Fig. 8.13(a)] is possible. The second case allows greater weight reduction because the rim is significantly thinner. Equations (8.21) through (8.24) are used to find the concentric mirror's second radius, the sagittal depth of each mirror, their edge thicknesses, and their volumes. Their weights and relative weights can then easily be obtained.

$$R_2 = R_1 + t_A, (8.21)$$

$$S_2 = R_2 - \left[R_2^2 - \left(\frac{D_G}{2} \right)^2 \right]^{1/2}, \qquad (8.22)$$

$$t_E = t_A + S_1 - S_2,$$
 (8.23)

$$V_{\text{MENISCUS}} = V_{\text{BASELINE}} - \pi \left(\frac{D_G}{2}\right)^2 S_2 + \left(\frac{\pi}{3}\right) \left(S_2^2\right) \left(3R_2 - S_2\right), \tag{8.24}$$

Examples 8.7 and 8.8 pertain to concentric and nonconcentric meniscus mirrors.

Example 8.7: Solid concentric meniscus mirror. (For design and analysis, use File 8.7 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(c) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). It has a radius of curvature R_1 of 72.000 in. (1828.800 mm) and a spherical back surface that is concentric with R_1 . (a) What are the mirror's R_2 and edge thickness, (b) what is its weight, and (c) how does that weight compare with the baseline mirror weight of 66.562 lb (30.192 kg)? Note: the baseline volume is 835.162 in.³ (13,685.853 cm³).

(a) From Eq. (8.21):
$$R_2 = 72.00 + 3.000 = 75.000$$
 in. (1905.000 mm).

From Eq. (8.22):
$$S_2 = 75.00 - \left[75.000^2 - \left(\frac{18.000}{2} \right)^2 \right] = 0.542$$
 in. (13.767 mm).
From Eq. (8.16): $S_2 = 72.000 - \left[72.000^2 - \left(\frac{18.000}{2} \right)^2 \right] = 0.565$ in. (14.351 mm).

From Eq. (8.23): $t_E = 3.000 + 0.565 - 0.542 = 3.023$ in. (76.784 mm).

(b) From Eq. (8.24):

$$V_{\text{CONCENTRIC}} = 853.162 - \left[\left(\pi \right) \left(\frac{18.000}{2} \right)^2 \right] \left[0.542 \right] + \left(\frac{\pi}{3} \right) \left(0.542^2 \right) \left[\left(3 \right) \left(75.000 - 0.542 \right) \right] \\ = 835.162 - 137.922 + 69.050 = 766.290 \text{ in.}^3 (12,557.240 \text{ cm}^3).$$

From Table B8a: ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³) Mirror weight = $(V_{\text{CONCENTRIC}})(\rho_{\text{ULE}}) = (766.290)(0.0797) = 61.073$ lb (27.703 kg). (c) The relative weight is $\left(\frac{61.073}{66.562}\right)(100) = 92\%$ of the baseline mirror weight

The mirror design in Fig. 8.13(e) is called the "single-arch" configuration. The concave back surface could have a parabolic or circular contour. In the former case, the axis of the parabola could be oriented parallel to the mirror axis (X) and decentered to locate its vertex at P_1 on the rim of the mirror. This is called an X-axis parabola. Alternatively, the parabola could be oriented radially with its vertex at P_2 on the mirror's back. This is called a Y-axis parabola. All three curves must pass through P_1 and P_2 . The radius of the circle is a design variable. These possibilities are drawn approximately to scale in Fig. 8.14 for the mirror considered in the examples. The circle shown is chosen to be parallel to the reflecting surface at the mirror's rim. The volume of material removed by contouring the back surface with either of the parabolic contours or a circular contour can be calculated by finding the sectional area to the right of the chosen curve in Fig. 8.14 and bound at the right by the vertical line A-B. That area is then revolved around the mirror's axis at the radius of the chosen circular contour than for either parabolic contour. This means that the mirror's relation of the mirror's relation of the sectional area is smaller for the Example 8.8: Solid meniscus mirror with $R_2 < R_1$. (For design and analysis, use File 8.8 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(d) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). It has radii of curvature R_1 of 72.000 in. (1828.800 mm) and R_2 of 14.746 in. (374.548 mm). (a) What is the mirror's edge thickness, (b) what is its weight, and (c) how does that weight compare with the baseline mirror weight of 66.562 lb (30.192 kg)? Note: the baseline volume is 835.162 in.³ (13,685.853 cm³).

(a) From Eq. (8.22):
$$S_2 = 14.746^2 - \left[14.746^2 - \left(\frac{18.000}{2} \right)^2 \right] = 3.065$$
 in. (77.851 mm).
From Eq. (8.16): $S_1 = 72.000^2 - \left[72.000^2 - \left(\frac{18.000}{2} \right)^2 \right] = 0.565$ in. (14.351 mm).

From Eq. (8.23): $t_E = 3.000 + 0.565 - 3.065 = 0.500$ in. (12.700 mm)

(b) From Eq. (8.24):

$$V_{R2$$

 $= 835.162 - 779.950 + 405.050 = 460.260 \text{ in.}^3 (7542.310 \text{ cm}^3).$

From Table B8a: ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³).

Mirror weight = $(V_{\text{CONCENTRIC}})(\rho_{\text{ULE}}) = (460.260)(0.0797) = 36.683 \text{ lb} (16.639 \text{ kg}).$

(c) The mirror's relative weight is (36.345/66.562)(100) = 55% of the baseline mirror weight.

weight reduction with either parabolic contour is greater than that with the circular back contour.

The sectional area, A_P , for either half-parabola is given by Eq. (8.25) where x_2 and y_2 are as illustrated in Fig. 8.14. The greatest volume is removed with the X-axis parabola because it has a slightly larger radius of revolution. Volumes for X axis and Y axis cases are given by Eq. (8.26).

$$A_{p} = \left(\frac{2}{3}\right)(x_{2})(y_{2}), \qquad (8.25)$$

$$V_{\text{S-ARCH}} = V_{\text{BASELINE}} - (A_{p})(2\pi)(y_{\text{CENTROID}}), \qquad (8.26)$$

where:

$$y_{\text{CENTROID Y}} = r_1 + \left(\frac{3}{5}\right)(y_2),$$
 (8.27)

$$y_{\text{CENTROID X}} = r_1 + y_2 - \left(\frac{3}{8}\right)(y_2).$$
 (8.28)

The dimension x_2 of the mirror is given by

$$x_2 = t_A + S_1 - t_E, (8.29)$$

$$y_2 = \left(\frac{D_G}{2}\right) - r_1. \tag{8.30}$$

The parabola with symmetry about the Y axis might be preferred because the mirror thickness then decreases monotonically with increasing distance from the axis. This is not necessarily the case for the X-axis parabola. As may be seen in Fig. 8.14, the minimum mirror thickness for this contour occurs well inside the rim.

Cho et al⁹ showed that the single-arch mirror with a Y-axis parabolic back surface gives the best compromise between self weight deflections when the mirror axis is oriented at the zenith and the horizon.

In Examples 8.9 and 8.10, we calculate the relative weights for single-arch mirrors with Y- and X-axis parabolic contours.

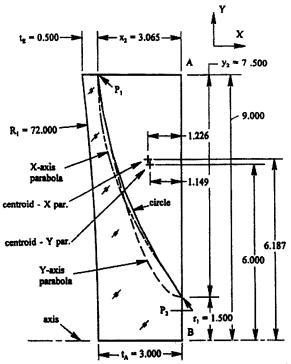


Figure 8.14 Three possible contours (X axis parabola, Y axis parabola, and circle) for the single-arch mirror plotted to the same scale. Dimensions are in inches and apply to Examples 8.9 and 8.10.

Example 8.9: Solid single-arch mirror with Y axis parabolic back, (For design and analysis, use File 8.9 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(e) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). It has a radius of curvature R_1 of 72.000 in. (1828.800 mm) and a single-arch Y-axis parabolic back contour with vertex at $r_1 = 1.500$ in. (38.100 mm). The edge thickness of the mirror is 0.500 in. (12.700 mm). (a) What is the mirror's weight, and (b) how does that weight compare with the baseline mirror weight of 66.562 lb (30.192 kg)? Note: the baseline volume is 835.162 in.³ (13,685.853 cm³). By Eq. (8.16): $S_1 = 72.000 - \left[72.000^2 - \left(\frac{18.000}{2}\right)^2 \right] = 0.565 \text{ in. (14.351 mm)}.$ By Eq. (8.29): $x_2 = 3.000 + 0.565 - 0.500 = 3.065$ in. (77.851 mm). By Eq. (8.30): $y_2 = \left(\frac{18.000}{2}\right) - 1.500$ in. = 7.500 in. (190.500 mm). By Eq. (8.25): $A_p = \left(\frac{2}{3}\right)(3.065)(7.500) = 15.325 \text{ in.}^2(9887.082 \text{ mm}_2).$ By Eq. (8.28): $y_{\text{CENTROID X}} = 1.500 + (3/5)(7.500) = 6.000$ in. (152.400 mm) By Eq. (8.26): $V_{\text{S-ARCH}} = 835.162 - (15.325)(2\pi)(6.000)$ $= 257.423 \text{ in.}^3 (4218.409 \text{ mm}^3)$ From Table B8a: ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³) Mirror weight = $(V_{\text{CONCENTRIC}})(\rho_{\text{ULE}}) = (257.423)(0.0797) = 20.516 \text{ lb.} (9.306 \text{ kg})$ (c) The mirror's relative weight is (20.516/66.562)(100) = 31% of the baseline mirror weight.

The double-arch mirror of Fig. 8.13(f) is thickest at a zone usually chosen as about 55% of the mirror's diameter.⁹ It is typically supported at three or more points within this zone. The rear surface is shaped as two parabolic curves with perhaps equal thickness at the rim and axis. The outer arch might well be a Y-axis parabola, while the inner arch might be an X-axis parabola. The latter choice is made to avoid an inflection point in the inner arch surface at the axis. Equation (8.25) can be used to calculate the sectional areas of either type of arch. We use x_2 and y_2 for the outer case and x_3 and y_3 for the inner case, as illustrated in Fig. 8.14.

The radius of revolution ($y_{CENTROID Y}$) for the outer arch and the sectional area of that arch of the mirror are given by Eqs. (8.27) and (8.25), respectively. Equation (8.31) defines the volume of the outer arch. The corresponding parameters for the inner arch are given by Eqs. (8.32) and (8.33).

$$V_{\text{OUTER ARCH}} = (A_{\text{p-OUTER}})(2\pi)(y_{\text{CENTROID Y}}), \qquad (8.31)$$

Example 8.10: Solid single-arch mirror with X axis parabolic back. (For design and analysis, use File 8.10 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(f) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). It has a radius of curvature R_1 of 72.000 in. (1828.800 mm) and a single-arch X axis parabolic back contour with its vertex at the right hand edge of the mirror rim (P_1 in Fig. 8.15) and passing through the point P_2 on the mirror back at $r_1 = 1.500$ in. (38.100 mm). The edge thickness of the mirror is 0.500 in. (12.700 mm). (a) What is the mirror's weight, and (b) how does that weight compare with the baseline mirror weight of 66.562 lb (30.192 kg)? Note: the baseline volume is 835.162 in.³ (13,685.853 cm³).

By Eq. (8.16):
$$S_1 = 72.000 - \left[72.000^2 - \left(\frac{18.000}{2} \right)^2 \right] = 0.565$$
 in. (14.351 mm).

By Eq. (8.30): $x_2 = 3.000 + 0.565 - 0.500 = 3.065$ in. (77.851 mm).

By Eq. (8.31): $y_2 = \left(\frac{18.000}{2}\right) - 1.500$ in. (38.100 mm) = 7.500 in. (190.500 mm). By Eq. (8.25): $A_p = \left(\frac{2}{3}\right)(3.065)(7.500) = 15.325$ in.² (9887.082 mm₂). By Eq. (8.31): $y_{\text{CENTROID X}} = 1.500 + 7500 - \left(\frac{3}{8}\right)(7.500) = 6.187$ in. (157.162 mm). By Eq. (8.32): $V_{\text{S-ARCH}} = 835.162 - (15.325)(2\pi)(6.187) = 835.162 - 595.745$ = 239.417 in.³ (3923.372 cm³).

From Table B8a: ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³). Mirror weight = (V_{CONCENTRIC})(ρ _{ULE}) = (239.417)(0.0797) = 19.081 lb. (8.655 kg).

(c) The mirror's relative weight is (19.081/66.562)(100) = 29% of the baseline mirror weight.

$$y_{\text{CENTROID X}} = \left(\frac{3}{8}\right)(y_3), \qquad (8.32)$$

$$V_{\text{INNER ARCH}} = (A_{\text{P INNER}})(2\pi)(y_{\text{CENTROID X}}), \qquad (8.33)$$

$$V_{\text{D-ARCH}} = V_{\text{BASELINE}} - V_{\text{OUTER ARCH}} - V_{\text{INNER ARCH}}.$$
(8.34)

The symmetrical double-concave (DCC) mirror configuration shown in Fig. 8.13(g) does not reduce substrate weight, but is included here for comparison. It is generally used only when the axis is horizontal, or nearly so, because gravity deflections then are symmetrical about the midplane and are smaller than with nonsymmetrical configurations. This mirror suffers excessively from surface deformation when the axis is vertical.⁹

Example 8.11: Solid double-arch mirror. (For design and analysis, use File 8.11 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(f) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). It has a radius of curvature R_1 of 72.000 in. (1828.800 mm) and a double-arch back contour with Y axis parabolic outer arch and X axis parabolic inner arch. Assume $t_E = t_A = 0.500$ in. (12.700 mm), $t_Z = 3.000$ in. (76.200 mm), $r_M = 0.550(D_G/2)$, and the annular zone width on the mirror's back = 0.600 in. (15.240 mm). (a) What is the mirror's weight, and (b) how does that weight compare with the baseline mirror weight of 66.562 lb (30.192 kg)? Note: the baseline volume is 835.162 in.³ (13,685.853 cm³).

(a)
$$r_M = (0.550) \left(\frac{18.000}{2}\right) = 4.950$$
 in. (125.730 mm).
By inspection of Fig. 8.14(g):
 $y_3 = r_M - \left(\frac{0.600}{2}\right) = 4.950 - 0.300 = 4.650$ in.(118.110 mm)
 $r_1 = r_M + \left(\frac{0.600}{2}\right) = 4.950 + 0.300 = 5.250$ in.(133.350 mm).

$$y_2 = \left(\frac{D_G}{2}\right) - r_1 = 9.000 - 5.250 = 3.750 \text{ in.}(95.250 \text{ mm}).$$

By Eq. (8.16):
$$S_1 = 72.000 - [72.000^2 - (18.000/2)^2]^{1/2} = 0.565$$
 in. (14.351 mm)

By inspection of Fig. 8.14(g):

 $x_2 = t_Z + S_1 - t_E = 3.000 + 0.565 - 0.500 = 3.065$ in. (77.851 mm), $x_3 = t_Z - t_A = 3.000 - 0.500 = 2.500$ in. (63.500 mm).

By Eq. (8.25):
$$A_{\text{p-OUTER}} = \left(\frac{2}{3}\right)(3.065)(3.750) = 7.662 \text{ in.}^2 (49.432 \text{ cm}^2),$$

 $A_{\text{p-OUTER}} = \left(\frac{2}{3}\right)(2.500)(4.650) = 7.750 \text{ in.}^2 (50.000 \text{ cm}^2).$

By Eq. (8.27):
$$y_{\text{CENTROID Y}} = 5.250 + \left(\frac{3}{5}\right)(3.750) = 7.500 \text{ in.}(190.500 \text{ mm}).$$

By Eq. (8.32): $y_{\text{CENTROID X}} = \left(\frac{3}{8}\right) (4.650) = 1.744 \text{ in.} (44.291 \text{ mm}).$ By Eq. (8.31): $V_{\text{OUTER ARCH}} = (7.662)(2\pi)(7.500) = 361.063 \text{ in.}^3 (5916.766 \text{ cm}^3).$

By Eq. (8.31): $V_{\text{DNRR ARCH}} = (7.750)(2\pi)(1.744) = 84.923 \text{ in.}^3 (1391.647 \text{ cm}^3).$ By Eq. (8.34): $V_{\text{D-ARCH}} = 835.162 - 361.063 - 84.923$ = 389.174 in.³ (6377.419 cm³).

From Table B8a, ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³). Mirror weight = ($V_{\text{D-ARCH}}$)(ρ_{ULE}) = (389.174)(0.0797) = 31.017 lb. (14.069 kg).

(c) The mirror's relative weight is (31.017/66.562)(100) = 47% of the baseline mirror's weight.

The edge thickness and volume of this type mirror are

$$t_{\rm E} = t_{\rm A} + S_1 + S_2, \tag{8.35}$$

$$V_{\rm DCC} = (\pi) (r_2^2) (t_E) - \left(\frac{\pi}{3}\right) (S_1^2) (3R_1 - S_1) - \left(\frac{\pi}{3}\right) (S_2^2) (3R_2 - S_2).$$
(8.36)

Example 8.12 uses these equations in a design for a symmetrical double-concave mirror as part of our group comparison.

Example 8.12: Solid double-concave mirror. (For design and analysis, use File 8.12 of the CD-ROM.)

A concave ULE mirror shaped as in Fig. 8.13(g) has a diameter D_G of 18.000 in. (457.200 mm) and axial thickness t_A of 18.000/6 = 3.000 in. (76.200 mm). Each of its two optical surfaces has a radius of curvature R_1 of 72.000 in. (1828.800 mm). The axial thickness is 3.000 in. (76.200 mm). (a) What is the mirror's weight, and (b) how does that weight compare with the baseline mirror weight of 66.562 lb (30.192 kg)? Note: the baseline volume is 835.162 in.³ (13,685.853 cm³).

(a) By Eq. (8.16): $S_1 = 72.000 - (72.000^2 - 9.000^2)^{1/2} = 0.565$ in. (14.351 mm). By Eq. (8.22): $S_2 = 72.000 - (72.000^2 - 9.000^2)^{1/2} = 0.565$ in. (14.351 mm). By Eq. (8.35): $t_E = 3.000 + 0.565 + 0.565 = 4.130$ in (104.902 mm). By Eq. (8.36): $V_{DCC} = (\pi) \left(\frac{18.000}{2}\right)^2 (4.130) - (2) \left(\frac{\pi}{3}\right) (0.565^2) [(3)(72.000) - 0.565]$ = 1050.957 - (2)(72.018) = 906.921 in.³ (14,861.772 cm³). From Table B8a, ρ for ULE = 0.0797 lb/in.³ (2.205 g/cm³). Mirror weight = $(V_{D-ARCH})(\rho_{ULE}) = (906.921)(0.0797) = 72.282$ lb. (32.787 kg). (c) The mirror weight has been *increased* by a factor of (72.282/66.562)(100) = 109% as compared to the baseline design.

8.6.2 Cast ribbed substrate configurations

Historically, early attempts to reduce the weight of large mirrors for astronomical telescopes involved casting pockets into the back surface of the substrate to eliminate material that contributed little or nothing to the strength or stiffness of the mirror. Notable in these early efforts was the casting by the Corning Glass Works of two blanks for the 200-in. (5.1-m) diameter Hale telescope that has been operational on Mt. Palomar in California since 1949. These blanks, which were cast before World War II, were made of a then new borosilicate crown glass (Pyrex) with CTE of $\sim 2.5 \times 10^{-6}$ /°C. To accelerate temperature stabilization, the structures cast into the substrates have ribs of approximately 4-in. (10.2-cm) maximum thickness. The overall edge thickness of the mirror used in the telescope is about 24 in. (61

cm); the central hole for light passage is about 40-in. (102-cm) in diameter. The weight of the mirror is about 20 tons (1.8×10^4 kg), representing a saving of about 50% over that of a solid disk with an equivalent self-weight deflection.^{10, 11}

A vast amount of material was removed from the blank for the Hale primary as it was ground to an f/3.3 parabola. A technique for making even larger cast mirrors is spin casting the glass in a mold containing numerous hexagonal shaped ceramic void formers (cores) arranged as the negative of the desired structure. The mold is located within a furnace that is slowly rotating about a vertical axis. After the raw glass is melted on top of the cores, centrifugal force creates a near net-shaped parabolic top surface, thereby minimizing subsequent material removal. Several large mirrors have been made by this basic process from Ohara E6 glass and Zerodur at the Steward Observatory Mirror Laboratory in Arizona and at Schott Glaswerke of Mainz, Germany, respectively.

Two especially large cast mirror substrates that have apertures of 8.41 m (26.54 ft), central holes of 0.889 m (2.92 ft) diameter, edge thicknesses of 0.894 m (2.93 ft), and weights of 16,000 kg (35,274 lb). These blanks are used in the Large Binocular Telescope (LBT) in the Mount Graham International Observatory in Southeastern Arizona. Figure 8.15 shows the first blank, while Fig. 8.16 shows a partial section view through the mold and furnace.^{12,13} The glass was slowly heated to and melted at 1180°C. Cooling and annealing took about 1 month thereafter. This particular blank had a small defect that was due to an inadvertent leak in the wall of the mold. This was repaired successfully by fusing additional glass onto the blank and reannealing it.

8.6.3 Built-up structural configurations

Figure 8.17 shows various construction configurations for machined and built-up lightweight mirrors.¹⁴ These include symmetrical and nonsymmetrical sandwiches, partly and fully open ("waffle") back designs, and foam-filled sandwich constructions. Each of these would have a characteristic areal density in pounds per square foot or kilograms per square meter depending on material type, material distribution, thickness of members (faceplate, backplate, and core webs), etc. In some designs, the core is integral with the front and back facesheets of the structure, while in others the parts are separate and partially attached together. The attachment means include thermal fusing, adhesive and frit bonding, and, for metal mirrors, brazing or welding. The pattern of cells in the core has a strong influence on the mirror's weight and stiffness. Triangular, square, circular, and hexagonal shapes are most commonly used in cells.

As noted earlier, a mirror lightweighted by removing nonessential material from within the substrate envelope is structurally more efficient than an equivalent-sized solid mirror. Since the material near the neutral plane^{*} contributes little to bending stiffness, it can safely be eliminated. This reduces weight so a desirably high stiffness-to-weight ratio can be provided. This results in some reduction of shear resistance. The manner in which the mirror is supported contributes strongly to the effects of gravity and other externally applied accelerations.

The neutral plane within a mirror divides the substrate so moments exerted by gravity on the front and back sections when the axis is horizontal are equal and opposite.

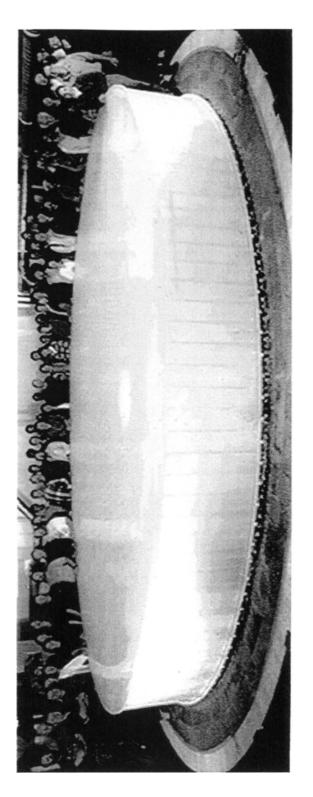


Figure 8.15 Photograph of the first 8.4 m (27.6 ft) diameter cast mirror substrate for the Large Binocular Telescope. (From Hill et al.¹²)

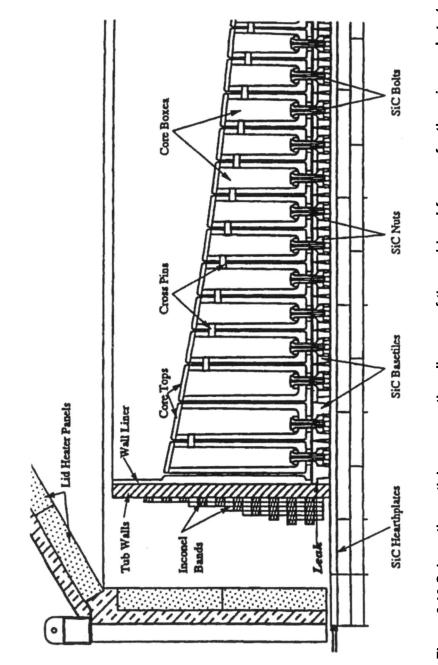






Figure 8.17 Cross section views for machined and built-up mirror substrates: (a) Symmetrical sandwich, (b) nonsymmetrical sandwich, (c) foam or fused-fiber core sandwich, (d) partially open back, and (e) open back. (From Seibert.¹⁴)

8.6.3.1 Egg crate construction

Figure 8.18 shows a classic type of built-up construction; it is called the "egg crate" configuration. The core is created as cellular "webs" made of thin slotted strips that interlock, but are not attached to each other. Front and back faceplates are fused to the top and bottom edges of the core to form the mirror substrate. The diameter-to-thickness ratio of such a mirror is typically about 7:1, so a 20-in. (50.8-cm) diameter mirror would be about 2.85-in. (7.239-cm) thick. Since not all parts of the core are connected, it is not as stiff as some of the more modern types, such as the fused monolithic structure.

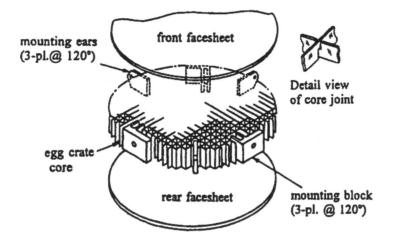


Figure 8.18 Construction details of a 32-in. (81-cm) diameter egg crate mirror prior to assembly. (Courtesy of Goodrich Corporation, Danbury, CT.)

An actual mirror with this type construction is shown in Fig. 8.19. It was used in the Orbiting Astronomical Laboratory Copernicus (OAO-C) launched in 1972 by NASA. It weighed approximately 105 lb (48 kg). The equivalent solid-disk mirror would have weighed 360 lb (164 kg).¹⁵

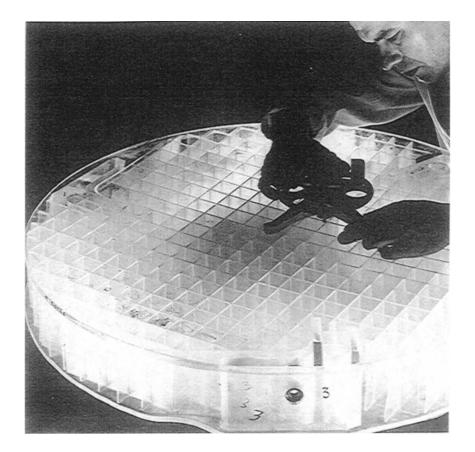


Figure 8.19 Photograph of the OAO-C lightweight primary mirror before coating. It was made in the egg crate configuration of Fig. 8.18. (Courtesy of Goodrich Corporation, Danbury, CT.)

8.6.3.2 Monolithic construction

In the 1960s, Corning Glass Works developed a technique for making monolithic mirror structures with improved shear resistance by fusing together many "ell"-shaped parts to create an egg crate core and then fusing on the front and back plates, in turn. Figure 8.20 shows schematically how two 90-deg joints of premachined parts (called "ells") are simultaneously torch-welded.¹⁶ In some designs, cylindrical rings are fused to the outer rim of the core to enclose it and to increase its stiffness. If the mirror is perforated, a ring may be fused into the central hole for the same reasons (see for example, Fig. 8.21).

When the entire core has been created, its top and bottom ends are usually ground flat and parallel. In either case, one facesheet is located on the core and heated in a furnace until they are fused together. It is then cooled slowly. The other facesheet is added in the same manner. If the assembly is then softened on a curved mandrel, the structure can be sagged to the meniscus shape usually desired for minimum glass removal in producing the optical surface. The substrate so created is monolithic and has the characteristics of the bulk material throughout.

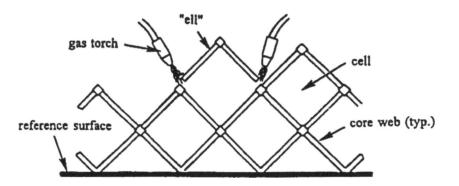


Figure 8.20 Corning's process for attaching 90-deg "ells" by torch welding to form a fused mirror core. (Adapted from Lewis.¹⁶)

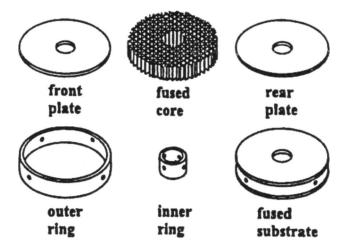


Figure 8.21 Basic parts of a typical perforated fused monolithic mirror substrate. (Adapted from Lewis.¹⁶)

During the fusing operation, the softened material usually distorts, resulting in shape defects, as indicated schematically in Fig. 8.22(a). The assembled mirror blank is carefully inspected to find the internal region nearest the front surface of the front facesheet with a minimum number of defects (bubbles and inclusions of impurities). Manufacture then proceeds into grinding to remove excess material as indicated in Fig. 8.22(b) and locate the surface to be polished within the so-called critical zone. The back surface of the mirror is also ground to produce a reasonably smooth contour.

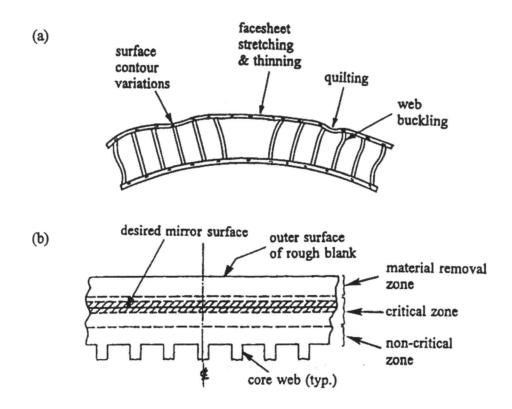


Figure 8.22 Details of the monolithic fused mirror blank. (a) Typical defects caused by heating the glass above the softening point in order to fuse the parts together. (b) Location of the mirror surface within the "best" region of the front sheet.

A conceptual sketch of a typical monolithic substrate is shown in Fig. 8.23. This is a 1.52-m (60-in.) diameter meniscus blank intended for use as the primary mirror of a telescope. The mounting for this mirror is described in Chapter 9. The detail view in the figure shows a mounting block fused into the rim of the structure. Three such blocks provide "strong points" for attaching the mirror with flexures to the mirror cell.

The fused monolithic construction is feasible only with materials of essentially zero CTE because they do not fracture from temperature gradient induced stresses when rapidly heated or cooled during the fusion process. Corning's ULE ceramic glass composed of about 92.5% SiO₂ and 7.5% TiO₂ is very well suited for this type of construction. Its CTE is predictably near zero in the temperature range 5 to 35°C. Furthermore, its actual CTE can be measured precisely by nondestructive testing using ultrasonic velocity measurement techniques.¹⁷ The characteristics of this material are listed in Table B8(a). Hobbs et al.¹⁸ reported that fused silica also can be fusion welded, but that the process requires higher temperatures so is more difficult.

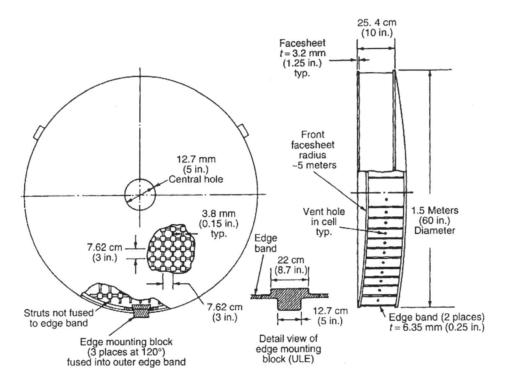


Figure 8.23 Conceptual layout of a typical 1.52-m (60-in.) diameter fused and slumped monolithic ULE mirror blank.

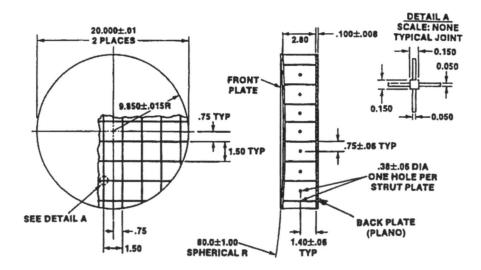


Figure 8.24 A mirror configuration suitable for assembly by frit bonding. (Adapted from Fitzsimmons and Crowe.¹⁹)

8.6.3.3 Frit-bonded construction

Making a core and attaching it to the facesheets using an assembly process similar to brazing can create another type of built-up mirror blank. All parts are attached with "frit." This is an "adhesive" made of an organic vehicle and powdered glass. Its CTE is controlled to not introduce excessive stress into the blank during or subsequent to application. The resulting blank is free of the defects shown in Fig. 8.22(a) because the mirror's previously annealed structural elements never reach their softening temperature when the frit melts. Figure 8.24 shows a typical configuration. Mirror blanks made by this process can have thinner webs and a higher diameter-to-thickness ratio than monolithic fused blanks. Tighter dimensional tolerances can be held in a frit-bonded substrate because the structural members do not distort. The frit-bonded mirror will weigh less and be more rigid than the monolithic construction. See the comparison in Table 8.1.

Characteristic	Fusion bonded	Frit bonded
Minimum core density	10%	3%
Mean bond strength	2500 lb/in. ² (17.2 MPa)	5000 lb/in. ² (34.5 MPa)
Mounting blocks	Fused in	Fused or frit bonded in
Maximum cell size	4 in. (10.2 cm)	(6 in. (15.2 cm)
Minimum rib thickness	0.150 in. (3.81 mm)	(0.050 in. (1.27 mm)
Average plate thickness for a given mirror diameter D:		
D < 30 in.	0.160 in. (4.06 mm)	0.10 in. (2.54 mm)
30 in. < D < 90 in.	0.38 in. (9.65 mm)	0.30 in. (7.62 mm)
D > 90 in.	0.60 in. (15.24 mm)	0.40 in. (10.16 mm)

Table 8.1 Design characteristics of fused and frit-bonded mirrors.

8.6.3.4 Hextek construction

Another technique involving fusion of separate parts to make a lightweight mirror substrate is used by the Hextek Corporation of Tucson, AZ.²⁰⁻²² Here, similar lengths of circular cross section glass tubing are placed on end between glass facesheets to form a sandwich configuration as shown in Fig. 8.25. The rear sheet is perforated with small holes, one for each tube. The assembly is sealed over a manifold in an oven. The top sheet is weighted down and air or another gas is admitted to the tubes under sufficient pressure to just balance the weighted top sheet. The oven temperature is raised until the tubes are fully fused to both sheets. Then the pressure is increased to press the softened tubes outward until they contact and fuse to the adjacent tubes. The assembly becomes monolithic with either a square [see Fig. 8.25(a)] or hexagonal [see Fig. 8.25(b)] core cell pattern. An integral sidewall also is formed on the core as the tubes expand. After annealing and cooling, finished substrates appear generally as shown in Fig. 8.26. The larger blank is 1 m (40 in.) in diameter by 17-cm (6.7-in.) thick and has been slumped to a f/0.5 sphere in a second firing over a convex refractory mold.

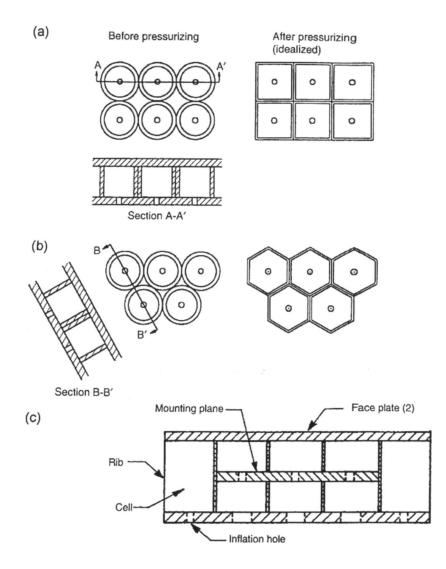


Figure 8.25 Tube placement patterns and resulting core cell configurations for the Hextek process to fabricate lightweight mirror substrates. (Courtesy of Hextek Corporation, Tucson, AZ)

This process is unique in that the pneumatic support of the structure during fusing permits use of substantially higher temperatures than would be possible without pressurization. Superior bonding of the tubes results without excessive sagging (quilting) of the top facesheet between the ribs formed by the tube walls. Typical cell size in a Hextek blank is 6.4 cm (2.5 in.) and facesheet thickness is on the order of 1 cm (0.4 in.).²¹ Use of thin wall tubing, wide separations, and very thin facesheets are all possible and result in ultra-light construction with reasonably uniform rib geometry. For example, the smaller blank shown in Fig. 8.26 is 0.45 m (17.7 in.) in diameter and 10-cm (3.9-in.) thick. Its areal density is 31.8 kg/m^2 .

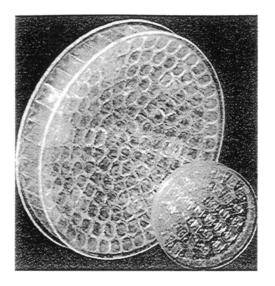


Figure 8.26 photographs of two fused monolithic mirror substrates made by the Hextek process. (Courtesy of Hextek Corporation Tucson, AZ.)

The Hextek process is reported to be rapid and relatively inexpensive, to make efficient use of materials, and to produce 100% fusion bonds between components. Borosilicate glass such as Corning Pyrex 7740, Schott Tempax (with a CTE of 3.2 ppm/°C), and Schott Borofloat glass is generally used, although Vycor and fused silica have also been employed. Acid etching of cut glass parts to remove surface impurities introduced during raw material production was found to be desirable for appearance reasons, but not essential to the technical quality of the blank.²¹ Blanks with central perforations and internal mounting bosses at the neutral surface of the blank [see Fig. 8.25(c)] can be made and either concave or convex facesheet contours can be created.

Voedodsky et al.²² indicated that this technology is capable of creating flat or curved (convex or concave) substrates as fast as f/0.5 and apertures as large as 2.0 m (78.7 in.). Further, it was reported by those authors to be capable of achieving areal densities as low as 15 kg/m².

8.6.3.5 Machined core construction

A fused silica lightweight mirror substrate with a core machined from a solid disk is illustrated in Fig. 8.27. This mirror has a symmetrical concave shape. Its core was machined from a solid blank by boring through holes of various shapes and grinding a concave surface on both sides. It was fused to preformed meniscus shaped facesheets. The 20-in. (50.8-cm) diameter mirror weighed about 16 lb (7.3 kg).²³ Figure 8.28 shows the pattern of holes produced by drilling and grinding with annular diamond bonded core and end mill tools. Cusps remaining after hole drilling were removed with grinding tools. The wall thicknesses after machining were 1 to 3 mm. After fusing, the substrate was monolithic. Mirrors of this shape are best used with their axes horizontal because significant self-weight deflections occur when the axes are vertical.

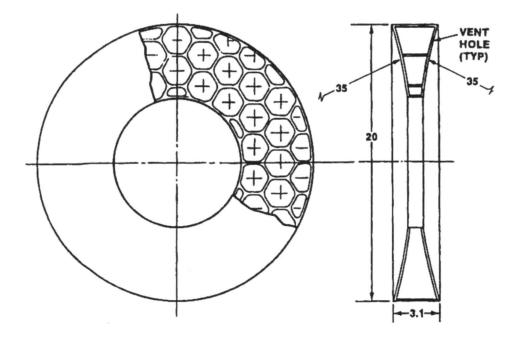


Figure 8.27 Configuration of a symmetrical concave mirror with core machined from a solid blank. The front and rear plates were preshaped (sagged over a convex mold) to meniscus form and fused to the core. Dimensions are in inches. (From Pepi and Wollensak.²³)

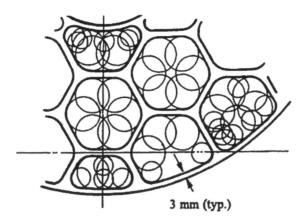


Figure 8.28 Typical core machining hole pattern for a mirror such as that shown in Fig. 8.27. (From Pepi and Wollensak.²³)

A more recent technique for machining a lightweight core from a solid blank uses an abrasive water jet (AWJ) cutting process developed by Corning Glass Works. Edwards²⁴ described the technique and apparatus used (see Fig. 8.29) as follows:

"The system is powered by two 250 horsepower motors driving hydraulic pumps, which in turn feed intensifiers that output over 60,000 psi water pressure. Passing through a 0.040- in. (1.02-mm) diameter sapphire jewel orifice, the water jet creates a vacuum, pulling abrasive garnet into the stream. After fully entraining the abrasive in a mixing tube, the water exits the nozzle at over Mach 2, capable of cutting through 30-cm thick glass. A tool check station provides alignment and process parameter calibration. The five axis head can compensate for slight changes in the shape of the cut due to wear of the jewel and mixing tube during jet on time as well as any slight part movement. The system manipulator positions the nozzle in the 150 in. × 250 in. × 48 in. workspace to an accuracy of better than ± 0.0025 in., repeatable to within ± 0.001 in. Each day the operator makes test cuts, which are measured to determine the health and contour of the jet. Ongoing verification throughout the water jet process ensures tight dimensional tolerances are maintained in the resultant lightweight cores."

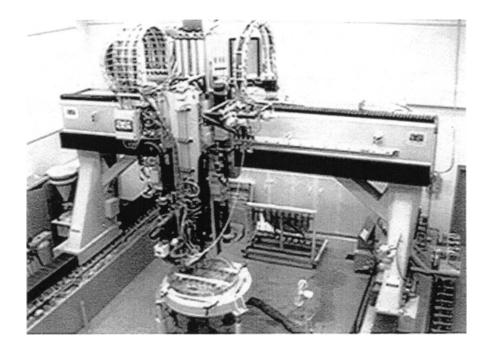


Figure 8.29 Photograph of the Corning abrasive water-jet machine used to make lightweight cores for mirror substrates. (From Edwards.²⁴)

Figure 8.30 shows a typical product made by the Corning AWJ process. This is a 1.1-m (43.3-in.) diameter core that was later incorporated into one of the tertiary mirrors for the 6.5-m (255.9-in.) aperture Magellan Telescopes. This particular design incorporated thicker webs at the outer edge and central mounting hub, a hexagonal pattern superimposed on an elliptically shaped blank, and a circular central through-hole.²⁴ The machine shown in Fig. 8.29 is capable of handling substrates up to about 3 m (118 in.) in diameter.

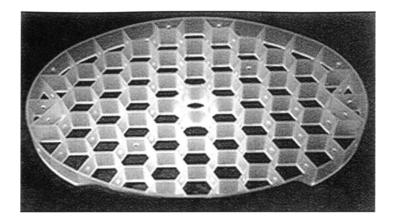


Figure 8.30 Example of a mirror core lightweighted by the Corning AWJ process. (From Edwards.²⁴)

8.6.3.6 Foam core construction

Conventional lightweight mirrors as discussed above face fundamental limitations on minimizing web thicknesses because of the resulting flexibility and susceptibility to distortion at high temperatures typically used for fusing these structural members to the other components of the mirror. To maximize weight reduction, the webs also must be separated by large distances relative to their thickness, thereby allowing the mirror facesheet to sag between the webs under gravitational load or under polishing forces (creating surface distortions called "print through" or "quilting"). Mirrors with foam cores can significantly reduce these problems because the facesheets are uniformly supported on a micrometer level rather than on a millimeter level. If the core is shaped to near net shape prior to adding the facesheets, some problems of thermal distortion during manufacture are eliminated. Weight reduction results primarily from the large percentage of open space (typically >90%) within that structure. Goodman and Jacoby²⁵ compared these and some other characteristics of conventional web and foam cores for mirrors.

Although fused silica foam was a fairly widely known material in the U.S. optical industry during the 1960s, attempts to build lightweight mirrors using this material as a structural core between facesheets were unsuccessful because the core was hard to shape, fuse to the sheets, and attach to a mount.^{26,27} The use of cellular metals such as aluminum for this purpose was investigated during the early 1980s with somewhat greater success.²⁸ The use of metallic foams as cores for mirrors is discussed in Section 8.9. We concentrate here on other materials—in particular, silicon.

In 1999, Fortini described research into the use of open-cell silicon foam as a core material for ultra-light mirrors with single crystal silicon facesheets.²⁹ See Fig. 8.31(a). At room temperature, silicon typically has a density of 2.3 g/cm³, a CTE of $2.6 \times 10^{-6}/K$, and a thermal conductivity (k) of 150 W/m-K. Figures 8.32(a) and (b) show the variations with temperature of CTE and k for silicon as compared with the same parameters for Be. (The data plotted in view (a) are from Paquin.³²)For cryogenic applications, the very low CTE and high k of Si would be favorable for mirror applications. The single crystal reflecting surface of a silicon mirror can be polished to an optical figure of typically $< \lambda/10$ p-v at $\lambda = 0.633$ µm wavelength and microroughness <5 Å rms.

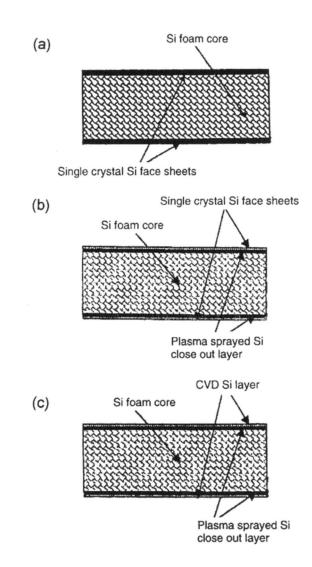


Figure 8.31 Schematic sectional views of mirrors with Si facesheets and foam cores; (a) first design (From Fortini.²⁹), (b) design with plasma sprayed layers (From Jacoby et al^{30}), (c) design with CVD Si layers (From Jacoby et al.³¹)

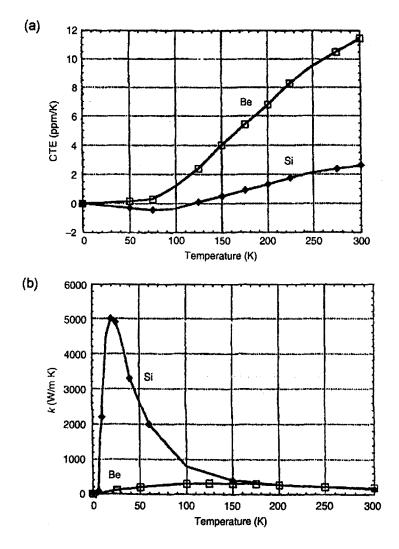


Figure 8.32 Comparisons of variations with temperature for (a) thermal expansion coefficients and (b) thermal conductivities of Be and Si. (From Fortini.²⁹)

Figure 8.33 shows a scanning electron micrograph of the typical open-cell microstructure of Si foam. Typically, it has cells of 65 pores per inch (ppi). Fortini²⁹ indicated that mirror substrates with diameters of 9.45 cm (3.72 in.) made with 5% dense foam combined with two 0.889- mm (0.035-in.) thick facesheets would produce a mirror with areal density of 15 kg/m². The resulting substrate would have stiffness equivalent to a 3.81 cm (1.50 in.) thick single crystal silicon monolith (diameter/thickness = 2.48). The latter substrate would, however, have an areal density six times greater than the foam structure. Bonding experiments reported by Fortini²⁹ indicated no significant effects on optical figure due to cycling between room and liquid nitrogen temperatures with facesheet-to-core bonds. Similarly, facesheet edge bonds did not seem to affect the figure under the same temperature changes. These results encouraged development of mirrors with significantly larger diameters.

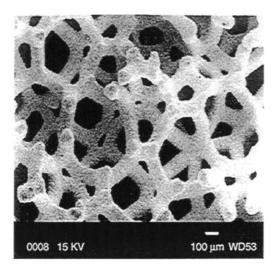


Figure 8.33 Scanning electron micrograph of a typical open cell Si foam structure (From Fortini.²⁹)

Jacoby et al.³³ reported experiments with silicon foam core mirrors at temperatures as low as -183° C. These mirrors had cores with 65 ppi cells and the architecture illustrated by Fig. 8.31(b). They differed slightly from the construction shown in Fig. 8.31(a) in that the cores were contoured to near-net-shape by controlled crushing between precision mandrels prior to silicon infiltration. Also, the faces of the silicon cores were plasma sprayed with layers of polycrystalline silicon typically 0.025- to 0.030-in. (0.635- to 0.762-mm) thick to close out the open structure locally. After annealing, they were polished flat before bonding to the facesheets. The polishing process smoothed the plasma sprayed surfaces to ensure a good bond.

A further development of mirrors with silicon foam cores that have the architecture shown in Fig. 8.31(c) was reported by Jacoby et al.³¹ The process comprises the following steps: (1) bring open-cell or reticulated vitreous carbon (RVC) foam to near-net shape by CNC machining, (2) plasma spray with polycrystalline silicon to build a layer 0.025- to 0.030-in (0.635- to 0.762-mm) thick through interparticle bonding and a sintering reaction, (3) lap surface to flatness and test interferometrically, (4) apply layer of highly densified polycrystalline silicon by a CVD process to form facesheets on the core totaling 1.0-mm (0.039-in.) thick (sprayed layer plus CVD layer), (5) superpolish one face to \leq 3.0-nm rms and figure quality <70-nm p-v, and (6) apply coating as appropriate. Inspection naturally takes place upon completion of each step. The authors indicated that this manufacturing process avoids the significant cost and technical problems of producing large single crystal facesheets as well as potential problems with bonding subdiameter crystals to form large facesheets.

Analyses based on both classical and FEA techniques²⁵ predicted that mirrors with diameters of at least 0.5 m (19.7 in.) and areal densities of 7.0 kg/m² could be made in the manner just described. Further, these mirrors should have high fundamental mode frequencies of about 447 Hz. Si foam core mirrors are especially suited for mounting in carbon-fiber-reinforced-silicon carbide structures.²⁵ Developments of silicon foam core

mirrors for specific space-application experiments were described by Jacoby et al.³³ and by Goodman et al.³⁴ Cryogenic (177K) and vacuum (10^{-5} torr) test results of a 6.0-in. (152.4-mm) diameter mirror were reported in those references. Optical figure stability while cycling from 300K to 177K to 300K was reported to be excellent.

Jacoby and Goodman³⁶ summarized measured properties of silicon and silicon carbide foam materials of differing densities (typically in the range of 8% to \sim 30%) relative to their solid counterparts. Young's modulus, Poisson's ratio, compressive strength, and tensile strength were shown to vary nearly linearly with density. Variations with temperature over the range of 25 to 300K of fundamental frequency (and hence, Young's modulus) and of damping effect were found to be small. The CTEs of bulk silicon and of silicon foam were found to be nearly equal over the temperature range of 120 to 280°C.

A lightweight 55-cm (21.6-in.) diameter, f/1 parabolic mirror subassembly of double-arch configuration made by 2005 silicon foam technology³⁷ for use in a high-energy laser application is described in Section 15.21.

8.6.3.7 Internally machined mirror construction

A fundamental method of lightweighting mirrors is to machine recesses into the back of a solid disk. A mirror made by this approach resembles that of the Hale Telescope primary or spin cast mirrors, which have cavities obtained by casting the glass around strategically placed cores. Similar recesses can be produced by sandblasting, by abrasive water-jet machining, or ground by a CNC milling machine using diamond-impregnated tools. Generally, mirrors with this type of construction have lower rigidity than designs with a back faceplate.

Stiffer mirrors are obtained if the cavities are created by a milling tool entering a solid plate through small blind access holes drilled into the back surface. The access holes have only a small effect on the blank stiffness. This technique is not new. A mirror design described by Simmons³⁸ illustrated by Figs. 8.34 and 8.35 had triangular internal cavities obtained by undercutting with a grinding tool through a series of blind access holes 2.5 in. (6.4 cm) in diameter drilled into the back of a 64-in. (1.62-m) OD by 12-in. (30.5-cm) thick solid disk. The ribs between cavities were 0.20-in. (5.1-mm) thick. All fillets at the intersection of ribs with one another and with the front and back plates had 0.75-in. (19-mm) radii. This left a large post of material at the intersection of each set of six triangles. Weight was reduced at some expense in stiffness by removing material from the centers of these posts by machining a 1.5-in. (3.8-cm) diameter cylindrical cavity into each of them. The center-to-center distance of these holes was 7.30 in. (18.5 cm). The height of each equilateral triangle was 5.25 in. (13.3 cm).

This blank contained 138 large, triangular-shaped cavities and 55 small, cylindrical cavities. When completed, it weighed ~1035 lb (~470 kg). If solid, it would weigh ~3475 lb (~1580 kg); this represented a weight reduction of 70%. Dimensional control of the internal surfaces created by this technique was comparable to that achieved with normal metalworking. After removal of the desired mass of material from the cavities, the mirror was machined to its final external dimensions. It was then acid-etched to remove surface imperfections and local stresses in the surfaces.

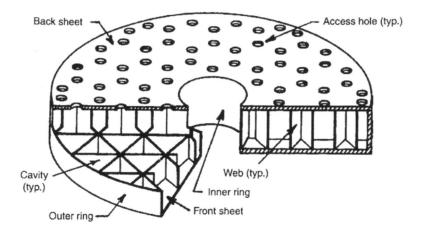


Figure 8.34 Cutaway conceptual sketch of a lightweighted mirror substrate containing triangular internal cavities (cells) undercut through multiple access holes through the back surface. (From Simmons.³⁸)

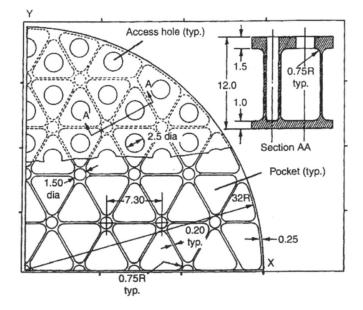


Figure 8.35 Configuration of a 64-in. (1.62-m) diameter Cer-Vit mirror machined in the manner of Fig. 8.34. Dimensions are inches. (Adapted from Simmons.³⁸)

A contemporary example of this type construction is shown in Fig. 8.36.^{39,40} This is a diagram of the back side of the 2.7-m (106-in.) diameter primary mirror for the Stratospheric Observatory for Infrared Astronomy (SOFIA) telescope that replaces NASA's Kuiper Airborne Observatory (KAO) telescope. The SOFIA mirror design is a plano concave structure with a heavily beveled rim featuring "flying buttress" lateral supports and a circular central hole. Hexagonal cells (essentially blind holes) with thin webs between them were machined into the plano concave Zerodur blank using diamond tools to undercut material and leave a nearly complete back plate.

Pertinent dimensions of the lightweighted structure are as follows: OD 270.5 cm, ID 42.0 cm, hexagonal pocket size 18.5 cm, front plate thickness 1.5 cm, total thickness 35.0 cm, back plate OD 230.0 cm, typical web thickness 0.7 cm, and back plate mean thickness 2.5 cm. The initial weight of the solid substrate was 3400 kg, while the weight of the finished mirror is 850 kg. This represents a reduction to 25% of starting weight.³⁹

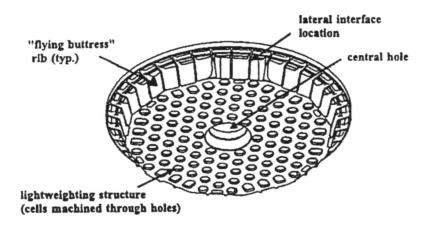


Figure 8.36 The back side of the SOFIA primary mirror that is lightweighted by machining pockets through holes in the back. (Adapted from Erdman et al.³⁹)

After machining, the substrate was acid etched to further reduce weight and to remove microscopic cracks created during the grinding operation. The supports for this mirror are described in Section 11.3.2.

8.7 Thin Facesheet Configurations

If the substrate thickness for a mirror is reduced drastically with respect to its diameter, inherent rigidity is impossible. Success in supporting such a substrate during manufacture, establishing and maintaining high-quality performance (i.e., optical figure) during use, and protecting it from damage caused by extreme vibration and shock depends on the ability of a companion structure to support the mirror adequately at all times. During use, this structure plays a strong role in controlling mirror shape. The optical component is then only one part of a much more complex system that can sense errors in the mirror's shape and apply the appropriate forces to the mirror's facesheet to correct those errors. This "adaptive optics" technology can improve the performance of increasingly large ground-based

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astronomical telescopes that cannot be built with the technology discussed above because weight and cost would increase beyond reasonable limits. Another motivation for using thin adaptive mirrors is their now proven ability to compensate for atmospheric "seeing" effects. Further, this technology allows quite large optical systems to be carried into space and operated from that vantage point.

Conceptually, the adaptive optic of a large reflecting telescope can be the primary, the secondary, or some smaller aperture optic located at a downstream image of the system aperture stop, i.e., at a pupil of the system. The aperture stop of most astronomical telescopes is at the primary mirror. The design of the Large Binocular Telescope (LBT) places the aperture stop at the secondary mirror. Consequently, that mirror is used to correct the wavefront. We here summarize some key aspects of the LBT adaptive secondary design.

The LBT secondary mirrors are 91.1-cm (35.9-in.) diameter Zerodur meniscus shells 1.52-mm (0.060-in.) thick. Manufacture started by grinding, polishing, and figuring the concave aspheric (ellipsoidal) optical surface on a 150-mm (5.9-in.) thick meniscus shell. The optical surface of the shell was then attached with pitch to a stiff blocking body of matching convex radius. The thickness of the shell was reduced by grinding and polished to the finished thickness. The shell was edged to OD, a center hole was cored out, and the rim was beveled and polished. The shell was then removed from the blocking body; the convex side was masked to protect locations for later attachment of magnets and aluminized. One mirror for each telescope plus a spare were manufactured.⁴¹

The design of the LBT adaptive secondary is based largely on a prior design created for the new MMT telescope, which has been proven successful.⁴² The basic concept for both systems is illustrated schematically in Fig. 8.37(a). Item 1 is a flange attached to the secondary support structure (at top) through a six-DOF hexapod mechanism. Item 2 represents three electronics/computer boxes. Item 3 is a thick meniscus aluminum support/cold plate. Item 4 is a set of 672 cold finger/actuators, one of which is shown in view (b). Each cold finger is attached to Item 3. Item 5 is a thick (50-mm) meniscus Zerodur reference plate. It is perforated to allow all the cold fingers to almost reach Item 6, which is the deformable shell. Between the flange (1) and the reference plate (3) are a set of passive force actuators, or astatic levers, that maintain the true shape of the reference surface within 100-nm rms during system operation.

The shell has a central hole to which a thin central membrane is attached to provide lateral and in-plane rotational constraint. This membrane is anchored to the telescope structure. An experimental prototype of this feature of the shell mounting is shown in Fig. 8.38. When the shell is not actively serving as the secondary mirror of the system, it is constrained axially by a set of mechanical stops located at the shell's inner and outer edges. During operation, the thin shell is suspended by reaction forces created between wire coils at the ends of the cold fingers and permanent magnets bonded to the back surface of the shell. The nominal spacing between the shell's inner (convex spherical) surface and the outer (concave spherical) surface of the reaction body is 50 μ m (0.002 in.). During operation, the separation between these surfaces is monitored in real time by 672 capacitive sensors capable of resolving 2- to 3-nm changes in location of the aluminum coating on the back of the shell relative to the reaction body. The sensing and control system operates at a bandwidth of at least 1 kHz in response to error correction commands from a sensor measuring errors in the telescope optical system's reflected

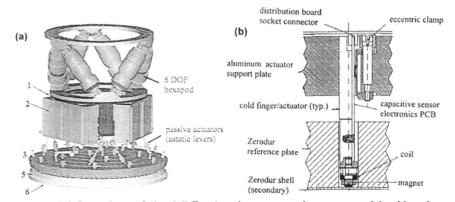


Figure 8.37 (a) Drawing of the LBT adaptive secondary assembly. Numbered items are identified in the text. (b) Schematic of one cold finger/actuator passing through the reference plate to near the back of the thin shell secondary mirror. (Adapted from Riccardi et al.⁴³)

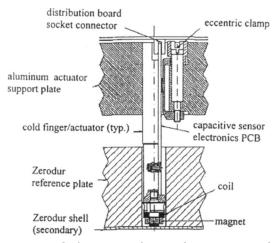


Figure 8.38 Schematic of the central membrane mounting for the LBT adaptive thin shell secondary. Dimensions are in millimeters. (Adapted from Riccardi et al.⁴³)

wavefront.⁴⁴⁻⁴⁶ The actuators have sufficient dynamic range ($\sim 0.1 \text{ mm}$) for the secondary to compensate low-order tilts caused by atmospheric and wind effects as well as to provide chopping capability.

8.8 Metallic Mirrors

The metals commonly used to make mirrors are listed with their key mechanical properties in Table B8b. The most common are wrought aluminum and beryllium, the latter being most popular in cryogenic space applications. Mirrors for use in high-energy laser applications or with high-power light sources need to be cooled. This frequently is done by circulating coolant through tubular passages machined into the mirror substrate. These are usually made of oxygen-free high-conductivity (OFHC) copper or TZM, an alloy of titanium, zirconium, and molybdenum.

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Fabrication of metallic mirrors typically involves most of the following steps: formation of the blank, geometric shaping, stress relieving, plating (usually with electroless nickel), optical finishing, and optical coating. Many materials can be cast; others are welded or brazed from components. Single-point diamond turning has found great application in creating fine quality optical surfaces in metals such as aluminum, brass, copper, gold, silver, electroless nickel (coating), and beryllium copper. The purity of the material is very important.⁴⁵ The surface finish of metal surfaces is inferior to glass materials, but is adequate for infrared systems and some visible-light applications. Figure 8.39 shows a variety of small metal mirrors made by one manufacturer using SPDT methods.

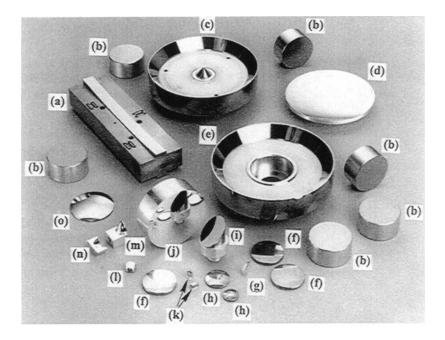


Figure 8.39 Photograph of optical components made by SPDT. (1) Al telescope mirror, (2) Cu Axicon mirror, (3) ZnSe diffractive aspheric lens, (4) Cu parabolic mirror, (5) Cu phase retarder, (6) Cu mirror, (7) 45-deg pressurecontrolled variable-radius mirror, (8) Cu waxicon, (9) water-cooled Cu mirror, (10) ZnSe lens with spiral phase steps, (11) Cu mirror polarization-sensitivecoated to reflect s- and absorb p-polarization, (12) Al parabolic mirror, (13) Cu rooftop beamsplitter, (14) Al off-axis parabolic mirror, (15) Al mirror, (16) replica parabolic mirror, (17) Ge asperic lens, (18) multispectral-ZnS negative aspheric lens, (19) multispectral-ZnS aspheric meniscus lens, (20) four Cu "button" mirrors, (21) two ZnSe transmissive-beam integrators, (22) Cu reflective-beam integrator, (23) Cu toroidal reflector, and (24) water-cooled Cu top-hat-shaped mirror. (Courtesy of II-VI, Inc., Saxonburg, PA.)

Figure 8.40 shows the back side of a typical metal mirror. It is the 7.3-in. (18.5-cm) diameter by 0.7-in. (1.78-cm) thick secondary mirror used in the infrared telescope of NASA's Kuiper Airborne Observatory.⁴⁶ Lightness of weight and low inertia were essential to the success of this equipment since the mirror moved mechanically in oscillatory tipping fashion to rapidly switch the field of view of the telescope from the target of interest to the sky background for calibration purposes.

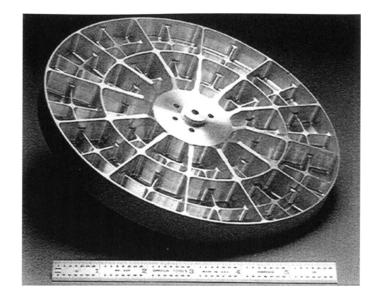


Figure 8.40 Photograph of the lightweighted aluminum scanning secondary mirror used in the Kuiper Airborne Observatory. (From Downey et al.⁴⁶)

The 7:1 diameter-to-thickness Type 5083-O aluminum substrate was lightened by machining open pockets into a solid blank. The total final weight was 1.1 lb (0.5 kg), representing a 70% reduction from a solid. The convex hyperboloidal optical surface (which was not electroless nickel plated) was created by SPDT machining to the final figure. The quality of the surface was about 0.67 λ p-v at 633-nm wavelength over 90% of the aperture. The final surface was coated with aluminum and silicon monoxide films. The mounting surfaces at the center of the mirror were diamond turned to facilitate accurate machining of the optical surface when the blank was reversed later. The surface figure achieved at the -40°C operating temperature was $\lambda/2$ at a 633-nm wavelength.

The mirror is shown mounted on its drive mechanism in Fig. 8.41. The square-wave response of the mirror and its drive mechanism for beam tilt angles up to \pm 23 arcmin was about 40 Hz. It was driven in orthogonal tilts by four electromagnetic actuators located symmetrically at the back of the mirror. The moving assembly [weight about 2 lb (4.4 kg)] tilted about its center of gravity on two axis flex pivot gimbals. The actuator coils were mounted to a stationary base plate that provided a conductive path for temperature control. The entire assembly could be moved axially by a motor-driven ball screw through a range of \pm 1.3 cm (0.51 in.) for focus adjustment during flight.

Many lightweight beryllium mirrors have been fabricated by techniques similar to those just discussed. Usually these were used in space applications, although some have been used in high-speed scanning applications where high stiffness and minimal weight are required to prevent surface distortion by centrifugal force. For wavelengths beyond ~3 μ m in the infrared, polished bare beryllium has high reflectance so it is not necessary to apply electroless nickel plating. This avoids thermal problems caused by bimetallic effects from a CTE mismatch.⁴⁷

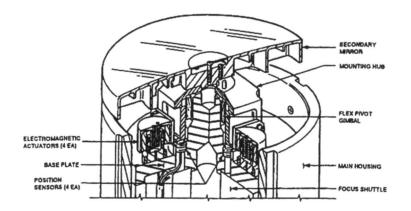


Figure 8.41 Mirror from Fig. 8.40 mounted on its drive mechanism. (From Downey et al.⁴⁶)

One very successful means for making beryllium mirrors is a powder metallurgy technique patented by Gould⁴⁸ and described by Paquin et al.⁴⁹ and Paquin.⁵⁰ In this process, high-purity beryllium powder is constrained within precision manufactured metal (such as low-carbon steel) containers of the desired dimensions and shape, outgassed at >670°C, sealed, and autoclaved at a pressure of ~103 MPa (~15,000 lb/in.²) and temperature of 850 to 1000°C. The latter process is called hot isostatic pressing (HIP). When returned to ambient temperature and pressure, the container is opened. This yields Be mirror blanks of near net shape with low porosity and few inclusions. Gould's process improvement included forming internal lightweighting pockets in the mirror blank by compressing the material around void formers made of leachable material (monel or copper) that could be removed after compacting. Figure 8.42 shows two such mirrors. They were 9.5-in. (24.1-cm) diameter by 1.2-in. (2.8-cm) monolithic closed sandwiches weighing 2.16 lb (0.98 kg). Hexagonal cells measuring 1 in. (2.5 cm) and webs 0.05-in. (1.3-mm) thick were formed in these mirrors. In the foreground of the figure, the back face of one mirror shows access holes for supporting the void formers and removing them later. After polishing, the front facesheets typically had figures of $\lambda/25$ p-v at a 633-nm wavelength. These experimental mirrors were extremely stiff, with the first resonance at about 8700 Hz. The manufacturing process proved to be scalable to larger mirror sizes and is the basis for manufacture of many of today's Be mirrors.

Geyl and Cayrel⁵¹ reported that the blanks for each of the four secondary mirrors for ESO's Very Large Telescope (VLT) were manufactured by the HIP process from I-220-H Be powder as plano-convex solids. They were rough machined into an open-back lightweight form with triangular cells having 70-mm (2.76-in.) inscribed diameters, ribs of 3-mm (0.12-in.) thicknesses, and front facesheet thickness of 7 mm (0.28 in.). The mirror specifications called for overall diameters of 1.12 m (44.09 in.), center thicknesses of 130 mm (5.12 in.), radii of curvature of 4553.57 \pm 10 mm 179.274 \pm 0.394 in.), weights ~42 kg (92.5 lb) with ELN coating, and hyperbolic figures. The machined blanks were heat-treated and acid-etched to remove surface stresses at appropriate times during fabrication, fine ground, ELN plated, and polished. A typical mirror exhibited a wavefront error of 349-nm rms, 1770-nm p-v, 0.22-arcsec surface slope error, and \leq 15-Å microroughness.

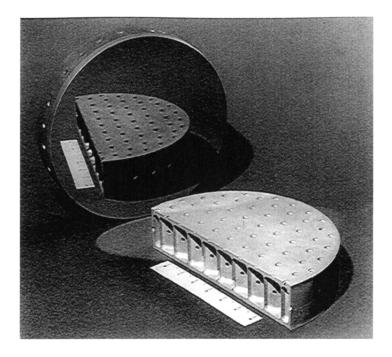


Figure 8.42 Photograph of 9.5-in. (24.1-cm) diameter monolithic, closed-sandwich beryllium mirrors made by the HIP process. (From Paquin.⁵⁰)

Figure 8.43 from Cayrel⁵² shows the back side of one mirror with its titanium support frame. A bayonet type of interface is provided at the mirror's center for temporary alignment, calibration, and observation devices. Six mount interfaces are machined into the mirror's core. Bipods at three of these interfaces are used to support the mirror at its neutral surface. The remaining three mount interfaces are for safety devices that prevent the mirror from falling in case of mount breakage.

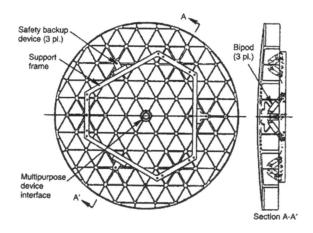
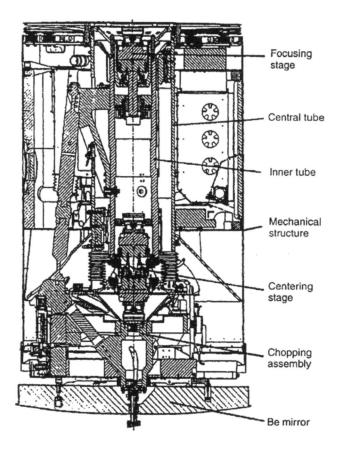
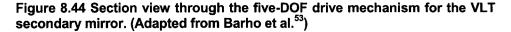


Figure 8.43 Schematic of the back side of the beryllium secondary mirror for VLT showing its support frame. (Adapted from Cayrel.⁵²)

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The mirror support frame is attached to a multipurpose drive unit as shown in Fig. 8.44 from Barho et al.⁵³ This unit provides a five-degree-of-freedom adjustment including focus along the telescope axis, centering during observation to compensate for varying gravity influences, tilt to stabilize the field of view, and chopping (oscillatory) motion to calibrate the system against the background sky. Details regarding the drive unit are given by Stanghellini, et al.⁵⁴





The primary for the James Webb Space Telescope (JWST) comprises 18 hexagonal beryllium segments measuring 1.32 m (48.17 in.) flat-to-flat. Together, they form a 25 m² contiguous optical surface measuring 6.6 m (259.842 in.) across (flat-to-flat). See Fig. 8.45. The beryllium is Brush Wellman O-30 Grade material. Each segment is lightweighted by precision machining 600 triangular pockets into the back surface. See Fig. 8.46. The alphanumeric designations indicate that the array is made up of six segments polished to each of three different aspheric contours. This is necessary because the three groups lie at differing distances from the optical axis.

The segments are being polished and figured on a series of proprietary computer controlled optical surfacing (grinding and polishing) machines at Tinsley Laboratories, Richmond, CA. Particular care is being taken to ensure personnel safety.⁵⁵ Metrology of the surfaces is being accomplished in several stages; first with a coordinate measuring machine, then with a Scanning Shack-Hartman System, and finally with visible light interferometry. Figuring and testing are both done under strictly controlled temperature ($20 \pm 2^{\circ}$ C) with horizontal laminar air flow through HEPA filters.

Specifications for the segments call for vertex radius of 15,899.915 \pm 1 mm (625.981 in.) with segment-to-segment matching within \pm 0.100 mm (0.004 in.), conic constant -0.99666 ± 0.0005 , surface figure error ≤ 20 nm rms (≥ 222 mm/cycle, and clear aperture to ≤ 5 mm (0.197 in.) of rim. Maximum rms surface figure errors are: 20 nm for mid frequencies (period ≥ 222 mm), 7 nm for high frequencies (222 mm > period > 0.080 mm), and 4 nm surface roughness (period $\leq 80 \ \mu$ m). Upon arrival from the machining supplier, the lightweighted mirror blanks are within 0.101 mm (0.004 in.) p-v of the true off-axis aspheric profile. Throughout the life of the substrates, shock loads from handling, transporting, and processing are limited to $a_G \leq 5$ so as not to exceed the microyield stress level for the beryllium. This will prevent creep or plastic deformation over time that could affect the mirror's long-term stability.⁵⁶ Special handling equipment has been designed, built, and qualified to meet this requirement.

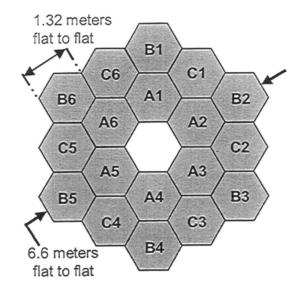


Figure 8.45 Array of hexagonal segments in the JWST primary mirror. (From Wells et al.⁵⁵)

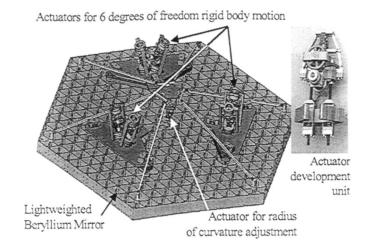


Figure 8.46 Schematic view of the back side of one segment from the JWST primary. The detail view shows an early version of one actuator. (From Wells et al.⁵⁵)

As may be noted from Fig. 8.46, three bipod actuators are attached to the back of each mirror segment. These provide six-DOF rigid-body motion of the segment. A seventh actuator is provided at the center of the substrate for radius of curvature adjustment. This actuator will be used during alignment and testing as well as on orbit to match all segments to the same radius within a close tolerance and, in concert with angular alignment achieved with the 6 DOF actuators, create the desired contiguous aspheric optical surface. To change the radius, the central actuator acts against a mechanical truss structure created by six struts (the white lines in the figure) attached to the outer end of that actuator and to the rim of the mirror. Radius change is independent of rigid-body alignment. The techniques to be used for mirror alignment are summarized in Chapter 12.

8.9 Metallic Foam Core Mirrors

The development of lightweight mirrors with metallic foam cores started soon after foamed aluminum became available as a material for heat exchangers. This material had density as low as 4% of the parent material and was low in cost, easily made, and easily brazed with minimum distortion to aluminum sheets. These favorable attributes led Pollard and co-workers to design and analyze a lightweight mirror made from this material.⁵⁷ Their model was a 12.0-in. (30.5-cm) diameter mirror comprising 0.12-in. (3.05-mm) thick concave faceplates and a core of nominal density: 10% that of the base material. Finite-element analysis and experiments did not have the expected correlation; this was attributed, in part, to differences between actual and anticipated density of the foam and to variations in other mechanical properties of that material.

Stone et al. gave results of an investigation of the shear modulus of foam materials.⁵⁸ New techniques were found to be needed since the ASTM standard for such measurements was not adequate. Ceramic (Amporox T and Amporox P), nickel, and aluminum/silicon carbide foams were tested. For materials of differing cell density, the measurements included weight density, density relative to that of the base material, shear modulus, and shear modulus relative to that of the base material. Finite-element analysis of a ~ 1.0 m (40 in.) diameter mirror substrate design utilizing foam material indicated the sensitivity of the design to variations of the various material properties. In the analysis, Poisson's ratio was assumed zero. Vukobratovich⁵⁹ pointed out that the so-called Ashby's relationship⁶⁰ for cellular solids did not agree with the results of experiments at the University of Arizona.

A design for an aluminum foam core/aluminum facesheet mirror described by Vukobratovich⁵⁹ is shown in Fig. 8.47. The use of aluminum/silicon carbide metal matrix composite (MMC) facesheets with a nickel foam core and MMC facesheets with a MMC foam core were suggested as possible improvements over the aluminum version.

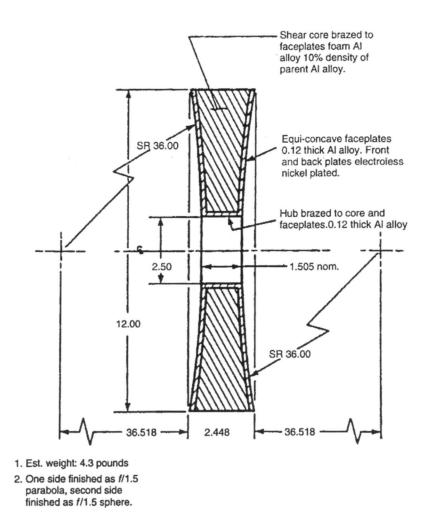


Figure 8.47 Design for a lightweight mirror with aluminum face plates and an aluminum foam core. Dimensions are in inches. (From Vukobratovich.⁵⁹)

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Mohn and Vukobratovich⁶¹ described a design for an all MMC telescope with a 0.3m (12-in.) aperture. Figure 8.48 shows this design schematically. The truss that supports the primary and secondary mirrors was made from 25-mm (1.0-in.) diameter extruded structural-grade MMC tubing with a 1.25-mm (0.05-in.) wall thickness. The secondary support was made from structural-grade MMC extruded bar stock. The secondary mirror was machined from optical-grade MMC, ELN plated, and polished. The double-concave shaped primary was fashioned from MMC core with MMC facesheets. It was ELN plated on both sides, thermally cycled for stability, and polished to an optical figure of about 1 visible light fringe. The entire telescope weighed 4.5 kg (10 lb).

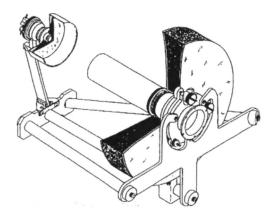


Figure 8.48 Schematic view of a 0.30-m (12-in.) aperture, *f*/5 Cassegrain telescope using the mirror shown in Fig. 8.47. (From Mohn and Vukobratovich.⁶¹)

A 0.4-m (15.75-in.) aperture Cassegrain telescope made completely of MMC and employing a single-arch form of primary mirror was described by Vukobratovich et al.⁶² This primary mirror weighed 3.2 kg (7.0 lb), which is about 43% of the weight of a solid mirror of the same shape. Its overall thickness was 83.57 mm (3.29 in.).

McClelland and Content⁶³ as well as Hadjimichael et al.⁶⁴ described ways for optimizing the design of aluminum foam core/aluminum facesheet mirrors for cryogenic applications. They cited the advantages of newly developed techniques for superpolishing bare aluminum surfaces to ~0.6-nm rms microroughness to eliminate the need for applying a polishable plating such as electroless nickel [see Lyons and Zaniewski⁶⁵] and athermal construction to allow manufacture and testing at room temperature and operation at cold temperatures with minimal mirror deformation. Sample concave spherical mirrors having diameters of 5 in. (127 mm) and clear apertures of 4 in. (101.6 mm) with high stiffness, low weight, and minimal print-through from the internal structure have been made and tested. The cores typically were fabricated from a 40 pore per inch (ppi) open-cell aluminum foam with a density ~8% of solid aluminum. Figure 8.49 shows a cross sectional view of one of these mirrors. An outer ring was provided to stiffen the mirror laterally. The mount was integrated into the back facesheet to simplify mounting of the mirror. The cores were brazed to the facesheets and outer ring by a proprietary process. The brazed assembly was annealed slowly to relieve stress. Areal density of the mirror was $<20 \text{ kg/m}^2$. The optical surface was diamond turned to the required contour after annealing.

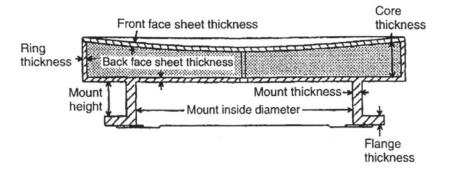


Figure 8.49 Schematic sectional view of an aluminum foam core/aluminum face plate mirror with integral mount. (From McClelland and Content.⁶³)

8.10 Pellicles

Very thin mirrors, beamsplitters, and beamcombiners can be made from films of material such as nitrocellulose, polyester, or polyethylene. Their thicknesses typically are 5 μ m (0.0002 in.) ± 10%, although 2 μ m (0.0008 in.) ± 10% thick films and special films up to 20- μ m (0.0008-in.) thick are also available. The surface qualities of standard varieties are usually better than 40/20 scratch and dig while the optical figure typically is 0.5 to 2 λ per inch. The base material transmits well (>90%) from 0.35 to 2.4 μ m, but has numerous deep absorbing regions beyond 2.4 μ m.⁶⁶ See Fig. 8.50 for a simplified representation of the transmission characteristics of a typical standard type pellicle. Pellicles can be coated to reflect, split, or combine light beams in the visible to near IR region with conventional or custom-designed coatings. Standard A/R coatings can be applied to the back side of the films.

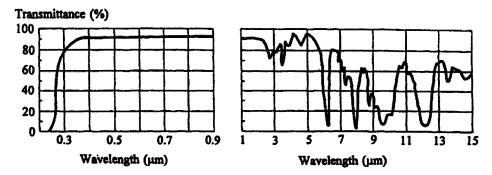


Figure 8.50 Simplified transmission characteristics of a standard nitrocellulose pellicle in the visible and infrared regions. (Courtesy of National Photocolor Corp., Mamaroneck, NY.)

A prime feature of the pellicle is the absence of ghost imaging since the first and second surface reflections at 45-deg incidence are so close together they appear superimposed. Interference effects are frequently seen. A pellicle with uncoated front and

MIRROR DESIGN

A/R coated back surfaces at 45-deg incidence can serve as a 4% beam sampler. If both surfaces are uncoated, it has about 8% total reflectance and transmittance of about (0.92)(0.90) = 83% in the visible spectral pass-band.

Since pellicles are so thin, they are more fragile than conventional plane-parallel plate optics. They are susceptible to the acoustic vibration of adjacent air columns, but work well in a vacuum. Some thicker varieties (notably, ones made of polyester films) can be used under water. The temperature range of their usefulness is about 40°C to +90°C. They can tolerate relative humidity to 95%.

Pellicles must be mounted so that the frames are not distorted, because that would distort the optical surfaces. They typically are supported by circular, square, or rectangular frames with beveled and lapped front surfaces to which the stretched film is attached. These frames usually are black anodized aluminum and have threaded holes for mounting. Special units can be made of stainless steel or ceramic. Figures 8.51 and 8.52 show a variety of nonstandard mounts and standard mounts, respectively, for pellicle products as supplied by one manufacturer.

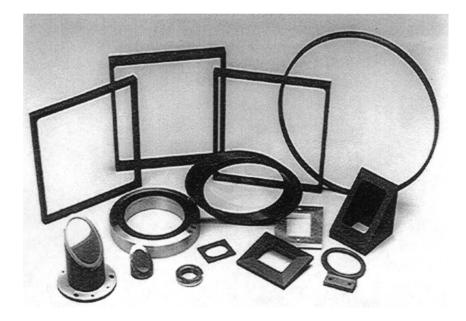
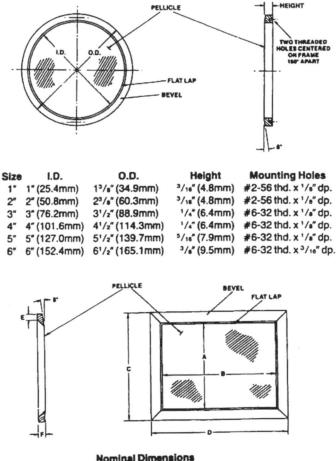


Figure 8.51 A variety of nonstandard mounted pellicles as supplied by one manufacturer. (Courtesy of National Photocolor Corp., Mamaroneck, NY.)



Nominal Dimensions		
	Inches	Millimeters
Α	5	127.0
в	7	177.8
С	65/.	168.3
D	8º/.	212.7
E	5/15	7.9
F	7/16	11.1

Figure 8.52 Some standard pellicle frame designs and dimensions. (Courtesy of National Photocolor Corp., Mamaroneck, NY.)

8.11 References

- 1. Hopkins, R.E., "Mirrors and prism systems," Chapt. 7 in *Applied Optics and Optical Engineering*, III, Academic Press, New York, 1965.
- 2. Smith, W.J., Modern Optical Engineering, 3rd ed., McGraw-Hill, New York, 2000.
- 3. Kaspereit, O.K., ORDM 2-1, Design of Fire Control Optics, U.S. Army Ordnance, Washington. 1952.
- 4. Jenkins, F.A., and White, H.E., *Fundamentals of Optics*, McGraw-Hill, New York, 1957.
- 5. Schubert, F., "Determining optical mirror size," Machine Des. 51, 1979:128.

- 6. Rodkevich, G.V., and Robachevskaya, V.I., "Possibilities of reducing the mass of large precision mirrors," *Sov. J. Opt. Technol.* 44, 1977:515.
- 7. Englehaupt, D., Chapt. 10 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, 1997.
- 8. Paquin, R., Chapt. 3 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, FL, 1997.
- Cho, M.K., Richard, R., and Vukobratovich, D., "Optimum mirror shapes and supports for light weight mirrors subjected to self-weight," *Proceedings of SPIE* 1167, 1989:2.
- 10. Florence, R., Perfect machine: Building the Palomar Telescope
- 11. Loytty, E.Y., and DeVoe, C.F., "Ultralightweight mirror blanks," *IEEE Trans.* Aerospace Electron. Syst., AES-5, 1969:300.
- Hill, J.M., Angel, J.R.P., Lutz, R.D., Olbert, B.H., and Strittmatter, P.A., "Casting the first 8.4 meter borosilicate honeycomb mirror for the Large Binocular Telescope," *Proceedings of SPIE* 3352, 172, 1998.
- Hill, J.M. and Salinari, P., "The Large Binocular Telescope Project," *Proceedings of* SPIE 3352, 1998:23.
- 14. Seibert, G.E., "Design of Lightweight Mirrors," SPIE Short Course Notes, SPIE, Bellingham, 1990.
- 15. Yoder, P.R., Jr., *Opto-Mechanical Systems Design*, 3rd ed., CRC Press, Boca Raton, 2005.
- 16. Lewis, W.C., "Space telescope mirror substrate," OSA Optical Fabrication and *Testing Workshop, Tucson*, Optical Society of America, Washington, 1979.
- 17. Hagy, H.E. and Shirkey, W.D., "Determining Absolute Thermal Expansion of Titania-Silica Glasses: A Refined Ultrasonic Method," *Appl. Opt.* 14, 1975:2099.
- 18. Hobbs, T.W., Edwards, M., and VanBrocklin, R., "Current fabrication techniques for ULE and fused silica lightweight mirrors," *Proceedings of SPIE* **5179**, 2003:1.
- 19. Fitzsimmons, T.C., and Crowe, D.A., "Ultra-lightweight mirror manufacturing and radiation response study," *RADC-TR-81-226*, Rome Air Development Ctr., Rome, 1981.
- 20. Angel, J.R.P. and Wagsness, P.A.A., U.S. Patent 4,606,960, 1986.
- 21. Parks, R.E., Wortley, R.W., and Cannon, J.E., "Engineering with lightweight mirrors," *Proceedings of SPIE* **1236**, 1990:735.
- 22. Voevodsky, M. and Wortley, R.W., "Ultra-lightweight borosilicate Gas-Fusion™ mirror for cryogenic testing," *Proceedings of SPIE* **5179**, 2003:12.
- 23. Pepi, J.W., and Wollensak, R.J., "Ultra-lightweight fused silica mirrors for cryogenic space optical system," *Proceedings of SPIE* 183, 1979:131.
- 24. Edwards, M.J., "Current fabrication techniques for ULE[™] and fused silica lightweight mirrors," *Proceedings of SPIE* **3356**, 1998:702.
- Goodman, W.A. and Jacoby, M.T., "Dimensionally stable ultra-lightweight silicon optics for both cryogenic and high-energy laser applications," *Proceedings of SPIE* 4198, 2001:260.
- 26. Noble, R.H., "Lightweight mirrors for secondaries," Proc. Symposium on Support and Testing of Large Astronomical Mirrors, Tucson, AZ 4-6 Dec. 1966, Kitt Peak National Observatory and Univ. of Arizona, Tucson, 1966:186.
- 27. Angele, W., "Main mirror for a 3-meter spaceborne optical telescope," Optical Telescope Technology, NASA SP-233, 1969:281
- Catura, R. and Vieira, J., "Lightweight aluminum optics," Proc. ESA Workshop: Cosmic X-Ray Spectroscopy Mission, Lyngby, Denmark, 24-26 June, 1985, ESA SP-2, 1985:173.

- 29. Fortini, A.J., "Open-cell silicon foam for ultralight mirrors," *Proceedings of SPIE* **3786**, 1999:440.
- 30. Jacoby, M.T., Montgomery, E. E., Fortini, A. J., and Goodman, W. A., "Design, fabrication, and testing of lightweight silicon mirrors," *Proceedings of SPIE* **3786**, 1999:460.
- 31. Jacoby, M.T., Goodman, W.A., and Content, D.A., "Results for silicon lightweight mirrors (SLMS)," *Proceedings of SPIE* **4451**, 2001:67.
- 32. Paquin, R.A., "Properties of Metals," Chapt. 35 in *Handbook of Optics*, Optical Society of America, Washington, 1994.
- 33. Jacoby, M.T., Goodman, W.A., and Content, D.A., "Results for silicon lightweight mirrors (SLMS)," *Proceedings of SPIE* **4451**, 2001:67.
- Goodman, W.A., Müller, C.E., Jacoby, M.T., and Wells, J.D. (2001). "Thermomechanical performance of precision C/SiC mounts," *Proceedings of SPIE* 4451:468.
- 35. Goodman, W.A., Jacoby, M.T., Krödel, M., and Content, D.A., "Lightweight athermal optical system using silicon lightweight mirrors (SLMS) and carbon fiber reinforced silicon carbide (Cesic) mounts," *Proceedings of SPIE* **4822**, 2002:12.
- 36. Jacoby, M.T. and Goodman, W.A., "Material properties of silicon and silicon carbide foams, *Proceedings of SPIE* **5868**, 2005: 58680J.
- 37. Goodman, W.A. and Jacoby, M.T., "SLMS athermal technology for high-quality wavefront control," Proceedings of SPIE **6666**, 2007: 66660Q.
- 38. Simmons, G.A. (1970). "The design of lightweight Cer-Vit mirror blanks, in *Optical Telescope Technology, MSFC Workshop, April 1969, NASA Report SP-233*: 219.
- Erdman, M., Bittner, H., and Haberler, P., "Development and construction of the optical system for the airborne observatory SOFIA," *Proceedings of SPIE* 4014, 2000:309.
- 40. Espiard, J., Tarreau, M., Bernier, J., Billet, J., and Paseri, J., "S.O.F.I.A. lightweighted primary mirror," *Proceedings of SPIE* **3352**, 1998:354.
- Martin, H.M., Zappellini, G.B., Cuerden, B., Miller, S.M., Riccardi, A., and Smith, B. K., "Deformable secondary mirrors for the LBT adaptive optics system," *Proceedings of SPIE* 6272, 2006:62720U.
- Brusa, G. Riccardi, A., Salinari, P., Wildi, F.P., Lloyd-Hart, M., Martin, H.M., Allen, R., Fisher, D., Miller, D.L., Biasi, R., Gallieni, D., and Zocchi, F., "MMT adaptive secondary: performance evaluation and field testing," *Proceedings of SPIE* 4839, 2003:691.
- Riccardi, A., Brusa, G., Salinari, P., Gallieni, D., Biasi, R., Andrighettoni, M., and Martin, H.M., "Adaptive secondary mirrors for the Large Binocular Telescope," *Proceedings of SPIE* 4839, 2003:721.
- 44. Gallieni, D., Anaclerio, E., Lazzarini, P.G., Ripamonti, A., Spairani, R., DelVecchio, C., Salinari, P., Riccardi, A., Stefanini, P., and Biasi, R., "LBT adaptive secondary units final design and construction, *Proceedings of SPIE* **4839**, 2003:765.
- 45. Dahlgren, R., and Gerchman, M., "The use of aluminum alloy castings as diamond machining substrates for optical surfaces," *Proceedings of SPIE* **890**, 1988:68.
- 46. Downey, C.H., Abbott, R.S., Arter, P.I., Hope, D.A., Payne, D.A., Roybal, E.A., Lester, D.F., and McClenahan, J.O., "The chopping secondary mirror for the Kuiper airborne observatory," *Proceedings of SPIE* **1167**, 1989:329.
- 47. Vukobratovich, D., Gerzoff, A., and Cho, M.K., "Therm-optic analysis of bi-metallic mirrors," *Proceedings of SPIE* **3132**, 1997:12.
- 48. Gould, G., "Method and means for making a beryllium mirror," U.S. Patent No. 4,492,669, 1985.

- 49. Paquin, R.A., Levenstein, H., Altadonna, L., and Gould, G., "Advanced lightweight beryllium optics," *Opt. Eng.* 23, 1984: 157.
- 50. Paquin, R.A., "Hot isostatic pressed beryllium for large optics," *Opt. Eng.* 25, 1986: 2003.
- 51. Geyl, R. and Cayrel, M. "The VLT secondary mirror a report," *Proceedings of SPIE* CR67, 1997:327.
- 52. Cayrel, M. "VLT beryllium secondary mirror No. 1 performance review," *Proceedings of SPIE* **3352**, 1998 721.
- 53. Barho, R., Stanghellini, S., and Jander, G., "VLT secondary mirror unit performance and test results," *Proceedings of SPIE* 3352, 1998 675.
- 54. Stanghellini, S., Manil, E., Schmid, M., and Dost, K., Design and preliminary tests of the VLT secondary mirror unit," *Proceedings of SPIE* 2871, 1996:105.
- Wells, C., Whitman, T., Hannon, J., and Jensen, A., "Assembly integration and ambient testing of the James Webb Space Telescope primary mirror," *Proceedings of* SPIE 5487, 2004:859.
- Cole, G.C., Garfield, R., Peters, T., Wolff, W., Johnson, K., Bernier, R., Kiikka, C., Nassar, T., Wong, H.A., Kincade, J., Hull, T., Gallagher, B., Chaney, D., Brown, R.J., McKay, A., and Cohen, L.M., "An overview of optical fabrication of the JWST mirror segments at Tinsley," *Proceedings of SPIE* 6265, 2006:62650V.
- 57. Pollard, W., Vukobratovich, D., and Richard, R., "The structural analysis of a lightweight aluminum foam core mirror," *Proceedings of SPIE* 748, 1987:180.
- Stone, R., Vukobratovich, D., and Richard, R., "Shear moduli for cellular foam materials and its influence on light-weight mirrors," *Proceedings of SPIE* 1167, 1989:37.
- 59. Vukobratovich, D., "Lightweight laser communications mirrors made with metal foam cores," *Proceedings of SPIE* **1044**, 1989:216.
- 60. Gibson, L.J. and Ashby, M.F., Cellular Solids, Pergamon Press, England, 1988.
- 61. Mohn, W.R. and Vukobratovich, D., "Recent applications of metal matrix composites in precision instruments and optical systems," *Opt. Eng.* 27, 1988:90.
- 62. Vukobratovich, D., Valente, T., and Ma, G. (1990). "Design and construction of a metal matrix composite ultra-lightweight optical system," *Proceedings of SPIE* **2542**, 1995:142.
- 63. McClelland, R.S. and Content, D.A., "Design, manufacture, and test of a cryo-stable Offner relay using aluminum foam core optics," *Proceedings of SPIE* **4451**, 2001:77.
- 64. Hadjimichael, T., Content, D., and Frohlich, C., "Athermal lightweight aluminum mirrors and structures," *Proceedings of SPIE* **4849**, 2002:396.
- 65. Lyons, J.J. III and Zaniewski, J.J., "High quality optically polished aluminum mirror and process for producing," U. S. Patent 6,350,176 B1, 2002.
- 66. Stern, A.K., private communication, 1998.

CHAPTER 9 Techniques for Mounting Smaller Nonmetallic Mirrors

The appropriateness of a mechanical mounting for a mirror depends on a variety of factors, including

- the inherent rigidity of the optic;
- the tolerable movement and distortion of the reflecting surface or surfaces;
- the magnitudes, locations, and orientations of the steady state forces (preloads) holding the optic against its mounting interfaces during operation;
- the transient forces driving the optic against, away from, or transversely to the mounting interfaces during exposure to extreme shock and vibration;
- the effects of steady state and changing temperatures;
- the number, shapes, sizes, and orientations of mounting interfaces between the optic and the mount;
- the rigidity and long-term stability of the mount;
- assembly, adjustment, maintenance, package size, weight, and configuration constraints; and
- affordability in the context of cost of the entire instrument.

In this chapter we address a variety of techniques commonly used to constrain mirrors in the size range from about 0.5 in. (1.27 cm) to about 35 in. (89 cm). At the small end of this range, where the mounts tend to be very simple, techniques used for mounting lenses may suffice. As would be expected, complexity increases with mirror size. The general techniques considered include mechanical clamping, elastomeric bonding, optical contacting, and mounting on flexures. Mountings appropriate for nonmetallic and metallic mirror substrates are included. In general, we progress from smaller to larger sized optics. Mountings for mirrors to be used in astronomical telescope applications are discussed in the next chapter. It is pointed out that many mounting problems sometimes thought to exist only with the largest mirrors also exist with small mirrors; the difference is one of scale. In some contemporary designs involving "small" size, but high performance, these same problems are of sufficient magnitude to warrant special consideration.

9.1 Mechanically Clamped Mirror Mountings

Figure 9.1 shows a very simple means for attaching a glass mirror configured as a plane parallel plate to a metal surface. The reflecting surface is pressed against three flat, coplanar (lapped) pads by three spring clips. The spring contacts are directly opposite the pads to minimize bending moments. This design constrains one translation and two tilts in semikinematic fashion. The posts that support the clips are machined to the proper heights for the clips to exert clamping forces (preload) of controlled magnitude normal to the mirror. Customized spacers may be used on top of the posts if desired. The ends of the clips are rounded cylindrically to obtain a line contact with the glass. Spherical pads could be used on the clips, but, as is discussed in Section 13.4, higher contact stresses will then result.

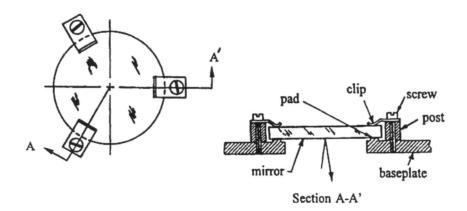


Figure 9.1 A simple spring clamped mirror mounting. (Adapted from Durie.¹)

As in the case of the similar mounting for a prism discussed in Section 7.2, the spring clips should be strong enough to restrain the mirror against the worst case shock and vibration acceleration to which the assembly may be subjected. These clips are designed as cantilevered beams of free length equal to the distance from the edge of the restraining means (the screw in Fig. 9.1) to the nearest edge of the contact area on the mirror. Equation (7.1) is used to compute the total preload P required of each clip.

$$P_i = \frac{Wa_G}{N},\tag{7.1}$$

where W is the weight of the mirror, a_G is the maximum expected acceleration normal to the pads, and N the number of springs. In metric units, include factor 9.807 in numerator.

Example 9.1: Clamping force required to constrain a mirror. (For design and analysis, use File 9.1 of the CD-ROM.)

What axial force is required of each of three springs to constrain a flat round mirror weighing 0.041 kg (0.090 lb) in the manner of Fig. 9.1 with a safety factor of 2 under acceleration of 15 times gravity directed normal to the reference pads?

From Eq. (7.1):
$$P_i = \frac{(0.041)(9.807)(15)(2)}{3} = 4.021 \text{ N} (0.904 \text{ lb}).$$

Check: Solving with USC units:

From Eq. (7.1): $P_i = \frac{(0.090)(15)(2)}{3} = 0.900 \text{ lb} (4.003 \text{ N}).$

We can determine the deflection required of each spring clip to provide a particular preload using²

$$\Delta x = \frac{\left(1 - v_M^2\right) \left(4P_i L^3\right)}{E_M b t^3},$$
(3.42)

where v_M is Poisson's ratio for the spring material, L is the free (cantilevered) length of the spring, E_M is Young's modulus for the spring material, b is the width of the spring, t is the thickness of the spring, and N is the number of springs.

The bending stress S_B created within the cantilevered spring may be calculated from Eq. (3.43), which is²

$$S_B = \frac{6PL}{bt^2 N}.$$
(3.43)

All parameters are as defined earlier.

This stress should be smaller than the yield stress for the material by a chosen safety factor f_s . The value of t that makes this happen is given by Eq. (7.8):

$$t = \left[\frac{6P_i L f_s}{bS_Y}\right]^{1/2}.$$
(7.8)

Typically, f_s would be at least 2. Note that, if the spring were attached to the mount by some means that did not require it to be perforated, the bending stress would be reduced by a factor of about 3 from that given by Eq. (3.43).²

Two lateral motions of the mirror on the pads and rotation in the plane of the pads are not constrained other than by friction in the design represented in Fig. 9.1. This may be acceptable because performance of a flat mirror is insensitive to these motions. Excessive lateral movement of the optic can be prevented by adding stops or, if the mirror is round, by sizing the supports to provide a specific small clearance to the mirror.

Figure 9.2 illustrates a less desirable mounting design in which the mirror rim rests directly on a supporting surface machined into the plate.¹ Spring clips provide clamping forces, as in Fig. 9.1, but unless the supporting surface is as flat as the mirror, minor irregularities can occur anywhere on that surface. Hence, bending moments can be introduced at the high spots and the reflecting surface may be deformed. Similar irregularities could result from foreign matter (such as dust) trapped between the mirror and the mounting surface. The likelihood of this happening with localized small pads is significantly less than with a continuous optic-to-mount contact. If there are multiple irregularities in the interface, there may also be uncertainty as to the orientation of the mirror if it shifts under vibration.

Example 9.2: Cantilevered spring constraint for a flat mirror. (For design and analysis, use File 9.2 of the CD-ROM.)

Assume that a flat round mirror is to be constrained in the manner of Fig. 9.1, but registered against six pads (lapped coplanar) and preloaded with six cantilevered springs directly opposite the pads. The mirror's weight is 1.900 lb (0.862 kg) and the acceleration normal to the plane of the pads is $a_G = 25$. Each spring free length L is 0.625 in. (15.875 mm), and its width b is 0.250 in. (6.350 mm). What should be the deflections of the springs if they are made of 6061-T6 aluminum and a safety factor of 2.0 on bending stress is desired.

From Table B12, for 6061-T6 aluminum: $E_M = 9.9 \times 10^6 \text{ lb/in.}^2, \quad v_M = 0.332, \quad S_Y = 38,000 \text{ lb/in.}^2$ From Eq. (7.1): $P_i = \frac{(1.900)(25)}{6} = 7.917 \text{ lb} (35.21 \text{ lb}).$ From Eq. (7.8): $t = \left\{ \frac{(6)(7.917)(0.625)(2.0)}{[(0.250)(38,000)]} \right\}^{1/2} = 0.079 \text{ in.} (2.008 \text{ mm}).$ From Eq. (3.42): $\Delta x = \left\{ \frac{(1-0.332^2)(4)(7.917)(0.625^3)}{[(9.9 \times 10^6)(0.250)(0.079^3)]} \right\}^{1/2} = 0.0056 \text{ in.} (0.142 \text{ mm}).$ From Eq. (3.43): $S_B = \frac{(6)(7.917)(0.625)}{[(0.250)(0.079^2)]} = 19,028 \text{ lb/in.}^2 (131.2 \text{ MPa}),$ $f_S = \frac{(38,000)}{(19,028)} = 1.997 \text{ (acceptable)}.$

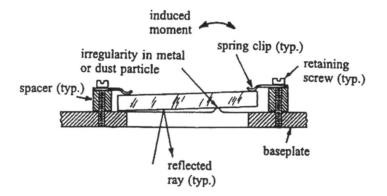


Figure 9.2 Illustration of the effect of pad irregularity or a particle in the mirror-to-mount interface. (Adapted from Durie.¹)

An arrangement sometimes used when mounting flat first surface mirrors to an unperforated baseplate is illustrated in Fig. 9.3. Here, the clips are solid so they do not bend. They might be machined integral with the baseplate. Compliance is built into the mount by inserting three small pads of soft material under the mirror opposite the clips.

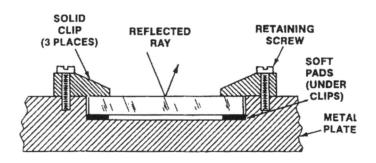


Figure 9.3 Mirror constraint using resilient pads as springs. (From Yoder.³ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

Compression of the pads accommodates thickness variations in the mirror. Wedge in the mirror substrate must be accounted for in the design. Selection of pad material is critical inasmuch as some types will, over time, be permanently deformed or become stiff. In either case, the preload is changed. It might be possible to adapt the design technique outlined in Section 7.2 for using Sorbothane or a similar material to design these pads.

A semikinematic mounting for a partially reflecting mirror used as a beamsplitter plate is illustrated in Fig. 9.4. This plate registers against three fixed "points" (actually small areas) and is spring loaded directly opposite these points. Here, and in any design with hard contacts against the reflecting side of the mirror, the location and orientation of that surface do not change with the temperature of the optic. Displacements of the mounting points caused by temperature changes may, of course, affect the location and orientation of that surface.

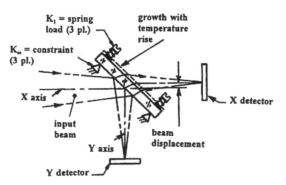


Figure 9.4 A semikinematic mounting for a beamsplitter plate. (Adapted from Lipshutz.⁴)

A mirror mount design concept that includes spring-loaded constraints normal to and in the plane of the reflecting surface is illustrated in Fig. 9.5. While compression coil springs are shown, cantilevered spring clips could be employed. This mount is semikinematic since all six degrees of freedom are constrained by spring loads and the contacts are small areas instead of points. Note that each pad contacting the back and rim of the mirror can align itself to the glass surface, thereby preventing the edges of the pads from creating stress concentrations. Alternatively, those pads might be portions of long radius spheres. Contacts somewhere on the curves would then occur even if the pad were tilted slightly.

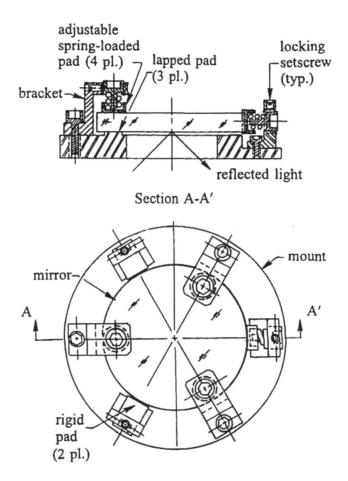


Figure 9.5 Concept for a spring-loaded mirror mounting. (From Yoder.)

A type of small mirror mount sold commercially by several suppliers has a cylindrical cavity slightly larger in ID than the OD of the cylindrical mirror to be mounted. Such a mount is illustrated schematically in Fig. 9.6. The mirror's rim rests on two plastic (usually Nylon or Delrin) rods imbedded into the lower wall of the mount. The rods are nominally parallel to each other and to the cavity axis. The mirror is pressed against the rods by gravity. This mounting configuration is known as a "Vee mount." It is considered in more detail in Section 11.1.1.

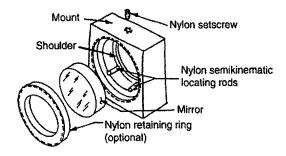


Figure 9.6 Schematic of a commercially available mount for small mirrors in which the mirror rests on two plastic rods. (From Vukobratovich.⁶)

The Nylon setscrew at the top of the mount can apply a light radial preload. Excess pressure will distort the mirror. If the retaining ring is tightened to squeeze the mirror against the flat shoulder of the mount, the setscrew should not be tightened as radial force could over constrain the mirror and distort it. The slight compliance of the Nylon components tends to prevent distortion or damage to the mirror when the temperature changes. However, performance at extreme temperatures should be checked to avoid surprises during use under such conditions.

The mounting for the secondary mirror of the Cassegrain telescope used in NASA's Geostationary Operational Environmental Satellite (GOES) is shown in Fig. 9.7. The aperture of the mirror is 1.53 in. (3.9 cm). Hookman⁷ reported that the ULE secondary mirror is mounted in an Invar cell, is supported radially and axially by pads of RTV566, and is registered against three 0.002 in. (0.05 μ m) thick Mylar pads equally spaced around the periphery of the mirror's aperture. The pads are bonded in place with epoxy to ensure that they do not move. The radial RTV pads are 0.200 in. (5.1 mm) in diameter and 0.01 in. (0.25 mm) thick, while the axial pads have the same diameters and are 0.025 in. (0.64 mm) thick. The Invar retaining ring is held by three screws to the back of the cell as shown in the exploded view of Fig. 9.8. When bottomed against the cell, the cured axial RTV pads are compressed 0.002 in. (0.05 mm) to preload the mirror nominally by about 2.15 lb 9.6 N). The radial pads are centered axially at the neutral plane of the mirror.

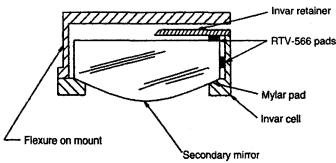
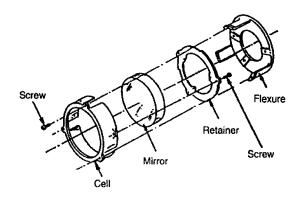
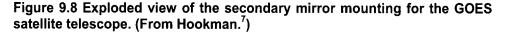
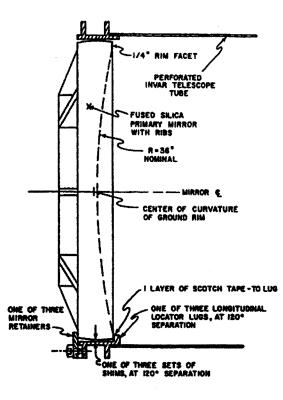


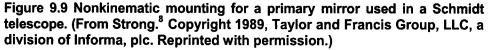
Figure 9.7 Partial section view of the mounting for the secondary mirror of the GOES satellite telescope. (Adapted from Hookman.⁷)

To minimize temperature effects caused by a mismatch of CTEs of the Invar mirror cell and the aluminum mounting plate, the cell is supported on the ends of three flexure blades machined integrally into that plate. The flexure blades are 0.5-in (12.7-mm) long, 0.32-in. (8.1-mm) wide, and 0.020-in. (0.5-mm) thick. Because of symmetry, temperature changes do not disturb the radial location or tilt of the mirror.









A nonkinematic mounting for a 16 in. (40.6 cm) diameter fused silica spherical primary mirror that John Strong used in a Schmidt telescope is shown in Fig. 9.9.⁸ The mirror rim was ground with a spherical contour, centered at the mirror's face, to avoid chipping when installed or removed for recoating. A narrow [0.25 in. (6.35 mm) annular width] flat bevel ground on the mirror's face was pressed against three steel pads (called "lugs" in the figure) inside an Invar mirror cell. The pads had previously been filed coplanar and perpendicular to the axis of the telescope tube.

Three shims with thicknesses of about 0.09 in. (2.3 mm) were inserted between the spherical ground rim of the mirror and the ID of the cell. The thickness of each shim was slightly larger than the radial clearance so they caused the cell to spring very slightly out of round. By successive adjustments of the thicknesses and locations of the shims, the mirror's central normal was brought coincident with the telescope axis.

This mirror was constrained axially by three retaining spring clips that clamped the mirror against the three axial reference pads. Friction prevented it from rotating about the telescope axis. One layer of thin plastic tape (Scotch tape) was used to isolate the mirror from the pads. This increased the mount's resistance to mechanical shock and provided some thermal isolation.

Vukobratovich⁶ suggested the kinematic configuration shown in Fig. 9.10 for mounting a rectangular mirror on edge. The mirror back is supported at three points located at the lower corners and at the midpoint of the top edge. In the vertical direction, the mirror is supported at two points located at distances of 0.22a from the ends, where a is the length of the longer (horizontal) edge of the mirror. These locations minimize the mirror's deflection due to gravity. Note that the mirror is not preloaded against the back supports and the only preload acting vertically is self-weight. If the bottom supports are located slightly in front of the plane containing the CG of the mirror, an overturning moment would be exerted. This moment would tend to press the mirror against the top back support. In the absence of friction, the mirror also would tend to slide on the lower edge point supports until it contacts the lower back supports. If the point contacts are changed to small area contacts to make the design semikinematic, friction would come into play and there would be no guarantee that the mirror would touch the lower back supports. If the mirror is intentionally moved by applying an external horizontal force, the desired contact with those supports could be established. Friction should then hold the mirror in place unless disturbed.

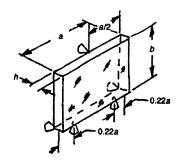


Figure 9.10 Concept for a kinematic mounting for a rectangular mirror supported on edge. (From Vukobratovich.⁶)

Circular, rectangular, or nonsymmetrically shaped mirrors can often be mounted in the same manner as a lens. Circular mirrors up to at least 4 in. (10.2 cm) diameter can be held with threaded retaining rings. Circular or noncircular ones can be held with flanges or cantilevered springs. The OD limit for a threaded mount is set primarily by the increasing difficulty of machining thin circular retaining rings with sufficient roundness in larger diameters.

Figure 9.11 shows the retaining ring concept as applied to a mirror. The optic is shown as a convex sphere although a mirror with an aspheric, or concave surface could be mounted similarly. The reflecting surface is registered (i.e., aligned) against a conical shoulder in the mount by an axial preload exerted by tightening the retainer. The ring typically has a loose fit (Class 1 or 2 per ANSI publication B1.1-1982) in the mount's thread ID. Contact occurs on the polished surface of the mirror to encourage precise centering of the curved surface on the mechanical axis of the mount because of balancing of radial components of the axial force. See Section 3.1.

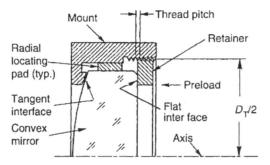


Figure 9.11 Conceptual configuration of a convex mirror secured in its mount with a threaded retaining ring.

Precise edging or close tolerances on the OD of the mirror are not required if custom fitted spacers are used as radial locating pads. Although it is not indicated in the figure, contact on the convex surface should occur at the same height from the axis as the center of the opposite contact area to minimize bending of the mirror. Sharp corner contact on the polished surface is shown. To minimize contact stress in the mirror (see Section 13.8), a tangential (conical) or toroidal (donut shaped) interface would be preferred. Section 3.9 describes the various shapes of mechanical interfaces for lenses. They are equally applicable to mountings for small mirrors.

As in the case for lens retainers, two or more holes are sometimes drilled into the exposed face of the retainer to accept pins on the end of a cylindrical wrench used to tighten the ring. Alternatively, one or two diametrical slots may be cut across the face of the retainer for this purpose. A flat plate that spans the retainer can be used as the wrench in the latter case. If a retaining force is applied to the back of the mirror and the fixed interface is at the front surface, the possibility of damaging the reflecting surface by the scraping action of the retainer while being tightened is reduced.

As in lens mounting, the magnitude of the total preload (P) developed in a threaded retainer lens mounting design with a specific torque (Q) applied to the ring at a fixed temperature can be estimated by the following equation:

$$P = \frac{5Q}{D_T},\tag{3.34a}$$

where D_T is the pitch diameter of the thread as shown in Fig. 9.1.

Note that the accuracy of this equation is subject to the same limitations discussed in Section 3.4.1 and depends primarily upon the coefficient of friction in the threaded joint, which is quite uncertain in real life. This means that the torque applied to a threaded retainer cannot be relied upon to produce a specific preload.

Another mounting for a small circular mirror, in this case a second surface type, is shown in Fig. 9.12. Here the mirror surface registers against a tangential interface while the flat bevel on the front of the mirror touches a toroidal interface on the retainer. Contacts occur at the same height on both sides. The choice of these interface shapes, the dimensions, and a "loose" fit in the retainer threads ensure minimal contact stress as well as minimal tendency for moments introduced by the mount to bend the mirror.

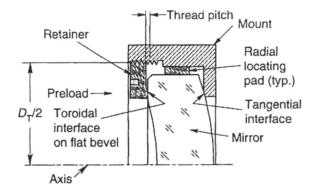


Figure 9.12 Conceptual configuration of a threaded retaining ring mounting for a second surface mirror.

A typical design for a circular mirror mounting involving a continuous flange is shown in Fig. 9.13. This type of retainer is most frequently used with mirrors larger than typically could be held with threaded rings or if a more precise axial preload is needed in the application. Several close fitting locating pads around the rim of the mirror help to center it to the mechanical axis of the mount. An annular locating land on the shoulder localizes contact on the mirror's flat back directly opposite the clamping (preload) force. The land surface should be lapped flat to minimize distortion of the mirror's reflecting surface by overconstraint. It also should be accurately perpendicular to the lens axis.

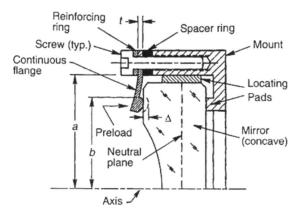


Figure 9.13 Schematic configuration of a circular flange type retainer axially constraining a concave mirror in a mount.

The interface between the flange and the flat bevel on the mirror is shown as toroidal to minimize contact stress. A flat surface on the flange would work well if it could always be aligned exactly to the bevel, but a sharp-corner contact and thus increased stress in the optic could result from machining errors or temperature changes.

Temperature changes also can create problems with regard to the fit of the radial locating pads and the constancy of axial preload in this and all the other mirror mounts discussed here because of differential expansion or contraction of the optical and mechanical parts. This topic is considered and corrective measures outlined in Chapter 14.

The function of the clamping flange in the design of Fig. 9.13 is the same as that of the threaded retainer described earlier. The magnitude of the preload exerted in this way can be determined fairly closely using Eqs. (3.38) through (3.40) which, according to Roark,² apply to a perforated circular plate with the outer edge fixed and an axially directed load applied uniformly along the inner edge to deflect that edge:

$$\Delta x = \left(K_A - K_B\right) \left(\frac{P}{t^3}\right),\tag{3.38}$$

where

$$K_{A} = \frac{\left\{3\left(m^{2}-1\right)\left[a^{4}-b^{4}-4a^{2}b^{2}\ln\left(\frac{a}{b}\right)\right]\right\}}{\left(4\pi m^{2}E_{M}a^{2}\right)},$$
(3.39)

$$K_{B} \frac{3(m^{2}-1)(m+1)\left[2\ln\left(\frac{a}{b}\right)+\left(\frac{b^{2}}{a^{2}}\right)-1\right]\left[b^{4}+2a^{2}b^{2}\ln\left(\frac{a}{b}\right)-a^{2}b^{2}\right]}{(4\pi m^{2}E_{M})\left[b^{2}(m+1)+a^{2}(m-1)\right]},$$
(3.40)

where P is the total preload, t is the flange thickness, a and b are the outer and inner radii of the cantilevered section, m is the reciprocal of Poisson's ratio (v_M) , and E_M is the Young's modulus of the flange material.

The spacer under the flange can be ground at assembly to the particular axial thickness that produces the predetermined flange deflection when firm metal-to-metal contact is achieved by tightening the clamping screws. Customizing the spacer accommodates variations in mirror thicknesses. The flange material and thickness are the prime design variables. The dimensions a and b, and hence the annular width (a - b), can also be varied, but these are usually set primarily by the mirror aperture, mount wall thickness, and overall dimensional limitations.

The stress, S_B , built up in the bent portion of the flange must not exceed the yield strength of the material. The following equations apply:

$$S_{B} = K_{C}P/t^{2} = S_{Y}/f_{S}, \qquad (3.41)$$

where

$$K_{C} = \left[\frac{3}{2\pi} \right] \left[1 - \frac{2mb^{2} - 2b^{2}(m+1) \ln(a/b)}{a^{2}(m-1) + b^{2}(m+1)} \right],$$
(3.42)

and P is the total preload, t is the flange thickness, a and b are the outer and inner radii of the cantilevered section, m is the reciprocal of Poisson's ratio (v_M) and E_M is the Young's modulus of the flange material.

The reader is referred to Sect. 3.6.2 for discussions of the use of these equations and for worked-out numerical examples illustrating their applications.

As in the previously considered case of flange constraint for refracting optics, the deflections Δ measured between the attachment points (screws) should be essentially the same as those existing at those points. This ensures uniform contact at the desired zonal height from the axis. This can be accomplished by machining the flexing portion of the flange as a thinned annular region in a thicker ring, thereby providing extra thickness at the clamped annular zone of the flange. It also could be done by reinforcing the flange with a stiff backup ring as shown schematically in Fig. 9.13.

Increasing the number of screws also tends to reduce the possibility of non-uniform preload around the mirror's edge. If we adopt the advice of Shigley and Mischke⁹ with regard to spacing of screw constraints on a gasketed flange for a high-pressure chamber and apply it to the mirror mounting case, the number of screws, N, should be:

$$3 \le \frac{\pi D_B}{(Nd)} \le 6,\tag{9.1}$$

where D_B equals the diameter of the "bolt circle" passing through the centers of the screws, and *d* equals the diameter of the screw head.

This criterion may be overly conservative in an optical application, especially if a stiff backup ring is employed or the flange is thickened in the region where it is clamped. Engineering judgment and possibly experimentation might well be applied here.

9.2 Bonded Mirror Mountings

First surface mirrors with diameters typically about 6 in. (15.2 cm) or smaller can be bonded directly to a mechanical support in much the same manner as described earlier for

prisms. The ratio of the largest face dimension to the thickness should be less than 10:1 and preferably no more than 6:1 in order for dimensional changes in the adhesive during curing or under temperature changes not to distort the mirror surface excessively. Figure 9.14 illustrates such a design. The determination of the required total bonding area, Q_{MIN} , follows the methods using Eq. (7.9) that were explained in Sect. 7.5.

$$Q_{\rm MIN} = \frac{Wa_G f_S}{J} \tag{7.9}$$

where: W is the mirror's weight, a_G is the worst-case acceleration factor, f_S is the desired safety factor (typically 2 to 5), and J is the adhesive's shear or tensile strength (usually approximately equal). Example 9.3 illustrates this calculation.

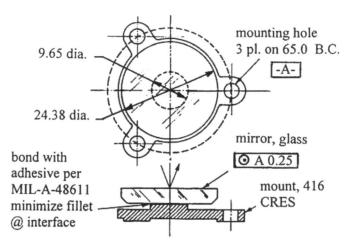


Figure 9.14 A typical bonded first surface mirror assembly. Dimensions are in millimeters. (Adapted from a U.S. Army drawing.)

Example 9.3: A mirror bonded on its back to a mount. (For design and analysis, use File 9.3 of the CD-ROM.)

A mirror is made of N-BK7 is 4.000 in. (10.160 mm) in diameter and is 0.750-in. (19.050-mm) thick (5.33:1 ratio). The mounting base is type 416 stainless steel. The bonding pad is circular and the adhesive is EC2216B/A epoxy. The acceleration is a_G = 15. What are (a) the minimum area of the bond for a safety factor f_S of 4 and (b) the minimum bond diameter?

From Table B1: the glass density is 0.091 lb/in.³. From Table B14: J = 2500 lb/in.² for 2216B/A epoxy.

The mirror weight is
$$\pi \left(\frac{4.000}{2}\right)^2 (0.750)(0.091) = 0.858 \text{ lb}(0.389 \text{ kg}).$$

(a) From Eq. (7.9): $Q_{\text{MIN}} = \frac{(0.858)(15)(4)}{2500} - 0.021 \text{ in.}^2 (13.285 \text{ mm}^2).$

(b) The minimum bond area = $\pi \left(\frac{D}{2}\right)^2 = Q_{\text{MIN}}$, where D is the bond diameter, so

$$D = \left| \frac{(2^2)(0.021)}{\pi} \right|^{\frac{1}{2}} = 0.164 \text{ in. } (4.153 \text{ mm}).$$

As pointed out earlier in discussions of bonded prisms, for maximum glass-to-metal bond strength, the adhesive layer should have a particular thickness. Experience has indicated that 3M EC2216-B/A epoxy should have a thickness of 0.075 to 0.125 mm (0.003 to 0.005 in.). Some adhesive manufacturers recommend thicknesses as large as 0.4 mm (0.016 in.) for their products, while some users have found success with 0.05 mm (0.002 in.) thicknesses. A thin bond is stiffer than a thick one. The resonant frequency of the bonded assembly would be higher with a thin bond than with a thick one.

One method of ensuring the right layer thickness is to place spacers (wires, plastic fishing line, or flat shims) of the specified thickness at three places symmetrically located on one bonding surface before applying the adhesive. Care must be exercised to hold the glass part against these spacers during assembly and curing. The adhesive should not extend between the spacers and either part to be bonded since this could affect the adhesive layer thickness. Another technique for obtaining a uniformly thin layer of epoxy between the glass and metal surfaces is to mix small glass beads into the epoxy before applying it to the surfaces to be bonded.¹¹ When the parts are clamped securely together, the largest beads contact both faces and hold those surfaces apart by the bead diameters. Since such beads can be procured with closely controlled diameters, the achievement of specific thickness joints is relatively simple. The glass beads have essentially no effect on bond strength.

Figure 9.14 represents a typical bonded assembly for a military application. It was designed long before Eq. (7.9) was published. Its bond diameter was probably chosen from experience with similar assemblies that did not fail during vibration and shock testing or in service. It is possible that the design was proportioned as shown because it "looked right" to the draftsman. In any case, from the dimensions given, its bond diameter-to-mirror diameter ratio is approximately 0.4. In Example 9.3, the ratio, based on Eq. (7.9), is 0.16/4.00 = 0.04. The factor of 10 difference between these designs may indicate an ultra conservative design philosophy in the case of the older design.

Because adhesives and metals typically have CTEs larger than those of glasses and other mirror substrate materials, differential dimensional changes at extreme temperatures can be significant. It is advisable to keep bond areas as small as possible while providing adequate strength. As in the case of prism bonding, the adhesive should, if possible, be distributed in small separated areas with the total area equal to at least the calculated minimum value for the anticipated shock and vibration loadings. This minimizes thermal expansion problems and helps secure the mirror in a more kinematic fashion. A pattern of three small bonds arranged in an equilateral triangle has been found to work well. A ring of small bonds has been used successfully with circular optics. The ring diameter should be about 70% of the mirror diameter in that case. See Fig. 9.15.

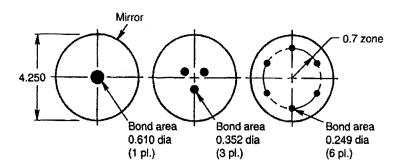


Figure 9.15 Schematics of equal total area bonds drawn to the same scale on the back of a first surface mirror. (a) Single centered bond, (b) three bonds in equilateral triangle pattern, (c) six bonds at ~0.7 zone. Dimensions are in inches. (From Yoder.³ Copyright 2005 Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

A point emphasized in connection with bonding prisms and that is also pertinent to bonding mirrors is that fillets of the adhesive should not be allowed to form at the edges of the bond area. See Fig. 7.20(a). Excess adhesive that might seep out of the joint before curing should be removed. A preferred bond configuration is shown in Fig. 7.20(b). This can be accomplished by controlling the quantity of adhesive applied.

Another way in which small mirrors can be attached to their mounts is to use an annular ring of elastomer as we discussed in Section 3.9 for mounting lenses. Figure 9.16 shows an example. The same design principles as discussed in the earlier section apply. The

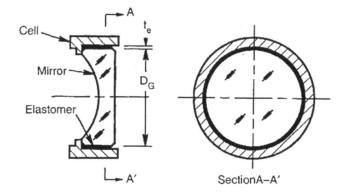


Figure 9.16 Schematic of a concave first surface mirror potted with an annular ring of elastomer into a cell in the same manner as shown in Fig. 3.36. (From Yoder.³ Copyright 2005 Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

design can be rendered radially athermal through application of Eqs. (3.59) and (3.60) or (3.61). Noncircular mirrors can be secured in place by this technique.

Figure 9.17(a), shows another technique for elastomerically potting a mirror into a mount.¹² The circular mirror has the dimensions indicated in the figure. It is attached to its mount with twelve discrete segments or pads of elastomer applied in the gap between the mirror OD and the cell ID. In this case, the mirror is fused silica ($\alpha_G = 3.22 \times 10^{-6}$ /°F), the cell is Kovar ($\alpha_M = 3.05 \times 10^{-6}$ /°F), and the elastomer is Dow Corning 6-1104 silicone ($\alpha_e = 261 \times 10^{-6}$ /°F). Assuming $\nu_e = 0.499$ and applying Eq. (3.60), the value for $\alpha_e^* = 4.34 \times 10^{-6}$ /°F. From Eq. (3.59), the nominal "athermal" thickness of the elastomer pads would be 0.914 mm (0.036 in.). The pad edge dimension (if square) or diameter (if circular) is d_e. This dimension is a design parameter.

A finite element analysis of vibrational modes for this design reported by Mammini et al^{12} indicated that the fundamental frequencies of the piston and tip/tilt modes varied with t_e . Figure 9.17(b) shows these variations as spline fits through the data points listed in the paper. The application required that these frequencies be at least 300 Hz. The long dashed lines show that d_e should then be at least 0.28 in. (7.11 mm). The actual dimension used was 0.289 in. (7.34 mm). Thermal analysis showed that an 18°F temperature change would cause an out-of-plane surface distortion of less than 1/300 wavelength at 633 nm over the entire mirror surface.

Vukobratovich⁶ described a technique for radially constraining a circular lens or mirror with a thin strip of Mylar located between the OD of the optic and the ID of the cell. See Fig. 9.18. The shim is perforated at three places located so the holes line up with radially directed holes through the cell wall. RTV compound is injected through those holes to reach the rim of the mirror. Pads of RTV formed after curing hold the mirror from rotating about its axis (clocking) and constrain the optic radially.

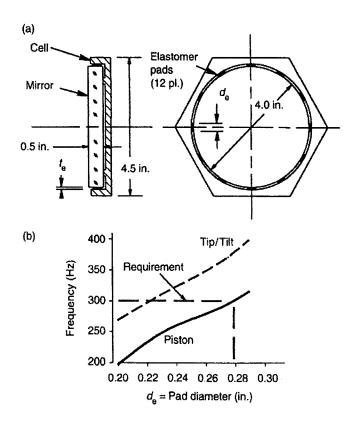


Figure 9.17 (a) A flat mirror mounted into a cell with multiple (12) discrete pads of elastomer of dimension d_e and thickness t_{e^*} (b) Plots of the fundamental frequency of the assembly determined by FEA for tip/tilt and piston vibration modes. The required frequency (for each mode) is shown. (Adapted from Mammini et al.¹²)

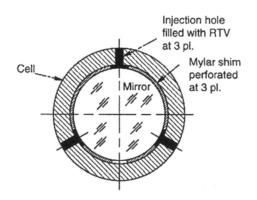


Figure 9.18 A mirror mounting concept in which three pads of elastomer are injected through three holes in the cell wall and in a Mylar shim strip. (Adapted from Vukobratovich.⁶)

9.3 Compound Mirror Mountings

Two or more mirrors attached together or to a common mount form optomechanical subassemblies that can serve some particular function not possible with a single mirror. For example, two flat mirrors oriented at 45 deg to each other can be used to deviate a light beam by 90 deg. If rigidly attached together, this penta mirror subassembly will serve the same function as a penta prism, but will not suffer light absorption losses within the glass. Hence, it can be used throughout the ultraviolet through infrared spectral regions. Of course, the reflecting coatings would have to reflect in the region of interest. The weight of a penta mirror is generally lower than that of a penta prism of equivalent aperture.

A significant problem in the design and fabrication of compound mirror subassemblies is how to establish and hold the mirrors in the proper relative orientation to maintain alignment stability over the long term and not distort the optical surfaces. One approach that has been used is to mechanically clamp the mirrors individually to a precision-machined metal block or to a built-up structure providing the appropriate angular and positional relationships between surfaces. Other approaches include bonding glass-to-glass or glass-to-metal parts and optical contacting glass parts. We will describe examples of these techniques.

Figure 9.19 shows a penta mirror constructed by mechanical clamping. Here, two gold coated first surface mirrors shaped generally as rectangles with rounded ends were each held by three screws to three coplanar lapped pads on either side of an aluminum casting that accurately provided the 45-deg dihedral angle. The screws passed through clearance holes in the mirrors; each screw compressed two Belleville washers to preload the mirror against a pad. Mrus et al.¹³ described how this hardware was used as part of an automatic theodolite system for prelaunch azimuth alignment of the Saturn V space vehicles. It was used in a generally stable environment inside a concrete bunker at Cape Canaveral.

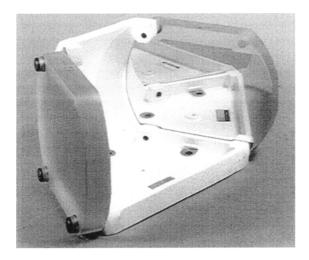


Figure 9.19 A penta-mirror subassembly made by clamping two mirrors to a precision metal casting. (Courtesy of NASA Marshall Space Flight Center.)

A bonding approach used successfully to make many stable penta mirror subassemblies for application in military optical rangefinders (see Patrick¹⁴) had glass mirrors optically cemented on edge to a glass baseplate as shown schematically in Fig. 9.20. This subassembly was attached to the optical bar of the rangefinder. Figure 9.21 shows an actual penta reflector of this general type. The base plate in this example is metal. The useful aperture is slightly over 50 mm (1.97 in.).

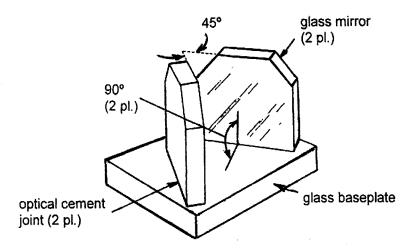


Figure 9.20 A penta-mirror subassembly made by cementing mirrors on edge to a glass baseplate forming accurate 45° and 90° angles as noted.

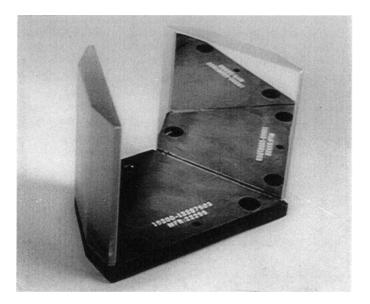


Figure 9.21 A penta mirror subassembly made by bonding glass mirrors on edge to a metal bracket. (Courtesy of PLX Corporation, Deer Park, NY.)

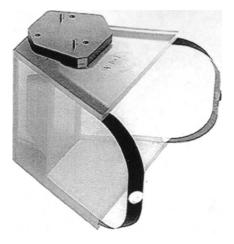


Figure 9.22 A 10-cm (3.94 in.) aperture penta mirror subassembly made by optically contacting Cer-Vit components. (From Yoder.¹⁵)

Figure 9.22 shows a penta mirror subassembly in which the polished faces of two flat Cer-Vit mirrors were optically contacted to a Cer-Vit angle block that had been ground and polished to within 1 arcsec of the nominal 45 deg.¹⁵ The angle block was hollowed out to reduce weight without reducing strength. Triangular Cer-Vit cover plates were then attached with optical cement to both the top and bottom of the assembly and a rectangular cover plate was cemented across its back. These three plates served not only as mechanical braces but also sealed the exposed edges of the contacted joints. With mirrors measuring approximately 11 by 16 by 1.3 cm (4.33 by 6.30 by 0.51 in.), the subassembly had a clear aperture of 10 cm (3.94 in.). The circular white spot that may be seen on the forward edge of the left mirror in Fig. 9.22 is a small flat mirror cemented in place to serve as an alignment reference during installation in a telescope. A second such reference mirror was attached to the other main mirror. A roof penta mirror assembly of similar construction and size (see Fig. 9.23) was also described in the referenced paper.



Figure 9.23 A 10 cm (3.94 in.) aperture roof penta subassembly made by optically contacting Cer-Vit components. (From Yoder.¹⁵)

To verify these optically contacted designs, a prototype of the penta mirror assembly was mounted in its housing (see Fig. 9.24) and then subjected to adverse thermal, vibration, and shock environments. First, it was temperature cycled several times from -2to 68°C (28 to 154°F) while monitoring the reflected wave front interferometrically. The test setup was capable of detecting changes of $\lambda/30$ and had an inherent error of less than $\lambda/15$ for $\lambda = 0.63 \mu m$. The maximum thermally induced distortion in the penta mirror was $\lambda/4$ peak to valley. This error was acceptable for the intended application. The assembly was then vibrated without failure at loadings up to $a_G = 5$ and frequencies of 5 to 500 Hz along each of three orthogonal axes. Resonances were noted at the higher frequency in two axes. Shock testing with peak loading up to $a_G = 28$ in 8 msec pulses along two directions also produced no permanent degradation of the device—as demonstrated by post-exposure interferometric evaluation. These adverse environmental conditions were representative of anticipated extremes during shipping and operation of the system.

A roof mirror that functions as a Porro prism is shown in Fig. 9.25. This assembly has an aperture of slightly over 1.75 in. (44.4 mm) by 4.0 in. (102 mm). Its mirrors are 0.5-in. (12.7-mm) thick Pyrex. These mirrors are bonded on one long edge to a Pyrex keel that is, in turn, bonded to a 0.125-in. (3.2-mm) thick stainless steel mounting plate. The stainless steel end plates are made with precision 90-deg angles. Each end plate is bonded to the top of one mirror and to the end of the other mirror. The mirrors are aligned at right angles within tolerances as small as 0.5 arcsec. The tolerance on mirror figure is as small as 0.1 wave at $\lambda = 0.63 \mu m$.

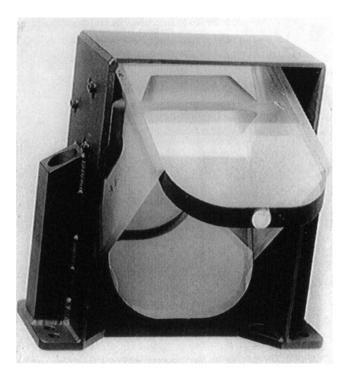


Figure 9.24 Photograph of the penta mirror subassembly of Fig. 9.22 mounted in its Invar housing. (From Yoder.¹⁵)

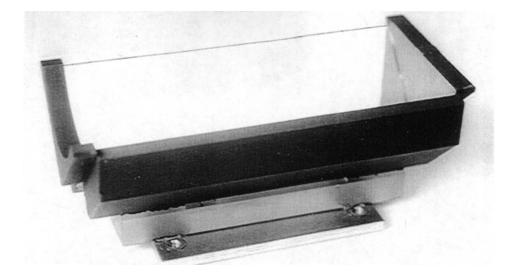


Figure 9.25 Photograph of a Porro mirror subassembly made by bonding two flat mirrors at 90 deg. (Courtesy of PLX Corporation, Deer Park, NY.)

Figure 9.26 shows front and back views of a hollow corner retroreflector (HCR), which is the mirror version of a cube corner prism (see Section 6.4.22). This subassembly comprises three, nominally square-faced Pyrex mirrors. The aperture of this unit is about 45 mm (1.77 in.). The edges of the mirrors are bonded to each other and supported as a subassembly in an aluminum housing configured for ease of mounting using an elastomeric material (white) surrounding three rubbery inserts (gray). The accuracy of the 180° light deviation for such units typically lies between 0.25 and 5 arcsec. Reflected wavefront error can be as small as 0.08 wave of visible light, with the actual value depending upon aperture size. Apertures exceeding 5 in. (12.7 cm) are available.

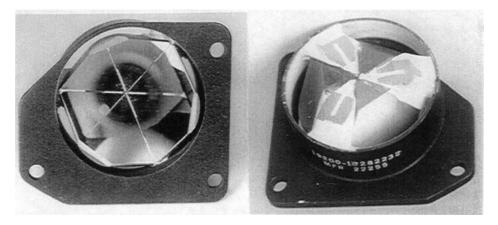


Figure 9.26 Front (a) and back (b) views of a HCR made by bonding three flat square mirrors in a mutually perpendicular fashion. (Courtesy of PLX Corporation, Deer Park, NY.)

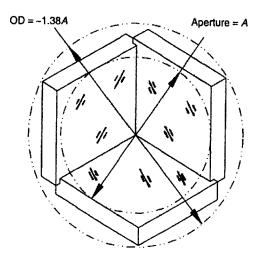


Figure 9.27 Front view schematic of another HCR. (Courtesy of PROSystems, Inc., Kearneysville, WV.)

Figure 9.27 shows schematically a front view of another commercially available HCR. According to Lyons and Lyons,¹⁶ each of the three mirrors in this unit is fashioned with a narrow 90-deg groove along one edge. Adjacent mirror edges are bonded together in these grooves. This design provides very small seam widths of 0.001 in. (25 μ m) at the mirror interfaces. The dimensions in the figure relate the clear aperture to the outside frontal diameter of the device.

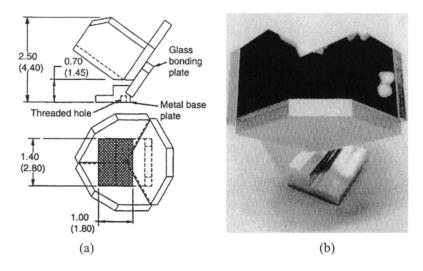


Figure 9.28 Schematic (a) and photograph (b) of a HCR bonded to a metal baseplate. Dimensions apply to models with ODs of 2.50 and (4.0) in. [6.35 and (10.0) cm]. (Courtesy of PROSystems, Inc., Kearneysville, WV.)

In the HCR subassembly shown in Fig. 9.28, a glass plate is cemented at one end to the back of one mirror of a bonded subassembly of the type shown in Fig. 9.27. The other end of the plate is bonded into a notch machined into a metal baseplate. This baseplate has a threaded hole for attachment to the structure of some hardware application. Units with apertures of 2.50 in. (6.35 cm) and 4.00 in. (10.16 cm) have dimensions as indicated in the diagram. Pyrex or Zerodur mirrors and aluminum or Invar bases are used depending upon the temperature range of the application and cost constraints. Testing has indicated some models to be stable in regard to optical and mechanical performance over temperature ranges as large as $\sim 200^{\circ}$ C ($\sim 360^{\circ}$ F). Some have operated well at 170 K.

Figure 9.29 illustrates a unique type of HCR in which the virtual apex of the three mirror array is located at the center of a metal sphere within typically 0.0001 to 0.0005 in. (2.54 to 12.70 μ m). This type device is commonly referred to as a "spherically mounted retroreflector" (SMR). Sphere diameters typically range from 0.5000 to 1.5000 in. (12.700 to 38.100 mm). Dimensions (in inches) of the SMRs represented in the figure are: A = 0.30, 0.50, and 1.00; B = 0.37, 0.60, and 1.15; C = 0.42, 0.73, and 1.23; and D = 0.52, 0.92, and 1.51 for sphere diameters of 0.500, 0.875, and 1.500 in., respectively.

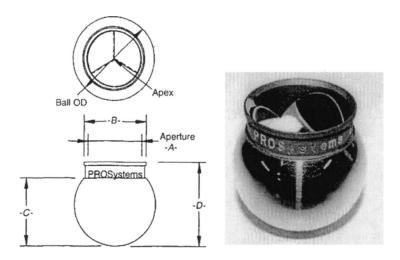


Figure 9.29 Schematic diagram and photograph of a spherically mounted hollow retroreflector (SMR). Dimensions are given in the text for units of diameters 0.5 to 1.5 in. (Courtesy of PROSystems, Inc., Kearneysville, WV.)

SMRs are used in the manufacturing and metrology industries as targets for tracking and measuring distances with electro-optical coordinate measuring machine (CMM) or target trackers employing laser beams. Bridges and Hagan¹⁷ described one tracker. Figure 9.30 is a schematic diagram of the function of such a system. Those authors reported that the device measures feature locations on objects as far away as ~35 m (~115 ft) with an accuracy of $\pm 25 \ \mu m (\pm 0.001 \ in.)$ at 5 m (16.4 ft) range. Key to success with such a device is absolute distance measurement in which round trip time of flight of a modulated IR beam is transmitted coaxially with the tracker beam to the SMR. This allows measurements to be resumed without recalibration if the beam is temporarily obscured during operation. It also allows the system to monitor slow alignment drifts. The bodies of the spherical SMR targets are typically made of Type 420 CRES for corrosion resistance and magnetic characteristics. Target surfaces are CNC machined, hand ground, and polished to a high degree of sphericity (typically true spheres within ± 0.000025 in. (0.64 µm).

In the application of Fig. 9.30, one of these SMRs is held magnetically in a three faced (kinematic) pocket on the tracker for system calibration. During operation, that target is attached to or held by hand in contact with the surface or feature to be measured. As the target is moved to selected points on the surface, the tracker determines the surface coordinates. The contour and/or relative locations of features on the surface are determined from multiple point sampling by an associated computer. The system's software automatically compensates for the radius of the sphere.

It is important in this application that errors in the dihedral angles and the maximum differences between those angles for a given target unit are small. As pointed out by

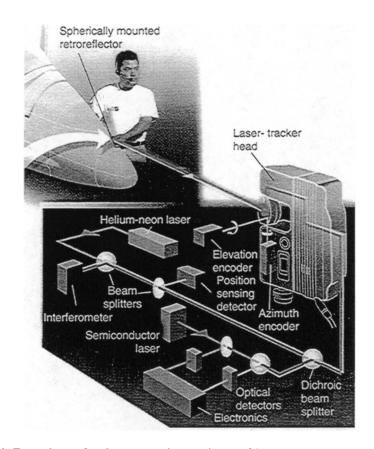


Figure 9.30 Function of a laser tracker using a SMR to remotely measure coordinates of selected points on a distant object. (From Bridges and Hagan.¹⁷ Copyright 2001, American Institute of Physics. Reprinted with permission.)

Yoder,¹⁸ angle errors from exactly 90 deg in a cube corner prism or in a HCR determine the absolute accuracy of 180-deg beam deviation. The return beam actually comprises six beams. Two come from each mirror. For any HCR, the worst-case deviation error δ is given by:

$$\delta = 3.26\varepsilon, \tag{9.2}$$

where ε is the error in dihedral angle, all such errors are equal, and all errors have the same sign. In a laser tracker application, if deviation errors are too large, the tracker will "jump" from one return beam to another at long target ranges because all six beams are not captured by the tracker aperture. The angle errors and the differences between angles determine the value for the constant in Eq. (9.2). SMRs typically have angle errors ranging from 3 to 10 arc sec and angle differences of 2 to 10 arc sec. Apex centration error is as small as 0.0002 in. (5.1 μ m).

An important characteristic of a HCR is the coating applied to the mirrors. Ordinary first surface mirror coatings could introduce a phase shift into the polarized light beam from the tracker. This would degrade the system's performance. Special coatings can be applied to the mirrors to minimize this problem. One type of coating that reduces this phase shift is the "zero-phase silver" coating offered by Denton Vacuum Company, Moorestown, NJ. Bridges and Hagan¹⁷ indicated that the actual residual phase shifts introduced by the coatings on a batch of mirrors should be measured and sets of mirrors with similar phase shifts selected for use in a given SMR unit. This tends to maximize the range at which the tracker can acquire and track that particular target.

A useful adaptation of the design of a HCR is the lateral transfer retroreflector (LTR). An example of a device of this type is shown in Fig. 9.31. It comprises a single mirror at one end of an elongated box and a roof mirror at the other end of that box. The roof edge is oriented at 45 deg so it functions in the manner of an Amici prism. All three mirror surfaces are mutually perpendicular so the device acts as a "slice" through a HCR of very large aperture. Lateral offsets of the beam up to 30 in. (76 cm) and apertures up to 2 in. (5 cm) are commercially available from at least one manufacturer.

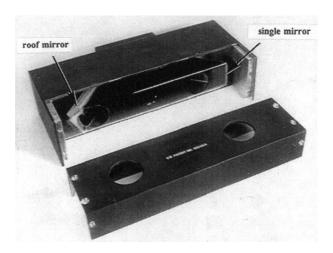


Figure 9.31 Photograph of a partially disassembled lateral transfer retroreflector (LTR). (Courtesy of PLX Corp., Deer Park, NY.)

9.4 Flexure Mountings for Smaller Mirrors

In optical instrument applications, flexures are passive mechanical devices used to isolate optical components from mechanical and thermally induced forces acting on the structural support system. They minimize the effects of those forces on optical component location and orientation as well as surface deflections. Flexures are free of stiction and friction effects that hamper the use of spherical ball joints and hinges in optical instruments. In most cases, flexures are designed with compliance in one direction, but stiffness in the two orthogonal directions.

Figure 9.32 illustrates the principle underlying the design of one type of flexure mounting for mirrors. The mirror is circular and mounted in a cell. That cell is suspended from three thin flexure blades. A curved arrow indicates the single direction of allowable motion for each flexure acting alone. For small deflections, these curved motion paths closely approximate straight lines. Ideally, these lines of freedom should intersect at a point and this point should coincide with the center of gravity of the mirror. The flexures should be made of the same material and their free lengths should be equal; the fixed ends of the three flexures form an equilateral triangle. The function of this system of flexures M and B will permit rotation only about point O, which is the intersection of flexure B with a line extending flexure A. With flexure C in place, rotation about O is prevented because flexure C is stiff in that direction. Although it is not apparent in the figure, the flexure blades have sufficient depth perpendicular to the mirror face to provide stiffness and prevent the mirror from translating axially.

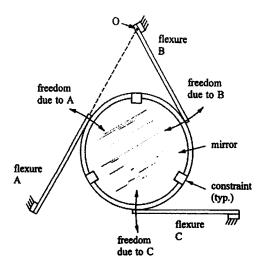


Figure 9.32 Concept for a flexure mounting for a circular mirror.

When temperature changes cause differential thermal expansion of the structure to which the flexures are attached relative to the mirror and cell assembly, radial motion of the mirror will be impeded without stressing the mirror. The only motion permitted because of expansion or contraction is a small rotation about the normal through the intersection of the lines of freedom. This occurs because of small changes in lengths of the flexures. The magnitude of this rotation, θ , in radians, may be approximated as:

$$\theta = (3^{1/2}) \alpha \Delta T, \tag{9.3}$$

where α is the CTE of the flexure material and ΔT is the temperature change.

Example 9.4: Rotation of a mirror on a flexure mounting about its axis as a result of temperature change. (For design and analysis, use File 9.4 of the CD-ROM.)

If the flexures of Fig. 9.32 are beryllium copper and ΔT is 10°F (5.5°C), what is the rotation θ ?

From Table B12: the CTE of BeCu is 9.9×10^{-6} /°F (17.8^{-6} /°C)

From Eq. 9.3: $\theta = (3^{1/2})(9.9 \times 10^{-6})(10) = 0.000017$ rad = 0.59 arcmin

This rotation is inconsequential in most applications.

Figure 9.33 shows a variation of the above flexure mounting concept in which the rim of a circular mirror is bonded centrally to three flexures oriented tangentially to that rim and anchored to the structure at both ends. Conceptually, this follows the same principle as flexure mountings for a lens discussed in Sect. 3.9. The flexure configurations of Figs. 3.43 through 3.48 are particularly similar. Minor bowing of the flexures may occur because of temperature changes if the flexure is not made of the same material as the mount or integral with that mount. In all cases, alignment of the flexures and of the mirror at the time of bonding would be facilitated by the design and use of appropriate fixturing.

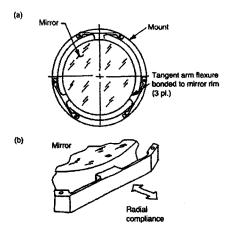


Figure 9.33 (a) Mounting for a circular mirror employing three tangential flexures supported at both ends. (b) Detail view. (Adapted from Vukobratovich and Richard.¹⁹)

Figure 9.34 schematically shows a possible interface between a cantilevered tangential flexure and the rim of a mirror. Here a square boss is attached (i.e., bonded) to the rim of the mirror. The free end of the flexure is bonded to this boss. In order to accommodate small misalignments between the bosses and the seats caused by manufacturing or assembly errors, sub-flexures are provided at four places in each flexure as indicated. The bonded interfaces to the bosses are at the squared-off ends of the subflexures.

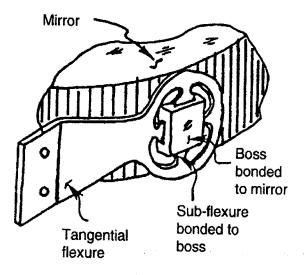


Figure 9.34 Concept for the interface between the free end of a cantilevered tangential flexure blade and a boss bonded to the rim of a circular mirror. (Adapted from Vukobratovich and Richard.¹⁹)

Figure 9.35 shows the mechanical design of a boss that might be bonded to the rim of a 15.0 in. (38.1 cm) diameter mirror. The material of such a boss should be selected to match its CTE with that of the mirror as closely as possible. For example, a boss made of Invar 36 might well be used with a ULE or Zerodur mirror. An adhesive such as EC2216B/A might be chosen for use with those materials. The adhesive would be injected through the 0.045 in. (1.14 mm) diameter access hole at the center of each boss. The adhesive thickness could be fixed by the use of shims or microspheres added to the adhesive. Flexures of the type shown in Fig. 9.34 might be interfaced with this boss.

Another configuration for a flexure that could interface with the boss of Fig. 9.35 is illustrated schematically in Fig. 9.36. This flexure might be made of 6Al4V titanium. The through hole at the left allows the flexure to be attached to a cylindrical structure (mirror cell). A small amount of rotation about the axis of that hole may be needed to align the square recess with the boss on the mirror. The square hole would be sized slightly larger than the mating boss to allow space for the appropriate thickness of adhesive (epoxy) to be inserted. Dimensions of the boss and the hole would need to be closely toleranced. A locating pin can be passed temporarily through the holes in the boss and at the center of the square recess to align the mating parts and to provide equal gaps on all sides of the boss. After the boss is fastened to the flexure, the pin would be removed so the adhesive can be injected into the joint to the mirror.

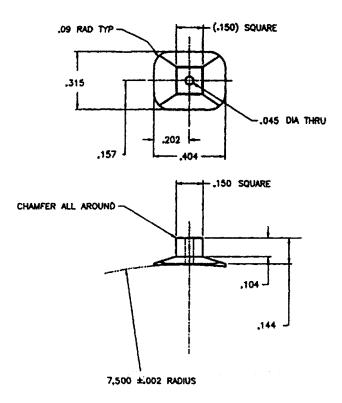


Figure 9.35 Design of a boss suitable for bonding to the rim of a 15.0 in. (38.1 cm) diameter mirror for mounting in the fashion depicted in Fig. 9.34.

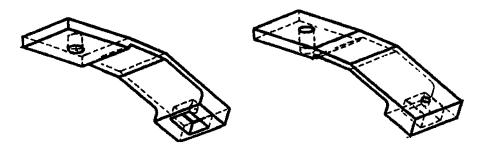


Figure 9.36 Schematic configuration of a cantilevered flexure shaped to interface with the boss of Fig. 9.35 and the cylindrical mounting for a circular mirror.

A mirror of rectangular shape might be supported in a cell attached to three deep cantilevered flexure blades as shown in Fig. 9.37. The dashed lines indicate the directions of freedom (approximated as straight lines). In this case, the intersection of these lines, which is stationary, does not coincide with the geometric center of the mirror or the center of gravity of this particular mirror and cell combination. By changing the angles of the corner bevels and relocating the flexures, the intersection point could be centralized and the design improved from a dynamic viewpoint. Differential thermal expansion across the mount-to-structure interface can then occur without stressing the mirror. Axial movement of the mirror is prevented by the high stiffness of the blades in that direction.

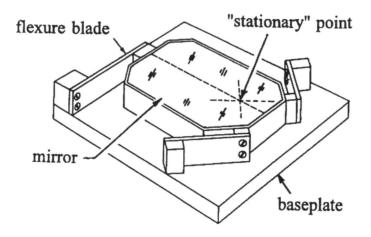


Figure 9.37 Concept for a flexure mounting for a rectangular mirror mounted in a cell with the cell supported from external structure by three flexures.

If the rectangular mirror is to be mounted without a cell, bosses configured as shown in Fig. 9.35, but with flat bonding surfaces might be attached directly onto the rim of the mirror. They would be attached in the same manner described for the circular mirror with bosses. A straight version of the flexure shown in Fig. 9.36 would then be used at three places to interface the mirror with structure.

Figure 9.38 shows schematically some other types of bosses, threaded studs, and flexures that have been successfully bonded to mirrors to allow them to be attached to the optical instrument structure. Those in View (a) are bonded into recesses or notches ground into the mirror substrate, while those in View (b) are bonded externally to the mirror surfaces.

Another concept for mounting a circular mirror with cantilevered tangential flexures is depicted in Fig. 9.39. Here the flexures are integral with the body of the ring shaped mounting. The flexures typically would be created by machining narrow slots using an electric discharge machining process. This mirror mounting is an extension of a design concept advanced for lens mounting discussed in Section 3.9. Once again, the blades are stiff in the tangential and axial directions and compliant radially as would be appropriate to negate decentrations that are due to temperature changes.

Modest-sized mirrors [for example, those in the 15- to 24-in. (38.1- to 61-cm) diameter range] or smaller mirrors used in high-precision, high-performance applications may benefit from being mounted in the manner shown in Fig. 9.40. Here a circular mirror with three bonded on bosses is attached to tangentially oriented arms containing dual sets of universal joint type flexures. Three axial metering rod-type supports that include flexures are also

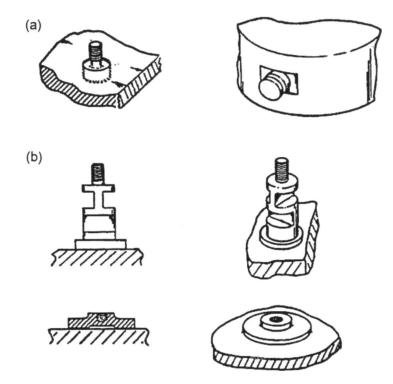


Figure 9.38 Illustrations of some bosses, threaded studs, and flexures that can be bonded to the rims or backs of mirrors for attachment to structure.

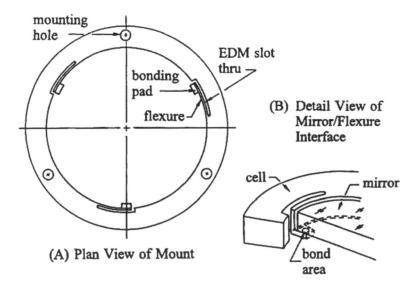


Figure 9.39 A mirror mounting with integral flexures. (Adapted from $\mathsf{Bacich.}^{^{20}})$

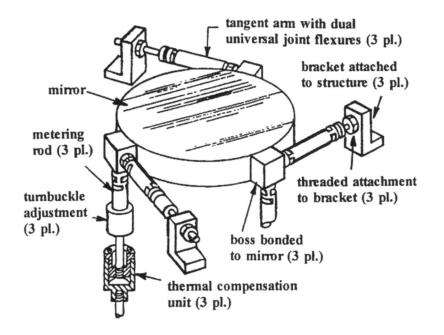


Figure 9.40 System of flexures configured to minimize displacement and/or distortion of the optical surface of a small to modest sized mirror caused by temperature changes and mounting forces. This mount also provides adjustments for all six degrees of freedom.

attached to the bosses. Such a mounting is essentially radially insensitive to temperature changes because of the action of the tangent arm flexures. The thermal compensation mechanisms shown in the axial supports make the design less sensitive in that direction to temperature changes. The latter mechanism consists of selected lengths of dissimilar metals arranged in a reentrant manner. The function of this type of athermalizing mechanism is explained in Section 14.1. Differential screws might be employed to advantage as the means for attaching the fixed ends of the tangent arms to the brackets in some applications. This would provide fine adjustment of the lengths of the tangent flexures. The "turnbuckle" mechanisms shown in the metering rods would facilitate axial adjustment. These could also be differential screws. Two-axis tilts of the mirror can be adjusted by differential motion of these axial mechanisms.

A quite different technique for mounting a mirror to a mechanical support is sketched in Fig. 9.41. Here a round mirror is bonded to three flexure blades that are in turn attached by screws, rivets, or adhesive to a circular mount of essentially the same diameter as the mirror. The flexures are flat so they can flex radially to accommodate differences in thermal expansion. They have the same free length, are of the same material, and are located symmetrically so thermally induced tilts and decentrations of the mirror are minimized. The local areas on both the mirror and mount where the flexures are attached are flattened in order to obtain adequate contact area for bonding and to prevent cupping of the springs. The blades should be as light and flexible as is consistent with vibration and shock requirements. Høg discussed a design of this type.²¹

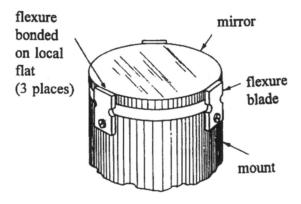


Figure 9.41 Sketch of a circular mirror mounting with axially directed, radially compliant flexures. (Adapted from Høg.²¹)

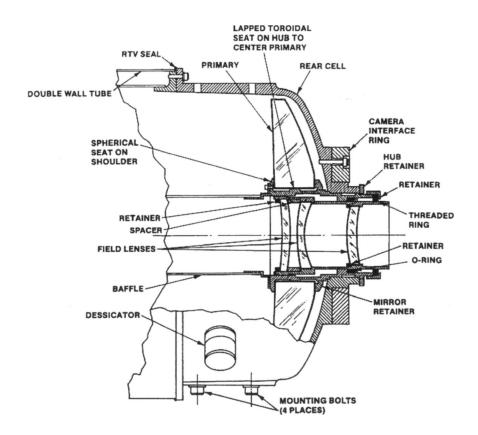


Figure 9.42 Sectional view of the back end of a catadioptric system featuring a hub mounting for the primary mirror. (From Yoder.³ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

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9.5 Central and Zonal Mountings

Some lightweighted mirrors are mounted on a hub that protrudes through a central perforation in the mirror substrate. An example is shown in Fig. 9.42. This is at the back end of a 150 in. (3.81 m) focal length (EFL), f/10 catadioptric objective that was used in a photographic application for tracking missiles.³ The system is discussed in Section 15.11. Both surfaces of the ~16 in. (~40.6 cm) diameter primary mirror are spherical; the first is the concave spherical reflecting surface while the second is shaped convex with radius smaller than that of the first to reduce weight. Figure 8.14(d) applies.

In this mounting design, the first mirror surface registers against a convex spherical seat on an integral shoulder of the hub. The seat radius is ground to match that of the mirror in the manner illustrated by Fig. 3.31(b). An annular toroidal land is provided on the cylindrical hub. The OD of this land is lapped to closely match the ID of the hole in the mirror at room temperature. A threaded retaining ring bearing against the flat bevel at the back of the mirror provides axial preload. In the original design of this instrument, Mylar shims were inserted into all axial glass-to-metal interfaces to help achieve close contact between the optical component surfaces and the less perfect mechanical surfaces. In Section 15.18, an alternative way to interface this mirror to its mount is explained.

Smaller sized single-arch lightweighted mirrors also are center mounted on hubs since their rims typically are very thin and so lack adequate strength to support the mirror. Designs typically follow the general lines of the mounting shown in Fig. 9.42. More sophisticated hub mounting designs might be used with mirrors in the 25 to 40 in. (0.6 to 1 m) range since those mirrors usually are more flexible and hence more susceptible to gravitational effects. Mirrors larger than this upper limit are generally not suited for central mounting.

Vukobratovich²³ reported that a significant drawback in hub mounting a single arch mirror may arise if the center of gravity is forward of the vertex of the optical surface. The hub mount then cannot provide support in the plane containing that important point and astigmatism is introduced at and near the axis horizontal position.

A concept for mounting a single arch glass mirror to a hub with a spherical clamp interfacing with a conical interface in the mirror is shown in Fig. 9.43. It was one of several concepts suggested for possible use in mounting the 85-cm (33.5-in.) diameter primary mirror of the Space Infrared Telescope Facility (SIRTF) [now the Spitzer Space Telescope]. At the time, both metallic and nonmetallic materials for the mirror were being considered. The mirror actually used in that telescope is beryllium. It is considered in Chapter 10.

The clamp constrains the mirror in all six degrees of freedom near its CG at its thickest and strongest section with a large area contact, thereby reducing stresses that could disturb the optical surface. A flat surface is provided on the back of the mirror; the axis of the cone is perpendicular to that surface and the apex of the cone coincides with it. A metal insert with an external conical surface and a concave spherical inner surface fits into the substrate's conical surface. The hub has a convex spherical surface forming a bearing in conjunction with the insert. This spherical bearing ensures contact of the conical surfaces. Slight slippage can occur between the conical surfaces, but expansion/contraction of the mount will not alter coincidence of the cone apex with the plane of the mirror's back.

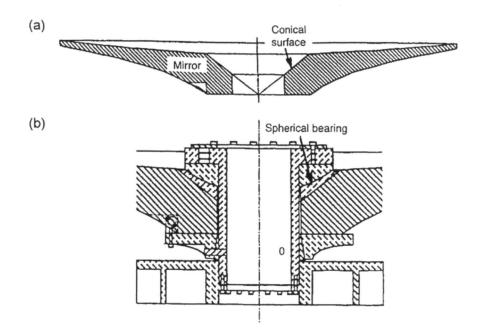


Figure 9.43 Schematic of a hub mounting for a single arch mirror featuring a conical optomechanical interface. (Adapted from Sarver et al.²²)

Double arch lightweighted mirrors are typically mounted on three or more supports attached to the back surface of the substrate at its thickest point. Figure 9.44 shows one design concept. The mirror is supported by a combined ring/point support system. In the zenith looking position, it rests on a three segment air bag support ring contacting the mirror back surface at the base of the thickest zone. As the mirror axis depresses toward the horizon, mirror weight is progressively picked up by three or more radial supports inserted into sockets in the mirror substrate. At the axis horizontal position, the mirror weight is supported entirely by the radial supports. They act in the plane through the mirror center of gravity to minimize optical surface deformation. The radial support shown here is of the general type discussed in more detail in Section 11.1. The air bag ring support is similar to ones discussed in Section 11.4.3.

A more sophisticated mounting for a double arch mirror is shown in Fig. 9.45. This design was created to support a 20-in. (50.8-cm) diameter double arch mirror with three equally spaced clamp and flexure assemblies oriented so the flexures were compliant in the radial direction, but stiff in all other directions. This allowed the aluminum mounting plate to contract differentially with respect to the fused silica mirror as the temperature was lowered to about 10K. Each clamp was a "tee" shaped Invar 36 part that engaged a conical hole in the annulus of the mirror's back surface. The twin parallel flexures were 91-mm (3.6-in.) long by 15-mm (0.6-in.) wide and were made of 0.04-in. (1.0-mm) thick 6Al4V-ELI titanium. The blades were separated by 25 mm (1in.). This mounting design was analyzed extensively and found to provide acceptable thermal performance and to withstand launch loads typical of the Space Shuttle as well as to survive (with some damage) a crash landing of the Shuttle.²³

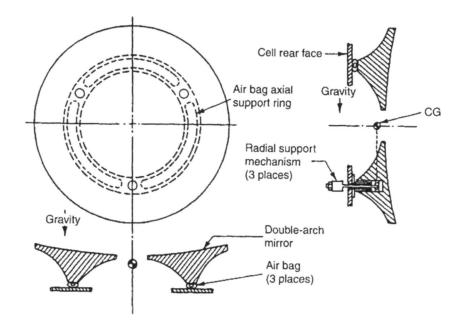


Figure 9.44 Schematic of a combined air bag axial and three point radial support system for a double arch mirror. (From Vukobratovich.⁶)

9.6 Gravitational Effects on Smaller Mirrors

So far in this chapter we have paid little attention to the effects of external forces such as gravity and operational accelerations on mirror surface figure. When the aperture is modest, the thickness and material choice conducive to stiffness, and the performance requirements not too high, the optic can be considered a rigid body and mounted semikinematically or even nonkinematically without excessive performance problems. If, however, any or all of these attributes do not prevail, we must consider the effects of external forces. Gravity is the most prevalent, so we will limit our discussion to that force, frequently called "self weight deflection." A special case is gravity release in space and the related problems of making and mounting a mirror so it does not become distorted when normal gravitational force is missing. That aspect of the problem will be considered in Section 11.4.

The largest gravitational disturbances occur when the mirror axis is vertical. The way the mirror is held then affects the magnitude of surface deformation and the resulting surface contour. Let us consider two cases: (a) a circular mirror simply supported around its rim and (b) a rectangular mirror simply supported at its rim. Using Roark's theory for flexure of simple unclamped plates under uniform gravitational load normal to the plate's surface,² we can derive the following equations for the deflections:

$$\Delta y_{\rm CIRC} = \frac{(3W)(m-1)(5m+2)(a^2)}{\left(16\pi E_G m^2 t_A^3\right)},\tag{9.4}$$

$$\Delta y_{\text{RECT}} = \frac{0.1442wb^4}{\left[E_G t_A^3 \left(1 + 2.21\xi^3\right)\right]},\tag{9.5}$$

where W is the total mirror weight, w is its weight/area, m is (1/Poisson's ratio), E_G is Young's modulus, a is the semidiameter or longest dimension, b is the shortest dimension, t_A is the thickness, ξ is b/a, and mirror area = πa^2 (if circular) or ab (if rectangular).

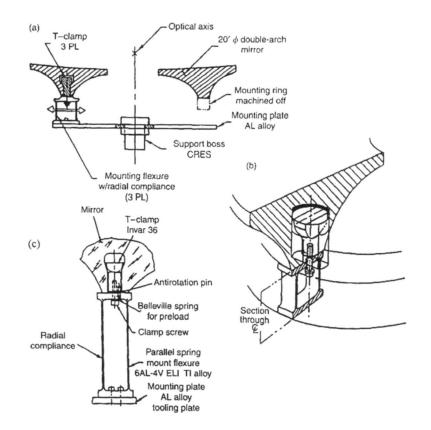


Figure 9.45 A mounting design for a double arch mirror. (a) Sectional view, (b) isometric view of one clamp and flexure mechanism, and (c) sectional view through the latter mechanism. (From Iraninejad et al.²⁴)

The induced sags (Δy_i) are measured at the mirror center and represent changes in sagittal depth of the optical surface if the mirror is not flat. Figure 9.46 shows the geometry. Examples 9.5 and 9.6 will help the reader assess typical self weight mirror deflection magnitudes for specific applications.

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If the mirrors are nominally flat, we would expect that contour lines of an equal change in sag on the deflected circular mirror's reflecting surface would be circles while those for the rectangular mirror would be generally elliptical. If the same circular mirror were supported at three points rather than all around the rim, and if those points were located at different radial zones of the mirror's aperture, the contours of the surfaces would appear essentially as indicated in Fig. 9.47. Crosses indicate the support points. Although originally drawn for a large perforated mirror, the same patterns would be expected in smaller mirrors; only the scale would change. The same general effects would be expected if the mirror were rectangular. The surface deformation contours would then be modified elliptical lines.

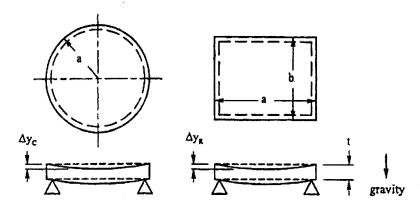


Figure 9.46 Geometry of circular and rectangular mirrors supported at their rims.

There is an optimum zonal radius for three point support of a circular mirror of constant thickness *t* that gives minimum surface deflection. A minor rewriting of an equation by Vukobratovich²⁵ gives Eq. (9.6) as an approximation for this deflection. He indicated that this condition occurs at $R_E = 0.68$ times the mirror radius, R_{MAX} . Supporting the back of an upward looking axis vertical mirror on a circle of this radius produces a "hole and downward rolled edge" contour under the influence of gravity. The surface also is deformed in a six lobed figure, generally as indicated in Fig. 9.47(b).

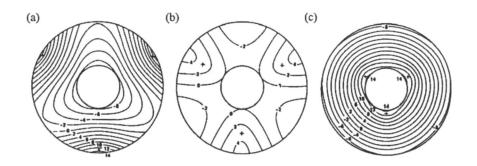


Figure 9.47 Contour patterns for a circular mirror supported on three points at different zonal radii: (a) At 96%, (b) at 73%, and (c) at 38%. Contour intervals apply to a 4 m (158 in.) diameter solid mirror. (From Malvick and Pearson.²⁶)

Example 9.5: Gravity-induced deflections of an axis vertical circular mirror. (For design and analysis, use File 9.5 of the CD-ROM.)

A 20.000 in. (50.800 cm) diameter (D_G) plane parallel, fused silica mirror is uniformly supported around its rim with its axis vertical. Assume diameter/thickness equals 6. Calculate its self weight deflection in waves of red light [$\lambda = 2.49 \times 10^{-5}$ in. (0.6328 µm)].

From Table B5, for fused silica: $\rho = 0.0796 \text{ lb/in/}^3 (2.202 \text{ g/cm}^3), E_G = 10.6 \times 10^6 \text{ lb/in.}^2 (7.3 \times 10^4 \text{ MPa}), \text{ and } \nu_G = 0.17.$ $m = \frac{1}{0.17} = 5.882, a = \frac{20.000}{2} = 10.000 \text{ in.} (25.4 \text{ cm}), t_A = \frac{20.000}{6} = 3.333 \text{ in.} (8.467 \text{ cm}),$ $W = \pi (10.000^2)(3.333)(0.0796) = 83.349 \text{ lb} (37.807 \text{ kg}).$ From Eq. (9.4): $\Delta y_{\text{CIRC}} = \frac{(3)(83.349)(5.882 - 1)[(5)(5.882) + 1](10.000^2)}{(16\pi)(10.6 \times 10^6)(5.882^2)(3.333^3)} = 5.44 \times 10^{-6} \text{ in.}$ $= 5.44 \times 10^{-6}/2.49 \times 10^{-5} = 0.22 \lambda_{\text{RED}}.$

Example 9.6: Gravity-induced deflections of an axis vertical rectangular mirror. (For design and analysis, use File 9.6 of the CD-ROM.)

A rectangular plane parallel, fused silica mirror with dimensions a = 20.000 in. (50.800 cm) and b = 12.5 in. (31.750 cm) is uniformly supported around its rim with its axis vertical. Assume largest dimension/thickness equals 6. Calculate its self weight deflection in waves of red light [$\lambda = 2.49 \times 10^{-5}$ in. (0.6328 µm)].

From Table B5, for fused silica: $\rho = 0.0796 \text{ lb/in}^3 (2.202 \text{ g/cm}^3) \text{ and } E_G = 10.6 \times 10^6 \text{ lb/in.}^2 (7.3 \times 10^4 \text{ MPa}).$

 $\xi = 12.500/20.000 = 0.625,$ $t_A = 20.000/6 = 3.333$ in. (8.467 cm), W = (20.000)(12.500)(3.333)(0.0796) = 66.327 lb (30.085 kg), and w = 66.327/[20.000)(12.500)] = 0.265 lb/in.². From Eq. (9.5):

$$\Delta y_{\text{RECT}} = \frac{(0.1442)(0.265)(12.500^4)}{(10.6 \times 10^6)(3.333^3)([1+(2.21)(0.625^3)])} = 1.52 \times 10^{-6} \text{ in.} (3.86 \times 10^{-5} \text{ mm}).$$
$$= 1.52 \times 10^{-6}/2.49 \times 10^{-5} = 0.06 \lambda_{\text{RED}}$$

$$\Delta y_{\rm MIN} = 0.343 \rho \left(R_{\rm MAX}^{4} \right) \frac{(1 - \nu^2)}{E_G t^2}, \qquad (9.6)$$

where ρ is density, R_{MAX} the mirror radius, ν is Poisson's ratio, E_G is Young's modulus, and *t* is thickness. Example 9.7 illustrates the use of this equation.

Example 9.7: Circular mirror surface sag condition for three point zonal support. (For design and analysis, use File 9.7 of the CD-ROM.)

A circular plane parallel mirror of diameter $D_G = 20.000$ in. (50.800 cm) is supported at three points equally spaced at the $0.68R_{MAX}$ zone. The diameter-tothickness ratio is 5. If the material is ULE, what would be the sag of the mirror at its center? Express the result in waves for $\lambda = 0.6328 \ \mu m (2.49 \times 10^{-6} \text{ in.})$.

From Table B8a: $\rho = 0.080 \text{ lb/in.}^3$, $v_G = 0.17$, and $E_G = 9.8 \times 10^6 \text{ lb/in.}^2$

$$=\frac{20.000}{1000}$$
 = 4.000in. (10.160 cm)

From Eq. (9.6):
$$\Delta y_{\text{MIN}} = \frac{(0.343)(0.080)\left(\frac{20.000}{2}\right)^4 (1-0.17^2)}{\left[(9.8 \times 10^6)(4.000^2)\right]}$$

$$= 1.7 \times 10^{-6} \text{ in.} = \frac{1.7 \times 10^{-6}}{2.49 \times 10^{-6}} - 0.68 \lambda_{\text{RED}}$$

If the three equally spaced zonal supports are moved to the edge of the mirror, the deflection at the mirror's center is increased from the minimum value derived from Eq. (9.6) by a factor of about 3.9. The shape of a nominally flat mirror would then resemble a dish, i.e., high at the rim and low at the center. The rim would also droop significantly between the supports, as indicated in Fig. 9.47(a).

If the three equally spaced zonal supports were moved close to the center of the mirror, its rim would droop nearly equally all around as indicated in Fig. 9.47(c). The almost symmetrical appearance of this pattern leads one to believe that refocusing of the system would at least partially compensate for the gravity effect and improve the image. It should be noted that gravity effects on circular mirrors vary approximately as the cosine of the angle between the local gravity vertical and the axis of symmetry of the mirror.

Adding axial support points would reduce the magnitude of the surface distortion for any mirror. Multipoint supports with nine, eighteen, thirty six, etc., support points acting through three or more symmetrically located lever mechanisms might be used. These mounts are called "Hindle" mounts in recognition of the 1945 contribution of J.H. Hindle.²⁷ He divided the frontal area of a uniform thickness circular plate into a central disk of one third the total area and an annulus of two thirds that area. Three supports from structure were placed near this interface. For a nine point mount, three and six supports lie on inner and outer circles of radius R_1 and R_0 , respectively. Equations (9.7) through (9.9) and (9.10a) then apply. Each set of three support points is connected by a triangular plate that is pivoted at a point one third of the way up the triangle's altitude from its base. Each contact then

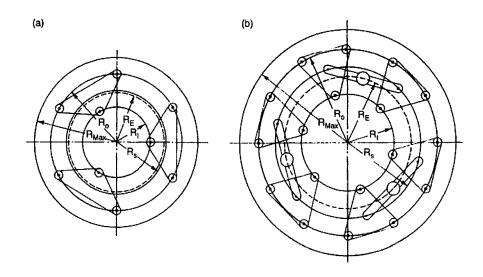


Figure 9.48 Multipoint (Hindle) mechanical axial support configurations for (a) nine-point and (b) eighteen-point mounts.

$$R_E = \left(\frac{1}{6}\right) (3)^{1/2} D_G = 0.2887 D_G, \tag{9.7}$$

$$R_{I} = \left(\frac{1}{2}\right) \left(\frac{1}{6}\right)^{1/2} \left(D_{G}\right) = 0.2041 D_{G},$$
(9.8)

$$R_o = \left(\frac{1}{2}\right) \left(\frac{2}{3}\right)^{1/2} \left(D_G\right) = 0.4082 D_G,$$
(9.9)

$$R_{s\,9} = \left[\left(\frac{1}{6}\right) \left(\frac{1}{6}\right)^{1/2} \left(D_G\right) \right] + \left[\frac{(2R_o \cos 30 \deg)}{3} \right] = 0.3037 D_G, \tag{9.10a}$$

$$R_{s\ 18} = \left[\left(\frac{1}{6}\right) \left(\frac{1}{6}\right)^{1/2} \left(D_G\right) \right] + \left[\frac{(2R_o\ \cos 15\ \text{deg})}{3}\right] = 0.3309 D_G, \tag{9.10b}$$

carries an equal share of the total weight. Equations (9.7) through (9.9) and (9.10b) apply to the eighteen point mount of Fig. 9.48(b). In both cases, $D_G = 2R_{MAX}$. Hindle's equations were refined slightly in 1996.²⁸ See also Mehta.²⁹ The lever mechanism used in the eighteen point mount is commonly termed a "whiffletree" from its similarity to the harness configuration used to tie two beasts of burden to a single load. See Fig. 9.49.

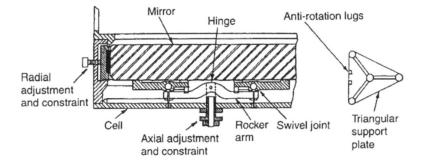


Figure 9.49 Sketches of one whiffletree mechanism for an eighteen point Hindle mount. (Adapted from Hindle.^{27,28} Copyright 1996 by Jeremy Graham Ingalls and Wendy Margaret Brown.)

Hindle mounts with 36-point supports are used to mount the segments of the Keck Telescopes and the primary mirror of the SOFIA Telescope. These systems are described in Section 11.2.

9.7 References

- 1. Durie, D.S.L., "Stability of optical mounts," Machine Des. 40, 1968:184.
- Roark, R.J., Formulas for Stress and St rain, 3rd ed., McGraw-Hill, New York, 1954. See also Young, W.C., Roark's Formulas for Stress & Strain, 6th ed., McGraw-Hill, New York, 1989.
- 3. Yoder, P.R., Jr., *Opto-Mechanical Systems Design*, 3rd ed., CRC Press, Boca Raton, 2005.
- 4. Lipshutz, M.L., "Optomechanical considerations for optical beamsplitters," *Appl. Opt.* 7, 1968:2326.
- 5. Yoder, P.R., Jr., Chapt. 6 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, 1997.
- 6. Vukobratovich, D., Introduction t o Opt omechanical Desi gn, SPIE Short Course SC014, 2003.
- Hookman, R., "Design of the GOES telescope secondary mirror mounting," *Proceedings of SPIE* 1167, 1989:368.
- 8. Strong, J., Procedures in Applied Optics, Marcel Dekker, New York, 1989.
- 9. Shigley, J.E. and Mischke, C.R., "The design of screws, fasteners, and connections," Chapter 8, in *Mechanical Engi neering Design*, 5th ed., McGraw-Hill, New York, 1989.
- 10. Yoder, P.R., Jr., "Nonimage-forming optical components," *Proceedings of SPIE* 531, 1985:206.
- 11. See, for example, certified particle products made by Duke Scientific Corp. (www.dukescientific.com).
- Mammini, P., Holmes, B., Nordt, A., and Stubbs, D., "Sensitivity evaluation of mounting optics using elastomer and bipod flexures," *Proceedings of SPIE* 5176, 2003:26.

- 13. Mrus, G.J., Zukowski, W.S., Kokot, W., Yoder, P.R., Jr., and Wood, J.T., "An automatic theodolite for pre-launch azimuth alignment of the Saturn space vehicles," *Appl. Opt.* 10, 1971: 504.
- 14. Patrick, F.B., "Military optical instruments," Chapter 7 in Applied Optical and Optical Engineering V, Academic Press, New York, 1969.
- 15. Yoder, P.R., Jr., "High precision 10-cm aperture penta and roof-penta mirror assemblies," Appl. Opt. 10, 1971:2231.
- 16. Lyons P.A. and Lyons, J.J., private communication, 2004.
- 17. Bridges, R. and Hagan, K., "Laser tracker maps three-dimensional features," *The Industrial Physicist* 28, 2001:200.
- 18. Yoder, P.R., Jr., "Study of light deviation errors in triple mirrors and tetrahedral prisms, J. Opt. Soc. Am. 48, 1958:496.
- 19. Vukobratovich, D. and Richard, R., "Flexure mounts for high-resolution optical elements," *Proceedings of SPIE* **959**, 1988: 18.
- 20. Bacich, J.J., Precision Lens Mounting, U.S. Patent 4,733,945, 1988.
- 21. Høg, E., "A kinematic mounting," Astrom. Astrophys. 4, 1975:107.
- 22. Sarver, G., Maa, G., and Chang, L., "SIRTF primary mirror design, analysis, and testing," *Proceedings of SPIE* 1340, 1990:35.
- 23. Vukobratovich, D., private communication, 2004.
- 24. Iraninjad, B., Vukobratovich, D., Richard, R., and Melugin, R., "A mirror mount for cryogenic environments," *Proceedings of SPIE* **450**, 1983:34.
- 25. Vukobratovich, D., "Lightweight Mirror Design," Chapter 5 in Handbook of Optomechanical Engineering, CRC Press, Boca Raton, 1997.
- 26. Malvick, A.J. and Pearson, E.T., "Theoretical elastic deformations of a 4-m diameter optical mirror using dynamic relaxation," *Appl. Opt.* 7, 1968:1207.
- 27. Hindle, J.H., "Mechanical flotation of mirrors," in *Amateur Telescope Making, Book One*, Scientific American, New York, 1945.
- 28. The three volume *Amateur Tel escope Maki ng* series was rearranged, discretely clarified, and republished in 1996 by Willmann-Bell, Inc., Richmond, VA.
- 29. Mehta, P.K., "Flat circular optical elements on a 9-point Hindle mount in a 1-g force field," *Proceedings of SPIE* **450**, 1983:118.

CHAPTER 10 Techniques for Mounting Metallic Mirrors

The design of metallic mirrors was discussed in Section 8.8 of this work. Because the design of the mirror itself and the design of its mounting are closely interrelated in many cases, examples of a few mountings were also described in that section. For example, Figs. 8.41 showed how the aluminum secondary mirror for the Kuiper Airborne Observatory was attached with screws to the hub of a tip/tilt drive mechanism. Providing a proper interface between the mirror and the hub and the requirement for minimal weight along with high stiffness so the mechanism could move the mirror at high acceleration rates were key features of the mirror design. Similarly, the mountings for the beryllium primary segments of the James Webb Space Telescope, described in conjunction with Fig. 8.45, are closely tied to the design of the mirrors.

In this chapter, we delve more deeply into the specifics of mounting metallic mirrors. First, we consider how metal mirrors and their interfaces with the mountings can be shaped precisely by single point diamond turning. Then, we consider integral mountings wherein features of the mirror substrate are configured to attach directly to the mechanical support. Provision of flexures in the mounting features of larger metallic mirrors is then described. These flexures serve to minimize optical surface distortion effects due to mounting forces. We consider how platings applied to the mirror surfaces to provide suitable material for polishing or diamond turning affect the optical behavior of the mirror when the temperature changes. Heat transfer through the mechanical interface plays a key role in such cases. Finally, we describe how metallic mirrors and their mounts can be configured to facilitate assembly and alignment.

10.1 Single Point Diamond Turning of Metallic Mirrors

Single crystal diamond cutting tools and specialized machinery are used to create flat or curved surfaces on a variety of materials by very accurately cutting away thin layers of the surface. This process is variously called "single point diamond turning," "precision machining," or "precision diamond turning." We here adopt the first terminology and abbreviate it as SPDT. The process has developed from crude experiments to fully qualified production processes since the early 1960s. See, for example, Saito and Simmons,¹ Saito,² Sanger,³ and Rhorer and Evans.⁴

The SPDT process generally involves the following steps: (1) preform or conventionally machine the part to rough shape with approximately 0.1 mm (0.004 in.) excess material left on all surfaces that will be processed, (2) heat treat the part to relieve stress, (3) mount the part with minimal induced stress in an appropriate chuck or fixture on the SPTD machine, (4) select, mount, and align the diamond tool on the machine, (5) finish machine the part to final shape and surface quality with multiple light cuts under computer control, (6) inspect the part (in situ, if possible), and (7) clean the part to remove cutting oils and/or solvents. For some applications, plating the surfaces following step (2) is required to provide an amorphous layer of material to be diamond turned. Sometimes step (7) is followed by polishing the optical surface or surfaces to smooth it or them. An appropriate optical coating may then be applied if required for the application.

SPDT equipment can be defined as an "instrument" since it unquestionably meets the classical definition of Whitehead⁵ which states: "an instrument may be defined as any mechanism whose function is directly dependent on the accuracy with which the component parts achieve their required relationships." In the present case, that accuracy is achieved, in part, from inherent mechanical rigidity and freedom from self generated and external vibration and thermal influences. Predictability and high resolution of rotary and linear motions and long lifetime characteristics of its mechanisms are inherent attributes of a good SPDT machine design.

The SPDT process creates a periodic grooved surface that scatters and absorbs incident radiation. Figure 10.1(a) is a greatly magnified schematic view of the localized contour of the turned surface in a machine functioning in a facing operation. Figure 10.2 illustrates this operation. The inherent roughness of the substrate surface before SPDT is depicted at left in Fig. 10.1(a). The diamond tool has a small curved nose of radius R. The motion of the tool across the surface creates parallel grooves as indicated on the right of that figure. The theoretical p-v height h of the resulting cusps is given by the following simple equation involving the parameters designated in Fig. 10.1(b):

$$h = \frac{f^2}{8R},\tag{10.1}$$

where f is the transverse linear feed of the tool per revolution of the surface. For example, if the spindle speed is 360 rpm, the feed rate is 8.0 mm/min, and the tool radius is 6 mm, the value for f = 8.0/360 = 0.0222 mm per revolution. Hence, $h = (0.0222)^2/[(8)(6)] = 1.03 \times 10^{-5}$ mm or 1.03 Å. Note that the width of each cusp equals f.

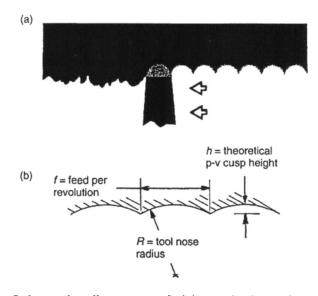


Figure 10.1 Schematic diagrams of (a) a single point diamond tool advancing from right to left across the surface of a substrate and (b) the geometry of the surface before (left side) and after (right side) passage of the tool in the SPDT process.

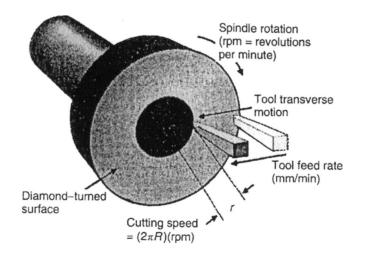


Figure 10.2 Representation of a SPDT facing operation.

The tool of a SPDT machine must follow an extremely accurate path relative to the surface being cut throughout the procedure. Rhorer and Evans⁴ listed several factors as sources of errors in the machined surface contour beyond the grooved structure just described. These are paraphrased as: (1) waviness from inaccuracies of travel of the mechanisms providing tool motions; (2) nonrepeatability of axial, radial, and/or tilt motions relative to the spindle rotation; (3) external and/or self generated vibration; (4) effects within the turned material wherein differential elastic recovery of adjacent material grains and/or impurities cause contour "steps" or "orange peel" appearance in the surface; and (5) contour defects within the cusps because of irregularities in the shape of the tool's cutting edge.

The earliest applications of SPDT were to IR optics because they could have rougher and less accurate surface contours than ones for use at shorter wavelengths. Recent advances in SPDT technology allow optics to be made smooth enough to be used in visual and lower performance UV instruments. Vukobratovich⁶ indicated that the surface microroughness typically achieved in quantity production on bare 6061 aluminum mirror surfaces is 80 to 120 Å rms while that achieved on surfaces plated with amorphous material is about 40 Å rms.

Materials that can be machined by the SPDT process with more or less success are listed in Table 10.1. Compatibility with the process is often an expression of practicality. Some materials such as ferrous metals, electrolytic nickel, and silicon *can* be diamond turned, but wear the cutting tools rapidly. The process is not generally considered economical for machining those materials. Some alloys of listed generic metals can be processed satisfactorily by SPDT while others are unsatisfactory. For example, good surfaces can be cut on 6061 aluminum while 2024 aluminum typically produces poor surfaces. Ductile materials (those hard to polish) are generally compatible with SPDT, whereas brittle materials polish easily but are not suitable for SPDT. In some cases, brittle materials can be processed to high precision on a SPDT machine by substituting a fine grinding head for the diamond cutter.

aluminum	calcium fluoride	mercury cadmium telluride
brass	magnesium fluoride	chalcogenide glasses
copper	cadmium fluoride	silicon (?)
beryllium copper	zinc selenide	polymethylmethacrylate
bronze	zinc sulfide	polycarbonate
gold	gallium arsenide	Nylon
silver	sodium chloride	polypropylene
lead	calcium chloride	polystyrene
platinum	germanium	polysulfone
tin	strontium fluoride	polyamide
zinc	sodium fluoride	ferrous metals (?)
ELN (K > 10%)	KDP	
EN (?)	KTP	

Table 10.1 Materials that usually can be diamond turned.

Note: Materials with (?) cause rapid diamond wear.

Dahlgren and Gerchman⁸ reported that plate, rolled, extruded, or forged wrought forms of metals are most commonly used for SPDT, but considerable success has been achieved in the diamond turning of carefully prepared near-net-shape castings of types 201-T7, 713-T5, and 771-T52 aluminum alloys. They also indicated that to present a homogeneous substrate to the diamond, a casting must be made of virgin, metallurgically pure raw material (<0.1% impurity), its hydrogen content must be <0.3 ppm, the material handling equipment and gating system must not increase the impurity level of the raw material, and the rate of casting solidification must be carefully controlled so cooling occurs isotropically from the optical surface inward. Similar success with SPDT machining of aluminum castings was reported by Ogloza et al.⁹ The latter paper included comparisons of different aluminum alloys and different operational set ups of the SPDT instrument.

Polycrystalline materials may not machine well because their grain boundaries may be emphasized by the cutting action of the tool. Gerchman and McLain¹⁰ described their investigation of SPDT results when diamond turning various forms of single crystal and polycrystalline germanium. For IR applications, the differences in results were found to be insignificant and the presence of grain boundaries did not seem to cause brittle fracture to surfaces.

Gerchman⁷ gave a fairly complete summary of specifications and manufacturing considerations for SPDT optics including guidelines for selecting and specifying material characteristics, translating optical surface descriptions from the lens designer's terminology into SPDT machine terminology, surface tolerancing, evaluation techniques (including the use of null compensators), controlling surface texture (tool marks) and lay orientations, minimizing and measuring surface errors and cosmetic defects, as well as clarification as to which U.S. MIL specifications to apply in measuring such flaws.

Sanger³ provided a very thorough review of SPDT developmental history and technology, including machine design features and capabilities, work piece support techniques, diamond tool characteristics, numerical (computer) control systems, environmental control, work piece preparation, machining operation guidelines (depth of cut, speeds and feeds, tool wear), and techniques for testing of finished surfaces.

There are two basic types of SPDT instruments: the lathe type, in which the work piece rotates and the diamond tool translates, and the fly cutter type, in which the tool rotates and the work piece translates. Parks¹¹ described fourteen different geometries of SPDT instruments capable of creating cylinders, exterior and interior cones, flats, spheres, toroids, and aspheres. Here, we will consider four of those geometries.

Figure 10.3(a) shows a lathe type SPDT instrument with the linear tool axis oriented parallel to the spindle axis. This resembles the conventional metal working lathe. The work piece is mounted between live and dead centers (as shown) or cantilevered from a faceplate on the spindle. By appropriate fixturing, the diamond tool can also be positioned to turn the inside diameter of a hollow cylinder. If the linear tool slide is rotated about a vertical axis to lie at a horizontal angle to the spindle axis, this instrument can be used to machine external or internal cones.

Figure 10.3(b) illustrates one form of fly cutting SPDT instrument. Here the work piece is a flat generated by a single or multiple series of slightly displaced parallel curved tool cuts. If the spindle axis is not accurately perpendicular horizontally to the linear ways, the surface becomes cylindrical. Faceted scanner mirrors are machined by a variation of this geometry in which the work piece is indexed about an axis inclined with

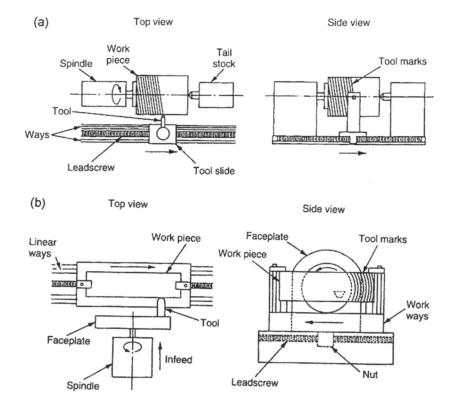


Figure 10.3 Schematics of (a) a lathe type SPDT machine used to turn cylindrical surfaces and (b) a fly cutter SPDT machine used to turn flat surfaces. (From Parks.¹¹)

respect to both the spindle and linear ways. The design and manufacture, by SPDT techniques, of precision polygon scanners were discussed in depth by Colquhoun et al.¹² The only practical method of machining scanner mirrors directly into a substrate with closely spaced, flat internal reflecting facets is through use of the SPDT process.

Another type of fly cutter SPDT instrument, this one designed to create spherical surfaces, is illustrated in Fig. 10.4. Here two rotary motions about coplanar and intersecting axes carry both the work piece and the tool. The radius *R* created equals $r/\sin \theta$, where these parameters are as depicted in the figure. The function is similar to that of a diamond cup surface generating machine as used in mechanical and optical shops. As shown, a concave surface is machined; a convex one is created if the point *C* representing the intersection of the two axes is moved behind (i.e., to the left of) the work piece. This is generally accomplished by supporting the tool on a yoke with arms passing on both sides of the work piece spindle. If the tool is mounted on a linear feed rotating about the axis through point *C* of this figure, the configuration is called a "*R*- θ " instrument. It cuts aspherical as well as spherical surfaces.

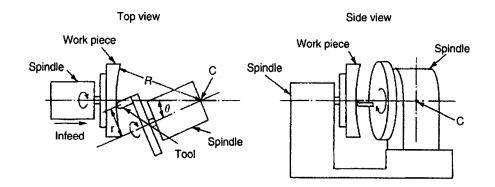


Figure 10.4 Schematic of a dual rotary axis fly cutting SPDT machine configured to cut concave spherical surfaces. (From Parks.¹¹)

Gerchman¹³ described the capability of a four axis system involving three linear motions (X, Z, and Z') plus rotation of the work piece with encoder readout of rotational position. The Z' axis of this system is a limited, rapid linear motion of the diamond tool. By coordinating the motion of the tool with the work piece's rotational position, a nonaxisymmetric surface can be generated. Gerchman¹³ also described how such a machine might be used to create off-axis aspheric mirror surfaces.

Figure 10.5 shows an instrument with four axes: two rotary and two linear, the latter being stacked vertically. Rotation of the tool occurs about the center of the circular cutting edge through action of a stepper motor. This motor is indexed after individual or a few rotations of the work piece spindle. This keeps the tool normal to the work piece and eliminates errors due to variation of radius, hardness, and finish along the tool's cutting edge. It also allows a tool with a shorter radius to be used, thus allowing greater slope variation on the machined surface. With this number of motions, convex and concave aspheric surfaces can be machined.

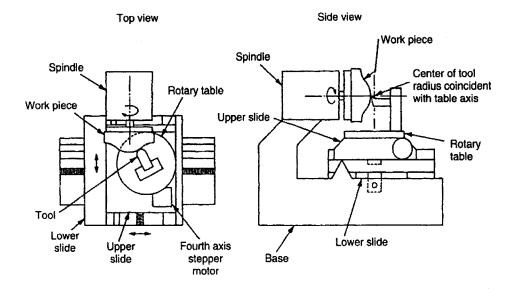


Figure 10.5 Schematic of a SPDT machine with four axes, stacked linear slides, and stepwise control of the diamond tool's orientation relative to the work piece. (From Parks.¹¹)

Figure 10.6 shows a way in which five-axis machining can be implemented in a SPDT instrument. The X and Y axes are separated rather than stacked as in Figure 10.5.

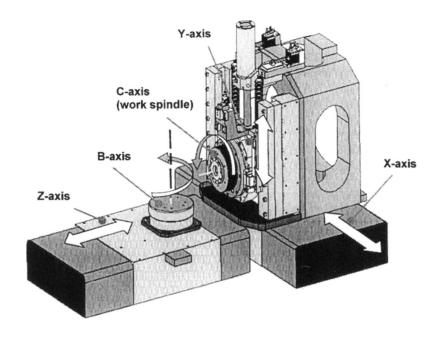


Figure 10.6 A five-axis SPDT instrument configuration.



Figure 10.7 Photograph of an ultra-precision, free-form SPDT/grinding machine, the Moore NANOTECH Model 350FG. (Courtesy of Moore Nanotechnology Systems, Keene, NH.)

A variety of single and multiple axis SPDT machines with different work piece size capacities and surface contour capabilities are available from various manufacturers worldwide. For example, Figure 10.7 shows a three- to five-axis SPDT instrument. It represents the current state of the art in commercially available production instruments. Key specifications and capabilities of this instrument, the NANOTECH 350FG Ultra-Precision Freeform Generator produced by Moore Nanotechnology Systems, LLC of Keene, NH, are summarized in Table 10.2.

Table 10.2 Key features and capabilities of the NANOTECH 350FG SPDT/grinding machine shown in Fig. 10.7.

Genera	ll Features:	
٠	Monolithic cast epoxy-granite structure with three-point vibration isolation	
•	Mounted in a NEMA 12 cabinet	
•	Delta Tau PC-based CNC motion contol, Windows operated	
•	Linear axis travels: $X = 350 \text{ mm}$; $Y = 150 \text{ mm}$; $Z = 300 \text{ mm}$	
•	Workpiece capacity: 500-mm diameter, 300-mm long	
•	Laser holographic linear axis, athermally mounted	
•	Closed-loop liquid coolant system for temperature control within ±0.5°F	
Perfor	mance:	
•	Motion accuracies: ≤ 25 nm (axial and radial)	
٠	Programming resolution: 1 nm (linear), 0.00001-deg rotary	
٠	Surface error: $\leq 0.15 \ \mu m \ (p-v)/75 \ mm \ on \ 250-mm \ diamert \ convex \ Al$	
	sphere	
٠	Surface finish: ≤ 3.0 nm (typically)	
Source: N	Aoore Nanotechnology Systems, LLC, Keene, NH	

^{*} NANOTECH is a registered trademark of Moore Nanotechnology Systems, Inc.

This machine can perform SPDT operations on axisymmetric and nonaxisymmetric work pieces. Precision grinding can be accomplished on materials not compatible with SPDT with optional equipment. Figure 10.8 shows a close up of a grinding spindle in use. It is mounted on the B-axis table. Typical optical-figure accuracy and surface roughness of the SPDT surfaces produced by these techniques are included in Table 10.2. These surfaces are adequate for IR applications and some visible light applications. With post-process polishing, surfaces can be brought to visible light quality standards.

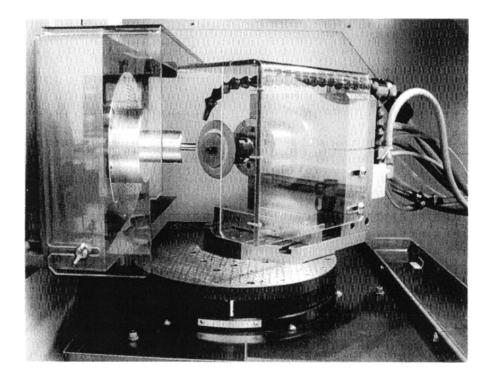


Figure 10.8 Close up photograph of the grinding spindle and wheel as used in the machine of Fig. 10.7 to grind precision surfaces on brittle materials. (Courtesy of Moore Nanotechnology Systems, Keene, NH.)

Single crystal diamond chips have unique characteristics that make them ideal for SPDT applications. When properly oriented, they are very hard, have low contact friction, are very stiff mechanically, have good thermal properties, and take an edge sharp to atomic dimensions. They also can be resharpened when wear becomes excessive. The cutting edge radius may vary for the particular application from infinity to as small as 0.030 in. (0.76 mm). Typical maximum defect depths in properly sharpened diamond tools are $<0.3\times10^{-6}$ in. (8×10^{-3} µm) as indicated by scanning electron microscope measurements. The radius can be held constant to about 6×10^{-6} in. (1.5 µm) for typical (short) radii. The diamond chips may be brazed to standard lathe tool bits as shown in Fig. 10.9 for physical support. When so mounted, they can easily be handled and attached to the SPDT instrument. Cubic boron nitride tools have proven effective for SPDT machining of bare beryllium substrates.¹⁵

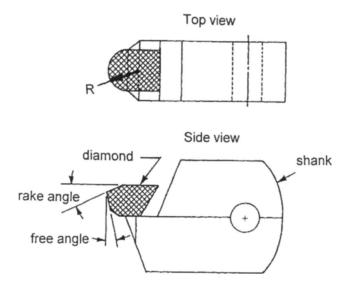


Figure 10.9 One type of diamond tool used for SPDT machining.

Figure 10.10 shows a multiple fly cutter tool head with three diamond tools installed so cuts can be made at different locations on the work piece, at different depths, or by different shaped tools during each pass so as to reduce the number of passes required to finish the surface.

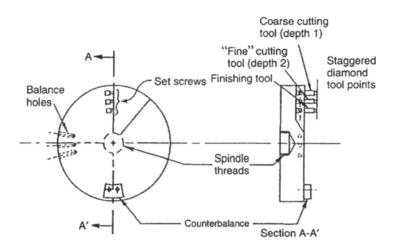


Figure 10.10 Schematic of a multiple fly cutter head with three diamond tools. (Adapted from Sanger. 3)

Stress free mounting of the work piece on the SPDT instrument is vital to achieving true surface contours and accuracy of machined dimensions. Techniques employed to ensure minimum stress include vacuum chucking, potting with elastomer, and flexure mounts. Figure 10.11(a) illustrates the vacuum technique as applied to a thin germanium lens element, whereas Fig. 10.11(b) illustrates a flexure technique as applied to an axicon mirror substrate. The use of centering chucks to facilitate SPDT operations on crystalline lenses and on optomechanical subassemblies is discussed in Section 12.1, in Erickson et al.¹⁷, and in Arriola.¹⁸

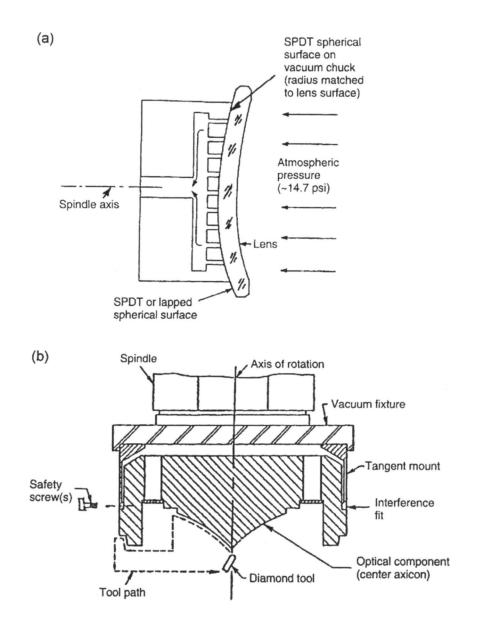


Figure 10.11 Schematics of (a) a vacuum chuck used to support a thin germanium lens (From Hedges and Parker.¹⁶) and (b) a flexure mounting for an axicon mirror during SPDT machining. (From Sanger.³)

While this discussion of SPDT and precision grinding of surfaces has concentrated on making optical surfaces, it is apparent that the same techniques apply to mechanical surfaces that form interfaces with optics and/or other mechanical parts. For example, the three mounting pads on the metal mirror shown in Fig. 10.12 are diamond turned in precise location and orientation with respect to the mirror surface. In fact, those surfaces are all machined without disturbing the setup of the substrate on the machine. Thus, no errors are created through misalignment from removing and reinstalling the optic. When the mirror is subsequently installed in an optical instrument, the optical surface is automatically aligned to the reference surfaces of the structure within the residual machining errors of the optical part.

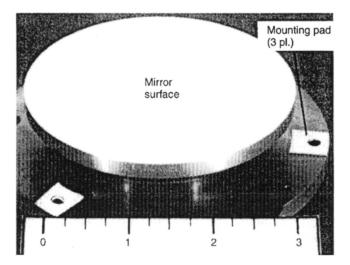


Figure 10.12 A metal mirror with front mounting pads created in the same SPDT setup as used for machining the optical surface. This maximizes alignment accuracy of the mounting surfaces. (From Zimmerman.¹⁹)

How best to design the interface between a metallic optic and its mount by SPDT machining of both components is illustrated in Fig. 10.13. Centering of one part relative to the other is controlled by SPDT machining the toroidal surface on the inner part for a close sliding fit into the cylindrical surface of the outer part. Axial location is controlled by SPDT machining the flat surface on the bottom of the inner part's flange perpendicular to the axis of that part. That surface mates with a partially relieved flat flange surface on the outer part. The latter surface is perpendicular to the axis of the outer part. When the two parts are drawn together by a series of bolts through the flanges, relative alignment of two axes within close tolerances and relative axial location of those parts are achieved.

An example of this design philosophy is given in Fig. 10.14. View (a) shows an aluminum mirror with the optical surface, the pilot diameter, and the axial interface all created by SPDT machining to close tolerances. Three axial locating pads are located just outside the cylindrical protrusion that forms the pilot diameter. These pads are machined coplanar and normal to the optical axis. As shown in view (b), they mate with the top surface of the back plate of the mount. The pilot diameter of the mirror slips snugly into the central recess machined into the mount. This design ensures that the mirror and the mount are properly aligned to each other *without adjustment* when assembled.

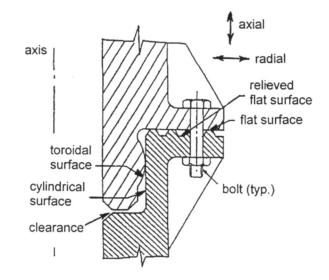


Figure 10.13 A typical interface between two SPDT components to ensure axial and radial alignment. (Adapted from Sanger.³)

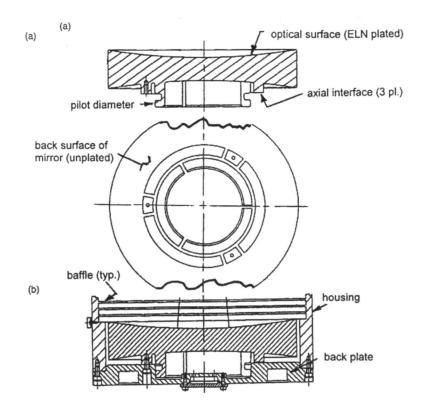


Figure 10.14 Optomechanical configuration of a SPDT mirror mounted in a SPDT machined housing. (Adapted from Vukobratovich, et al.²⁰)

10.2 Integral Mounting Provisions

Small and moderate-sized metal mirrors can often be mounted using the same techniques discussed for nonmetallic mirrors if there are no unusual requirements inherent in the application, such as extreme temperatures (e.g., cryogenic applications), exposure to highenergy thermal radiation (such as that from lasers or solar simulators), or extreme shock or vibration. The prime differences between metallic and nonmetallic mirror types have to do with differences in key mechanical properties, such as density, Young's modulus, Poisson's ratio, thermal conductivity, CTE, and specific heat. We can take advantage of these differences by using metals whose unique properties allow significant improvements in performance, weight, environmental resistance, etc.

A preferred method for supporting these metal mirrors involves mounting provisions built into the mirrors themselves. We illustrate a simple case in Fig. 10.15, which shows a section through a mirror with machined slots that isolate the mounting ears from the main part of the mirror. Forces exerted when attaching the mirror to the mount shown at the bottom of the figure then are not transmitted to the optical surface.¹⁹

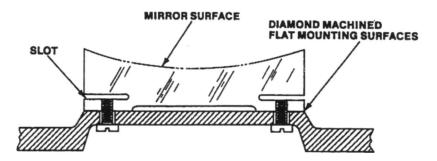


Figure 10.15 Diagram of a strain-free mounting for a small metal mirror. (From Zimmerman.¹⁹)

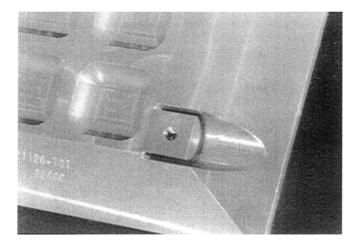


Figure 10.16 Closeup view of a mounting ear (flexure) machined into a metal mirror. (From Zimmerman.¹⁹)

Figure 10.16 shows a close-up photograph of the back side of a rectangular metal mirror having the same type of mounting interface feature as just outlined. In this case, the mounting ears have been machined into the mirror by core cutting parallel to the deeply beveled mirror back surface at multiple locations. A hole in each ear is threaded for a mounting screw. This feature is repeated at three places on the back of this mirror. When interfaced with a lapped or SPDT machined flat surface on the instrument structure little or no mechanical strain is created in the mirror to distort the optical surface. This is because the mounting ears are mechanically isolated from the mirror substrate by the slightly flexible residual material linking the ears to the substrate.

SPDT machining of the circular mirror shown in Fig. 10.17 might well proceed as follows: the optical surface is cut on the front side of the mirror blank, the necked-down groove is cut, the slot for an O-ring seal and the pilot diameter for centration are cut, and finally the inner (left) side of the flange is cut to serve as the mounting interface. All these operations are done without disturbing the alignment of the part on the spindle axis. Hence, they all have minimum relative errors. The pilot diameter is slightly larger than the OD of the mirror. This allows the mirror portion to be inserted carefully through the mating ID of the mount and the pilot diameter on the mirror to be engaged without striking the mirror on the mount. An O-ring is inserted into the groove before assembly to seal the component in place. This configuration is sometimes called a "mushroom" mirror because of its reduced diameter stalk that acts as a flexure to isolate the optical surface from mounting forces.

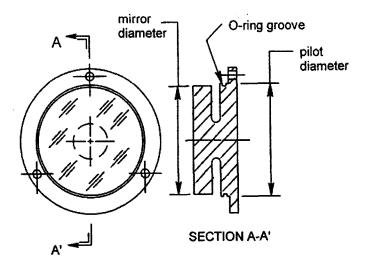


Figure 10.17 A small metallic mirror with optical and mechanical interfaces cut without removing the part from the SPDT machine. (Adapted from Addis.²¹)

10.3 Flexure Mountings for Metallic Mirrors

The mirror mount with integral flexure arms sketched in Fig. 10.18 was designed to provide stress-free support for a rigid metal mirror. The concept has been used successfully to make several beryllium mirrors.¹⁵ The mating surfaces of the three arms and the surface to which they were attached were precision lapped to minimize distortion

of the mirror surface when clamped in place. The mirror supports were not sufficiently stiff to hold the mirror during rough machining or grinding. The substrate was held by the cylindrical ring on the back of the mirror during these operations. It later was transferred to the flexure arms for final figuring when the forces exerted would be smaller.

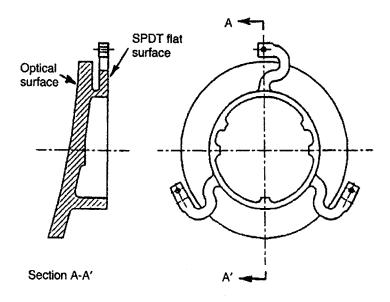


Figure 10.18 A beryllium mirror with integral flexure arm supports that interface with an external mounting surface. (From Sweeney.¹⁵)

Figures 10.19(a) and (b) show mounting flexure tab features machined into the rear surfaces of two metal mirrors intended for use in the Infrared Multi-Object Spectrograph (IRMOS) developed as a facility for the Kitt Peak National Observatory's 3.8-m (12.5-ft) Mayall Telescope.²² The mirror in view (a) is an off-axis section of a concave prolate ellipsoid measuring 264×284 mm (10.39 × 11.18 in.). Pockets are machined into the rear surface of this mirror to reduce its weight. The mirror in View (b) is an off-axis section of a convex oblate ellipsoid measuring 90×104 mm (3.54×4.09 in.). It is not lightweighted. Both mirrors are made of 6061-T651 aluminum with a 6:1 diameter to thickness ratio. They are stress relieved by aging at 175° C, then cycling the temperature between 83K, 23° C, and 150° C several times. All mounting and optical surfaces of both mirrors are finish machined by SPDT processing.

To facilitate assembly and alignment, the flexure tabs are cut into the mounting surfaces by a plunge EDM process to form the shapes indicated schematically in Fig. 10.19(c). The flexures minimize optical surface deformation due to mounting forces by bending to translate as much as ± 0.025 mm (± 0.001 in) and/or tilt by ± 0.1 deg. A threaded hole for attaching screws is provided in each tab.

The spectrograph is to operate at 80K. Its structure is fabricated from the same type of material (Al 6061-T651) as the mirrors to form an athermal assembly. Alignment of the system is facilitated by several crosshair fiducial marks scribed into the mirror rear and side surfaces during SPDT machining. One of these is pointed out in Fig. 10.19(b).

The slot and pinhole indicated in that same view serve as alignment references when the substrate is attached to the SPDT machine.

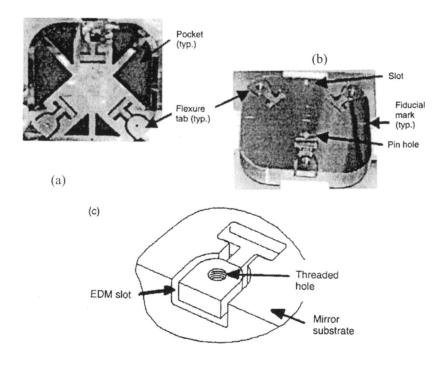


Figure 10.19 (a) and (b) Rear mounting and alignment surfaces of two aspheric aluminum mirrors featuring integral flexure tabs to minimize surface deformation from mounting forces. (c) Schematic of one flexure tab with its threaded mounting hole. (From Ohl et al.²²)

The flexure mounting for a mirror made from a solid HIPed beryllium cylinder and lightweighted by milling pockets in the substrate's back side was described by Altenhof.²³ The finished mirror is shown in Fig. 10.20. This mirror posed some significant design and fabrication challenges due to its unusual "kidney" shape, its large face dimensions of 65×40 in. (165×102 cm), a weight constraint of 118 lb (54 kg), optical figure requirements of $\lambda / 12$ rms at $\lambda = 0.63 \mu m$, operational temperature exposure range of 300 to 150 K, 15 times gravity vibration exposure, and a need for a natural frequency >50 Hz. The square lightweighting pocket pattern was chosen after a trade-off analysis to provide the minimum weight consistent with stiffness requirements to prevent deformation during fabrication and under the gravity loading of ground testing.

Manufacture posed some problems, including the need for successive heat treatments between rough machining steps and the need to remove subsurface damage layers as deep as 0.010 in. (0.25 mm) prior to the finish machining operation. This subsurface damage was removed by chemical etching to a depth of about 0.02 in. (0.5 mm). Critical surfaces were then finish machined, followed by a light chemical etch of 0.005 in. (0.13 mm) to remove residual subsurface damage. Thermal stabilization by repeated high and low temperature cycling completed the substrate processing.

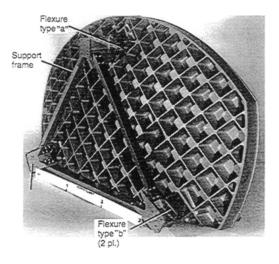


Figure 10.20 Photograph of the back side of a lightweighted HIPed beryllium mirror. Adapted from Altenhof.²³)

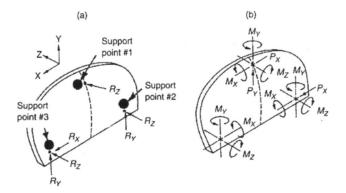


Figure 10.21 Concept for supporting the mirror of Fig. 10.20: (a) reaction forces applied and (b) unit moments applied. (From Altenhof.²³)

The mirror was designed to be mounted semikinematically with the degrees of freedom indicated in Fig. 10.21(a). The support attached at point No. 1 reacted only axial $(\pm Z)$ loads while that attached at point No. 2 reacted axial and vertical $(\pm Z, \pm Y)$ loads. The support at point No. 3 reacted loads in all three axes $(\pm X, \pm Y, \pm Z)$. This arrangement ensured that no pair of supports could restrain the mirror on a line connecting their centers and (theoretically) could not induce strain in the mirror. Support was to be at the mirror's neutral plane. Frictionless connections at the interfaces with the external supporting structure and infinitely compliant links to the structure were required.

The mounting design developed for this application utilized flexures with cruciform cross sections, as shown in Fig. 10.22. Each flexure link had to fit within a $4 \times 4 \times 3.5$ in. $(10 \times 10 \times 8.9 \text{ cm})$ pocket and was to accommodate local thermally induced deflections between the beryllium and dissimilar (CRES and Ti) structural materials.

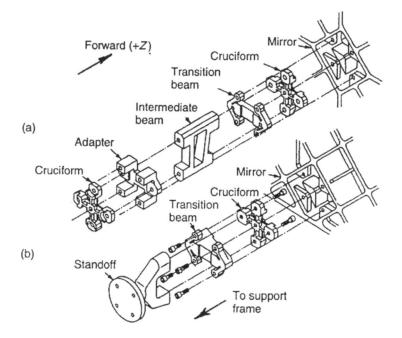


Figure 10.22 Exploded views of the flexure links used to support the mirror shown in Fig. 10.20: (a) as used at point No. 1 and (b) as used at points Nos. 2 and 3. (From Altenhof.²³)

Finite element analysis of proposed flexure configurations and materials indicated the suitability of 6AL-4V titanium because of its high spring merit factor (yield stress/elastic modulus), good thermal match to the beryllium mirror, and low density. Other materials considered and rejected were stainless steel, beryllium, aluminum, and beryllium-copper. Figure 10.23 is a photograph of one of the three flexures used.

To determine whether the redundant forces and moments that would result from use of these mirror supports would be acceptable, the finite-element model was used as follows: (1) unit forces (P) and moments (M) were applied singly at each support point of Fig. 10.21(a) in the directions indicated in Fig. 10.21(b). These loads represented inputs to the mirror due to uncorrected errors in spring rates and positioning of the flexures. (2) The worst case total deflections resulting from moment, force, and gravity terms in the above analysis were determined. These are given in Table 10.3 for what were believed to be the worst nodes. A deflection tolerance of 13 µin. (0.33 µm) over any instantaneous subaperture during self-weight testing was applied to this design. The right hand column of the table indicates the corresponding computed values. All deflections are well within the tolerable value. Moment and force values corresponding to this tolerable deflection were determined to be 0.5 lb-in. (0.056 N-m) and 0.5 lb (0.2 kg), respectively. These inputs were assumed to exist at each support point simultaneously.

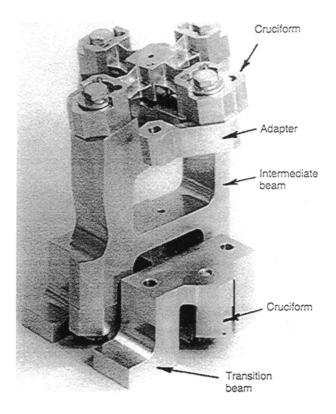


Figure 10.23 Photograph of the flexure link sketched in Fig. 10.22(a). (From Altenhof. 23)

Table 10.3 Worst case summations of mirror deflections with unit loads applied at support points of the mirror modeled in Fig. 10.22.

Node number	Deflections due to moments (µin.)	Deflections due to forces (µin.)	Deflections due to gravity (µin.)
9*	7.24	14.2	4.8
56*	-1.71	-7.0	4.6
31	1.33	3.9	1.9
1	1.89	6.3	-2.3
60	0.56	1.3	2.0
4	4.88	15.7	2.0
66	0.73	-0.50	2.9

* These nodes represent peak deflection points. Source: Adapted from Altenhof.²³ Another metal mirror of classic design mounted on flexures was the 27.8-lb (12.6 kg), 24.4-in. (61.0-cm) diameter, f/2 beryllium primary of the highly successful Infrared Astronomical Satellite (IRAS) orbited by NASA in 1983. The optical system was of the Ritchey-Chretien form. Figure 10.24 shows the optomechanical schematic of the telescope. It operated in the 8 to 120 μ m spectral region at a temperature of 2K and at an orbital altitude of 900 km. All structural elements connecting the two mirrors and the mirrors themselves were beryllium with the same CTE. The system was therefore athermal and temperature changes would not degrade optical performance.

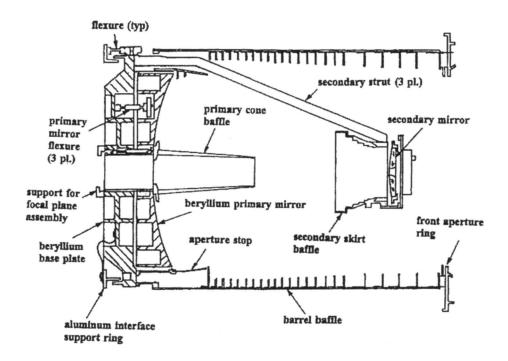


Figure 10.24 Optomechanical configuration of the 24-in. (61-cm) aperture, all-beryllium telescope for the Infrared Astronomical Satellite (IRAS). (Adapted from Schreibman and Young.²⁴)

The mirror was made from a pressing of Kawecki-Berylco HP-81 beryllium with CTE inhomogeneity specified to be no greater than 7.6×10^{-5} m/(m-K). It was lightweighted by machining pockets into the back surface. See Fig. 10.25. The structure remaining after machining comprised four circumferential rings with radial ribs at 20-deg intervals. Nominal wall thicknesses of the rings and ribs were 0.2 in. (0.51 cm). The face sheet was a minimum of 0.25 in. (6.35 mm) thick. The design was optimized by finite element analysis for minimum susceptibility to mount-induced and gravity-released distortions. A constraint on the design was that cryogenic testing (at 40K) was to be conducted in an available chamber that accommodated the mirror only in the axis horizontal orientation. Asymmetric distortion due to gravity was therefore of great significance. The analytical model consisted of 336 nodes and 252 plate elements representing the mirror's front face supported by 276 beam elements for the radial and

circumferential ribs. The computed rms surface deformation due to gravity effects after removal of defocus, decentration, and tilt was 0.020λ at $\lambda = 0.633 \mu$ m. The system error budget allowed 0.1λ wave for this deformation. Figuring at room temperature reduced the surface contour errors revealed during cryogenic testing. The cryogenic test/figure cycle was repeated until the mirror was deemed acceptable.

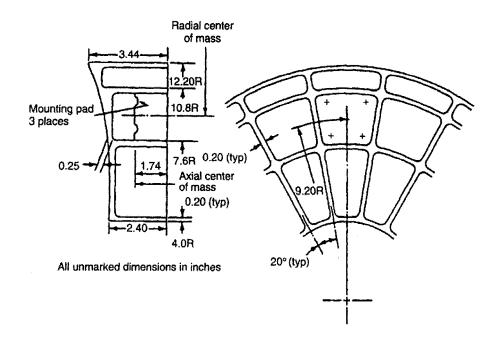


Figure 10.25 Detail views of the IRAS primary mirror. (Adapted from Young and Schreibman.²⁵)

The mirror was cantilevered from a large beryllium baseplate in the telescope structure by three flexure links of the "T-shaped" configuration shown in view (a) of Fig. 10.26. Each had a cruciform section attached to the mirror in a radial orientation. This part of the link accommodated errors in coplanarity of the pads on the link and on the mirror. The blade section of the link was oriented to allow relative radial morion between the mirror and the baseplate. These links were located at 120-deg. intervals on a 9.2-in. (23.4-cm) radius as shown in view (b). The stiff and compliant axes of the links were as indicated in the latter view. The points of attachment of the flexures were in the neutral planes of both the baseplate and the mirror. In the case of the mirror, this plane was 1.74 in. (4.42 cm) from the back surface. The links were made of 5A1-2.5Sn ELI titanium alloy^{*} with a coefficient of thermal expansion closely matching that of the beryllium.^{**}

^{*} ELI means "extra-low interstitial."

^{*} Vukobratovich et al.²⁶ indicated that Ti-6Al-4V ELI would be a better material for such flexures because of the low values of fracture toughness of the former material as reported by Carman and Katlin²⁷ and other researchers.

under the worst condition (cool down) would not exceed 385 lb/in.² (2.65 MPa). Assuming a microyield strength at cryogenic temperature of 3500 lb/in.² (24.1 Mpa), a respectable safety factor of \sim 9 would be obtained.

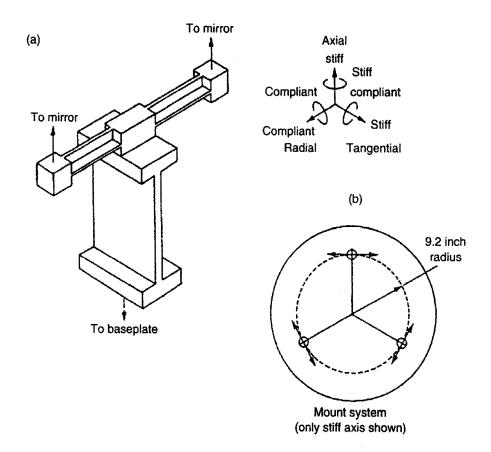


Figure 10.26 (a) Schematic of one flexure link used to support the IRAS primary from the telescope baseplate. (b) Frontal view of the mirror showing the orientation of the links. The stiff tangential axis is indicated for each link. (Adapted from Schreibman and Young.²⁴)

The general design principles underlying all these mirror examples with integral or separate flexures are the following: (1) mounting stresses are isolated from the mirror surface by incorporating flexure arms, geometric undercuts, or slots that create a form of flexure mounting; (2) the mirror is designed to be a stiffer spring than the interfacing mounting structure. Deformations then occur in the mount rather than in the mirror substrate; (3) the mirror should, if possible, be held during machining in precisely the same manner as it will be held during operation. Then, mounting strains will be the same in both conditions; (4) the mounting surfaces should be machined flat and parallel to the same degree of precision as the optical surface(s).

It should be noted that compliance with the second of these principles might result in a design in which rigid body displacement or tilt of the optical surface can occur during mounting. In this case, a provision would, be made for an alignment adjustment subsequent to installation. A further possible consequence of adhering to this principle is elimination of the need for adherence to the final principle.

10.4 Plating of Metal Mirrors

Because of the inherent crystalline structure, softness, and ductility of the most frequently used materials for metallic mirrors, aluminum and beryllium, it is practically impossible to achieve a high quality optical finish directly on the base metal. Both materials benefit by having thin layers of a metal (such as nickel) plated onto the base metal and the optical surface of these layers SPDT machined and/or polished to achieve a smooth finish. Similarly, the smoothness of optical surfaces on substrates made of chemical-vapor-deposited (CVD) or reaction-bonded (RB) silicon carbide can be improved with a thin layer of vapor-deposited pure copper added. Gold can be plated onto various types of metal substrates to form good IR reflecting mirrors.

Similar improvements in surface smoothness can be achieved in some cases with plated layers of the same material as the substrate. The proprietary AlumiPlate process for plating amorphous aluminum on aluminum is a prime example. The smoothness and thermal damage threshold of copper and molybdenum mirrors used in high energy laser beams can be improved if a thin amorphous layer of the base metal is deposited onto the substrates before polishing.

The most frequently used plating material for mirrors is nickel. Two basic processes are available for this purpose: electrolytic and electroless plating. Electrolytic nickel (here abbreviated EN) can be plated to a thickness of 0.030 in. (0.76 mm) or more. It has a Rockwell hardness of 50 to 58. The process is simple, but slow. It does not require precise temperature control. Typically, $140\pm15^{\circ}$ F ($60\pm8^{\circ}$ C) is adequate. Uniformity of coating thickness is not easily attained with this process. Electroless nickel (here abbreviated ELN) is an amorphous material with phosphorus content in the range of 11% to 13%. It can be plated more evenly, is more corrosion-resistant, and its application process is less complex mechanically and electrically than EN. On the negative side, the maximum practical thickness of the ELN layer is about 0.008 in. (0.20 mm), so the substrate must have very close to the proper contour before plating. The process temperature of ~200°F (93°C) for ELN plating is higher than that for EN. It must be controlled to $\pm 5^{\circ}$ F ($\pm 3^{\circ}$ C). Rockwell hardness of ELN plating is typically 49 to 55, but it can be increased somewhat by heat treating. An excellent detailed discussion of ELN was given by Hibbard.²⁸

Mismatch of thermal expansion characteristics of the plated layer and the base metal of the mirror substrate is one cause of dimensional instability in the completed optic. For nickel on beryllium, this mismatch is about 2×10^{-6} m/(m-K), whereas that for nickel on aluminum is about five times larger. The resultant bimetallic effect may be quite significant in high performance systems. Vukobratovich et al.²⁰ investigated possible approaches to minimizing bimetallic effects for an ELN plated aluminum 6061 concave mirror with diameter 18.0 cm (7.09 in.). The mirror configurations studied are shown in Fig. 10.27. Design variations from the baseline plano concave shape [view (a)] included (1) increasing the substrate thickness to resist bending [view (c)], (2) meniscus shape [view (d)], (3) designing the substrate with a symmetric cross section to produce equal

and opposite bending effects [view (e)], and, for all configurations, plating both sides of the substrate with equal and unequal thicknesses of nickel. The plane parallel configuration of view (b) was included for general information in regard to the effect of front to back surface plating thickness differences.

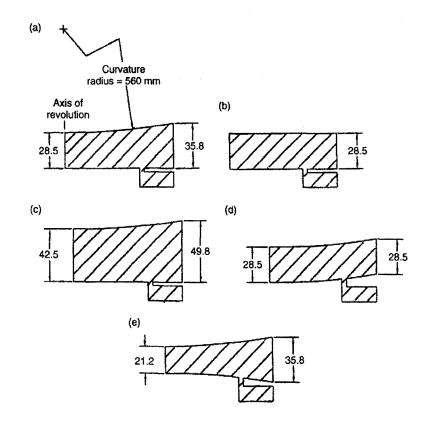


Figure 10.27 Mirror configurations investigated by Vukobratovich et al. and Moon et al. to determine bimetallic bending effects of various ELN platings. (From Vukobratovich et al.²⁰)

Both closed-form analyses using a method from Barnes³⁰ and FEA were conducted. Vukobratovich et al.²⁰ concluded: (1) the closed-form results did not correlate well with the FEA results, the FEA results being considered more accurate; (2) surface deformations comprising both correctable aberrations (piston and focus) and uncorrectable aberrations need to be determined rather than just surface departure from nominal; (3) because of mounting constraints on the mirror's back, equal thickness platings front and back may increase rather than decrease bimetallic bending; (4) increasing mirror thickness does not help; and (5) symmetric shaping of the substrate significantly reduces bending even if the back surface is not plated. The mirror and mounting design adopted by Vukobratovich et al.²⁰ upon completion of their investigation is discussed in Section 10.1 and shown in Fig. 10.14.

Moon et al.²⁹ extended the work of Vukobratovich et al.²⁰ to include both aluminum and nickel plating on aluminum and beryllium substrates. The mirror configurations of Fig. 10.27 were again investigated. The results of this later effort indicated that aluminum

plating on aluminum surfaces works best with equal thicknesses applied to front and back surfaces and that ELN plating on aluminum surfaces works best with only the front surface plated for the baseline and double concave mirror configurations. The best plating arrangement for the thick and meniscus substrates were found to be equal thicknesses front and back. With ELN plating on beryllium substrates configurations, the best arrangement would be plating on the front surface only. This conclusion applied to all mirror configurations.

A factor contributing to the long-term stability of ELN coated metallic mirrors is the stress inherent in the coating. Paquin³¹ discussed the dependence of this internal stress on the phosphorus content of the deposited nickel. In most cases, it is possible to specify a phosphorous content (typically ~12%) for zero stress when annealed. Hibbard³²⁻³⁴ considered this factor, among others, in discussions of means for minimizing dimensional instability of ELN plated mirrors. Varying the chemical composition and heat treatment applied allows the zero residual stress level to be located at the center of a given operational temperature range. Hibbard³³ provided information with regard to the dependence of CTE, density, Young's modulus, and hardness of ELN coating with phosphorus content. These parameters are important in modeling mirror designs.

The stress in plated coatings is easily measured by plating one side of thin metal strips as witness samples of the base material to be coated. Hibbard³³ described the technique. Generally, these strips measure 4-in. (102-mm) by 0.40-in. (10.2-mm) by approximately 0.03-in. (0.76-mm) thick with the latter dimension dependent on the particular material. The opposite faces of the strips are ground flat and parallel with inherent bending not exceeding 0.001 in. (25 μ m). Release of residual stress due to the plating bends the strip. The magnitude of the bending, and hence the stress, is determined by placing the strips on edge under a microscope and measuring contour departure from a straight line along the long dimension. Typical bending magnitudes for ELN coatings on aluminum are in the 0.010- to 0.015-in. (0.25- to 0.38-mm) range, so are easily measured with reasonable accuracy.

10.5 Interfacing Metallic Mirrors for Assembly and Alignment

As described in Section 10.2, one significant advantage of SPDT over other techniques for machining precision optical components is the ability to integrate locating, i.e., interface, and optical surfaces directly into each work piece of multiple component systems during fabrication, frequently without removing the work piece from the SPDT machine. This ensures maximum alignment accuracy of the optical surfaces to other portions of the overall system.

Figure 10.28, from Gerchman,³⁵ shows schematically six types of optical assemblies featuring this coordinated type of construction. Each assembly has at least one SPDT mechanical interface between separate optomechanical components. Each of these components is SPDT machined in a way that accurately aligns its optical surface(s) to the mechanical interfaces. A curved arrow indicates the axis of symmetry of each system. All but one system (e) has a mirror machined integrally with a spider support. The system in view (a) involves two centered conical (axicon) reflecting optical surfaces; hence the name reflaxicon. The fast Cassegrain telescope shown in view (b) requires only two components and it is quite short, while the longer slow Cassegrain telescope of view (c)

is most conveniently made with three components: two mirrors and a spacer. The reflecting (Schwarzschild) microscope objective system of view (d) also has a spacer that allows focus to be established. It is assembled with two threaded retaining rings. Note that the light path is from right to left in this view. The three-mirror system of view (e) has separate, off-axis optical components that need to be SPDT machined with reference mechanical surfaces or locating pins to facilitate rotational alignment about the axis. It also has integral stray light baffling provisions. The relatively complex four-mirror system of view (f) embodies features of all the other systems.

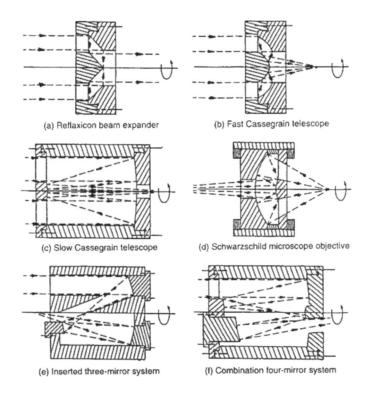


Figure 10.28 Six types of metallic optomechanical assemblies made by SPDT machining of optical and interfacing surfaces to facilitate assembly with little or no alignment required. (From Gerchman.³⁵)

The mechanical interfaces in systems such as are shown in Fig. 10.28 might well be configured generally as shown in Fig. 10.13 in order to provide centering and axial positioning. Minimal stress is introduced into the components if the radial interface involves close sliding contact, all surfaces providing the axial interface are coplanar flats and accurately normal to the component axes or a toroidal surface contacting a flat, and the bolt constraints are centered on the contacting pads.

A detailed description of the design, fabrication, assembly, and testing of an unobscured aperture 10 power afocal telescope assembly comprising two parabolic mirrors (one of which is off-axis) and a housing with two integral stray-light baffles was given by Morrison.³⁶ Figure 10.29 shows a sectional view of that system. Figure 10.30 is a drawing of the primary mirror while Figure 10.31 shows the secondary mirror.

Each of these mirrors has a flat flange on the reflecting surface side of the substrate that interfaces with the parallel ends of the housing. The interfacing surfaces and the optical surfaces are SPDT machined to high accuracy in regard to location and minimal tilt with respect to the optical axes. The flat surfaces also serve as alignment references during set up for testing. The length of the housing controls the vertex-to-vertex separation of the mirrors. The end surfaces are flat to $\lambda/2$ at 0.633 µm, parallel to 0.5-arc sec, and separated by the nominal length ±0.005 in. (±0.127 mm). The actual length is measured to ±10×10⁻⁶ in. (±0.25 µm). All parts are serialized for identification.

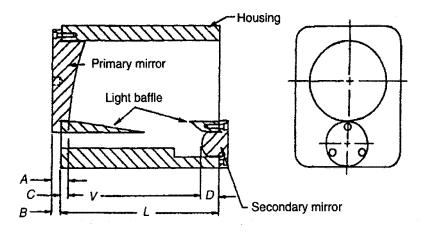


Figure 10.29 Example of a 10 power, afocal telescope comprising primary and secondary mirrors plus a housing with two light baffles. Both optical surfaces and all mechanical interfaces are SPDT machined for accurate inherent alignment. (From Morrison.³⁶)

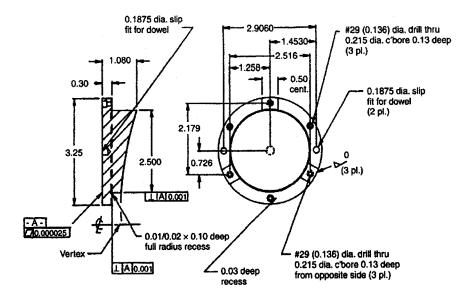


Figure 10.30 Drawing of the primary mirror for the telescope of Fig. 10.29. (Adapted from Morrison.³⁶)

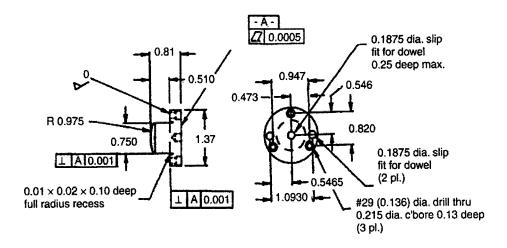


Figure 10.31 Drawing of the secondary mirror for the telescope of Fig. 10.29. (Adapted from Morrison. 36)

The primary mirror is attached to a sub plate (fixture) for diamond turning. This sub plate is vacuum chucked to the SPDT instrument and diamond turned flat to $\langle \lambda/2 \rangle$ at 0.633 µm. The rim of the sub plate is then turned to a roundness of $\pm 5 \times 10^{-6}$ in. ($\pm 0.13 \mu$ m) to provide an accurate reference for centering six precision jig bored dowel holes in pairs 2.906 in. (51.692 mm) apart on a 2.000-in. (50.800 mm) bolt circle to match the dowel pin holes on the primary mirror. A central dowel hole also is bored at this time. Three mirror blanks are mounted on the sub plate as shown in Fig. 10.32 for simultaneous machining. Note that the set up in this figure is functionally the same as that in Fig. 10.6.

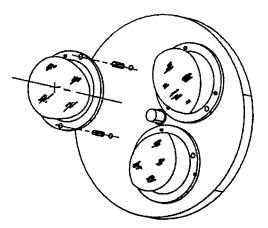


Figure 10.32 Schematic of a fixturing configuration for simultaneous SPDT machining of three off-axis primary mirrors for the telescope of Fig. 10.29. (Adapted from Morrison.³⁶)

After the optical surfaces of a set of primary mirrors are completed, their actual axial thicknesses are measured to 1×10^{-6} in. (0.025 µm) and recorded as dimension A of Fig. 10.29. Nominally, this dimension is 0.550 in. (13.970 mm). The mirrors are then

individually mounted on a vacuum chuck using a central dowel for centering. The mounting flange is turned until $A - B = 0.250 \pm 0.002$ in. (6.350 ± 0.051 mm). The actual dimension C, to the nearest microinch, is then recorded. The length L of the housing is also measured to the nearest microinch.

The secondary mirror is mounted individually on a vacuum chuck using a central dowel for centering and the optical surface diamond turned. Its flange is then machined in the same set-up so that its dimension D (see Fig, 10.29) equals L - V - C. All mirrors are machined in the same manner so they automatically are positioned correctly upon installation.

Because all critical dimensions have been machined to close tolerances, no adjustments are needed during assembly. The fabrication process has determined the optical alignment. Morrison³⁶ indicated that a telescope could be completely assembled by a technician within 30 minutes.

Another telescope featuring similar SPDT machined features to facilitate alignment was described by Erickson et al.¹⁷ Figure 10.33 shows the telescope schematically. All components were made of 6061 aluminum to render the system athermal. All surfaces marked SPDT were diamond turned as described below. The 8-in. (20.3-cm) diameter primary mirror had an integral mounting flange isolated from the optical surface by a necked-down flexure region similar in principle to that of Fig. 10.17. The primary and secondary mirrors had integral spherical reference surfaces diamond turned concentric with the nominal telescope focal point as indicated in Fig. 10.33. We here summarize major details of the component manufacturing process to show how these surfaces were used to facilitate accurate machining and assembly.

After conventional machining to near net shape and size, the secondary mirror was mounted on a SPDT instrument for diamond turning the back (nonoptical) surface. The substrate was then turned over and attached to a vacuum chuck for diamond turning the mirror's OD and ID, the convex aspheric optical surface, the concave spherical reference surface, and the mechanical interface for the focus spacer.

The following surfaces on the primary mirror were diamond turned in a single machine set-up: the flat mounting flange surface, the concave spherical reference surface, the convex spherical mirror back surface, and the mirror's OD and ID. It was then turned over and mounted by its flange to the SPDT faceplate. The substrate was centered to the spindle axis by minimizing runout of the precision OD. The concave aspheric optical surface and the axial interface for the secondary support were then turned and the mirror's ID was matched to the conventionally machined OD of the secondary support. Without removing the primary mirror from the spindle, the secondary support was attached with screws (not shown in Fig. 10.33) and the OD and axial interface for the secondary mirror axes.

After removal of the primary/secondary support subassembly from the spindle, the focus spacer was ground to thickness and parallelism and the secondary mirror installed. When the axial separation of the optical surfaces was correct, fringes could be observed between an auxiliary reference surface concentric with the focal point and both diamond-turned reference surfaces on the mirrors. The authors indicated that no subsequent alignment was needed to achieve < $\lambda/4$ reflected wavefront accuracy at $\lambda = 0.633 \mu m$ from production telescopes.

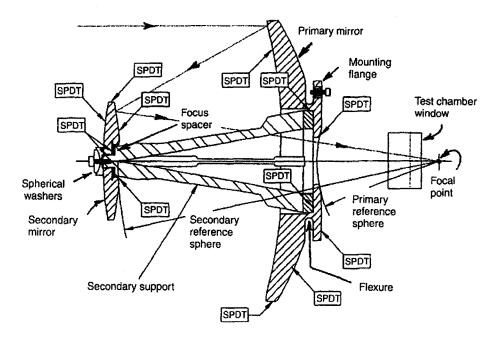


Figure 10.33 Optomechanical schematic of an all aluminum telescope with optical surfaces, mechanical interfaces, and alignment reference surfaces SPDT machined for simple assembly without adjustment. (Adapted from Erickson et al.¹⁷)

10.6 References

- 1. Saito, T.T. and Simmons, L.B., "Performance characteristics of single point diamond machined metal mirrors for infrared laser applications," *Appl. Opt.* 13, 1974:2647.
- 2. Saito, T.T., "Diamond turning of optics: the past, the present, and the exciting future," *Opt. Eng.* 17, 1978:570.
- 3. Sanger, G.M., "The Precision Machining of Optics," Chapt. 6 in *Applied Optics and Optical Engineering*, 10 (R. R. Shannon and J. C. Wyant, eds.), Academic Press, San Diego, 1987.
- 4. Rhorer, R.L. and Evans, C.J., "Fabrication of optics by diamond turning," Chapt. 41 in *Handbook of Optics*, 2nd ed., Optical Society of America, Washington, 1994.
- 5. Whitehead, T.N., *The Design and Use of Instruments and Accurate Mechanism*, *Underlying Principles*, Dover, New York, 1954.
- 6. Vukobratovich, D., Private communication, 2003.
- 7. Gerchman, M., "Specifications and manufacturing considerations of diamondmachined optical components," *Proceedings of SPIE* **607**, 1986:36.
- 8. Dahlgren, R. and Gerchman, M., "The use of aluminum alloy castings as diamond machining substrates for optical surfaces," *Proceedings of SPIE* **890**, 1988:68.
- Ogloza, A., Decker, D., Archibald, P., O'Connor, D., and Bueltmann, E., "Optical properties and thermal stability of single-point diamond-machined aluminum alloys," *Proceedings of SPIE* 966, 1988:228.

- Gerchman, M. and McLain, B., "An investigation of the effects of diamond machining on germanium for optical applications," *Proceedings of SPIE* 929, 1988:94.
- 11. Parks, R.E., "Introduction to diamond turning," SPIE Short Course Notes, SPIE, Bellingham, 1982.
- 12. Colquhoun, A., Gordon, C., and Shepherd, J., "Polygon scanners—an integrated design package," *Proceedings of SPIE* **966**, 1988:184.
- 13. Gerchman, M., "A description of off-axis conic surfaces for non-axisymmetric surface generation," *Proceedings of SPIE* **1266**, 1990:262.
- 14. Curcio, M.E., "Precision-machined optics for reducing system complexity," *Proceedings of SPIE* **226**, 1980:91.
- 15. Sweeney, M.M, "Manufacture of fast, aspheric, bare beryllium optics for radiation hard, space borne systems," *Proceedings of SPIE* **1485**, 1991:116.
- 16. Hedges, A.R. and Parker, R.A., "Low stress, vacuum-chuck mounting techniques for the diamond machining of thin substrates," *Proceedings of SPIE* **966**, 1988:13.
- 17. Erickson, D.J., Johnston, R.A., and Hull, A.B, "Optimization of the optomechanical interface employing diamond machining in a concurrent engineering environment," *Proceedings of SPIE* CR43, 1992:329.
- 18. Arriola, E.W., "Diamond turning assisted fabrication of a high numerical aperture lens assembly for 157 nm microlithography," *Proceedings of SPIE* **5176**, 2003:36.
- 19. Zimmerman, J., "Strain-free mounting techniques for metal mirrors," Opt. Eng. 20, 1981:187.
- 20. Vukobratovich, D., Gerzoff, A., and Cho, M.K., "Therm-optic analysis of bi-metallic mirrors," *Proceedings of SPIE* **3132**, 1997:12.
- 21. Addis, E.C. "Value engineering additives in optical sighting devices," *Proceedings* of SPIE **389**, 1983:36.
- Ohl, R., Preuss, W., Sohn, A., Conkey, S., Garrard, K.P., Hagopian, J., Howard, J. M., Hylan, J., Irish, S.M., Mentzell, J.E., Schroeder, M., Sparr, L.M., Winsor, R.S., Zewari, S.W., Greenhouse, M.A., and MacKenty, J.W., "Design and fabrication of diamond machined aspheric mirrors for ground-based, near-IR astronomy," *Proceedings of SPIE* 4841, 2003:677.
- 23. Altenhof, R.R., "Design and manufacture of large beryllium optics," *Opt. Eng.* 15, 1976:2
- 24. Schreibman, M. and Young, P., "Design of Infrared Astronomical Satellite (IRAS) primary mirror mounts," *Proceedings of SPIE* **250**, 1980:50.
- 25. Young, P. and Schreibman, M., "Alignment design for a cryogenic telescope," *Proceedings of SPIE* **251**, 1980:171.
- 26. Vukobratovich, D., Richard, R., Valente, T., and Cho, M., Final design report for NASA Ames/Univ. of Arizona cooperative agreement No. NCC2-426 for period April 1, 1989-April 30, 1990, Optical Sci. Ctr., Univ. of Arizona, Tucson, 1990.
- 27. Carman, C.M. and Katlin, J.M, "Plane strain fracture toughness and mechanical properties of 5Al-2.5Sn ELI and commercial titanium alloys at room and cryogenic temperature," *Applications-Related Phenomena in Titanium Alloys*, ASTM STP432, American Society for Testing and Materials, 1968:124-144.
- 28. Hibbard, D.L., "Electroless nickel for optical applications," *Proceedings of SPIE* CR67, 1997:179.
- 29. Moon, I.K., Cho, M.K., and Richard, R.M., "Optical performance of bimetallic mirrors under thermal environment," *Proceedings of SPIE* 4444, 2001:29.
- 30. Barnes, W.P., "Some effects of the aerospace thermal environment on high-acuity optical systems," Appl. Opt. 5, 1966:701.

- 31. Paquin, R.A., "Metal Mirrors," Chapt. 4 in *Handbook of Optomechanical Engineering*, CRC Press, Boca Raton, 1997.
- 32. Hibbard, D., "Dimensional stability of electroless nickel coatings," *Proceedings of SPIE* **1335**, 1990:180.
- 33. Hibbard, D., "Critical parameters for the preparation of low scatter electroless nickel coatings," *Proceedings of SPIE* **1753**, 1992:10.
- 34. Hibbard, D., "Electrochemically deposited nickel alloys with controlled thermal expansion for optical applications," *Proceedings of SPIE* **2542**, 1995:236.
- 35. Gerchman, M., "Diamond-turning applications to multimirror systems," *Proceedings* of SPIE **751**, 1987:113.
- Morrison, D., "Design and manufacturing considerations for the integration of mounting and alignment surfaces with diamond-turned optics," *Proceedings of SPIE* 966, 1988:219.

CHAPTER 11 Techniques for Mounting Larger Nonmetallic Mirrors

In this chapter, we continue our considerations of mountings for mirrors by discussing techniques that can be used to mount nonmetallic mirrors in the size range from about 0.89 m (35 in.) in diameter to 8.4 m (331 in.). Weight minimization is increasingly important as mirror size increases. With the exception of space-borne applications, the mirrors considered here are too flexible for 3-point, rim, or hub mounting during use and so must be supported at many points. Axial supports, generally applied to the back of the mirror, radial supports, generally applied to the rim of the mirror, and "defining supports" (locating and orienting the mirror) pose major design issues. Some mirrors have these forces applied within the interior of the substrate at the neutral surface where gravitational moments acting on the localized volumes are balanced. Most of the large mirrors mentioned here are intended for astronomical applications as scientists begin to reap the benefits of new design, manufacturing, and control technologies that break the size limitations previously imposed on ground-based systems by residual manufacturing errors, gravitational effects, and atmospheric turbulence. Mountings for selected examples of historically important telescope mirrors as well as operational and developmental telescope mirrors are considered. Mountings for mirrors used or tested with their axes in fixed orientations, horizontal or vertical, also are discussed. Some challenges of mounting large (~8-m) aperture, thin mirrors on active mechanisms that maintain required optical performance under the command of control systems with optical surface figure or image quality sensors are pointed out. An example of such an "adaptive" mirror (for the Gemini telescopes) is discussed. Finally, some unique features of the mountings for large mirrors in two highly successful space borne telescopes (Hubble and Chandra) are reviewed.

11.1 Mounts for Axis-Horizontal Applications

A mirror with its axis fixed in the horizontal direction or a variable orientation mirror with its axis temporarily oriented horizontal suffers surface deformations that are not rotationally symmetrical about the mirror axes. An example of a fixed orientation application is in laboratory test equipment. If the mirror is always used in the same orientation relative to gravity as it is when tested, the errors due to gravitational forces can, for the most part, be removed during the polishing operation. This cannot be done if the mirror orientation changes.

In a classical paper on mirror flexure due to gravity, Schwesinger¹ explained the two types of forces exerted on the edge of a mirror supported with its axis horizontal. Radially directed boundary forces, tensile or compressive, that vary in magnitude around the periphery of the mirror are the first type. These generally are distributed uniformly as indicated in Fig. 11.1(a) and are transmitted through the mirror to support the weight of each volume element. Potential deformations at a point on the surface of the mirror are V_R radially, V_{φ} tangentially, and V_Z axially. The mirror diameter is D_{G_i} its axial thickness is t_A , its edge thickness is t_E and its weight is W.

If the cross section of the mirror is not of uniform thickness, i.e., concave [as in Fig. 11.1(b)] or convex, the transmitted forces also produce moments that tend to bend the

mirror. The formation of such a moment is illustrated schematically in view (c), which shows a volume element of average thickness t + (dt/2) with the transmitted radial forces applied to the opposite faces. The upward and downward resultants of these forces are axially displaced by (dt/2), thereby producing an elemental moment. When integrated over the entire mirror, these elements have a resultant force equal to $W\xi$, where ξ is the distance from the center of mass to the midplane [dashed vertical line in Fig. 11.1(b)]. This resultant is balanced by a distribution of bending moments m_R , at the mirror edge as indicated at the bottom of view (b). These moments form the second type of boundary forces due to gravity. They tend to bend the mirror surface. The bottom edge is tilted upward and the top edge is tilted downward. The horizontal edges are not tilted. This produces a generally cylindrical deformation of the mirror surface.

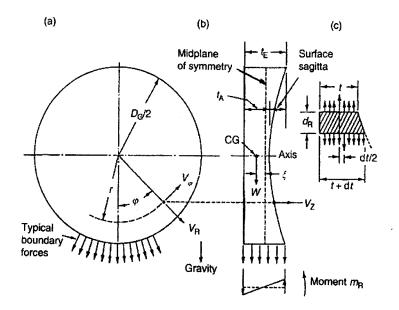


Figure 11.1 Gravity effects on the optical surface of an axis-horizontal concave mirror. (From Schwesinger.¹)

Schwesinger provided a theory for calculating the surface deformations and the resulting reflected wavefront errors caused by these two types of forces. Shear stresses and the effects of central perforations in the mirrors were not considered. Subject to these limitations, we will apply that theory to explain the advantages and disadvantages of some mechanical mounts commonly used to support mirrors of various configurations and sizes with axis horizontal.

11.1.1 V-mounts

Figures 11.2 through 11.4 illustrate three versions of a mount type in which the weight of an axis-horizontal mirror is supported radially by line contact of the cylindrical rim against two parallel horizontal cylindrical posts located symmetrically with respect to the mirror's vertical centerline and below its horizontal centerline. The contacts are similar to that for a cylinder resting in a V-block. It is advantageous to use a thin sleeve of plastic material such as Kevlar as the contacting surface on the posts to provide a slightly resilient interface and some thermal insulation as well as to reduce friction. Rollers also are sometimes used with large mirrors to minimize friction. The contacts between the posts and the mirror rim are between different diameter parallel-axis cylinders if the mirror is circular. A rectangular mirror also can be supported in such a mount. The contacts at the posts are then between cylinders and a flat surface. In each case, the axial position of the mirror is maintained by clamping its rim very lightly with clips on each of the lower posts. Each clip and the pad immediately behind it on the back plate of the mount contact only small areas on the mirror to minimize localized bending moments should the interfacing surfaces not be exactly parallel. Springs may be used to provide the clamping forces. In Fig. 11.2, a third post is provided at top center of the mirror. It normally does not contact the mirror but supports a clip extending in front of the mirror's rim to prevent the mirror from falling forward if bumped.

Support against gravity is at ± 60 -deg to the vertical centerline of the mirror in the design shown in Fig. 11.2. In the design of Fig. 11.3, the radial support is at ± 45 -deg These designs may be referred to as 120-deg and 90-deg V-mounts, respectively. Commercial mounts of the type shown in Fig. 11.2 accommodate mirrors in the 3.5-in. (9-cm) to 9.8-in. (25-cm) diameter range while those of the type shown in Fig. 11.3 have been made to accommodate mirrors in sizes ranging from 4-in. (10-cm) to 30-in. (76-cm) diameter. All these designs provide means for adjusting tilt of the mirror axis. Two axis translational adjustments are also provided in the design shown in Fig. 11.3.

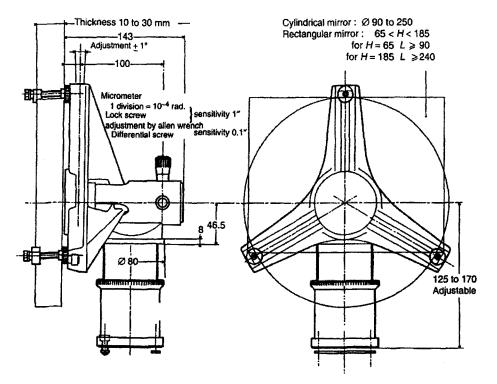


Figure 11.2 A commercial V-mount for mirrors to 250-mm diameter. (Courtesy of Newport Corporation, Irvine, CA.)

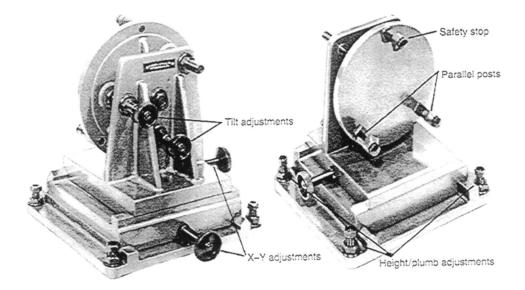


Figure 11.3 A larger commercial V-mount. (From literature provided by John Unertl Optical Company, Pittsburgh, PA.)

The commercial mirror mount shown in Fig. 11.4 (and sketched in Fig. 9.6) is much smaller than those described above, but is still a V-mount. Here, mirrors with diameters of about 25.4 mm (1.0 in.) rest on two parallel horizontally oriented plastic rods (typically made of Nylon or Delrin) inserted into recesses in the ID of a hole bored into a mounting plate. The mirror is secured in place by a Nylon setscrew pressing gently against the top center of the mirror. Usually, the mirror is located axially in such a mount by manually registering it very lightly against a shoulder or pads machined into the plate as the setscrew is tightened. Friction then constrains the mirror against axial motion.

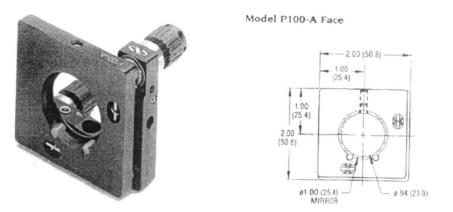


Figure 11.4 A commercial V-mount for 25.4-mm (1.0-in.) diameter mirrors. (Courtesy of Newport Corporation, Irvine, CA.)

An axis-horizontal mirror mounted in any of these V-mounts will exhibit surface deformations due to gravitational effects. These effects are usually found to be significant only if the mirrors are large. Axial constraints can also deform the mirror, but if they are carefully located and the axis is truly horizontal, the forces imposed in a benign environment such as a laboratory should not be large.

According to Schwesinger¹, the root mean square departure δ_{rms} from perfection (in waves) of the surface of a gravitationally deformed axis-horizontal mirror can be computed from:

$$\delta_{\rm rms} = \frac{C_{\kappa} \rho D_G^2}{\left(2E_G \lambda\right)},\tag{11.1}$$

where C_{κ} is a computed factor given by Schwesinger for each of six specific types of mounts used to support a mirror, ρ and E_G are the density and Young's modulus respectively of the mirror material, D_G is the mirror diameter, and λ is the wavelength of the reflected light. The rms error of the reflected wavefront would be twice the surface error $\delta_{\rm rms}$. Table 11.1 gives Schwesinger's values for C_{κ} for three mount configurations of greatest interest here and for specific values of the factor κ defined (after minor adaptation of nomenclature from Schwesinger's definition) by Eq. (11.2):

$$\kappa = \frac{D_G^2}{\left(8t_A R\right)},\tag{11.2}$$

where R is the optical surface radius of curvature and the other terms are as previously defined. Schwesinger limited his numerical considerations (and hence the data in Table 11.1) to the common case where $D_G = 8t_A$. Then, $\kappa = 0.5/(\text{mirror } f\text{-number})$.

Table 11.1 Values for Schwesinger's factor C_{κ} to be used in Eq. (11.1) for circular mirrors with axis horizontal in three types of mount and for particular values of the factor κ .

	к =	0 (flat)	0.1	0.2	0.3
Mount type:	<i>f</i> -number =		<i>f</i> /5	<i>f</i> /2.5	<i>f</i> /1.67
±45 deg V-mount	$C_{\kappa} =$	0.0548	0.0832	0.1152	0.1480
"Ideal" mount	$C_{\kappa} =$	0	0.0018	0.0036	0.0055
Strap mount	$C_{\kappa} =$	0.00743	0.0182	0.0301	0.0421

Vukobratovich² gave the following series expansion as an approximation for Schwesinger's factor C_{κ} .

$$C_{\kappa} = a_0 + a_1 \gamma + a_2 \gamma^2. \tag{11.3}$$

Here, the constants a_i are as listed in Table 11.2 and γ equals κ . Vukobratovich indicated that his constants for the strap mount (single asterisk) were derived by fitting to experimental data. The other constants for that mount (double asterisk) were derived by fitting to Schwesinger's value C_{κ} . The last two columns of the table compare values of C_{κ} for $\kappa = 0.2$ as computed with Vukobratovich's equation to those from Schwesinger's paper.

Table 11.2 Values for Vukobratovich's constants to be used in Eq. (11.2) for rms deflections of circular mirrors with axis horizontal in five types of mirror mount.

				C_{κ} for $\kappa = 0.2$	
Constant				Per	Per
Mount type:	a_0	<i>a</i> ₁	<i>a</i> ₂	Vukobratovich	Schwesinger
One point at $\varphi = 0 \text{ deg}$	0.06654	0.7894	0.4825	0.244	0.246
±45-deg V-mount	0.05466	0.2786	0.1100	0.1148	0.1152
±30-deg V-mount	0.09342	0.7992	0.6875	0.6348	
Strap mount*	0.00074	0.1067	0.0308	0.0340	0.0301
Strap mount**	0.00743	0.1042	0.0383	0.0421	0.0421

Source: Vukobratovich,² except as noted.

* Based on experiments at University of Arizona.

** Based on theory presented by Schwesinger.1

Figure 11.5 shows graphically the variation of $2\delta_{rms}$ in waves of green light vs. mirror diameter for each of the values of κ given in Table 11.1. The material here is Pyrex. Its Poisson's ratio is 0.2, which is essentially the same as that assumed by Schwesinger. Calculations to obtain this graph are as in Example 11.1. The vertical dashed line corresponds to $2\delta_{rms} = \lambda/14 = 0.071 \lambda$, which, according to Maréchal³ as explained by Born and Wolf,⁴ is the Rayleigh diffraction limit. We observe that a perfect flat mirror made of Pyrex mounted in this type of mount may be as large as 56.7-in. (144-cm) in diameter for diffraction-limited performance if gravitational deformation is the only error source. The diameter of a concave Pyrex mirror with $\kappa = 0.3$ (corresponding to f/1.7 for the 8:1 diameter-to-thickness ratio considered here) should be no larger than 34.6 in. (87.9 cm) for this same performance level.

The calculations leading to the graphs in Fig. 11.5 have been repeated for ULE and Zerodur materials to show the variations due to the different Young's modulus and density values of those materials as compared to Pyrex. All other parameters were unchanged. Figure 11.6 shows the diameter vs. rms wavefront error relationships. Zerodur appears to be the best material. This is attributed primarily to the higher Young's modulus for that material $(13.6 \times 10^6 \text{ lb/in.}^2)$ as compared to $(9.1 \times 10^6 \text{ lb/in.}^2)$ for Pyrex.

Malvick⁵ studied the theoretical elastic deformations of two solid centrally perforated mirrors, one a 230-cm (90.6-in.) diameter primary mirror of a stellar telescope at Steward Observatory and a 154-cm (60.6-in.) diameter biconcave mirror used at the University of Arizona's Optical Sciences Center for experimental purposes. One of the cases he considered was the larger mirror supported on edge by two pads located at ± 30 -deg from the vertical centerline. If the pads were located axially in the plane containing the mirror's center of curvature, the gravitationally induced surface deflections would be as

Example 11.1: Surface deflection of a circular flat mirror in a \pm 45-deg V-mount. (For design and analysis, use File 11.1 of the CD-ROM.)

A Pyrex mirror with $D_G = 64$ in. and $t_A = 8$ in. is mounted in a ±45-deg V-mount with axis horizontal. (a) Using theory from Schwesinger¹, what is the expected rms wavefront error in green light if the mirror is flat? (b) If the mirror is a f/2.5 concave sphere, what is its wavefront error?

From Table B8(a): Poisson's ratio is 0.2, $\rho = 0.081$ lb/in.³ (2.23 g/cm³), and E_G is 9.1×10^6 lb/in.² (6.3×10⁴ MPa). Green light wavelength = 0.546 µm (21.5×10⁻⁶ in.). The wavefront error is $2\delta_{rms}$.

(a) From Table 11.1, $C_{\kappa} = 0.0548$. From Eq. (11.1):

$$2\delta_{\rm rms} = \frac{(2)(0.0548)(0.081)(64^2)}{(2)(9.1 \times 10^6)(21.5 \times 10^{-6})\left(\frac{1}{25.4}\right)} = 0.093 \text{ wave.}$$

(b) From Table 11.1, $C_{\kappa} = 0.1152$. From Eq. (11.1):

$$2\delta_{\rm rms} = \frac{(2)(0.1152)(0.081)(64^2)}{(2)(9.1 \times 10^6)(0.000546)\left(\frac{1}{25.4}\right)} = 0.195 \text{ wave.}$$

shown in Fig. 11.7(a). Changing the support angle to ± 45 -deg would change the surface contours to appear as shown in Fig. 11.7(b). The inherent astigmatism of the surface is reduced by a factor on the order of three in the latter case, but the contours are more complex.

The pads supporting the above-described larger mirror were at ± 30 -deg from the vertical and located about 5 cm (2 in.) in front of the center of gravity (i.e., toward the mirror face). This presumably was done to ensure against the mirror's accidentally falling forward out of the mount. Malvick analyzed the effect of this shift and found that gravity plus the reactions of the mirror's back supports to the moment introduced by the offset radial supports produced the surface contours illustrated in Fig. 11.7(c). The deformations are about six times larger than those shown in Figs. 11.7(a) or (b).

From these theoretical evaluations, we learn why the simple V-mounts described here work reasonably well for modest sized solid mirrors. Of course, these performance predictions assume that the mirror axis remains exactly horizontal. Tipping the mirror in the terrestrial gravity field changes the supporting force conditions, and more sophisticated radial mounting arrangements might then be necessary because the influences of axial force components must be considered.

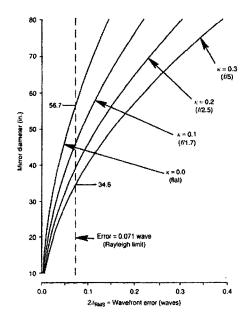


Figure 11.5 Variation of gravitationally induced rms wavefront error in green light for axis-horizontal circular Pyrex mirrors in \pm 45-deg V-mounts as a function of mirror diameter for different values of Schwesinger's factor κ . Mirror thickness is diameter/8.

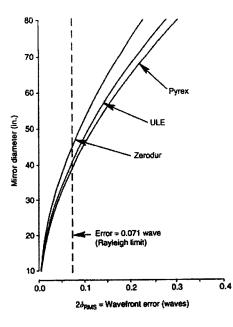


Figure 11.6 Variation of gravitationally induced rms wavefront error in green light for axis-horizontal circular solid Zerodur, ULE, and Pyrex mirrors in \pm 45-deg V-mounts as a function of mirror diameter for Schwesinger's factor $\kappa = 0.2$ (*f*/2.5 sphere). Mirror thicknesses are diameter/8.

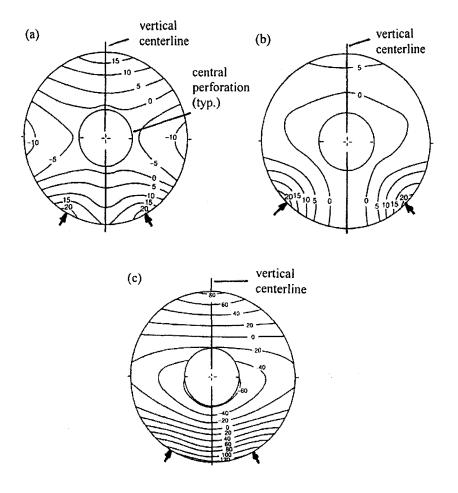


Figure 11.7 Surface deformation contours for a 91-in. (230-cm) diameter solid mirror from gravity effects when in a V-mount with pads at (a) \pm 30-deg and (b) \pm 45-deg from the vertical. Radial support is in the plane of the mirror's CG. (c) Surface contours for the same mirror as (a), but with the pads 5 cm (2 in.) in front of the CG. (Adapted from Malvick.⁵)

11.1.2 Multipoint edge supports

Vukobratovich,^{6,7} suggested that mechanical support to an axis-horizontal mirror can be obtained through a system of lever mechanisms applying forces normal to the lower portion of the mirror's rim at the neutral surface. A circular mirror with such a support is shown schematically in Fig. 11.8. Each mechanism is a whiffletree arrangement.^{*} Constraints are evenly distributed spatially at eight points located at 180-deg/7 = 25.7-deg intervals in the configuration depicted in the figure. Support at fewer points could be obtained with a simpler design while support at additional points could be obtained by adding more whiffletrees.

^{*} The whiffletree is defined in Section 9.6.

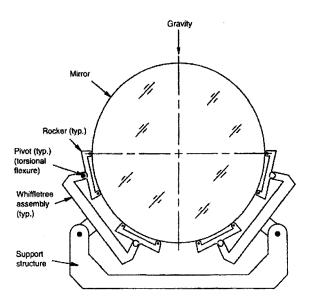


Figure 11.8 Multipoint (whiffletree) edge support for an axis-horizontal circular mirror. (From Vukobratovich.⁷)

If the mirror is rectangular and its axis is always horizontal, vertical support can be furnished at multiple points across the bottom as illustrated schematically in Fig. 11.9 for two- through five-point cascaded supports. Once again, increasing the complexity of the design could provide additional supports. According to Vukobratovich,⁷ for a given mirror length L_M , the optimum separation S of the support points is given by:

$$S = \frac{L_M}{\left(N^2 - 1\right)^{\frac{1}{2}}},\tag{11.4}$$

where N is the number of supports. Applying Eq. (11.4) to the four cases shown in Fig. 11.9 and for a constant L_M of 3.674 in. (93.320 mm), we obtain the values of S as listed for each case.

In the absence of friction, each of the lever mechanisms delivers force uniformly at the discrete contacts. If the areas of contact are small, the configuration can be considered semikinematic. Friction causes the mirror to become astigmatic. Rollers at the contacts can reduce frictional distortion of the optical surface.

11.1.3 The "ideal" radial mount

The "ideal" mount for axis-horizontal circular mirrors of larger size was defined by Schwesinger¹ as one in which the disk is balanced by radially directed push and pull forces around its periphery. The magnitudes of these forces vary as the cosine of the polar angle φ measured from the downward-pointing centerline of the disk. The radial forces are maximum compressive at the bottom of that centerline, decrease to zero on both sides at the horizontal centerline, and then change sign to tensile forces of increasing magnitude, reaching a maximum at the top of the disk. Figure 11.10 illustrates this

concept for a large 4-m (157-in.) diameter perforated mirror analyzed by Malvick and Pearson.⁸ The contour lines represent equal mirror surface deformations and indicate that the surface becomes astigmatic due to edge moments generated by gravity. Similar deformations, of lesser magnitude of course, occur in smaller solid mirrors when similarly mounted.

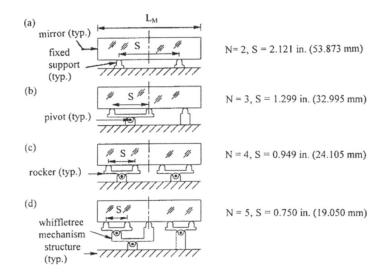


Figure 11.9 Family of cascaded whiffletree mechanisms serving as vertical supports for an axis-horizontal rectangular mirror with $L_M = 3.674$ in. (93.320 mm). (Adapted from Vukobratovich.⁷)

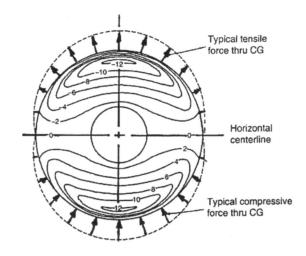


Figure 11.10 Surface deformation contours caused by gravity in a 4-m (157in.) solid mirror with axis horizontal in a mount that approximates the ideal mount as defined by Schwesinger.¹ The contour interval is 10^{-6} cm. (From Malvick and Pearson.⁸)

The analytical method used by Malvick and Pearson⁸ included shear effects as well as those of a large, central hole in a solid disk mirror. Using an analytical method called "dynamic relaxation" as developed by Day,⁹ Otter et al.,¹⁰ and Malvick¹¹ in a tensor formulation, Malvick and Pearson⁸ expressed the three-dimensional equations of elasticity as three equilibrium equations and six stress displacement equations. The body of the mirror was divided into a "reasonable" number of nonorthogonal curvilinear elements. Normal stresses were defined at the centers of the elements, shear stresses were defined at the centers of the element edges, and displacements were defined at the centers of the element faces. The equilibrium equations were set equal to the sum of acceleration and viscous damping terms. Initial stress, displacement, and velocity distributions at a time t_0 , were assumed. All three distributions at a later time t_1 were predicted mathematically. The process was iterated until the element velocities damped out to negligible values, leaving the static surface displacements corresponding to equilibrium in the three-dimensional body.

The analytical method given by Schwesinger¹ and applied earlier to the V-mount is also applicable, within limits, to the ideal mount design having a cosine distribution of radial forces. Equations (11.1) and (11.2) and data from Table 11.1 are used. As noted earlier, Schwesinger's method did not include shear stresses so this conclusion is overly optimistic. It is apparent, however, that perfect flat- and curved-surface mirrors could be quite large while providing subdiffraction limited performance *if* we could provide an "ideal" mount. Unfortunately, the physical realization of the ideal mount is much more difficult than its conception, and compromises must be made.

Figure 11.11 shows the force distribution of a mounting for an 18.1-in. (46-cm) diameter solid-meniscus Zerodur mirror that approximates the "ideal" mount, which was successful in a very demanding application. Although not a "large" mirror, it is included here as a good example of the type of mounting under discussion.

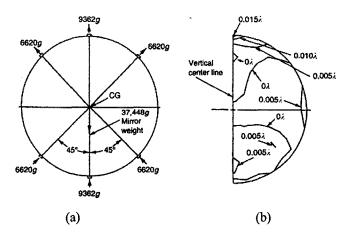


Figure 11.11 (a) Distribution of radial forces used to support a meniscus mirror in a nearly "ideal" axis-horizontal mounting. (b) Surface deformation contours with interval of $\lambda/200$ at 633-nm wavelength. (Courtesy of ASML Lithography Corporation, Wilton, CT.)

Three push and three pull forces were applied to the bottom and top of the rim of the mirror through six metal flexures bonded to the rim of the mirror at polar angles of $\varphi = 0$ -deg and ± 45 -deg from the vertical centerline. All the flexures were located at the neutral surface of the mirror. These flexures were flexible in all directions perpendicular to the radial direction and attached to a "rigid" cell. (See Fig. 11.12.) The mirror is shown mounted for interferometric testing, i.e., not in its final system installation, but those configurations were functionally equivalent. The mirror had a spherical reflecting surface radius of curvature of about 24 in. (61.0 cm) and nominally weighed 37.45 kg (82.6 lb).

The deformations of the reflecting surface from the best-fitting reference sphere are shown by the contour lines in Fig. 11.11(b). The wavelength interval is $\lambda/200$ for $\lambda = 633$ nm. It may be noted that the surface is essentially undistorted over most of the aperture. By careful choice of the radial forces applied, this level of performance was achieved in large production quantities of the mirrors.

11.1.4 Strap and roller chain supports

Figure 11.13 shows a typical strap mount for an axis-horizontal mirror. This is a commercial mount in which the mirror's rim rests in a sling supported at both upper ends from a vertical plate. The strap mount was first described by Draper¹² as a means of reducing the astigmatism of a mirror supported on edge. This type mount was first used to support mirrors used to test other optical components and is still used for that purpose. It has never been very successfully applied to variable orientation telescopes because it is not suited for systems involving changes in elevation angle of the mirror axis.

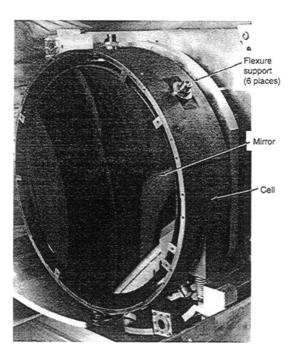


Figure 11.12 Photograph of the mirror and mounting from Fig. 11.11. (Courtesy of ASML Lithography Corporation, Wilton, CT.)

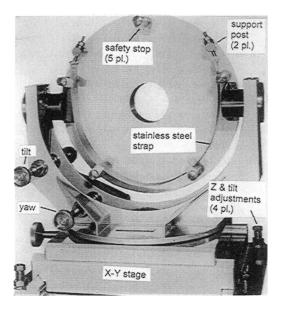


Figure 11.13 A typical commercial strap mount for axis-horizontal mirrors. (Adapted from literature from John Unertl Company, Pittsburgh, PA.)

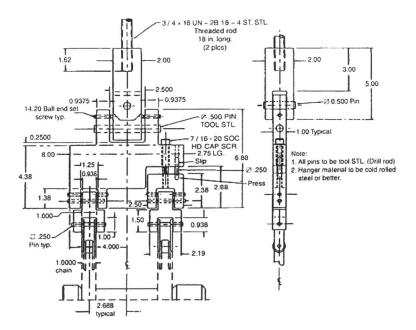


Figure 11.14 An adjustable mechanism for attaching a dual roller chain to a mirror support frame. (From Vukobratovich and Richard.¹⁴)

Schwesinger¹ gave values for C_{κ} as shown in Table 11.1 for various values of κ for this type mount. With Eqs. (11.1) and (11.2) and these data, one can approximate the rms wavefront error for a given mirror diameter, neglecting shear effects. Figure 11.14 shows the surface deformations typical of a large solid mirror in a simple strap mount as computed by dynamic relaxation including shear. Vukobratovich¹³ reported that observed deflections for mirrors larger than about 1.5 m (59.0 in.) mounted in this manner were somewhat larger than predicted by Schwesinger's 1954 equations. At least some of the discrepancy can be attributed to friction between the strap and the mirror's rim. Because the strap mount offers the dual advantages of good performance and simplicity, it is quite popular for commercial and custom fixed horizontal axis applications.

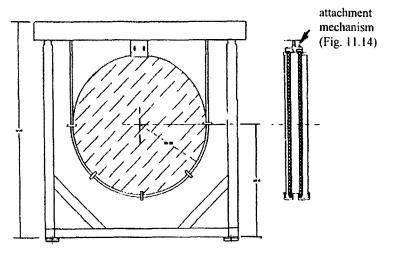
Dual steel cables have been used successfully as a strap for larger mirrors. Malvick ⁵ investigated the advantages of this splitting of the strap support into two narrower and separated straps to give more localized support to the mirror's rim. He showed that, by carefully adjusting the locations of these two supports in the axial direction, surface roll-off effects at the mirror's edge could be minimized. This type of support offers the advantages of reduced friction, ability to rotate the mirror about its axis, and stability without the need for axial support. Vukobratovich and Richard¹⁴ described the technique, in part, as follows:

"Roller chains are preferred over conventional bands primarily due to reduction in friction between the edge of the mirror and the chain. It is a mistake to use either plastic rollers or an insulating elastic layer between the rollers and the mirror edge. Plastic rollers will take a permanent deformation or set with the passage of time, increasing friction. Friction is also increased with the use of an elastic layer between the mirror edge and roller chain. Conventional roller chain, sold as conveyor chain with oversize rollers, employing steel rollers, is preferred.

"An important advantage of the roller chain is the commercial availability of roller chain. A wide variety of chain sizes and load capacities are available, and are relatively low in cost. Special chain links are available to permit the attachment of spacers and safeties to the roller chain. Roller chain supports are very compact, taking space around the mirror edge equal to the chain thickness. For optical shop testing, a roller chain permits ease of rotation of the mirror in its support to test for astigmatism.

"Point contact between rollers and the mirror edge, with resulting high stress and possible local fracture, is a drawback of the roller chain support. Careful installation and adjustment of the roller chain minimizes potential fracture at the mirror edge....

"A chain hanger provides termination of the chains, permits adjustment of the chain with respect to the mirror, and connects the support to the rest of the mirror mount. [It] provides three adjustments: location of the centroid of the two chains along the axis of the mirror, axial spacing between the two chains, and a vertical adjustment for mirror wedge. A standard hanger design for a 1.5-m [59.05-in.] mirror incorporating the above adjustments is shown in figure 5 [here Fig. 11.14]. A universal joint is provided at the top of the chain hanger to insure static determinacy of the support. Two chain hangers, one on each side of the mirror, are provided. The



chain hangers are attached to the mirror mount; for shop testing this is a large steel weldment called an easel, as shown in figure 6 [here Fig. 11.15]."

Figure 11.15 Diagram of a mirror in a typical dual-roller chain support. (From Vukobratovich and Richard.¹⁴)

The surface distortion of a 1.54-m (60.63-in.) diameter solid Cer-Vit mirror when mounted with axis horizontal in a dual-roller chain mount was analyzed by Malvick.⁵ Figure 11.16 shows the results graphically. Vukobratovich and Richard¹⁴ reported that a mirror of this design was tested and found to have an rms figure error of 0.078 waves.

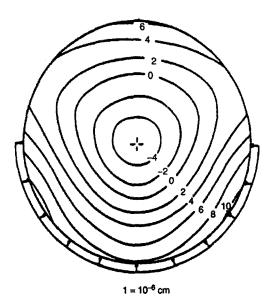


Figure 11.16 Surface deformation contours distorted by gravity for a large solid mirror mounted in a dual-roller chain support. (From Malvick.⁵)

11.1.5 Comparison of dynamic relaxation and FEA methods of analysis

The dynamic relaxation (DR) analytical method used by Malvick and Pearson⁸ to determine the surface deformations of large mirrors due to gravity effects was, for many years, the only available method to predict proposed or actual mirror/mount designs. The results of their analyses using this method have proved extremely useful to engineers and astronomers in evaluating mirror and mount designs in the 2- to 4-m (79- to 157-in.) class. This information is believed to be useful in evaluations of smaller or larger mirrors mounted in the same types of mounts since; in general, the contours remain the same as magnitudes scale approximately with size.

In order to determine if the same technical results would be obtained through application of finite element analysis (FEA) and DR methods to these design problems, Hatheway et al.¹⁵ recomputed the deflections of a mirror that was analyzed by Malvick and Pearson⁸ when supported in a strap mount. Figure 11.17 shows the mirror model analyzed. It has twenty equal 18-deg angular sectors, ten annular rings, and five nearly flat layers (front to back). The entire model has 1000 structural elements, each with 8 nodes and 6 sides. Dihedral symmetry was assumed to apply in both cases. The FEA model was processed using the MSC/O-POLY preprocessor and MSC/NASTRAN software. This allowed up to 100 Zernike polynomials of the surface to be evaluated, as well as to predict the surface deformations. The results of the FEA analysis are shown in view (b) of Fig. 11.18. The shapes and magnitudes of the surface deflections should be compared with those shown in view (a) of that figure, which represents Malvick and Pearson's results for the same mirror/mount combination.

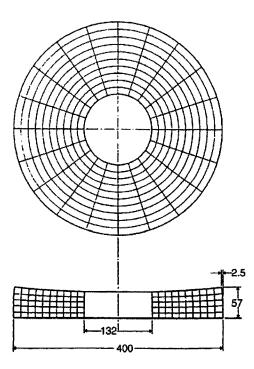


Figure 11.17 Analytical model used in DR and FEA analyses of a large solid mirror in a strap mount. (From Hatheway et al.¹⁵)

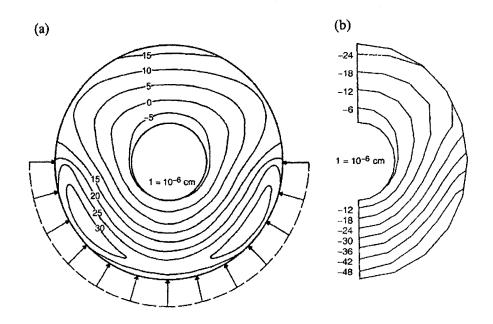


Figure 11.18 Applied loads and the resulting surface contours of the 4-m (157-in.) diameter mirror in a strap mount as determined by (a) dynamic relaxation (From Malvick and Pearson.⁸) and (b) FEA (From Hatheway et al.¹⁵).

Hatheway and his co-workers evaluated the study results as follows:

- (1) The algebraic signs of the results are reversed. This resulted from coordinate system redefinition.
- (2) The locations of the zero contours are slightly different. This was attributed to differences in the support points used to control rigid-body motions.
- (3) In general, the shapes of the contours are similar. The "lobing" (hexifoil deviation from circular contours) is reduced in the FEA results as compared to the DR results. This was attributed to elimination in the FEA model of a slight conical shape (drift angle) for the mirror's rim in the DR model due to the necessity in the FEA case to balance axial forces resulting from uniform pressure over the rim.
- (4) The peak-to-valley ranges of the displacement fields for the two cases are very close (50×10^{-6} cm for DR and 54×10^{-6} cm for FEA).

With these small differences recognized, Hatheway et al.¹⁵ concluded that the two methods give essentially the same results. This is reassuring since any other conclusion would cast doubts on the many design decisions made in the past based on Malvick and Pearson's pioneering work and greatly reduce the willingness of designers to use that work as the basis for future designs.

A significant advantage of the FEA method for analyzing mirrors and their mounts is the ability to present the results as Zernike coefficients. Figure 11.19 shows the magnitudes of the first 100 coefficients for the mirror analyzed by Hatheway et al.¹⁵ The expected concentration of errors in the first twenty terms is seen, but two spikes at terms 85 and 92 are noticeable. The reason for these spikes is not fully understood, but their significance is small if one considers the ratio of the areas under them to the total area under the spikes for the first twenty terms.

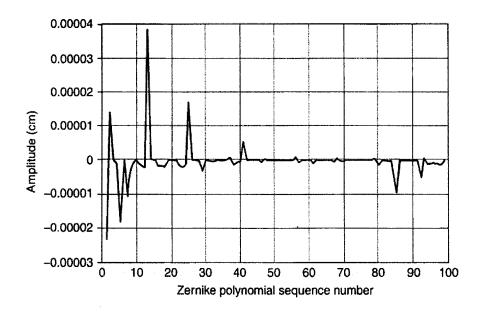


Figure 11.19 Amplitudes of the first 100 Zernike polynomials corresponding to the surface deformation of the mirror surface depicted in Fig. 11.18. (From Hatheway et al.¹⁵)

11.1.6 Mercury tube supports

Another approximation to the "ideal" mount is to support the mirror's edge by radially directed compressive forces of variable magnitude proportional to $(1 + \cos \phi)$, where ϕ is the polar angle measured from the downward pointing radius. Figure 11.20, from Malvick and Pearson,⁸ illustrates the forces applied and the resultant surface contour deformations for the same 4-m (157-in.) diameter mirror as formed the basis for the ideal mount analysis shown in Fig. 11.10. This type of force field results approximately when the mirror is supported within an annular mercury filled tube located between the mirror rim and a rigid cylindrical cell wall. The width of the tube is chosen so that, when it is nearly full of mercury, the mirror will float. Typical designs call for flattened neoprene coated Dacron tubes to hold the mercury. The axial location of the tube center should coincide with the plane through the center of gravity of the mirror so that overturning moments are avoided. Axially spaced dual-mercury tubes have also been used successfully. The mercury tube positions the mirror laterally without the need for hard radial defining point supports. A distinct advantage of the mercury radial support is that stress is minimized owing to the relatively large area over which the force is distributed.

According to Chivens,¹⁶ mirrors as large as 60-in. (1.5-in.) in diameter have been held centered within 0.0005 in. (0.012 mm) in astronomical telescopes with mercury tube

radial supports. Vukobratovich and Richard¹⁴ indicated some practical difficulties encountered with this type mount. The mirror contour is somewhat affected by irregularities in the tube such as seams, fill ports, and wrinkles. The fluid also tends to slosh from side to side under vibration so is usable only in relatively benign environments. In addition, one must consider the potential for a human health hazard due to the mercury itself.

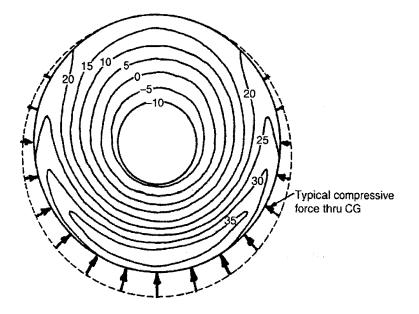


Figure 11.20 Surface deformation contours for a large solid mirror supported in a mercury-tube edge mount. Contour interval is 10^{-6} cm. (From Malvick and Pearson.⁸)

11.2 Mounts for Axis Vertical Applications

11.2.1 General considerations

As explained in connection with mounts for mirrors with fixed horizontal axes, the mounts for mirrors with fixed vertical axes have prime applications to test equipment for component or system evaluation or in laboratory apparatus. When the axis of a circular mirror is vertical, gravity acts symmetrically around the mirror axis. The deflections of the surface are then generally symmetrical unless asymmetric inhomogeneities in the mirror material's mass properties or asymmetry in mass distribution of built-up structure in the substrate occur. Further, with a fixed orientation of the mirror with respect to gravity during manufacture, testing, and use, surface figure errors can be polished out—or at least greatly reduced.

In this section, we consider different ways in which to support the axis vertical mirror. The air bag axial mount is examined first. This gives fairly uniform support over the entire area of the mirror's back. Ring supports made up of discrete air bags are also described. Then, we look at multiple point supports, which can be pneumatic, hydraulic, or lever mechanisms. Examples of each type are discussed. Applications of the Hindle mount to large mirrors are explained.

11.2.2 Air bag axial supports

Axial supports using air-filled bags or bladders have been installed in many astronomical telescope primary mirror mounts as a means of distributing axial forces over the back surface of the mirror.¹⁶ They are of two general types: those with large area contact provided by circular bladders and those with annular area contacts at two or more selected radial zones. These types are illustrated schematically in Fig. 11.21. Systems in which axial support is provided to a mirror by a series of discrete, usually circular, air bags that act as pistons pressing against localized areas of the mirror's back surface function in the manner of the multiple-point supports discussed later.

Air bags typically are made of two sheets of neoprene or neoprene-coated Dacron cemented together at the edges. Vukobratovich indicated that a special variety of ozone-resistant neoprene is needed if the air bag is to be used at a high altitude, such as in mountain top observatories.¹⁷ The mirror's location and orientation usually are referenced to three or more hard points projecting from the back plate of the mirror cell through sealed holes through the bag. A low-pressure pump supplies air to the bag through a pressure regulator. Safety provisions usually are in the form of nearby multiple soft supports that hold the mirror when air pressure is not supplied and it lowers onto these auxiliary supports.

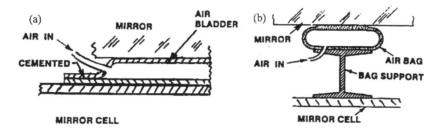


Figure 11.21 Schematics of (a) bladder type and (b) ring type air bag mirror supports.

It is difficult to design a single air bag that will produce the required force distribution for uniform support of mirrors with a thickness varying significantly in the radial direction. Multiple annular air bags can have different pressures, thereby accommodating this variation. Control of pressure to the required degree of precision is no small task in any air bag system; the problem is compounded if multiple bags are involved.

A limitation on dynamic performance of an air bag-type support system results from the low air pressure needed to support the mirror. Proper axial support to a mirror cannot be provided during rapid changes in elevation of the mirror's axis because the air pressure within the support(s) cannot be changed fast enough at such low pressure. This problem does not occur in static installations; hence, the air bag support has been used most successfully in fixed orientation applications, such as polishing and testing a mirror with its axis vertical.

During the manufacture of a 1.8-m (70.9-in.) diameter, f/2.7 paraboloid to be used as a spare mirror for the original MMT telescope, this 1200 lb (544 kg) slumped fused silica egg crate mirror was supported on its back with approximately 93% of its weight on a full

diameter neoprene air bag and the remaining 7% supported by three swivel-defining pads. That bag and the pads are shown in Fig. 11.22. The pressure needed to support the mirror was only about $(0.93)(1200)/(\pi)(35.45)^2 = 0.28$ lb/in.² (1930 Pa). The mirror was first ground and polished to a sphere. This allowed the choice of weight distribution between bag and small-area support to be based on measurement of the effects of varying pressure vs. the surface deformation. The pad "print-through" was also adequately minimized at that weight distribution.¹⁸

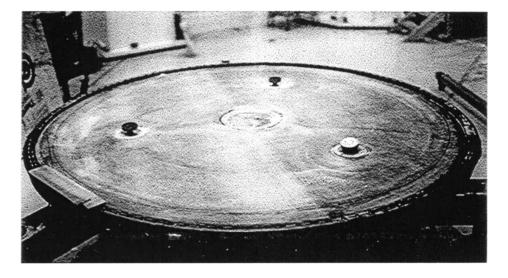


Figure 11.22 Photograph of a full-diameter bladder-type air bag used during polishing and testing of a 1.8-m (70.8-in.) diameter mirror. (Courtesy of the College of Optical Sciences, University of Arizona.)

An air bag or bladder in contact with the back of a mirror tends to thermally insulate that mirror from its surroundings. This might be undesirable for stability. It is obvious that this effect needs to be considered during the design of the temperature control system for any application where temperature varies significantly.

A bladder can also be constructed in the form of three or more pie-shaped sectors connected in parallel to the air manifold¹⁶ or differentially inflated to support a mirror with nonuniform weight distribution. (See Fig. 11.23.) In such designs, the hard points can be located at the radial intersections of the sectors. The lack of support at the narrow regions between the sectors can usually be tolerated.

Doyle et al.¹⁹ indicated that an FEA model created during design of an air bag support must take into consideration the shape of the interface between the bag and the mirror at the mirror's rim. If the mirror is rotationally symmetric, the pressure should be adjusted so the bag is tangent to the mirror at the rim as shown in Fig. 11.24(a). Those authors showed how corrections are applied if this is not the case, i.e., the bag is over inflated or under inflated as shown in Figs 11.24(b) and (c) respectively. They also showed how to adjust the FEA model if the mirror is not symmetrical. One case of such a mirror configuration is the off-axis paraboloid illustrated in Fig. 11.24(d).

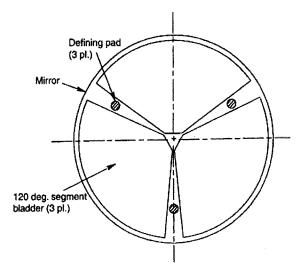


Figure 11.23 Schematic rear view of a segmented air bladder support for a large mirror comprising three 120-deg segments. Defining points also are indicated. (Adapted from Chivens.¹⁶)

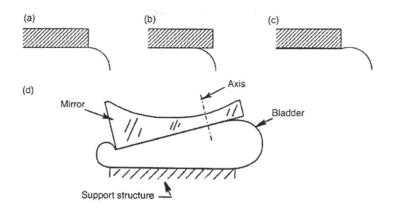


Figure 11.24 Four possible edge conditions for air-bladder axial support of a mirror: (a) tangent, (b) over inflated, (c) under inflated, and (d) exaggerated diagram for a nonsymmetrical (off-axis paraboloid) mirror supported by an air bladder. (Adapted from Doyle et al.¹⁹)

A variation on the concept of the air bag uses the mirror disk itself as a piston with one or more flexible gaskets or an O-ring to seal it at or near its rim to a closed cell. (See Fig. 11.25.) Creating a partial vacuum with a pump reduces the air pressure within the sealed region. The atmospheric pressure difference between the front and back of the mirror then supports the weight of the mirror and holds it against three defining pads (one shown in the figure). Chivens¹⁶ mentioned this approach. Several telescopes have been built with this type of support. One such design is used to support the primary mirror in the Gemini telescope. Multiple axial actuators provide partial support for this mirror. It is used in a variable orientation telescope so it is described in Section 11.3.

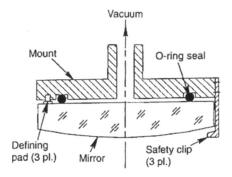


Figure 11.25 Conceptual schematic of a plano-convex mirror axially supported facing downward by atmospheric pressure in the manner of a negative air bladder support. (From Vukobratovich.⁷)

Baustian²⁰ described a double annular air bag support as used with a 150-in. (3.8-m) diameter mirror. The design radii for the zonal contacts were $0.48R_{MAX}$ and $0.85R_{MAX}$, respectively, where R_{MAX} is one-half the disk diameter. During operation, the axial location of the mirror was fixed by three "defining units" (hard points) at the $0.722R_{MAX}$ zone. Figure 11.21(b) shows a sectional view sketch of one ring support while Fig. 11.26 shows the layout of the mirror in its cell. The annular widths of the inner and outer air bags were 4.8 in. and 5.1 in. (12.2 cm and 13.0 cm) respectively. As may be seen in the front view, the annular bags were made in sections. This was for reasons of cost and ease of installation.

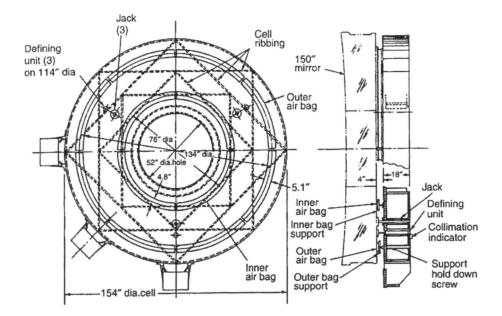


Figure 11.26 Layout of a double annular air bag support for a 150-in. (3.8-m) diameter mirror. Dimensions are inches. (Adapted from Baustian.²⁰)

Figure 11.27, from Malvick and Pearson,⁸ shows the computed symmetrical surface deformation typical of a 4-m (158-in.) diameter mirror when supported with axis vertical by two concentric rings as with the double annular air bag support. The ring radii in this example were nearly the same $(0.51R_{MAX} \text{ and } 0.85R_{MAX})$ as those provided by the design of Fig. 11.26. The peak-to-valley surface deformation is 3×10^{-6} cm (0.06 wave in green light) over most of the aperture.

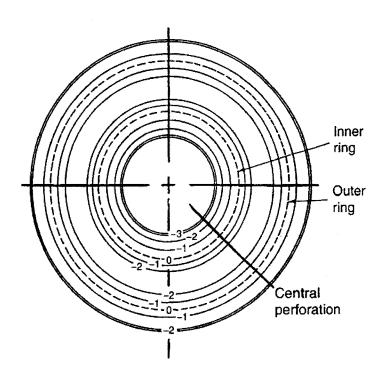


Figure 11.27 Computed surface deformation of a 4-m (158-in.) diameter axis vertical mirror supported on two annular rings located at the dashed lines. The contour interval is 10^{-6} cm. (Adapted from Malvick and Pearson.⁸)

11.2.3 Metrology mounts

Figure 11.28 illustrates a generic type of support that can be used beneath a vertical-axis mirror to provide localized support to one region on a large mirror. It consists of a pneumatic cylinder sealed at the top with a rolling diaphragm that supports a metal pad that in turn supports the mirror. An array of these devices can serve as a mounting for a large-axis vertical mirror. A hydraulic version of this device can also be used for the same purpose. Air or oil pressure can be differentially adjusted to provide the correct force to support local areas of a mirror with nonuniform weight distribution. Figure 11.29 shows an array of supports of this general type having circular diaphragms attached to cylindrical pressurized housings. Metal plates ride on the tops of the diaphragms and interface with the back of the mirror (not shown).²¹

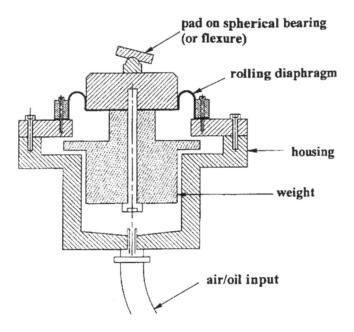


Figure 11.28 Schematic of a generic rolling diaphragm-type pneumatic or hydraulic actuator that can be used for axial support of a localized area on a large mirror.

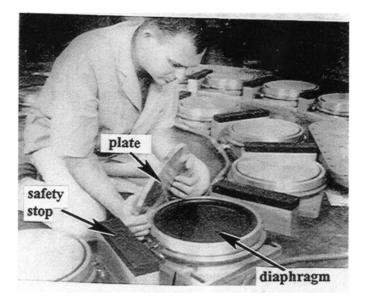


Figure 11.29 Photograph showing several rolling-diaphragm-type supports used for axial support of a 3.8-m (150-in.) diameter mirror during manufacture and test with axis vertical. (Adapted from Cole.²¹)

Hall²² gave an empirical equation for the number of support points N needed to limit the mirror surface deflection between discrete supports of solid mirrors as large as ~100 in. (2.54 m) to a tolerable value δ (in waves). Slightly simplified, this equation is:

$$N = \left(\frac{0.375D_G^2}{t_A}\right) \left[\frac{\rho}{(E_G\delta)}\right]^{\frac{1}{2}}.$$
(11.5)

All parameters are as previously defined. This equation is applied in Example 11.2.

Example 11.2: Number of discrete points needed to support a large axisvertical mirror. (For design and analysis, use File 11.2 of the CD-ROM.)

A plane-parallel fused-silica mirror with diameter 39.370-in. (100.000-cm), thickness of 5.625 in. (14.288 cm) is to be supported on N small areas with axis vertical. How many support points are needed to ensure that the surface deflection does not exceed 0.01 wave at 0.633-µm wavelength?

From Table B8a: $\rho = 0.080$ lb/in.³ and $E_G = 10.6 \times 10^6$ lb/in.²

By Eq. (11.5):

$$N = \left[\frac{(0.375)(39.370^2)}{5.625}\right] \left[\frac{0.080}{(10.6 \times 10^6)(2.492 \times 10^{-7})}\right]^{\frac{1}{2}} = 17.98 \text{ or } 18 \text{ points}$$

Because the mirror must be relatively stiff and approximate a plane-parallel plate for this equation to apply, it should be used only for mirrors having uniform or nearly uniform thickness (plane-parallel or meniscus) and thickness-to-diameter ratios of 6:1 or less. Surface deflections under gravity- and mount-induced forces for mirrors meeting these criteria, but having curved optical surfaces and flat backs, meniscus mirrors (which behave as shells), and all larger mirrors would best be analyzed by FEA techniques. Many of these mirror configurations can also be evaluated by closed-form methods described by Mehta.²³

A mount used to support a mirror during testing in the final stages of manufacture is frequently referred to as a "metrology mount." Generally, the mirror's axis is vertical and its mount must accurately locate the mirror surface relative to the optical test equipment and provide a stable, predictable, and repeatable support for the mirror. The metrology mount most frequently employed simulates a zero gravity environment. Typically, such a mount supports the mirror at many points so that self weight deflections from three strategically located position/orientation defining points and self weight deflections between support points are within specification. If the same mount is to be used to support a mirror during polishing as well as during testing, the additional loading due to polishing tools and auxiliary weights must be considered when computing the number of support points using Eq. (11.5).

The most common types of metrology mounts are those using pneumatic or hydraulic actuators and those using mechanical levers with counterweights and/or springs. A 36-point pneumatic mount suitable for supporting a 150-in. (3.8-m) diameter mirror was described by Cole.²¹ It used the support system shown, in part, by Fig. 11.29.

Figure 11.30, from Cole,²¹ shows the integrated fabrication/test facility used to manufacture the plano concave mirror. The metrology mount was located between the back surface of the mirror and the table of a grinding/polishing machine. This mount provided mirror support both during fabrication and during tests. It was fitted with arrays of 36 circular actuator pads, 36 rectangular rubber cushion blocks, and three racetrack shaped air bearings.

During optical tests, the mirror rested on the 36 pads. Three pads from the outer ring were deflated and thin spacers inserted between their pistons and the mirror to serve as hard location-defining points. The maximum air pressure was about 8 lb/in.², slightly different in the two rings, and adjusted so that all 36 pads supported essentially equal parts of the mirror weight.

For polishing, the three spacers were removed and the mirror floated on all 36 pads. The 9000 lb (~4100 kg) weight of the polishing tool pressed the mirror's back against the 36 rubber cushion blocks. Cole pointed out that special precautions were necessary in order for the tool weight to be supported equally by the blocks. The rear surface of the mirror and top surface of the table were lapped to match and the blocks ground to nominally the same height. To distribute the effects of residual small height discrepancies, the mirror was periodically rotated on the mount. To do this, the mirror was lifted by the three air bearings so that it could be moved on a thin layer of air. After rotating, it was lowered back onto the blocks and polishing resumed. After a polishing run was complete, the mirror was cleaned and refloated on the 36 pads for testing. This process was repeated until the tests indicated the mirror to be finished.

In preparation for building the Hubble Space Telescope (HST) primary mirror, NASA authorized Perkin-Elmer Corporation to demonstrate their proposed metrology mount design, integrated polishing/metrology setup, and computer controlled polishing process. A solid ULE mirror of 60-in. (1.5-m) diameter having a 10-in. (25-cm) central hole was prepared as a meniscus with a thickness of 3.82 in. (9.70 cm) so that it simulated the speed (f/2.3) and structural flexibility of the full-sized lightweighted (monolithic) HST primary. The surface figure goal for the demonstration was $\lambda/60$ rms at $\lambda = 0.633 \mu m$. Montagnino et al.²⁴ described the mount (see Fig. 11.31) that was designed to allow that specification to be met. Figure 11.32 shows the mirror on its mount as it was positioned in the test tower for in-process interferometric evaluation. The mount was attached to rails that allowed the mirror and mount to be transported easily from the test station to the fabrication station without disturbing the relative alignment of those components.

The base of the mount was a 60-in. (152-cm) square by 1-in. (2.5-cm) thick cast and annealed aluminum jig plate. Aluminum was selected for dimensional stability, low cost, and lightness of weight. Parallel ribs, 4-in. (10.2-cm) high on 8-in. (20.4-cm) centers were mounted to the upper surface of the baseplate. These provided mounting surfaces for the axial force mechanisms and stiffened the baseplate. Four additional ribs were bolted to the bottom surface of the plate perpendicular to the upper ribs to increase the cross axis stiffness.

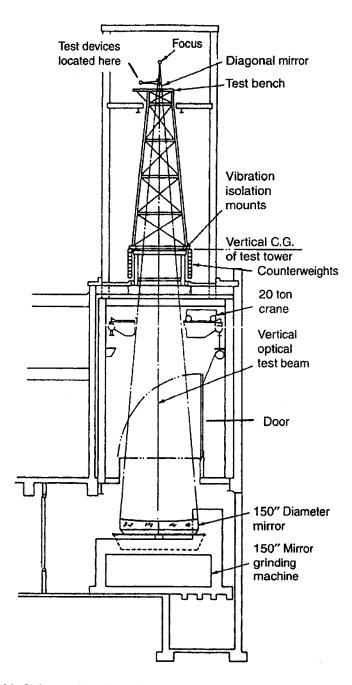


Figure 11.30 Schematic of an integrated polishing/metrology vertical test chamber facility designed for in-situ finishing and testing large mirrors. (Adapted from Cole.²¹)

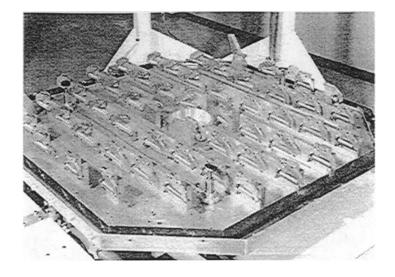


Figure 11.31 Configuration of the 52 point-polishing/metrology mount used to support the 60-in. (1.5-m) diameter mirror during demonstration of polishing/testing of the HST primary mirror. (From Montagnino et al.²⁴)

The force mechanisms, one of which is shown in Fig. 11.33, were designed to provide a very low spring rate. This assured that, after each mechanism was adjusted to a precise force, that force would not be altered by a minor change in mirror position or baseplate deflection. The low spring rate was achieved by use of a nonlinear linkage loaded by a conventional extension spring. The negative force gradient of the linkage was designed to nearly cancel the positive force gradient of the spring. This mechanism could have been designed to have a positive, negative, or zero net spring rate over the normal travel range. In practice, it was determined that a positive spring rate of 2 to 3 lb/in. provided best performance. This made it possible to control precisely the net force reaction at three position control (hard) points by a slight adjustment in vertical position of the mirror. Each linkage was mounted on a flexure pivot of the general type shown in Fig. 11.35 for minimum friction and low hysteresis. The vertical force developed by each force mechanism was transmitted to the mirror by a ball bearing mounted at the end of each lever. Horizontal force components and moments transmitted to the mirror were minimized by this design.²⁵

The bottom surface of the mirror was spherical. Cer-Vit buttons 1.25 in. (32 mm) in diameter were bonded to the spherical surface at each support point to provide a horizontal interface surface for the bearing. This was required to avoid lateral force components that would result if the bearing contacted a sloped surface. Cer-Vit was selected to minimize thermal stress at the ULE mirror interface. Since the bond was compression-loaded, a soft material (RTV silicone rubber) was used as the adhesive to minimize stress in the mirror from bond-curing and to provide for easy removal of the buttons at the completion of the fabrication cycle. A screw adjustment was provided in each mechanism for precise adjustment of spring force. The adjusting screws were located to be accessible for adjustment with the mirror installed on the mount.

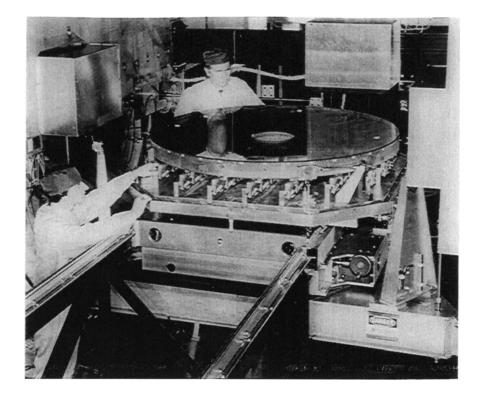


Figure 11.32 Photograph of the 1.5-m (60-in.) diameter mirror and multipoint mount used to prove the manufacturing and testing processes for making the full-sized HST primary mirror. (From Babish and Rigby.²⁶)

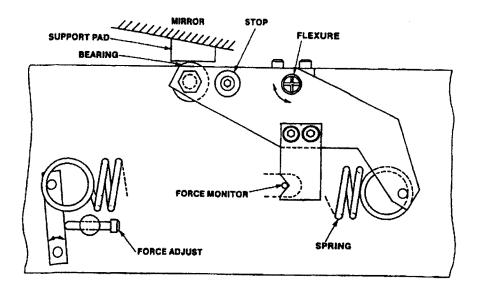


Figure 11.33 Schematic of support lever mechanisms as used in the mount of Fig. 11.32. (Courtesy of Goodrich Corporation, Danbury, CT.)

The compliant force mechanisms did not ensure stability of mirror position. Vertical displacement of the mirror would also affect the calibration precision of the support force mechanisms. Thus, a second condition, stable mirror position, was required for precise metrology operations. This was achieved with three hard points, equispaced around the outer edge of the mirror. It was necessary to monitor vertical force as well as position at these points. With the mirror position restrained at the three locating points, the algebraic sum of the force errors at each of the support force points reacted at the position control points. This required precise calibration of the support forces. The position monitoring points were instrumented to measure the force reaction. This made it possible to trim local support forces with the mirror in place in order to meet the force limits specified at the position control points to limit figure errors due to local bending of the mirror.

A mirror deflection sensitivity analysis indicated that a maximum force reaction on the mirror of ± 0.25 lb (0.11 kg) at each of the three position control points was required to limit local mirror deflection. Conformance to this specification required precise calibration of the support force mechanisms and precise initial centering of the mirror on the mount. Final force balance was achieved by mirror position bias and slight trimming adjustments of support forces near the position control mechanisms. These operations required the position/force monitors to have high force gradients to restrain position and be capable of measuring force over the range of 0 to 6 lb (2.7 kg).

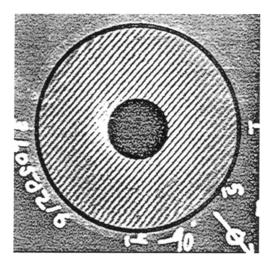


Figure 11.34 Interferogram of the 60-in. (1.5-m) diameter mirror that served as a subscale demonstration model for manufacture and testing of the HST primary mirror. Quality of the mirror was measured as λ /60 rms at 633 nm wavelength. (From Montagnino et al.²⁴)

The matrix of forces applicable over the surface of the mirror was computed from the measured weight of the mirror by a three-dimensional FEA analysis. A series of error analyses defined the tolerable errors in support-force calibration, geometric parameters, thermal distortion, and bearing friction. It was concluded that, with reasonable tolerances on all these variables, the mount would be capable of supporting the mirror adequately for figure measurement to $< \lambda/60$ rms as specified. Figure 11.34 shows an interferogram of the full aperture of the completed mirror. The used clear aperture was 57 in. (145 cm)

and the linear central obscuration 30%. Computer analysis of this interferogram indicated that the specified quality had been attained over this prescribed annular aperture.

Precise control of the lateral position of the mirror during polishing and testing was provided by three tangent bars attaching the mirror to mount structure at symmetrically located position control points. The tangent bars had universal flexures at each end to minimize vertical or lateral force reactions that would affect mirror figure. Both the axial and lateral forces imposed by the computer-controlled polishing technique (see Babish and Rigby²⁶) used to figure both the simulated and actual Hubble Telescope primaries were inherently much lower than with conventional polishing techniques.

In order to minimize lateral shifts of a large axis-vertical mirror due to horizontal forces exerted during conventional polishing on a metrology mount, constraints must be built into the mirror support mechanism. Hall²² reported success with a polishing mount having an array of calibrated compression springs, some or all of which had been damped by submersion in very soft pitch. Figure 11.35 shows one such support.

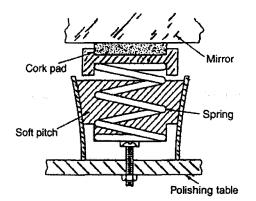


Figure 11.35 Schematic of a damped-spring mechanism that has proven successful in constraining lateral motion of a mirror while being polished conventionally on an integrated manufacturing/metrology mount. (From Hall.²²)

11.3 Mounts for Axis Variable Applications

11.3.1 Counterweighted lever-type mountings

Figure 11.36 illustrates the geometry of a counterweighted-lever mirror-flotation mechanism as used with a typical large solid mirror substrate. Arrays of these mechanisms located strategically (usually symmetrically) about the mirror's back and rim provide axial forces proportional to $\sin \theta$ and radial forces proportional to $\cos \theta$, where θ is the inclination angle of the mirror's axis. Counterweights W_1 and W_2 act through levers hinged to the mirror's cell structure at H_1 and H_2 , respectively. Each of the N mechanisms supports ~1/N times the mirror's weight with lever mechanical advantages of y_2/y_1 and x_2/x_1 , respectively. Typically, the latter ratios range from 5:1 to 10:1. As the elevation angle changes, the support forces automatically switch from one lever system to the other.

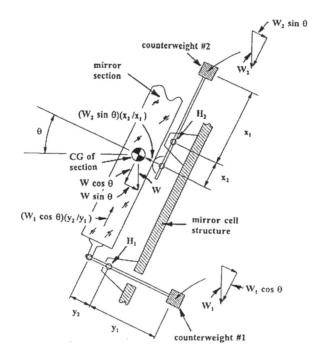


Figure 11.36 Geometry of typical lever mechanisms used for axial and radial flotation of a mirror. The force vector diagrams are not to scale.

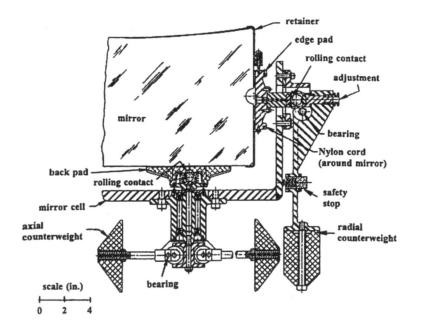


Figure 11.37 Schematics of the mechanisms used to support a 2.08-m (82-in.) diameter primary at the McDonald Observatory. (Adapted from Meinel.²⁷)

Figure 11.37 sketches the lever mechanisms used to support a 2.08-m (82-in.) diameter solid primary mirror in a telescope at McDonald Observatory.²⁷ The axial and radial supports have rolling contacts and bearings under significant loads. The rolling contacts are intended to move when the temperature changes. These contacts and the bearings are plagued by frictional effects (stiction) that resist the small movements that occur during operation.

Franza and Wilson²⁸ pointed out that mirror support levers should function as astatic mechanisms; i.e., the force exerted should be essentially constant in the presence of small changes in the location of the lever fulcrum caused by structural or temperature changes. To illustrate, if a fulcrum of a typical axial support moves a distance of δy as indicated in Fig. 11.38, the angular motion $\delta\theta$ of the lever will be arcsin ($\delta y/x_1$). The corresponding change δF in force F will be $F(1 - \cos \theta)$. For $\delta y = 1 \text{ mm} (0.0394 \text{ in.})$ and $x_1 = 100 \text{ mm} (3.937 \text{ in.})$, δF will be only 0.005% of F. The delivered force remains essentially fixed.

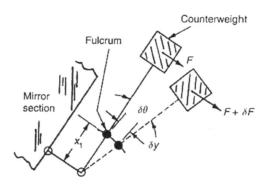


Figure 11.38 Geometry creating a force error δF when the fulcrum of a lever mechanism is mislocated by δy . (Adapted from Franza and Wilson.²⁸)

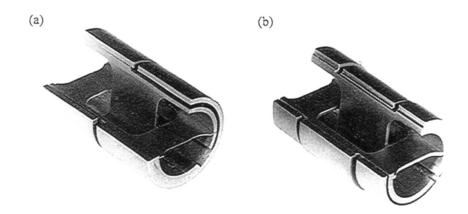


Figure 11.39 Schematics of typical flexure pivots: (a) cantilevered design and (b) double ended design. (Courtesy of Riverhawk Company, New Hartford, NY.)

A potentially serious problem with any lever mechanism is friction in the hinges. The ball and roller bearings used in early designs caused asymmetry and lack of repeatability in the forces applied to the mirrors. This was primarily because those bearings tend to stick when trying to undergo infinitesimal rotations and astigmatism in mirror surfaces typically results. The problem was significantly reduced when flexure bearings of the type shown in Fig. 11.39 became available in about 1960. Some early mirror mountings were modified to use this new technology. This resulted in significantly improved telescope performance.

The flexure bearing was originally invented by Bendix Corporation. Essentially the same product line has been manufactured by a sequence of companies over the years. It currently is available from Riverhawk Company of New Hartford, NY. A typical product has crossed flat flexure blades connecting concentric sleeves. One sleeve is attached to structure while the other is attached to the moveable member. Deflection ranges are typically ± 7.5 , ± 15 , and $\pm 30^{\circ}$. Cantilevered (single ended) and double ended versions are available. They typically are made of 400 series CRES, but other materials are available for special applications. For load and deflection combinations not exceeding 30% of specified maximums, these devices provide essentially infinite life. They do show very small amounts of hysteresis and transverse axis shift as results of angular deflections.

Some ribbed back mirrors have both axial and radial supports built into the same lever mechanism. Supports of this general type were used with large diameter primaries in a telescope on Kitt Peak and the Hale Telescope on Mt. Palomar. These designs are described below. They are included here primarily for their historic value.

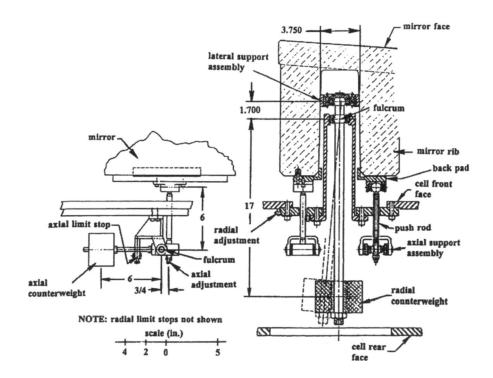


Figure 11.40 Diagrams of axial (back) and radial supports for a large primary mirror used at Kitt Peak National Observatory. (Adapted from Meinel.²⁷)

The Kitt Peak Telescope: Figure 11.40 is a schematic diagram of one of the lever support mechanisms for the 2.13-m (84-in.) diameter primary of a telescope operated some time ago on Kitt Peak. Separate counterweights were used in the axial and radial supports. Figure 11.41 shows a photograph of a similar mechanism used in a different telescope. The function of the latter, and hence of both designs, was described by Baustian²⁹ as follows: "The radial components are transmitted through the ball bearing head located at the upper end of the unit and are carried by a central lever arm whose disc shaped counterweight is visible at the bottom of the unit. The axial component of the mirror weight is supported on the flange located at the central section, with the load being transmitted through three push rods to their individual counterweight levers, located below the mounting flange of the support unit. The cylindrical counterweights are auxiliary balance weights to neutralize the weight of the flange bearing the axial load so as to neutralize its tendency to shift the center of gravity of the mirror." (Ref. 28, p. 16)

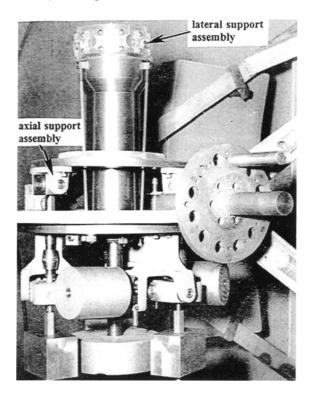


Figure 11.41 Photograph of axial- and radial-support hardware functioning in the manner of the mechanisms shown in Fig. 11.40. (Adapted from Baustian.²⁹)

The 5.1-m (200-in.) Hale Telescope: When the 5.1-m (200-in.) diameter Hale telescope was designed prior to World War II, it was realized that its "lightweight" mirror with D/t = 8.33 would not be stiff enough to be supported at just a few points. The Pyrex mirror blank (see Fig. 11.42) was cast around void formers to create many pockets in the back surface. In this design, radial support would occur deep inside the 36 circular pockets shown in the photograph near the plane of the mirror's center of gravity. Axial support would occur on

annular areas on the mirror's back surface surrounding these holes. Figure 11.43 schematically shows one of the mechanisms used.

The mirror here is looking at the zenith. The following is a description of the function of this mechanism paraphrased from Bowen:³⁰

The support ring, B, makes contact with the mirror in a plane normal to the optic axis through the center of gravity of the mirror. As the telescope turns from the zenith, the lower end of the support system, including the weights, W, attempts to swing about the gimbals, G_1 , and thereby exerts a force on the ring B through the gimbals G_2 in a direction normal to the optic axis. The weights and levers arms are adjusted so that the force exerted balances the component in the opposite direction of gravity acting on the section of the mirror assigned to this support. Likewise, the weights W pivot about bearings, P, in such a way as to exert a force along the rod, R, which is transmitted to the ring, S, by the gimbals, G_2 . These weights and lever arms are likewise adjusted so that the force exerted balances the component parallel to the optic axis of the pull of gravity on this same section of the mirror. The mirror is therefore floating on these supports, and no forces are transmitted across the mirror.

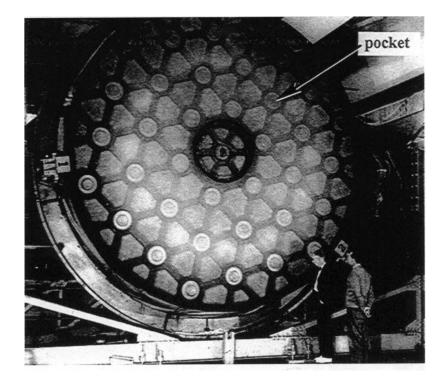


Figure 11.42 Photograph showing the ribbed structure of the 200-in. (5.08-m) diameter Hale Telescope primary mirror. (Adapted from Bowen.³⁰)

To define the orientation of the optic axis of the mirror and the axial location of the mirror, three of the weights located at 120-deg intervals in the outer ring of supports are locked in a fixed position. In the radial direction, the mirror is defined by four pins mounted on the tube that extends through the central hole in the mirror to support the Coudé flat. These pins bear on the inside of the 40-in. diameter central hole in the mirror. They are designed and constructed of materials that compensate for differences in the thermal expansion of Pyrex and steel and operate through ball bearings to eliminate the transmission of forces parallel to the axis.

11.3.2 Hindle mounts for large mirrors

The 10-m (394-in.) Keck Telescope primary: The general characteristics of the Hindle mount were discussed in Section 9.6 in the context of mirror sizes to about 89-cm (35-in.) diameter. Larger mirrors mounted in the same manner need supports. For example, the primary mirrors of the two 10-m (394-in.) aperture Keck telescopes on Mauna Kea contain 36 hexagonal segments, each supported axially by a 36-point Hindle mount. Each segment is Zerodur, 7.5-cm (3-in.) thick, has a 1.8-m (72-in.) point-to-point circumscribed diameter, and has a D/t ratio of 24:1. The optical surfaces have concave radii of curvature of approximately 35 m (1378 in.), but the actual shape of each surface is an asphere.

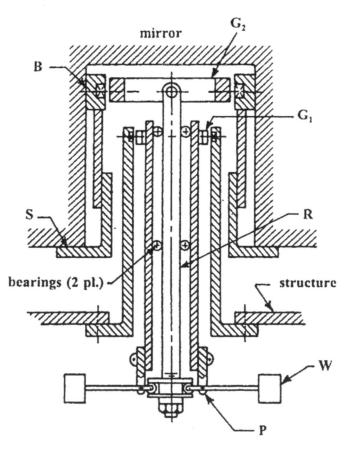


Figure 11.43 One of 36 combined axial/radial supports for the Hale Telescope primary. (Adapted from Baustian.²⁹)

Figure 11.44 shows the mounting layout for one segment while Fig. 11.45 is a photograph of the back of a typical segment. The three whiffletrees each interface with the structure at three points and with the segment at twelve points. Flexure rods that penetrate blind holes bored into the back of the mirror to the neutral surface create the latter interfaces. Because of the meniscus shape of the mirror, this plane is 9.99 mm (0.39 in.) in front of the midplane of the shell. All hinges linking components of the whiffletrees are flexure pivots that allow the required rotations without friction.

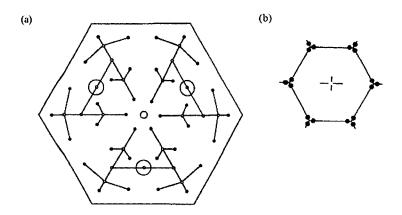


Figure 11.44 (a) Schematic of axial supports for one segment of the Keck Telescope primary mirror. The three large circles represent axial actuators, the 42 small open circles represent flexure pivots, and the solid circles represent the 36 supports. (b) Locations of edge sensors on the segment. (From Mast and Nelson.³¹)

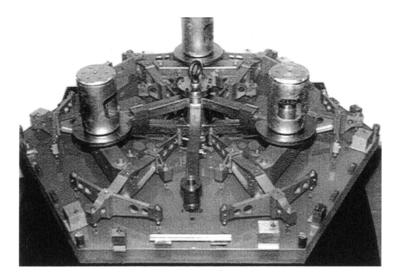


Figure 11.45 Photograph of the back of one segment of the Keck primary mirror showing the three actuators (cylindrical housings) and levers (radially and tangentially oriented bars). A hoisting structure is attached at the center. Portions of edge sensors may be seen around the edge of the mirror. (Courtesy of Terry Mast, University of California, Lick Observatory.)

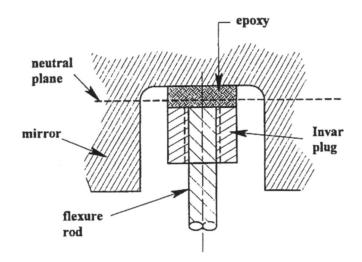


Figure 11.46 Geometry of the axial support interfaces deep within a Keck primary segment. (Adapted from Iraninejad et al.³²)

The locations of the mount-to-mirror attachment points and the geometry of each whiffletree were optimized for minimum rms mirror deflection under gravity loading in the axial direction.³¹ At the bottom of each axial support hole, the flexure rods are attached to Invar plugs that in turn are epoxied to the Zerodur mirror (see Fig. 11.46). Iraninejad et al. showed that the thickness of the epoxy layer was critical in terms of minimizing mirror surface deformation and would be optimum at a 0.25-mm (0.010-in.) thickness.³²

During manufacture, the segments started as uniform thickness, 1.9-m (74.8-in.) diameter by 7.5-cm (2.95-in.) thick meniscus shaped disks. These were ground and polished using stressed mirror techniques developed especially for this purpose.³³⁻³⁵ To contour the optical surface, the blank was supported on its back in the grinding and polishing machine. Lever mechanisms bonded to the rim of the mirror were loaded with weights to apply specific moments to the substrate at specific locations around the mirror's rim. A cross section view of the fabrication station is shown in Fig. 11.47. After being polished to a sphere, the mirror was removed from the machine and cut to the required hexagonal aperture shape. Theoretically, the mirror then would naturally assume an aspheric contour closely, but not exactly corresponding to the intended location for that segment in the aperture of the composite mirror. Sets of six different nonrotationally symmetric aspheres were required to fill the telescope's aperture.

Because of residual global stresses within the substrate that were released during cutting, the figure of each hexagonal mirror had to be corrected in order to achieve the required performance. Since there was good repeatability in the spring back after cutting, localized polishing processes were supplemented by attaching a set of springs collectively called a "warping harness" to elastically correct the residual contour errors of the mirror segment when installed in its 36-support axial mount.³¹ Figure 11.48 shows the locations of the set of springs on one whiffletree. Similar sets are attached to the other two whiffletrees in each segment mount. Each spring was an aluminum bar about $4 \times 10 \times 100$ mm (0.158× 0.394×3.94 in.). Moments were applied to each radial beam pivot by two springs, and an

additional moment was applied by another spring to each tangential beam pivot. These moments were set by manually adjusting screws while measuring the imposed force with a strain gauge bonded to each bar. Each adjustment was locked after setting. The design goal for stability of the adjustment was better than 5% for at least one year over the normal temperature range of $2^{\circ}C \pm 8^{\circ}C$ with full gravity direction variation from zenith to horizon. The total time required to adjust the 18 springs on each segment was typically 45 min.³¹

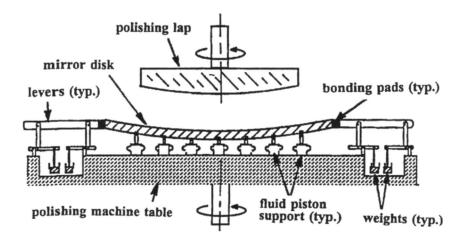


Figure 11.47 Schematic diagram of the stressed mirror grinding and polishing station used to fabricate the Keck primary mirror segments. (Adapted from Mast and Nelson.³¹)

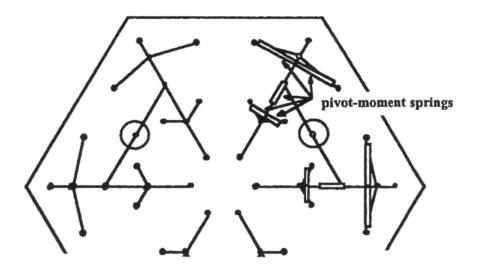


Figure 11.48 Diagram showing the locations of leaf springs added to the whiffletrees of Fig. 11.44 to create a warping harness that allows adjustment of the final figure of the polished mirrors after mounting. (From Mast and Nelson.³¹)

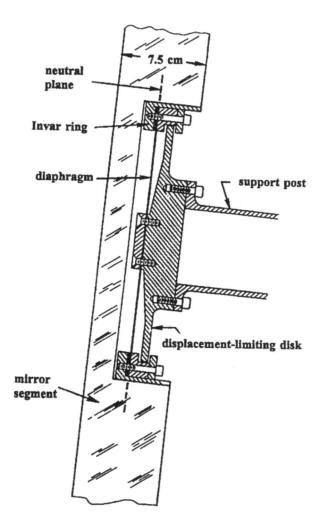


Figure 11.49 Basic concept for radial support of the Keck primary segments. Tangential flexures interfacing the ring to the recess in the mirror are not shown. (Adapted from Iraniejad et al.³²)

Because the axial supports are designed to be soft in the radial direction so as not to impart moments to the optical surface, separate supports are needed to constrain the mirrors laterally. Iraninejad et al.³² defined the design of the radial supports for the Keck segments. A conceptual sectional view of one of these supports is shown in Fig. 11.49. With the telescope axis horizontal, the weight of the segment is supported by a 0.25-mm (0.010-in.) thick flexible stainless steel diaphragm centrally attached to a rigid post extending from the telescope's structure. The edge of the diaphragm is clamped to an Invar ring approximately 10-mm (0.4-in.) thick that is bonded into a circular blind hole of ~254-mm (~10-in.) diameter recessed into the center of the segment. The interface between the cylindrical wall of this recess and the ring is through six 0.4-mm (0.016-in.) thick tangential flexures attached to six Invar pads, as indicated in Fig. 11.50. This feature of the design prevents excessive distortion of the mirror by temperature fluctuations. The pads are epoxied to the ID of the hole with thin adhesive joints. The centers of the flexures and the diaphragm are

located in the plane of the mirror's center of gravity, which is 2.2 mm (0.087 in.) in front of the mirror's midplane. This location does not exactly coincide with the neutral plane of the mirror, so a small (but tolerable) moment is exerted on the mirror when its axis is horizontal.

The use of a diaphragm to support the segment radially allows it to move axially or tilt in any direction by small amounts as needed to align it with respect to its neighbors and to create the required contiguous mirror surface. The orientation and axial position of each segment are measured with twelve edge sensors as illustrated schematically in Fig. 11.51; they are located on the back of the segments, as indicated in Fig. 11.44(b). The sensor body is attached to one mirror and the drive paddle is attached to the adjacent mirror. Narrow air spaces on either side of the paddle are carefully controlled by close tolerancing of parts and careful alignment at assembly. Motions of the drive paddle relative to the sensor body are sensed as a change in capacitance. Signals from a preamplifier and analog-to-digital converter are processed into drive commands for the actuators on the adjacent mirror segments. Tests indicated that the measurement errors were about 9-nm rms.³⁶ This was well below the budgeted error. Since the sensors are mounted to the mirrors, their weights were minimized so as not to significantly affect optical performance as a function of telescope orientation relative to gravity.

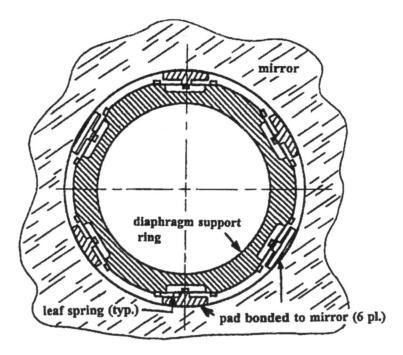


Figure 11.50 Flexure interfaces between the diaphragm support ring and the central recess in the Keck primary segment. (Adapted from Iraninejad et al.³²)

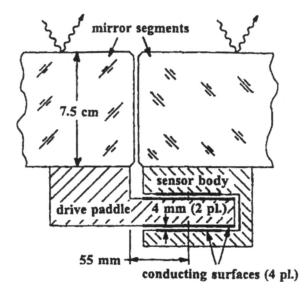


Figure 11.51 Schematic of one edge sensor used to measure the alignment error between adjacent segments of the Keck primary mosaic. (Adapted from Minor et al.³⁶)

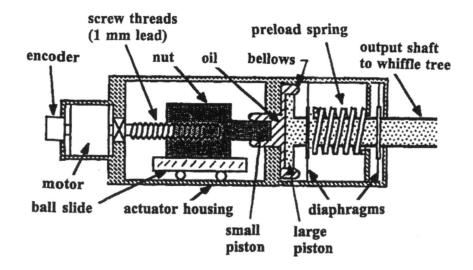


Figure 11.52 Schematic of one actuator used to align the Keck primary segments. (Adapted from Meng et al. 37)

One of the actuators used to align the mirror segments to each other to form a contiguous full-aperture primary mirror for the telescope is illustrated schematically in Fig. 11.52. A shaft extends from a 10,000 position encoder at left through a dc servomotor and

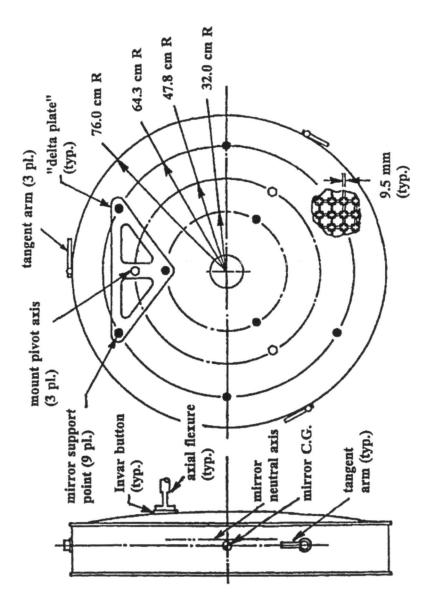
bearing to a 1-mm pitch-threaded interface with a nut that rides on a ball slide. At the right end of this nut is a small piston that presses into a volume of mineral oil encased in a bellows. A larger piston is spring loaded against the right end of the bellows. Rotation of the motor driven shaft advances the large piston and hence the output shaft as well as the center of the associated whiffletree at a rate of 4 nm per encoder increment. A relative mirror positional accuracy of better than 7-nm rms is achieved.³⁷

A laser beam expander: The conceptual design for a 1.52-m (60-in.) diameter fused and slumped monolithic ULE mirror substrate intended for use as the primary mirror of a high energy infrared laser beam expander telescope is shown in Fig. 8.23. Its components are as indicated in Fig. 8.21. The core cells are 7.6-cm (3-in.) squares. The meniscus shaped mirror is 25.4-cm (10-in.) thick, so the D/t ratio is 6:1. Outer and central rings (or edge bands) are provided to increase stiffness and, in the case of the outer ring, to provide a means for attaching three tangent rms to the substrate to serve as radial supports.

This mirror is much stiffer than the Keck telescope primary segments, and the tolerances for surface figure errors here are more lenient than those for an astronomical instrument, so a nine-point Hindle axial support is adequate. The mount configuration is shown in Fig. 11.53. Nine Invar bosses of the general type shown in Fig. 9.38(b) are epoxied to the back of the mirror at the locations indicated by black spots for attachment to the triangular (delta) plates of the whiffletrees. The core elements are thickened in the regions surrounding the nine attachment points for added strength. Dual axis flexures are incorporated into these bosses; one compliant motion is oriented toward the mirror center to accommodate temperature changes while the other bending axis is perpendicular to the first. These flexures act as a "universal joint." A similar dual axis flexure is located at the other end of a short rod connecting the boss to the delta plate corner. In combination, these flexures accommodate minor misalignment errors and/or displacements caused by externally applied acceleration forces.

Three tangent arms support this mirror laterally, as shown in Fig. 11.53. These arms have universal joint flexures at each end. Titanium is typically used in such flexures because of its very high yield stress and excellent fatigue life characteristics. The tangent arms usually are attached to the mirror in the plane of the mirror's center of gravity.

The SOFIA Telescope: As a final example of a Hindle mounting, we show in Fig. 11.54 a back view of the mounting for the 2.7-m (106.3-in.) diameter f/1.19 lightweighted parabolic Zerodur primary mirror for the SOFIA telescope (see sketch of the mirror in Fig. 8.36). This mounting is an 18-point axial support system with three whiffletrees, one of which is shown in Fig. 11.55. The entire mount is shown in exploded fashion in Fig. 11.56. The support rods attach to bosses (or pads) epoxied to the mirror's back surface in equilateral triangle patterns. Universal joint flexures can be seen at each end of these rods. The lower end of each rod attaches to a triangular support structure (called "star panel" in the figure) that serves as load spreaders. These star panels are attached through flexures, each with two degrees of freedom, to a center panel that, in turn, is attached at its center of gravity through a one-DOF pivot bearing to the center of one of three mirror cell support beams. The latter beams are attached to the shear box. That box is attached to the telescope structure at its center of gravity through one DOF flexures to the centers of three mirror support beams. The latter beams are rigidly attached to the shear box, which is, in turn, attached to the main structure of the telescope.





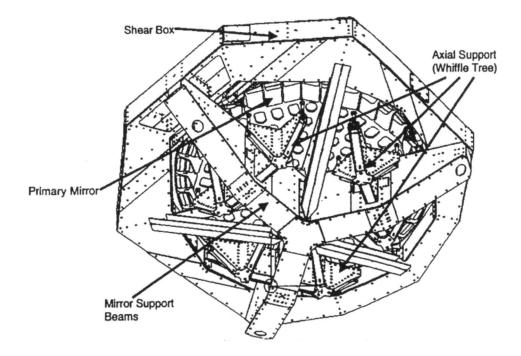


Figure 11.54 Back view of the SOFIA primary in its Hindle mount. (From Erdmann et al.³⁹)

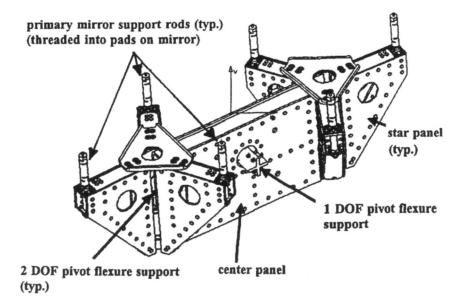


Figure 11.55 One whiffletree from the SOFIA mirror mount of Fig. 11.54. (From Erdmann et al.³⁹)

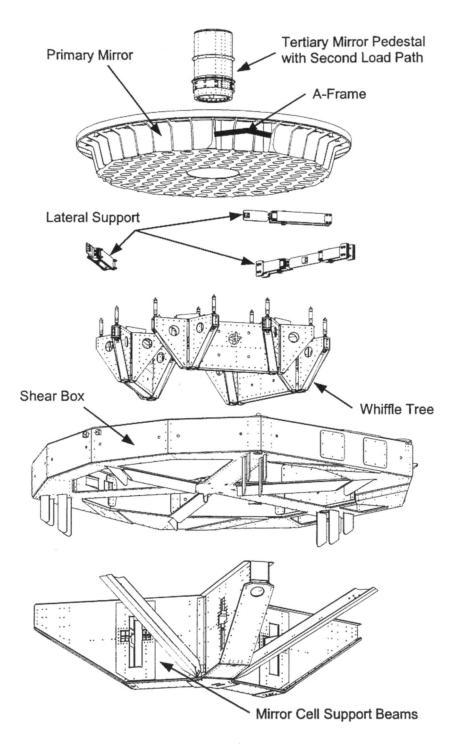


Figure 11.56 Exploded view of the SOFIA primary mirror mount. (From Bittner et al. 40)

Three tangentially oriented arms provide lateral support for the mirror. These are shown in Fig. 11.57. Each arm is attached through radially compliant flexures to brackets attached, in turn, to the shear box. Attached to the pad at the center of each arm is a curved stainless steel bipod. Figure 11.58 shows FEA models of the mirror with the bipods attached to the mirror's rim and the FEA model used to design and characterize the bipod. Necked down regions provide the desired flexibility in all directions except that tangent to the mirror.⁴¹ Each end of each bipod is attached by four screws to an Invar pad that is bonded to the mirror rim.

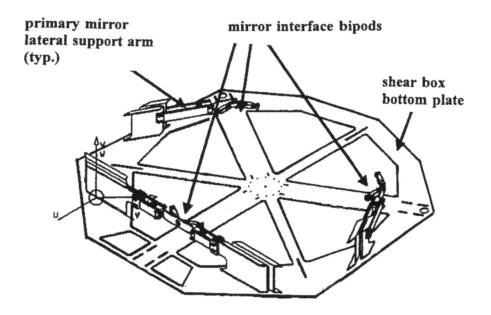


Figure 11.57 Three lateral support arms for the SOFIA primary attached to the shear box. (From Erdmann et al.³⁹)

Because the normal mode of operation of the SOFIA telescope is in a Boeing 747SP aircraft with no window, air stream turbulence will cause extreme vibrational disturbance at frequencies up to 100 Hz. The telescope and all its parts must therefore be of high stiffness. This is the reason for the complex designs of both the axial and radial supports for the primary mirror. Analyses indicated the lowest resonance of the mirror in its supports to be about 160 Hz. This favorable performance is due to a large degree to the use of a carbon fiber reinforced composite material with high stiffness, low density, and low CTE in the mounting. Other materials used are steel and titanium. The optomechanical design is athermalized by judicious choice of materials, component dimensions, and intercomponent interfaces. Only low outgassing materials are used in the mirror mounting so the mirror can be cleaned and recoated without removing it from the cell.

Wavefront error maps and interferograms of the mounted mirror at zenith and horizon orientations are reproduced in Figs. 11.59(a) and (b). The computed rms wavefront errors in these orientations are 278 nm and 283 nm, respectively. These values are considered excellent.⁴¹

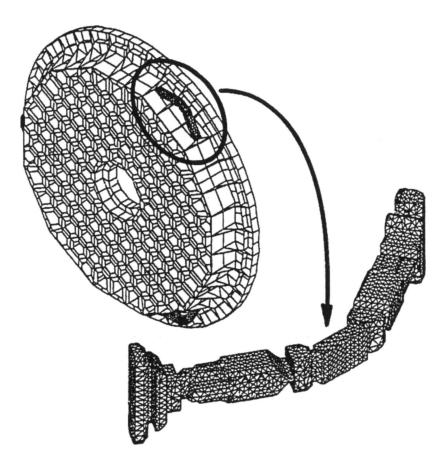


Figure 11.58 FEA models of the SOFIA primary with three bipods attached to provide lateral support and one of those bipods. Note the multiple flexures in the bipod. (From Geyl et al.⁴¹)

As a safety precaution, a tubular structure with a flange has been designed into the SOFIA primary mirror cell to protrude through the mirror's central hole. This feature of the structure (see Fig. 11.60) does not normally touch the mirror, but provides a mechanical constraint about 0.5 mm (0.020 in.) away that would catch and hold the mirror if the bonded axial and radial supports discussed earlier were to fail during an emergency landing of the aircraft.³⁹

11.3.3 Pneumatic and hydraulic mountings

The supports for large mirror substrates described in this section use pneumatic and/or hydraulic actuators to apply forces at multiple points on the mirror. Since the mirrors are not stiff and their mass distribution is not uniform, the force delivered by any one actuator will typically be different from the others in a given mount. Each force is usually controlled by monitoring it directly and "closing the loop" of an associated servo system so that the actual force corresponds to the unique value needed at each support point. Means for locating and orienting the mirror are also needed. These usually are independent of the axial and radial supports. The mounts should be astatic so that small temporary or permanent errors in alignment or externally induced dimensional changes, such as thermal effects, do not adversely affect the mount's performance.

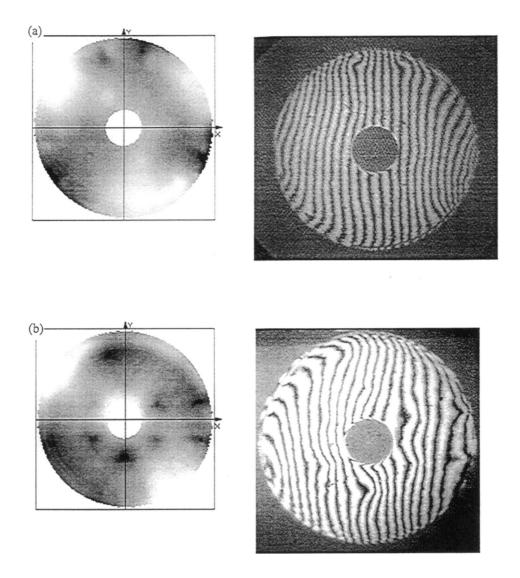


Figure 11.59 Wavefront error maps and interferograms of the mounted SOFIA primary oriented (a) with axis vertical (zenith) and (b) horizontal. (From Bittner et al.⁴⁰)

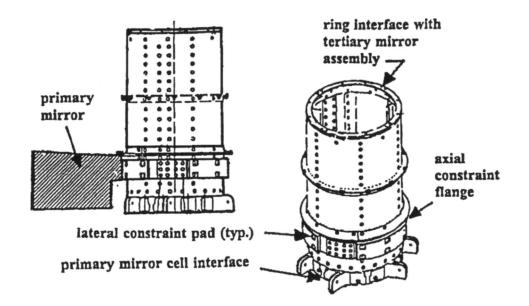


Figure 11.60 Features built into the SOFIA tertiary mirror support structure to provide an emergency landing safety constraint for the primary mirror. (From Erdmann et al.³⁹)

A large pneumatically supported telescope primary: One of the few mirrors of this type for which technical information is available in the open literature is the 4.2-m (165.4-in.) diameter plano concave solid primary for an astronomical telescope designed in the UK for use at the Spanish International Observatory in the Canary Islands. It also was one of the first mirror mountings designed with the aid of FEA. With a D/t ratio of 8:1, it was supported axially by a three-ring array of pneumatic actuators and radially by a series of identical counterweighted levers (see Fig. 11.61).⁴²

The axial support system consisted of rings of 12, 21, and 27 supports at radii of 0.798 m (31.4 in.), 1.355 m (53.3 in.), and 1.880 m (74.0 in.), respectively. These supports were circular pads of 298.5-mm (11.75-in.) diameter to distribute the forces over significant areas on the back of the mirror. Analysis predicted the stress distribution that was due to the axial mount through a section of the mirror as shown in Fig. 11.62. Estimates of the peak deviation from the true parabola and the change in focal length were 3 nm and <0.01 μ m, respectively. This performance was judged satisfactory.

The radial supports were arranged to exert parallel push pull forces on the mirror's rim as indicated in the FEA model of Fig. 11.63. These forces were all the same in magnitude and supported twelve vertical slices of equal weight. These forces were directed toward the centers of gravity of the individual slices, taking into account the curvature of the optical surface. Analysis showed that the positive deformations induced by gravity into the lower half of the mirror were balanced by the equal negative deformations of the upper half. This caused the reflected wave front to be tilted slightly in the vertical plane, but the peak departures of the surface from a true parabola would not exceed a tolerable 0.03 μ m. Figure 11.64 shows how the induced stresses were concentrated at the lower extreme of the aperture when the telescope axis was horizontal.

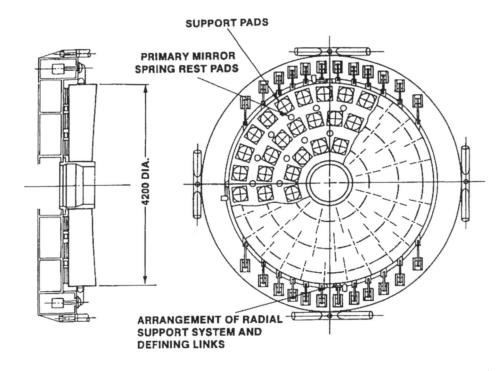


Figure 11.61 Schematic of the mount for an early 4.3-m (165.4-in.) diameter mirror. (Adapted from Mack.⁴²)

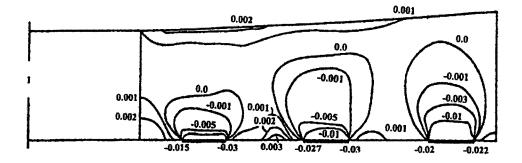


Figure 11.62 Analytically estimated distribution of stress in the mirror in Fig. 11.61 resulting from its ring supports with the axis vertical. (Adapted from Mack.⁴²)

The converted "multiple mirror" telescope: The multiple mirror telescope (MMT) on Mt. Hopkins originally had six mirrors of 1.8-m (70.9-in.) diameter arranged in a ring to achieve an effective aperture of 4.5 m (177.2 in.). As mirror-making technology improved, it was decided to redesign the telescope to utilize the largest single mirror that would fit inside the existing elevation yoke and a slightly modified observatory building.⁴³ A 6.5-m (256.5-in.) diameter borosilicate honeycomb mirror with relative aperture of f/1.25 was spin cast for this purpose. This mirror was mounted in a cell that served multiple structural purposes as well as holding the mirror.

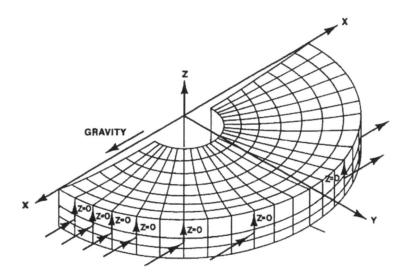


Figure 11.63 The three-dimensional FEA model used to analyze surface deformations of the mirror in Fig. 11.61 caused by gravity. (Adapted from Mack.⁴²)

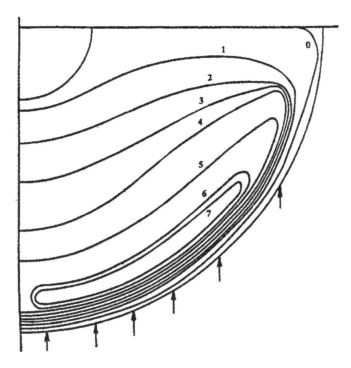


Figure 11.64 Estimated stress distribution in the mirror in Fig. 11.61 caused by gravity when the axis is horizontal. (Adapted from Mack⁴².)

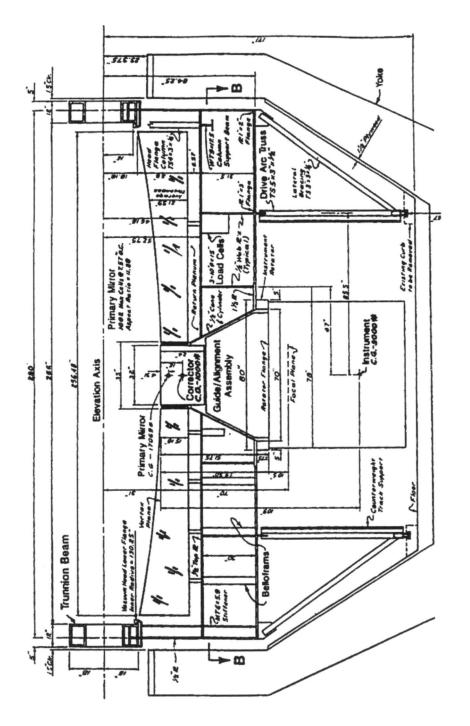
Figure 11.65 shows the new plano concave mirror in its cell with the associated components aft of the elevation axis. The mirror is supported axially and radially by 104 pneumatic actuators (called "Belloframs" in the figure). Figure 11.66(a) shows part of the honeycomb structure with actuators acting independently and through double and triple whiffletree load spreaders at the points indicated.⁴⁴ Figure 11.66(b) shows (as rectangles) typical locations of the supports for the actuators on the octagonal-shaped mirror cell. This cell consists of a top plate reinforced by a grid of 30-in. (762-mm) high webs that form compartments.⁴⁵

These compartments contain the actuators and other mechanisms and are closed by removable covers, but are interconnected by holes through the webs to form part of the thermal control system.⁴⁵ A return plenum for this system is formed by the space between the back of the mirror and the top plate of the cell as shown in Fig. 11.67.

Pressurized air from an off-board chiller and blower is forced through each of many ejector nozzles and passes through jet ejectors drawing air from the mirror cells and into the input plenum. The mixture of new air and the air from the mirror cells exhausts from that plenum back into the mirror cells via a series of ventilation nozzles. About 10% of the air volume escapes from the cell to allow space for the pressurized air input. This forced air ventilation through the honeycomb cells of the mirror keeps that optic within 0.15°C of ambient and isothermal to 0.1°C.⁴⁶⁻⁴⁸ This is consistent with the findings of Pearson and Stepp⁴⁹ and Stepp⁵⁰ regarding thermal gradient effects on telescope image quality. Details of the design of the thermal control system and of the temperature-sensing system for the new MMT mirror may be found in Lloyd-Hart⁴⁸ and in Dryden and Pearson.⁵¹

As indicated earlier, most of the actuators contact the mirror through load spreaders that function exactly as their name suggests. Diagrams of these mechanisms are shown in Fig. 11.68. The actuator attachment is at the center of each device. The frames for these load spreaders are made of Invar and steel with dimensions to match thermal deformations of the Ohara E6 glass in the mirror. Contact is through 100-mm (3.94-in.) diameter pucks made in two parts from the same batch of steel so the CTEs are the same. The lower part is a conical annulus to minimize weight and optimize distortions induced by the loading. The upper part has a necked rod flexure to decouple the puck from twisting of the load spreader frame. Each puck is attached to the mirror with a 2-mm (0.078-in.) thick layer of silicone rubber adhesive (Dow Corning Type 93-076-2) whose compliance absorbs thermally induced stresses and cushions the load.

Also shown in Fig. 11.68 are rubber static stops that are spaced at short distances from the corners of the load spreaders to serve as mirror constraints if the air pressure to the actuators were to fail during operation or when the system is inactive. These are commercial engine mounts; they consist of rubber "donuts" bonded to steel shafts. Shoulder bolts connected to the corners of the load spreaders limit shear and axial tension forces when the stops are in use.





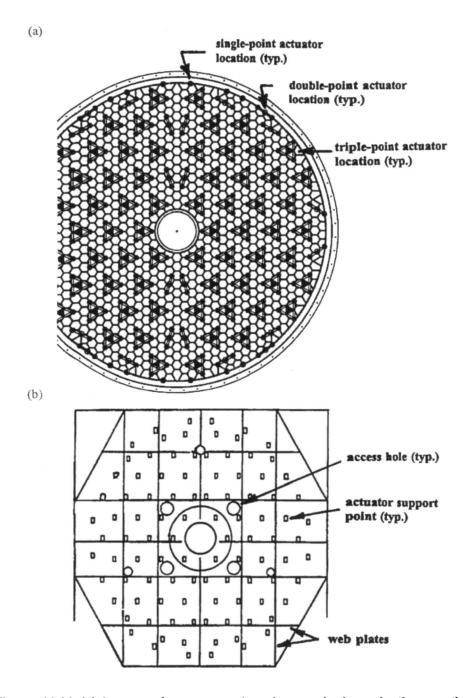


Figure 11.66 (a) Layout of support points (some single and others acting through double and triple load spreaders) on a portion of the MMT honeycomb mirror substrate. (Adapted from Gray et al.⁴⁴) (b) Section view B-B' from Fig. 11.65 passing through the support mechanisms. (Adapted from Antebi et al.⁴³)

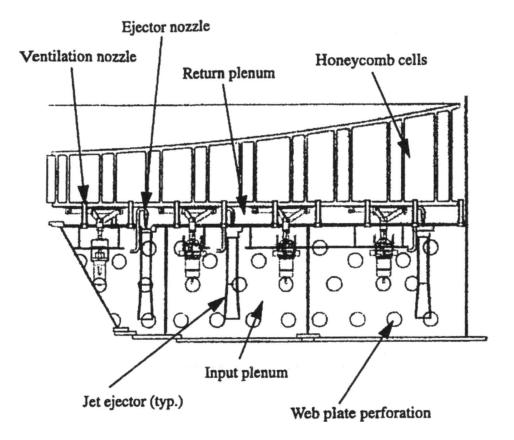


Figure 11.67 Layout of thermal control system components in the MMT mirror cell. (From West et al.⁴⁵)

The actuators themselves consist of pneumatic cylinders with pressure regulators, a load cell for force feedback, and a ball decoupler to eliminate transverse forces and moments. Figure 11.69 shows the two basic configurations. At left is a single-axis actuator with a double load spreader. It provides axial force only. The arrangement at right has two actuators, one working axially and the other working at 45 deg to the mirror's back surface. There are 58 of the latter type of devices. They apply radial forces near the back plate of the mirror and therefore produce moments as well as deflections. An enlarged view of the two-actuator device is shown in Fig. 11.70.

The MMT mirror is constrained in its cell as a rigid body by hard point supports as indicated in Fig. 11.71. Each of these supports is adjustable, but it becomes a stiff strut connecting the back plate of the cell to the back plate of the mirror when clamped. These struts are arranged as three bipods so orientation and location of the mirror is completely determined. Each strut includes a load cell that provides information that is fed back to the actuators. Adjustments are made so that the force exerted at each hard point is nearly zero.

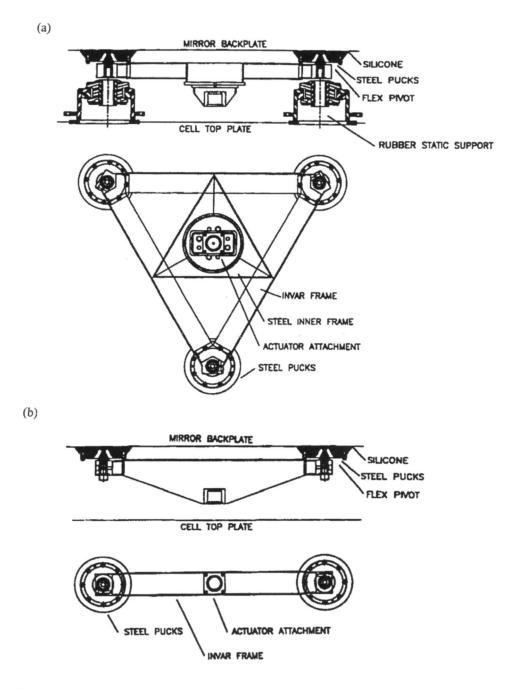


Figure 11.68 (a) Triple and (b) double load spreaders for the new MMT primary mirror. (Adapted from Gray et al. 44)

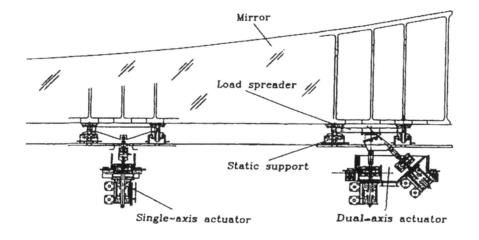


Figure 11.69 Schematics showing (a) a single-axis actuator and (b) a dual-axis actuator as used to support the new MMT mirror. (From West.⁴⁵)

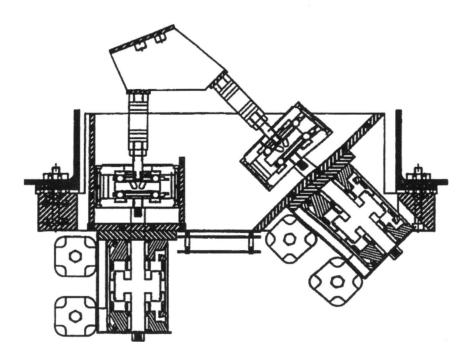


Figure 11.70 Details of the dual-axis actuator used at 58 locations on the new MMT primary mirror. (From Martin et al. 52)

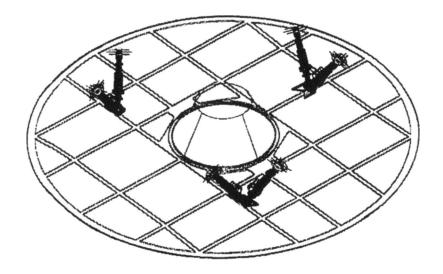


Figure 11.71 Configuration of six struts in bipod arrangements to provide hard points for determination of location and orientation of the new MMT mirror. (From West et al.⁴⁵)

The Gemini Telescopes: The pneumatic mounting for each of the two 8.1-m (318.9-in.) diameter ULE meniscus primary mirrors in the Gemini telescope is of distinctly different design from those previously discussed. The axial support consists of uniform air pressure enclosed by seals at the outer and inner edges of the 220-mm (7.87-in.) thick mirror plus 120 mechanisms that include a passive hydraulic cylinder and an active pneumatic actuator.⁵⁴ The radial support has 72 hydraulic mechanisms located around the rim of the mirror. Both axial and radial supports use hydraulic whiffletree systems to define the position of the mirror. These are adjusted as the telescope changes orientation, introducing small controlled mirror translations and tilts to maintain alignment with the rest of the telescope optics. The mounting system also enables compensation for thermally induced surface deformations; errors in force magnitude, angle, and position; radial support errors; and air pressure errors. In addition, the system can compensate for gravity sag of the secondary mirror and, if needed, change the primary figure from the parabola of the Cassegrain mode to the aspheric of the Ritchey-Chretien mode.⁵⁴

Most of the weight of the mirror is supported axially by a mechanical "air bag" for which the mirror acts as one wall and the cell makes up the other wall. Flexible rubber seals at the inner and outer edges of the mirror complete enclosure of the pressurized region. The pressure required to float most of the mirror's weight is approximately 3460 Pa (0.5 lb/in.²). This air pressure, combined with the force exerted by the seals, produces a small amount (~100-nm rms) of spherical aberration in the mirror's surface. The active support system easily compensates for this error.

About 20% of the mirror's weight is carried by the 120 support and defining mechanisms. This means that the actuators operate in push mode only and do not need to be connected (by bonding) to the mirror. Removal of the mirror from its cell for recoating is significantly simplified by this design choice.

Figure 11.72 shows the 120 support points arranged in five rings of 12, 18, 24, 30, and 36 contacts. The localized forces of magnitudes between 285 N (64 lb) and 386 N (86.7 lb) produce bumps on the surface, as indicated by the contour maps, but their maximum heights are only about 10-nm rms. Since these errors are fixed on the surface, they can be compensated for by localized polishing in the zenith-looking orientation during manufacture so the "print-through" pattern disappears. Throughout the operational inclination range of 0.5 deg to 75 deg, the air pressure is controlled, so the errors are tolerable.⁵⁴

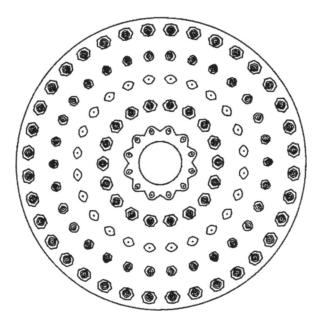
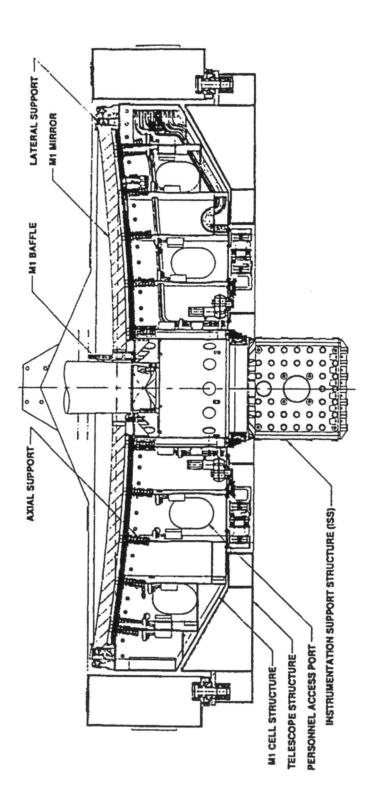


Figure 11.72 Surface contours of the Gemini primary mirror under 80% uniform pneumatic axial support plus 20% localized axial support at 120 points. The effects of air pressure seals are included. Contour interval is 10 nm, surface figure is 54 nm p-v (10-nm rms). (From Cho.⁵⁴)

The Gemini primary mirror assembly consisting of the cell, mirror, and axial and radial actuators is shown in Fig. 11.73. The welded steel mirror cell was designed in the manner of a honeycomb mirror structure to ensure stiffness without adding excessive weight (see Fig. 11.74). It is supported from the telescope structure on four bipods oriented at 45-deg to the elevation axis at a radius of about 60% of the cell radius as illustrated in Fig. 11.75 This bipod orientation was chosen because any distortion of the cell from horizon-pointing loading is symmetrical about the Y axis and anti-symmetrical about the X axis. This minimizes flexure of the mirror. Furthermore, under normal loading conditions, flexure of the telescope will not bend the mirror. Typical worst-case distortion contours of the cell's top surface (to which the mirror is attached) at the zenith and horizon are shown in Fig. 11.76. FEA analysis indicated that the expected cell distortions are within the allowable error budget for that portion of the system.^{53,54}





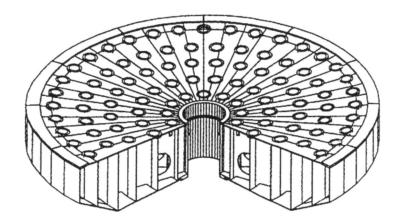


Figure 11.74 Cut-away sketch of the honeycomb structure of the Gemini mirror cell. (From Stepp et al. 53)

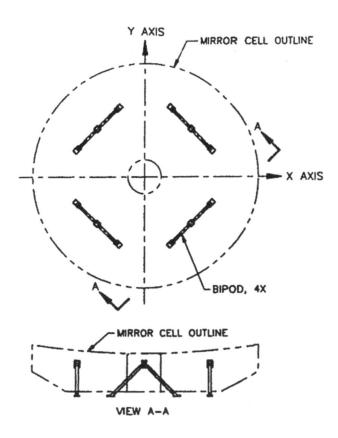


Figure 11.75 Schematic of the bipod support for the Gemini mirror cell. (From Stepp et al. 53)

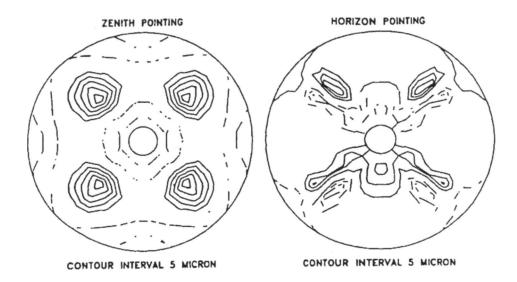


Figure 11.76 Anticipated extreme gravitational deflections of the top surface of the Gemini mirror cell on the four-bipod mounting. (From Stepp et al.⁵³)

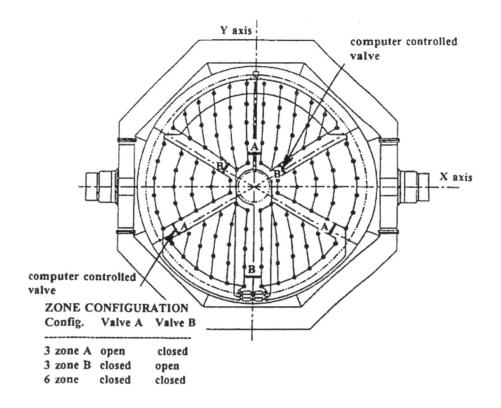


Figure 11.77 Schematic of the three-zone and six-zone hydraulic system modes in the Gemini primary mount. (From Huang.⁵⁵)

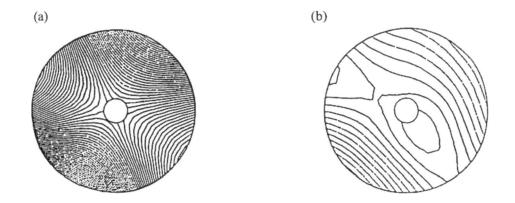


Figure 11.78 Gemini mirror surface deformation under a typical uneven wind load with axial support operating in (a) three-zone (semikinematic) and (b) six-zone (overconstrained) modes. (From Huang.⁵⁵)

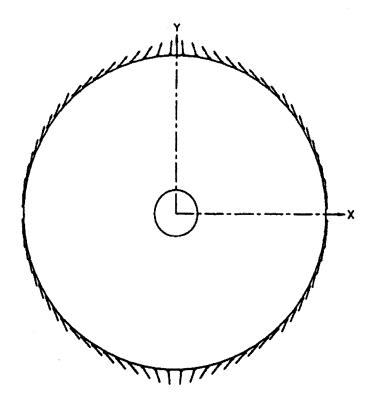


Figure 11.79 Distribution and resultant directions of radial support forces applied to the rim of the Gemini mirror. (From $Cho.^{53}$)

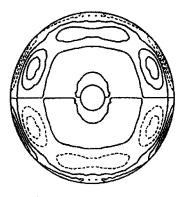


Figure 11.80 Gemini mirror surface contours with typical radial support optimization. Contour interval is 5 nm surface figure is 38 nm p-v (5-nm rms). (From Cho.⁵³)

11.4 Supports for Large, Space-borne Mirrors

The major differences in large mirror mounting design for a space application are the large accelerations experienced during launch, release of gravity effects as orbit is achieved, and thermal effects. The first of these generally requires a means for locking or "caging" the mirror mount so that shock and vibrations do not damage the mechanisms or optics, while the second requires that the optics be supported in a different manner during operation than was used during manufacture and testing on Earth. The operational temperature distribution may vary from that predicted in advance. Techniques for accommodating these conditions are discussed in the following hardware example.

11.4.1 The Hubble Space Telescope

The primary mirror for the Hubble space telescope is a 2.49-m (98 in.) diameter by 30.5-cm (12 in.) thick fused monolithic structure similar in construction to the mirror shown in Fig. 8.23. It is made of Corning 7941 ULE. The clear aperture of the mirror is 2.4 m (94.5 in.). The central hole is 71.1-cm (28-in.) in diameter. The front and back facesheets are nominally 2.54-cm (1.0-in.) thick and are separated by 25.4-cm (10.0-in.) by 0.64-cm (0.25-in.) thick ribs on 10.2-cm (4.0-in.) centers. Inner and outer edge bands also 0.64-cm (0.25-in.) thick and equal in depth to the ribs form circumferential reinforcements. Three localized areas within the core where the flight supports discussed below are attached have somewhat thicker ribs for added strength. The mirror weighs 4078 kg (1850 lb); this is approximately 25% of the equivalent solid structure.

The concept for the multipoint mounting arrangement used to support the primary mirror during manufacture and testing with the axis vertical was described in Section 11.2.3 in the context of the preparation and testing of a scaled-down version of the flight mirror.

Additional details about the 134 point-metrology mounting for the flight mirror were given by Krim.²⁵ Analysis indicated that this number of support points was sufficient to simulate the gravity free condition of operation. The actual thicknesses of the mirror's components were mapped ultrasonically to an accuracy of ± 0.05 mm (± 0.002 in.) as required inputs to a detailed FEA model of the substrate that was used to determine the distribution of forces required to support the mirror.

After polishing was completed, the mirror was transferred from the metrology mounting to its flight mounting. There the mirror is supported axially by three stainless steel links that penetrate the substrate at the three locations indicated in Fig. 11.81. The cell structure of the mirror can also be seen in this photograph. It is supported radially by three tangent arms attached to Invar saddles bonded to the back of the mirror. These supports are hidden in Fig. 11.81.



Figure 11.81 Front view of the primary mirror for the Hubble Space Telescope during preparation for coating. The internal cell structure and the forward ends of the flight axial supports can be seen. (Courtesy of Goodrich Corporation, Danbury, CT.)

Figure 11.82 is a schematic of the rear surface of the mirror. The detail view shows one tangent arm saddle and the clevis that connects it to a bracket. This bracket is, in turn, attached to a main box ring outside the mirror. This ring is attached with tangent arms and axial links to the spacecraft structure. Figure 11.83 shows a schematic partial section view through one of the mirror axial supports. The relationship between the mirror and the main ring is more apparent in this view. The radial supports (tangent arms) for the main ring are not shown.

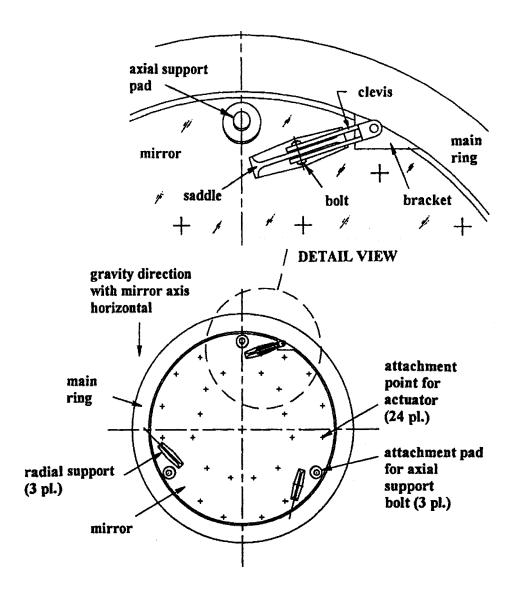


Figure 11.82 Schematic of the rear surface of the Hubble Space Telescope primary mirror showing axial and radial supports as well as attachment points for actuators. (Adapted from drawings provided by Goodrich Corporation, Danbury, CT.)

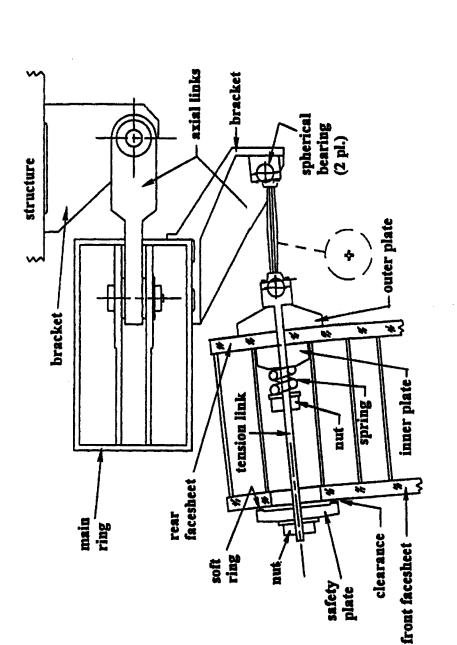
Also shown in Fig. 11.82 are the locations of 24 bosses bonded to the back of the mirror to act as interfaces to actuators provided for limited on-orbit reshaping of the optical surface. These actuator mechanisms used stepper motors to drive precision ball screws to apply localized forces to the mirror. They were not intended for real time control of the mirror's figure, but were provided as a means to correct minor astigmatism anticipated because of gravity release in space. Because the error did not develop, these actuators were not used. Unfortunately, owing to the square core configuration used in the substrate and the limited range of the adjustment, it was not possible to correct mirror curvature or spherical aberration with this figure correcting system. Had means for adjustment of those parameters been provided, on orbit correction of the asphericity error problem accidentally built into the mirror during manufacture might have been possible.

An explanation of the design of the axial supports to the mirror is in order. Referring to Fig. 11.83, we see that a flexure link with a cruciform cross section lies between two spherical bearings. One bearing is attached to the bracket on the main ring and the other to the back of the mirror's rear facesheet. That facesheet is clamped between outer and inner plates that span a cell in the mirror core. Preload is applied by a threaded nut acting through a spring. This mechanism, repeated three places at ~ 120 -deg intervals on the mirror's surface, is all that holds the mirror axially. The design provided sufficient rigidity to hold the mirror during coating, installation in the telescope, shipping, integration into the spacecraft, and launch. It still functions well in space where the gravitational environment is quite benign. This design is also capable of supporting and protecting the mirror during the rigors of Space Shuttle landing in case return of the telescope to earth is attempted. The safety plate and nut outside the front facesheet of the mirror would serve as a stop to hold the mirror axially during the rapid axial deceleration during such a landing. Note the small clearance between the safety plate and the mirror surface and the soft ring pad provided to soften the interface. Forward motion of the mirror would be constrained safely, the tension links transferring force to the bracket and thence to the main ring and structure.

11.4.2 The Chandra X-Ray Telescope

The Chandra Telescope [formerly called the Advanced X-ray Astrophysics Facility (AXAF)] was launched by NASA in July of 1999. It has two major modular assemblies: the optical bench assembly (OBA) and the high resolution mirror assembly (HRMA). The OBA contains the main conical structural component supporting the 1588-kg (3500-lb) HRMA at the forward end and the 476-kg (1050-lb) integrated science instrument module (ISIM) at the aft end. The OBA also contains light baffles, strong magnets that deflect electrons away from the X-ray sensors in the ISIM, electronics, heaters, and wiring.^{56,57}

The optical system, shown in Fig. 11.84, has four concentric cylindrical mirror pairs (paraboloids followed by hyperboloids) that intercept incoming X-rays at grazing incidence (between 0.5-deg and 1.5-deg) and focus them at the focal surface 10 m (394 in.) away. The configuration is known as Wolter I geometry.⁵⁸ The diameter of the largest mirror is 1.2 m (47.24 in.) while that of the smallest mirror is 0.68 m (26.77 in.). The mirrors are 0.84-m (33.1-in.) long. All mirrors were made of Zerodur, chosen for its low CTE $[0 \pm 0.05 \times 10^{-6}/^{\circ}C (0 \pm 0.03 \times 10^{-6}/^{\circ}F)]$, high polishability (better than 7 Å rms surface roughness), and compatibility with fabrication in the required cylindrical configuration. The mirrors were coated with iridium to enhance reflectance of X-rays.





Each mirror is supported at the plane of its axial centroid by twelve titanium flexures oriented as indicated in Fig. 11.85 and attached to Invar pads bonded to the mirrors with epoxy.⁵⁹ It will be noted that this figure shows six nested pairs of mirrors for a total of twelve. The number of mirrors was reduced to four pairs subsequent to publication of the figure. The flexures are bonded with epoxy to the ends of mirror support sleeves made of graphite epoxy. These sleeves are in turn attached to an aluminum central aperture plate. This plate has multiple rings of annular slots for passage of the X rays. Aluminum inner and outer cylinders enclose the mirrors after installation. The outer cylinder interfaces with three sets of bipods (not shown) that link the optical assembly to the optical bench.

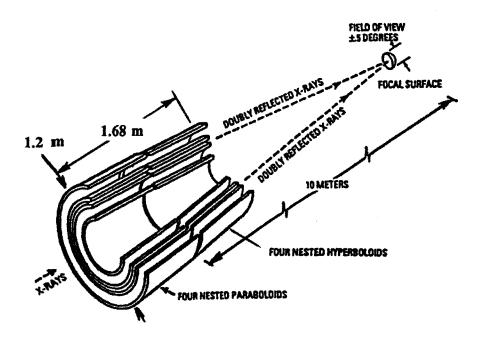


Figure 11.84 Optical system configuration for the Chandra Telescope. (Adapted from Wynn et al.⁵⁶)

The assembled telescope was required to image collected X rays from all four pairs of mirrors at an axial point, with 90% of the energy falling into a circle smaller than 0.05-mm (0.002-in.) diameter. This demanded that the mirrors be aligned within 0.1-arc sec in tilt and centered to a common axis within 7 μ m. To mount the mirrors to the flexures without residual gravity-induced strain, each was first attached to a system of gravity off loaders. These off loaders were attached to a cradle equipped with precision stepper motor-driven actuators capable of moving the mirror in all six degrees of freedom in 0.1- μ m increments. When aligned, the mirrors were tack bonded in place with epoxy to prevent motion caused by hydrostatic pressure from the bonding gun, followed by full bonding and curing. The instrumentation used to sense alignment errors was capable of measuring tilt errors smaller than 0.01 arcsec and lateral displacements of 1 μ m.^{57,60}

The temperature control system for the telescope was designed to actively maintain the internal optical cavity at a constant temperature of 69.8°F (21°C), which duplicates the temperature at assembly. The rest of the telescope is maintained at 50°F (10°C) during observations. Temperature is adjusted by on-board computer controls using radiant heater plates at either end of the optical assembly, and temperature-controlled light baffles. Thermal isolation is provided by an external covering for the HRMA with multilayer insulation (MLI) and the OBA with MLI plus an external layer of silver-coated Teflon film. Both insulation types serve to reject solar radiation. A full aperture door at the forward end of the HRMA served as a contamination shield during launch and, once opened on orbit, shields the optics from direct sunlight beyond 45 deg from the line of sight. The ISIM also is insulated and precisely temperature controlled.⁶¹

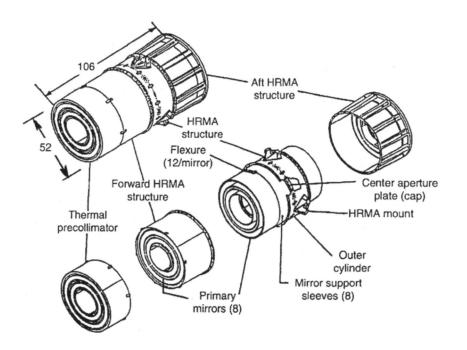


Figure 11.85 Optomechanical configuration of the Chandra Telescope's high resolution mirror assembly (HRMA). (Adapted from Olds and Reese.⁵⁷)

The entire telescope was designed to withstand the rigors of space shuttle launch, during which vibrational reaction loads as great as 30,000 lb (133,450 N) can be encountered. Static and dynamic FEA analyses were conducted throughout the design process to ensure stability of the structure. Measurements of the modal dynamic response of simulated critical components of the telescope to external driving forces verified the models for these analyses.

11.5 References

- 1. Schwessinger, G., "Optical effects of flexure in vertically mounted precision mirrors," J. Opt. Soc. Am. 44, 1954:417.
- 2. Vukobratovich, D., "Optomechanical system design," Chapter 3 in *Electro-Optical* Systems Design, Analysis, and Testing, 4, ERIM, Ann Arbor and SPIE, Bellingham, 1993.

- 3. Maréchal, A., "Etude des effets combinés de la diffraction et des aberrations géométriques sur l'image d'un point lumineux," *Rev. Opt.* 26, 1947:257.
- 4. Born, M. and Wolf, E., *Principles of Optics*, 2nd. ed., Macmillan, New York, 1964:468.
- 5. Malvick, A.J., "Theoretical elastic deformations of the Steward Observatory 230-cm and the Optical Sciences Center 154-cm mirrors," *Appl. Opt.* 11, 1972:575.
- 6. Vukobratovich, D., "Optomechanical design principles," Chapter 2 in *Handbook of Optomechanical Engineering*, A. Ahmad, ed., CRC Press, Boca Raton, 1997.
- 7. Vukobratovich, D., *Introduction t o Opt omechanical Desi gn*, SPIE Short Course SC014, 2003.
- 8. Malvick, A.J., and Pearson, E.T., "Theoretical elastic deformations of a 4-m diameter optical mirror using dynamic relaxation," *Appl. Opt.* 7, 1968:1207.
- 9. Day, A.S., "An introduction to dynamic relaxation," *The Engineer*, 219, 1965:218.
- 10. Otter, J.R.H., Cassel, A.C., and Hobbs, R.E., "Dynamic relaxation," Proc. Inst. Civil Eng., 35, 1966:633.
- 11. Malvick, A.J., "Dynamic relaxation: a general method for determination of elastic deformation of mirrors," *Appl. Opt.*, 7, 1968:2117.
- 12. Draper, H., "On the construction of a silvered glass telescope, fifteen and a half inches in aperture and its use in celestial photography," *Smithsonian C ontributions t o Knowledge*, 14, 1864.
- 13. Vukobratovich, D., personal communication, 2004.
- 14. Vukobratovich, D., and Richard, R.M., "Roller chain supports for large optics," *Proceedings of SPIE* **1396**, 1991:522.
- 15. Hatheway, AE., Ghazarian, V., and Bella, D., "Mountings for a four meter class glass mirror," *Proceedings of SPIE* **1303**, 1990:142.
- 16. Chivens, C.C., "Air bags," A Symp osium on S upport and Test ing of Larg e Astronomical Mirrors, Crawford, D.L., Meinel, A.B., and Stockton, M.W., eds., Kitt Peak Nat. Lab. and the Univ. of Arizona, Tucson, 1968:105.
- 17. Vukobratovich, D., private communication, 1992.
- Crawford, R. and Anderson, D. "Polishing and aspherizing a 1.8-m f/2.7 paraboloid," Proceedings of SPIE 966, 1988:322.
- 19. Doyle. K.B., Genberg, V.L., and Michels, G.J., *Integrated Optomechanical Design*, SPIE Press, Bellingham, 2002.
- 20. Baustian, W.W., "Annular air bag back supports," A Symposium on Support and Testing of Large Astronomical Mirrors, Crawford, D.L., Meinel, A.B., and Stockton, M.W., eds., Kitt Peak Nat. Lab. and the Univ. of Arizona, Tucson, 1968:109.
- Cole, N., "Shop supports for the 150-inch Kitt Peak and Cerro Tololo primary mirrors," Optical Tel escope Technol ogy Workshop, N ASA Rept. SP- 233, NASA, Huntsville, 1970: 307.
- 22. Hall, H.D., "Problems in adapting small mirror fabrication techniques to large mirrors," *Optical Tel escope Technol ogy Workshop, NASA Rept . SP-233*, NASA, Huntsville, 1970: 149.
- 23. Mehta, P.K., "Nonsymmetric thermal bowing of flat circular mirrors," *Proceedings of SPIE* **518**, 1984:155.
- 24. Montagnino, L., Arnold, R., Chadwick, D., Grey, L., and Rogers, G., "Test and evaluation of a 60-inch test mirror," *Proceedings of SPIE* 183, 1979; 109.
- Krim, M.H., "Metrology mount development and verification for a large space borne mirror," *Proceedings of SPIE* 332, 1982; 440.

- Babish, R.C., and Rigby, R.R., "Optical fabrication of a 60-inch mirror," *Proceedings* of SPIE 183, 1979:105.
- 27. Meinel, A.B. "Design of reflecting telescopes," *Telescopes*, G.P. Kuiper and B.M. Middlehurst, eds., Univ. of Chicago Press, Chicago, 1960:25.
- 28. Franza, F., and Wilson, R.N., "Status of the European Southern Observatory new technology telescope project," *Proceedings of SPIE* **332**, 1982:90.
- 29. Baustian, W.W., "The Lick Observatory 120-Inch Telescope," *Telescopes*, G.P. Kuiper and B.M. Middlehurst, eds., Univ. of Chicago Press, Chicago, 1960:16.
- 30. Bowen, I.S., "The 200-inch Hale Telescope," *Telescopes*, G.P. Kuiper and B.M. Middlehurst, eds., Univ. of Chicago Press, Chicago, 1960:1.
- 31. Mast, T., and Nelson, J., "The fabrication of large optical surfaces using a combination of polishing and mirror bending," *Proceedings of SPIE* **1236**, 1990:670.
- 32. Iraninejad, B., Lubliner, J., Mast, T., and Nelson, J., "Mirror deformations due to thermal expansion of inserts bonded to glass," Proceedings of SPIE **748**, 206, 1987.
- 33. Lubliner, J., and Nelson, J.E., "Stressed mirror polishing:1. A technique for producing non-axisymmetric mirrors," *Appl. Opt.* 19, 1980:2332.
- 34. Nelson, J.E., Gabor, G., Lubliner, J., and Mast, T.S., "Stressed mirror polishing:2. Fabrication of an off-axis section of a paraboloid," *Appl. Opt.* 19, 1980:2340.
- 35. Pepi, J.W., "Test and theoretical comparisons for bending and springing of the Keck segmented ten meter telescope," *Opt. Eng.* 29, 1990:1366.
- Minor, R., Arthur, A., Gabor, G., Jackson, H., Jared, R., Mast, T., and Schaefer, B. "Displacement sensors for the primary mirror of the W. M. Keck telescope," *Proceedings of SPIE* 1236, 1990:1009.
- Meng, J., Franck, J., Gabor, G., Jared, R.; Minor, R., and Schaefer, B., "Position actuators for the primary mirror of the W. M. Keck telescope," *Proceedings of SPIE* 1236, 1990:1018.
- 38. Yoder, P.R., Jr., Principles for Mounting Optics, SPIE Short Course SC447, 2007.
- 39. Erdmann, M., Bittner, H., and Haberler, P., "Development and construction of the optical system for the airborne observatory SOFIA," Proceedings of SPIE **4014**, 2000:302.
- 40. Bittner, H., Erdmann, M., Haberler, P., and Zuknik, K-H., "SOFIA primary mirror assembly: Structural properties and optical performance," *Proceedings of SPIE* **4857**, 2003:266.
- 41. Geyl, R., Tarreau, P., and Plainchamp, P., "SOFIA primary mirror fabrication and testing," *Proceedings of SPIE* 4451, 2001:126.
- 42. Mack, B., "Deflection and stress analysis of a 4.2-m diam. primary mirror of an altazimuth-mounted telescope," *Appl. Opt.* 19, 1980:1000.
- 43. Antebi, J., Dusenberry, D.O., and Liepins, A.A., "Conversion of the MMT to a 6.5-m telescope," *Proceedings of SPIE* 1303, 1990:148.
- 44. Gray, P.M., Hill, J.M., Davison, W.B., Callahan, S.P., and Williams, J.T., "Support of large borosilicate honeycomb mirrors," *Proceedings of SPIE* **2199**, 1994: 691.
- 45. West, S.C., Callahan, S., Chaffee, F.H., Davison, W., DeRigne, S., Fabricant, D., Foltz, C.B., Hill, J.M., Nagel, R.H., Poyner, A., and Williams, J.T., "Toward first light for the 6.5-m MMT telescope," *Proceedings of SPIE* **2871**, 1996:38.
- 46. Siegmund, W.A., Stepp, L., and Lauroesch, J., "Temperature control of large honeycomb mirrors," *Proceedings of SPIE* **1236**, 1990:834.

- Cheng, A.Y.S., and Angel, J.R.P., "Steps towards 8 m honeycomb mirrors VIII: design and demonstration of a system of thermal control," *Proceedings of SPIE* 628, 1986:536.
- 48. Lloyd-Hart, M, "System for precise thermal control of borosilicate honeycomb mirrors," *Proceedings of SPIE* **1236**, 1990:844.
- 49. Pearson, E., and Stepp, L., "Response of large optical mirrors to thermal distributions," *Proceedings of SPIE* **748**, 1987:215.
- Stepp, L., "Thermo-elastic analysis of an 8-meter diameter structured borosilicate mirror," NOAO 8-meter Telescopes Engineering Design Study Report No. 1, National Optical Astronomy Observatories, Tucson, 1989.
- 51. Dryden, D.M., and Pearson, E.T., "Multiplexed precision thermal measurement system for large structured mirrors," *Proceedings of SPIE* **1236**, 1990:825.
- 52. Martin, H.M., Callahan, S.P., Cuerden, B., Davison, W.B., DeRigne, S.T., Dettmann, L.R., Parodi, G., Trebisky, T.J., West, S.C., and Williams, J.T., "Active supports and force optimization for the MMT primary mirror," *Proceedings of S PIE* **3352**, 1998:412.
- 53. Stepp, L., Huang, E., and Cho, M., "Gemini primary mirror support system," *Proceedings of SPIE* **2199**, 1994:223.
- 54. Cho, M.K., "Optimization strategy of axial and lateral supports for large primary mirrors," *Proceedings of SPIE* **2199**, 1994:841.
- 55. Huang, E.W., "Gemini primary mirror cell design," *Proceedings of SPIE* 2871, 1996: 291.
- Wynn, J.A., Spina, J.A., and Atkinson, C.B., "Configuration, assembly, and test of the x-ray telescope for NASA's advanced x-ray astrophysics facility," *Proceedings of* SPIE 3356, 1998:522.
- 57. Olds, C.R., and Reese, R.P., "Composite structures for the advanced x-ray astrophysics facility (AXAF)," *Proceedings of SPIE* **3356**, 1998:910.
- 58. Wolter, H., Ann. Phys. 10, 1952:94.
- Cohen, L.M., Cernock, L., Mathews, G., and Stallcup, M., "Structural considerations for fabrication and mounting of the AXAF HRMA optics," *Proceedings of SPIE* 1303, 1990:162.
- Glenn, P., "Centroid detector system for AXAF-I alignment test system," *Proceedings* of SPIE 2515, 1995:352.
- 61. Havey, K., Sweitzer, M., and Lynch, N., "Precision thermal control trades for telescope systems," *Proceedings of SPIE* **3356**, 1998:10.

CHAPTER 12 Aligning Refracting, Reflecting, and Catadioptric Optics

Lower performance optical instruments and ones to be mass-produced are generally assembled by the "drop-in" technique described in Section 4.2. Essentially, one puts the optics in place, secures them by some means, and accepts whatever performance results. If higher performance is needed, one might tighten the tolerances on optics and/or mechanical parts. Other instrument designs keep the tolerances relatively loose and adjust alignment to improve performance. The highest performance instruments, such as optical projection systems for microlithography, set the tolerances to the highest level feasible *and* fine-tune alignment of carefully selected elements to achieve the maximum possible performance.

The premise upon which this chapter is written is that adjusting the alignment of an optic or system of optics and then securing that alignment is a valid method for boosting performance. Naturally, a trade-off exists between the costs of tight tolerances and the costs of adjustment mechanisms, tools, fixtures, and labor. To assist in making this choice, here we consider various aspects of alignment technology. We deal first with techniques that may be used to align individual optical components to their mounts. That discussion relates closely to and extends the discussion of centering techniques given in Section 2.1.2. We then consider alignment of systems comprising lenses, mirrors, and combinations thereof. Space constraints prohibit an exhaustive treatment of these subjects. Instead, we summarize some techniques that have proven useful. Most of these have been described in the literature. References are given wherever possible so the reader can find more details about topics of interest.

There are two closely related aspects of alignment: measuring errors and using mechanisms of some sorts to reduce those errors to acceptable magnitudes. We therefore consider both of these aspects for each technique considered. A final topic receiving special attention is the creation of prealigned modules needing no further adjustment when installed into an instrument. At the single-lens level, the module is sometimes referred to as a "poker chip." When two or more lenses are involved, we have a modular subassembly. Modular design reduces the time and effort required at assembly. It is most frequently applied in cases where the increased design, tooling, and fixturing costs of modularization can be amortized over large quantity production.

12.1 Aligning the Individual Lens

In Section 2.1.2, we described ways in which cylindrical rims, bevels, and other features can be ground on a lens while it is mounted on and aligned to the axis of a precision spindle on a centering machine. We also discussed several techniques for measuring centration errors of the lens during the edging process. The importance of centering the rim to the lens's axis is greatest if that optic is to fit closely into a bore machined into the mount. The resulting "rim-contact" design (see Fig. 2.12) obtains its alignment by virtue of this fit. A "surface-contact" design, on the other hand, obtains lens alignment by preloaded annular contact with a shoulder or similar reference feature within the mount (see Fig. 2.14).

In order for the surface-contact lens mounting to succeed, it is necessary that the design provide sufficient radial clearance around the rim so that rim does not touch the ID of the bore before the lens axis is properly aligned to the mount (see Fig. 2.15). When preload is applied to the lens to hold it against the reference interface, that preload should be uniformly distributed around the edges of both the front and back lens surfaces. Most these designs assume that the lens will be adjusted to align its axis to the mount before the preload is applied. In cases where preload is not provided, such as the elastomeric ring mount described in Section 3.9 or flexure mounts described in Section 3.10, the lens would be aligned before the sealant or adhesive is applied and held in alignment until curing is complete.

It should be noted that one cannot rely on radial components of axial preload to center the surface-contacted lens perfectly because friction is hard to overcome when the differences between the opposing radial forces applied to curved surfaces become small. Smith indicated the lower limit on residual centration error in such cases to be 0.0005 in. $(12.5 \ \mu m)$.¹ In some of the most accurate centering techniques, the lens is held in a fixture of some sort and adjusted to the proper orientation and location relative to some external reference such as a mechanical surface, to a prealigned light beam, or to an interferometer cavity. Once the associated metrology instrumentation verifies adequate alignment, it is clamped mechanically, potted into the mount with elastomer, or bonded to previously aligned flexures or to the mount to preserve that alignment. The fixture can then be stemoved.

12.1.1 Simple techniques for aligning a lens

The simplest way to center a lens to the axis of a cylindrical cavity machined into an instrument housing is to insert three shims or feeler gauges of equal thickness between the lens OD and the cavity ID to equalize the gap around the lens (see Fig. 12.1). If the lens itself has previously been centered so its rim is aligned properly with respect to its optical axis, alignment is complete. If an external alignment-monitoring device is used to define the desired lens location laterally with respect to an external reference, shims can still be used to fix the lens temporarily within the cavity ID, but those shims may need to be of differing thicknesses. In this case, the cavity does not serve as the prime reference for lateral positioning of the lens.

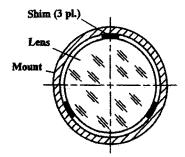


Figure 12.1 Centering a lens in a mount or housing by inserting equalthickness shims into the gap around the lens. The position of the lens is then secured.

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A commonly used means for adjustment of lens centration relative to a lateral positioning reference is illustrated in Fig. 12.2. Here, four setscrews pass through radially directed threaded holes in the housing and bear gently against the rim of the lens. Using the aforementioned external alignment-monitoring device as the measuring tool, the screws are used in push-push mode to center the lens. Once aligned, the lens is secured with a threaded retaining ring, by a flange, by an elastomer ring, or some other technique to hold it in place. Note that four setscrews allow adjustment that is more independent in two orthogonal directions than three setscrews.

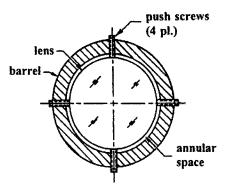


Figure 12.2 Use of four radially oriented setscrews to center a lens in a barrel or housing.

Figure 12.3 shows a sectional view through one objective of the 7×50 M17A1 military binocular designed in the early 1940s and built in large quantity for military use during World War II and the Korean War. It is in many ways similar to designs used in both military and consumer binoculars today. The 7.598-in. (192.989-mm) focal length, 1.969-in. (50.000-mm) aperture (f/3.86) doublet, made of glass types 511635 and 617366, has an OD of 2.048 + 0 – 0.004 in. (52.019 + 0 – 0.102 mm), an axial thickness of 0.571 ± 0.020 in. (14.503 ± 0.508 mm), and a nominal edge thickness of 0.406 in. (10.312 mm).

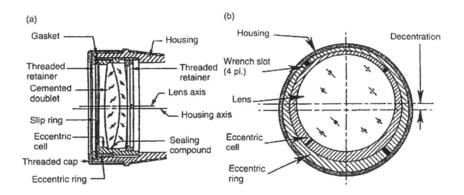


Figure 12.3 Sectional view of the objective lens mounting in a military binocular of World War II vintage. (Adapted from a U.S. Army drawing.)

This doublet is mounted against burnished sharp-corner interfaces in an aluminum cell with stepped ODs machined eccentrically with respect to the centerline of its ID. This cell is inserted into an aluminum ring that has a hole machined eccentrically with respect to its OD. The latter ring is mounted into a recess machined into a cast aluminum telescope housing assembly forming one half of the binocular. A 0.035-in. (0.889-mm) thick flat rubber gasket seals the outer edges of the lens cell, eccentric ring, and housing. Axial preload to hold the lens into its cell is provided by a threaded aluminum retainer, while the entire assembly, including the gasket, a thin aluminum slip ring, and a threaded retainer, is covered by a threaded cap. The thin annular slip ring serves to prevent torsional distortion of the gasket as the retainer is tightened. Sealing compound is used to seal gaps.

During assembly, the axis of the lens is adjusted laterally by differential rotation of the eccentric parts to align the lines of sight of the individual telescopes parallel to each other and to the hinge of the instrument within specified vertical and horizontal angular tolerances. Concentric tubular tools are used to rotate the eccentrics. In some instrument designs, the eccentric rings are matched for fits to each other and to the objective cell by selection from stock having slight variations due to manufacturing tolerances. These parts are thereafter treated as a set.

Usually, no means is provided for axial (focus) adjustment of the binocular objective. It bottoms against the shoulder in the cell and the cell bottoms against the shoulder in the body. In military binoculars having a reticle, focus is usually established by tight tolerancing of pertinent dimensions of parts contributing to the distance from the flint element vertex to the reticle pattern, or by providing a threaded axial adjustment for the reticle. With this adjustment, objectionable focus error can be removed at final assembly by observing parallax between the image of a distant target and the center of reticle pattern. Out-of-specification parallax is corrected by focusing the reticle.

12.1.2 Rotating spindle techniques

A more precise method for alignment of the single lens involves the use of a rotating spindle. This is usually an air or hydraulic bearing device that turns with minimal wobble. We consider here four basic techniques for precision centering of individual lenses using this basic method.

Technique No. 1 is illustrated schematically in Fig. 12.4. The meniscus lens shown in view (a) is to be mounted within the lens barrel. That barrel has been attached to the table of the spindle. The toroidal surface within the barrel is to serve as the mechanical reference for the lens so its alignment relative to the spindle axis is checked with a dial gauge or, preferably, some more precise indicator, such as an air gauge or a capacitance sensor. Alternatively, the barrel OD can be centered relative to the spindle axis and then a finish cut made on the toroidal surface to true it to the rotation axis. The single-point diamond turning (SPDT) method produces the most accurate surfaces. The lens is lowered onto the toroidal reference surface. Its lower surface is then automatically aligned to the spindle axis. The lens's lateral position is adjusted in orthogonal directions until the top lens surface ceases to wobble as the spindle is rotated slowly.

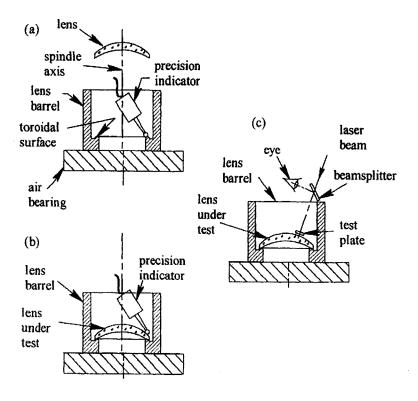


Figure 12.4 Alignment of a single lens using Technique No. 1.

The surface wobble can be measured with a precision indicator [Fig. 12.4(b)]. This can be a high-quality dial gauge or an electronic indicator. Bayar² reported that lens edge runout of 5 µin. (0.13 µm) can be measured with an electronic (capacitance) gauge. An interferometric method also might be used. The latter method is illustrated in Fig. 12.4(c). The interferometer is the Fizeau type with a cavity formed between the lens surface and a nearby test plate that has a curved surface with radius nearly equal to that of the lens surface. The separation between the test plate and the lens surface is exaggerated in the figure. The shape of the fringe pattern seen by eye (or preferably by way of a video camera for safety) is not critical. When the spindle is rotated slowly, the fringe pattern moves. A mark on the monitor can be used as a fixed reference. As the lens is adjusted (using a mechanism perhaps as simple as the four-screw device of Fig. 12.2), the fringe pattern motion becomes smaller. When the motion is too small to detect, the lens is aligned to the spindle axis. It then is secured in place by means such as a retainer or elastomer.

In many cases where an elastomer constraint is employed, success has been achieved by first securing the lens with a few small localized dabs of UV-curing epoxy and, following quick curing of that adhesive, more permanently bonding the lens with roomtemperature-curing epoxy. This two-step bonding technique allows the lens and barrel to be carefully removed from the spindle before the second application of epoxy has fully cured, thus freeing the spindle for another use. Another method for judging when the lens is aligned is illustrated in Fig. 12.5. This is an autocollimation method. The arrows represent the interface between the lens and the barrel. Here, the beam from back-illuminated crosshair reticle₁ passes through a beamsplitter, is collimated by lens₁, and focused by lens₂ toward the center of curvature C_1 of the surface R_1 of lens₃ that is to be centered. The portion of the beam reflected from R_1 is recollimated and focused by lens₂ and lens₁ respectively and reflected by the beamsplitter to form an image of reticle₁ at crosshair reticle₂. The eyepiece then recollimates that beam so the eye can observe it. If the surface R_1 wobbles as the subassembly turns on a spindle, the image of reticle₁ will move with respect to reticle₂ thereby indicating a centration error. Note that a hollow-shaft type of spindle is needed if the beam is to pass through that component.

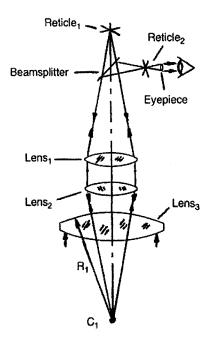


Figure 12.5 An autocollimation setup for sensing alignment errors of a lens rotating on a spindle. (Adapted from Bayar.²)

A more complex setup and technique for aligning a lens in a lens barrel is illustrated schematically in Fig. 12.6. Here, the lens is to be inserted between three flexures that are mounted inside the lens barrel. Only one of these flexures and one plane through the alignment mechanism are shown in the figure. A mechanism that provides horizontal translation and tilt in the plane perpendicular to the plane of the figure is required. Before inserting the lens, the barrel is adjusted so the pads on the flexures are concentric with the spindle axis. The lens is then attached to this five-axis alignment fixture by way of a vacuum chuck. It is moved carefully to a position above the flexures and lowered to the correct axial position. Using the Fizeau interferometer as the test means, the orientation of the lens is fine tuned until the top surface does not wobble as the spindle is turned. Note that a full rotation of the spindle is not possible because of mechanical interference from the fixture. Fortunately, motion of the fringe pattern can be detected with a small rotation.

The next step in the alignment process is to check the alignment of the lower surface of the lens. A second Fizeau interferometer with an auxiliary lens to refocus the beam through the test lens to that surface might be employed. This apparatus is not shown in Fig. 12.6. By iteration, the alignment of both surfaces of the test lens can be achieved. When that has been accomplished, adhesive is inserted through access holes (not shown) in the flexure pads and cured to permanently attach the lens. After curing, the subassembly can be removed from the fixture.

In Technique No. 2, a lens is conventionally mounted into a cell with the usual care one exercises in assembling such components. This subassembly is to be aligned to and secured inside a lens barrel. Figure 12.7 shows, in view (a), the subassembly in place on the alignment fixture and inside the barrel, but not final aligned. The pins and clearance holes in these parts indicate that, after alignment, the cell is to be secured to the barrel with epoxy inserted around the pins. Alignment of the subassembly is accomplished as in Technique No. 1 [see Fig. 12.7(b)]. Then, the epoxy is added to fill the holes around the pins and cured. The barrel/cell/lens subassembly can then be removed from the fixture. This process for attaching the parts is sometimes called "liquid pinning" or "plastic dowelling."

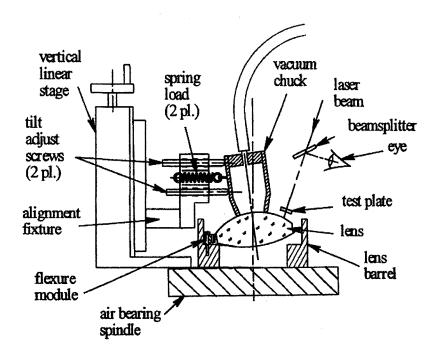


Figure 12.6 Schematic of a technique for aligning a lens with respect to flexures in a barrel using Technique No. 1.

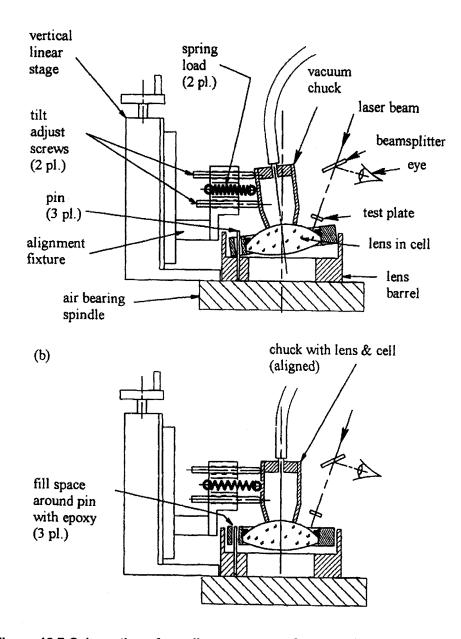


Figure 12.7 Schematics of an alignment setup for centering a lens and cell subassembly to a barrel as in alignment Technique No. 2: (a) Initial installation, (b) cell aligned and ready for liquid pinning.

In Technique No. 3, a cell is prepared by finish machining its surfaces (OD, rim circularity, thickness, and parallelism and flatness of the cell's end faces) to tight tolerances. Usually, a suitable interface for the lens also is machined into the cell in the same machining setup. This machining is best done on a single-point diamond-turning (SPDT) machine. This ensures that the cell can fit closely into the lens barrel or optical instrument in which the lens is to be used. The cell is mounted on and aligned to the precision spindle

as shown in Fig. 12.8(a). The lens is then inserted as shown in Fig. 12.8(b), aligned as in Technique No. 1, and secured in place. The resulting subassembly is sometimes called a "poker chip" because it has its alignment built in. The poker chip is assembled (without further alignment) into a lens barrel or instrument and secured in place. View (c) shows the resulting assembly.

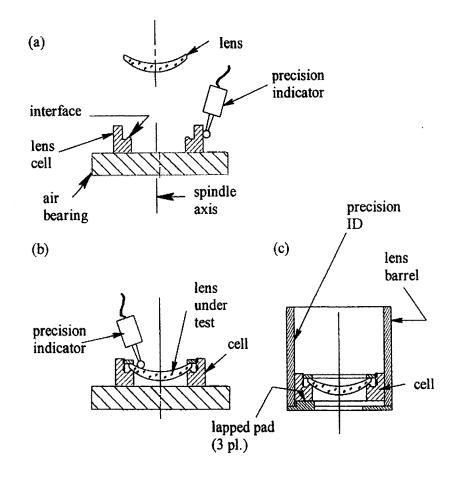


Figure 12.8 Schematics of an alignment setup for centering a lens and cell to a barrel as in alignment Technique No. 3.

In Technique No. 4, the cell is partially machined to final dimensions as indicated in Fig. 12.9(a). It is then mounted on and centered to a precision spindle. Usually this spindle is that of a SPDT machine because the unfinished surfaces of the cell are final machined in situ after the lens is installed and aligned to the spindle axis. The "spacer" shown in the figure indicates one way to ensure clearance below the cell for the diamond tool to pass. The lens is installed and centered to the spindle axis as described previously. Machining of the cell is then completed. Dimensions achieved by this machining operation are indicated in the figure. The resulting "poker chip" is ready to be installed.^{*}

^{*} Additional techniques for creating "poker chip" lens subassemblies using SPDT machining are described in Section 15.3.

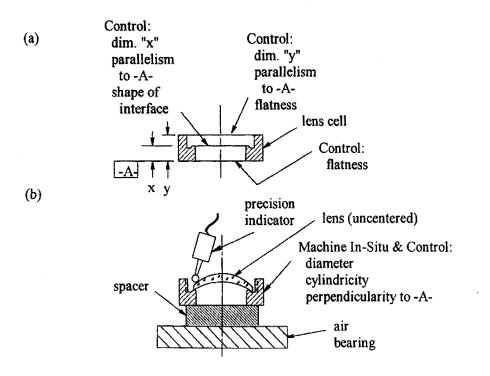


Figure 12.9 Schematics of an alignment setup for centering a lens and cell to a barrel as in alignment Technique No. 4.

12.1.3 Techniques using a "Point Source Microscope"

Parks and Kuhn³ and Parks⁴⁻⁶ have described several techniques for using a "point-source microscope" (PSM) as an alignment device for optics. Figure 12.10 is an exterior view of the instrument. Its optical schematic is shown in Fig. 12.11. There are two light sources. The one in the top portion of the instrument is a ~4.5-µm diameter, ~f/5 divergence point source created by a single-mode optical fiber pigtailed to a laser diode (not shown) operating at 635-nm wavelength. The collimating lens following the point source creates a diffraction limited "artificial star" at infinity for the microscope objective. A point image of the source is then produced at the focus of that objective. The source in the center portion of the PSM is a diffuse extended source (ground glass disc) back-illuminated by a red light-emitting diode. A condenser lens provides Köhler illumination by reimaging the extended source to fill the pupil of the microscope objective. A beamsplitter combines these two beams and a second beamsplitter folds both beams into the objective.

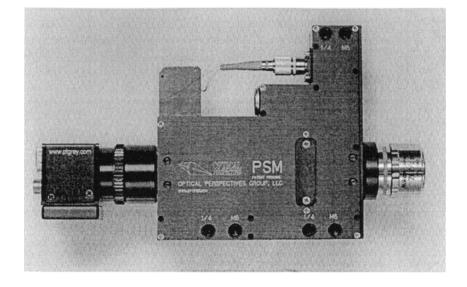


Figure 12.10 Photograph of the point-source microscope. (Courtesy of Optical Perspectives Group, LLC, Tucson, AZ.)

The lowest sketch in Fig. 12.11 shows the beam reflecting from a test surface located at the focus of the microscope objective to a CCD video camera. This is often called a "cats eye" reflection. If the surface of the test object is flat, specular, and approximately normal to the axis, the beam from the point source is returned as a "point" image at the CCD camera. If the test object surface is a convex sphere (or a paraxial region of an aspheric), specular, and located with its center of curvature at the objective focus, the beam from the point source is returned as a "point" image at the CCD camera by retroreflection. If the test object surface is flat and nonspecular, the beam from the extended source is returned to form an image of that surface at the camera. The camera can be adjusted to center any of these images to an electronically generated crosshair on a video monitor.

Usually, the PSM is mounted on a three-axis linear stage so it can be aligned to the test object and moved axially to access various images. The axial stage is provided with means to measure distances moved so the PSM can be used to measure surface radii. For example, view (a) of Fig. 12.12 shows a convex spherical surface positioned at the objective focus to produce a "cats eye" reflection. View (b) shows the PSM moved closer to the surface so the retroreflection from the center of curvature is in focus at the camera. The distance moved is the radius of the surface. The radius of a concave surface can be measured by a simple adaptation of this procedure. This function of the apparatus is especially useful as a means for non-contacting inspection of optics and for calibrating optical test plates.

The PSM can be used to monitor alignment errors during assembly of lenses by focusing it on each center of curvature in sequence. If the lens is on a precision spindle, a misaligned surface will return an image that nutates as the spindle rotates. It is necessary for either the PSM or the lens to move axially to access the two returned images.

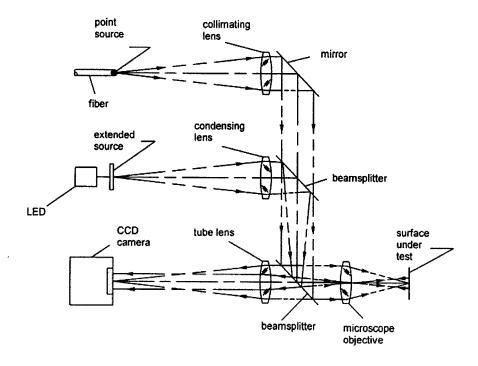


Figure 12.11 Optical schematic of the point-source microscope. (From Parks and Kuhn.³)

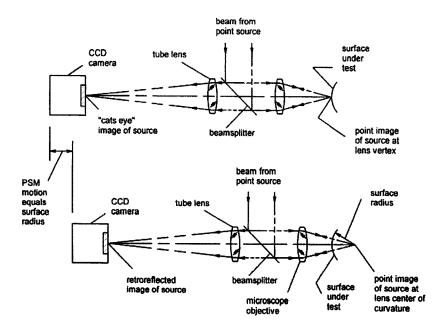


Figure 12.12 Schematic of the use of the PSM to measure the radius of curvature of a convex spherical surface. (Adapted from Parks.⁵)

If two PSMs are used, both images from a lens can be monitored without moving the lens or either PSM. We illustrate this technique in Fig. 12.13. Here, a single biconvex lens element is being aligned to the axis of a lens barrel on a spindle. PSM #1 is positioned to obtain a retroreflection from R_1 , while PSM #2 is positioned to obtain a retroreflection from R_2 . As the spindle rotates slowly, both images will nutate, indicating misalignment of both surfaces. To correct the errors, the lens is translated laterally and/or tilted within the barrel until the image motions are minimized. Then, the lens is secured in that aligned position. This method is ideal for use in production because both images are seen simultaneously in the same setup.

Note that the procedures just described are similar in principle to the visual test that would be used with the autocollimation setup of Fig. 12.5. Because the 1024×760 -pixel camera and the computer software associated with the PSM typically allows detection of ~0.1-pixel image movement, the accuracy of the latter instrument significantly exceeds the capability of the visual test. Many useful applications of the PSM instrumentation beyond those described here are explained in the referenced publications.³⁻⁶

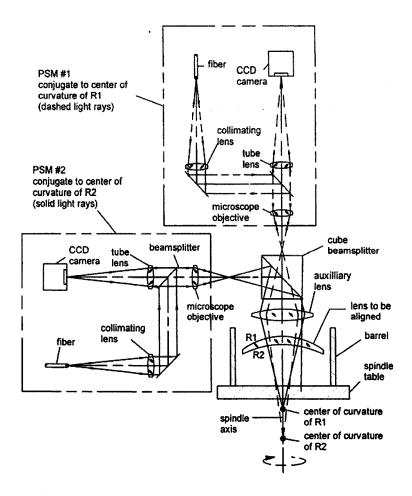


Figure 12.13 Two PSMs used to sense alignment errors of both surfaces of a lens simultaneously. (From Parks.⁵)

12.2 Aligning Multiple Lens Assemblies

Most of the principles and techniques used for aligning a single lens apply also to the alignment of multiple lenses to a common axis in more complex designs. In most precision assemblies, the glass-to-metal interfaces are on the polished optical surfaces rather than on secondary (ground) surfaces. Figure 12.14, from Hopkins,⁷ illustrates an extreme case of an air-spaced triplet in which all lenses and spacers have wedge, the lens rims are not cylindrical, the spacers contact spherical surfaces, and those contacts are cut spherical to match the concave lens surfaces or conical to ensure tangent contact on convex surfaces. In spite of these errors, the centers of curvature for all surfaces lie on a common axis. Hence, the lenses are aligned correctly. All that is needed to make this happen is a mount (not shown) that provides suitable axial interfaces at the exposed surfaces of lens A and lens C, means to move all three lenses laterally in that mount, and means for measuring alignment errors. Hopkins⁷ also points out that, although it is preferable that the spacers be round, this is not essential if the air spaces are measured and adjusted to minimum values at assembly. The latter requirement is not always easy to achieve. In such cases, we should ensure circularity of the spacers.

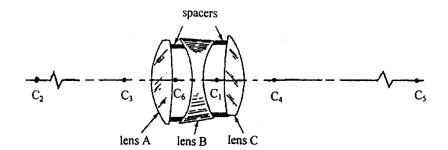


Figure 12.14 A triplet lens that is perfectly centered in spite of mechanical errors. (From Hopkins.⁷)

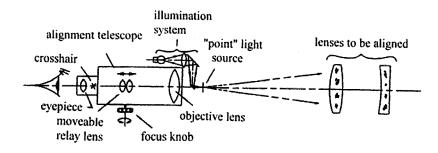


Figure 12.15 Schematic of an alignment telescope as it is used to detect alignment errors of an air-spaced doublet. (From Yoder.⁸)

12.2.1 Using an alignment telescope

Figure 12.15 illustrates schematically a technique that has been used successfully in aligning multiple lenses to a common axis in a housing. The alignment telescope shown there is a commercially available device featuring a moveable relay lens with unusually large dynamic range so the telescope can focus on targets at distances ranging from infinity down to zero, i.e., at the objective lens, and even behind the observer's head. The focus mechanism is very accurately made so the line of sight does not wander significantly as the focus distance changes over this full range. A typical telescope of this type is shown in Fig. 12.16(a). It is mounted in an adjustable mount such as that shown in Fig. 12.16(b). This mount provides orthogonal tilts in three axes to facilitate pointing of the telescope line of sight. Vertical and horizontal translations of the telescope also are provided by some means (not shown). In this instrument, line of sight wanders with focus smaller than 0.5 arcsec. A crosshair reticle is provided at the eyepiece focal plane. For the present application, the device is modified slightly by adding an external illumination system that creates a "point" source on axis at the objective. A tungsten filament lamp with optics to illuminate a pinhole is shown in the figure. A visible laser diode with a single mode fiber, such as used in the PSM described in Section 12.1.3, would provide a brighter source and reduced beam obscuration. In either case, the source illuminates the lenses to be aligned.

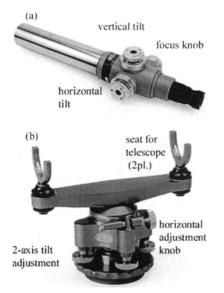


Figure 12.16 Photographs of (a) an alignment telescope and (b) an adjustable mount for that telescope. (Courtesy of Brunson Instrument Company, Kansas City, MO.)

Figure 12.17 shows schematically (and not to scale) how the apparatus is used. In view (a), one ray from the beam incident upon the first lens is shown reflecting from R_1 . As it enters the alignment telescope, the reflected ray appears to come from image₁. The telescope is focused on that image and tilted/translated so the image is centered on the telescope crosshairs. The telescope is then refocused on image₂ from R_2 [see view (b)] and the alignment of the telescope adjusted further until both image₁ and image₂ are centered to the crosshairs. The telescope axis is then coincident with the axis of lens₁.

The focus of the telescope is then adjusted to image₃, which comes from R_3 , and the tilt and/or lateral location of lens₂ adjusted independently from lens₁, so image₃ is also on the crosshairs. See view (c). Finally, the focus is adjusted to image₄ and the orientation of lens₄ is refined so that image appears centered to the crosshairs. See view (d). If the focus is now changed to observe each image in turn, all should appear to be centered. This indicates that the axis of lens₁ coincides with that of lens₂.

The simplest way to know which image comes from which surface is to raytrace the nominal (i.e., centered) system of lenses to be aligned, treating each surface in turn as a mirror and noting the sequence in which the images appear as the telescope focus is changed from infinity toward zero. Usually, the paraxial approximation is sufficient. In Fig. 12.17, the right-to-left reflected image sequence is 1-4-2-3.

Much more complex systems than shown here can be aligned by this method. The use of a laser as the source may be necessary if the Fresnel reflections from the various surfaces are dim because of the high efficiency of the antireflection coatings on those surfaces. Caution must be exercised to not exceed safety limits for laser beam intensity at the eye if that type of source is employed.

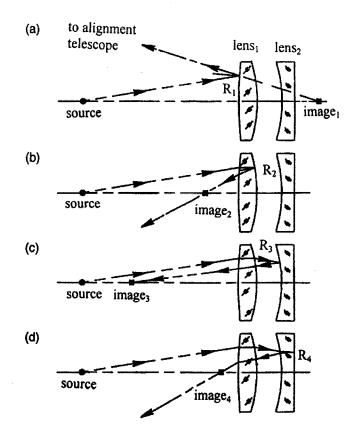


Figure 12.17 Schematics of the point source and the reflected images from lens surfaces 1 through 4 using the setup of Fig. 12.15 and the alignment telescope and mount of Fig. 12.16. (From Yoder.⁸)

12.2.2 Aligning microscope objectives

The lenses (and sometimes mirrors) used in microscope objectives are extremely sensitive to decentration and/or tilt because of their short focal lengths and short radii. Some axial air spaces also are critical to achieving maximum performance so they must be controlled with tight tolerances or adjusted during assembly. Referring to Fig. 12.18, we summarize an explanation from Benford⁹ of these processes for a typical refracting objective.

The three lenses are burnished into their cells in the manner described in Section 3.4. After cleaning, these subassemblies are inserted along with a first spacer (of nominal length) into the ID of the main barrel. The first two cells fit snugly into this ID, while the third cell has significant radial clearance. A temporary version of the sleeve labeled "parfocality adjustment" is screwed onto the barrel to provide an axial reference for the stack of lens cells. A second spacer is installed on top of the third cell and held temporarily with the threaded parfocality lock nut. The quality of the aerial image of an artificial star (an axial point object located in the object plane) is examined under high magnification and the first spacer (located between the first two cells) is selected by successive approximations from a stockpile of spacers with slightly different lengths. The choice is based on minimizing spherical aberration in the image. The test setup is sketched in Fig. 12.19.

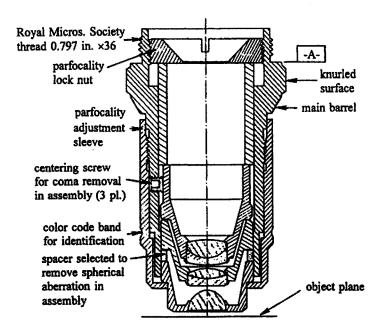


Figure 12.18 Construction of a typical microscope objective. (Adapted from Benford.⁹)

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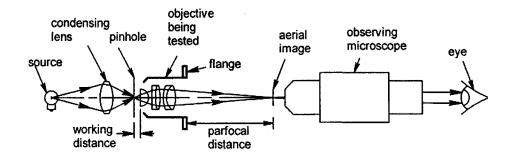


Figure 12.19 Schematic of the test setup used to align the microscope objective of Fig. 12.18.

A standard microscope slide is placed between the objective under test and the pinhole if the objective is intended to be used with such a slide. A laser diode with pigtailed single mode fiber may be used instead of the pinhole. Appearances of the star images at, inside, and outside best focus with the correct spacer (a) and with an incorrect spacer (b) are shown in Fig. 12.20.

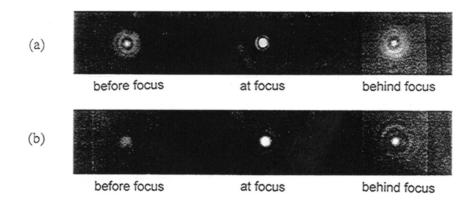


Figure 12.20 Appearances of artificial star images observed behind, at, and before best focus while adjusting a typical microscope objective for minimum spherical aberration: (a) lens correctly spaced (out-of-focus details nearly identical); (b) lens incorrectly spaced (the ring structure is visible behind focus, but not before focus). (Adapted from Benford.⁹)

The temporary version of the parfocality adjustment sleeve in place at this time provides access to three radially oriented centering screws that bear against the sides of the third lens cell. The lateral location of the third lens cell is adjusted by turning these screws while observing the magnified aerial image until that image appears symmetrical. This setting minimizes coma. After this adjustment is completed and the screws tightened firmly, the temporary sleeve is replaced with a permanent version (without access holes) and tightened to hold the setting. The parfocality adjustment sleeve is adjusted until the image is located at the standard distance from shoulder -A- on the main barrel. All lenses adjusted in this manner are "parfocalized," i.e., they have a common distance from flange to image. The parfocality lock nut is then tightened. The assembly process is then complete and the lens undergoes final inspection to confirm performance.

Reflecting objectives for microscopes are generally simpler than their refractive counterparts since they consist only of two mirrors in the Schwarzschild configuration. Figure 12.21 illustrates a typical (generic) optomechanical design. The setscrews pressing against the conical surface of the secondary mirror cell are used to adjust the centration of that short focal length element to optimize performance. Although normally designed for use without cover glasses over the sample, more complex mechanical designs feature a graduated external (knurled ring) adjustment of axial spacing of the mirrors to compensate (within limits) for cover-plate thickness.

Sure et al¹⁰ described some of the problems associated with assembly and alignment of high-power, high numerical aperture microscope objectives to be used in the UV at and below 248-nm wavelength in systems for inspection of state of the art semiconductor chips and in other applications involving measurement at nanometer scale. Particularly difficult is achieving the proper air spaces between lenses and assessing the performance of the system while adjusting positions of laterally adjustable elements. Because of the short wavelength and high photon energy of the transmitted beam, cemented lenses cannot be used in these objectives. Figure 12.22 shows a sectional view through a typical high performance objective, the Leitz $150 \times DUV$ -AT with NA of 0.9. This objective has 17 air-spaced singlet lenses made of fused silica and calcium fluorite and is capable of resolving details measuring 80 to 90 nm in an object. Obtaining this level of performance from production quantities of this objective requires the application of the following special in-process inspection and alignment techniques.

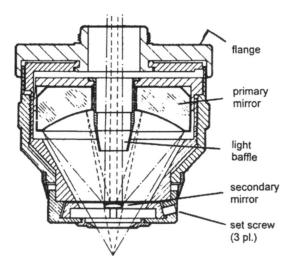


Figure 12.21 Optomechanical configuration of a typical reflecting microscope objective.

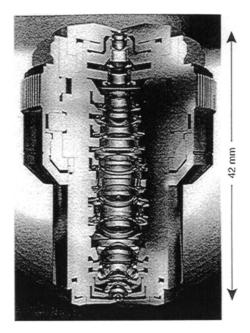


Figure 12.22 Cut-away sectional diagram of the Leitz $15 \times DUV-AT$ microscope objective with N.A. of 0.9. (From Sure et al.¹⁰)

The tolerances assigned to this lens design are listed in the right hand column of Table 12.1. Meeting these "limiting" tolerances entails great expense as compared to the "typical" values applied to more common microscope objectives. For instance, in order to achieve ± 2 -µm maximum air space thickness errors between elements, the location of each lens must be controlled with respect to its associated mechanical mount to ± 1 µm. This requires the use of interferometric techniques.

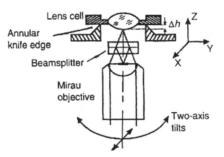
For each lens	Typical	Limiting
Radius error	5 λ*	0.5λ
Surface error	0.2 λ	0.5 λ
Surface roughness (rms)	5 nm	0.5 nm
Center thickness error	20 µm	2 μm
Refractive index error	2×10 ⁻⁴	5×10 ⁻⁶
Abbe number error	0.8%	0.2%
For the assembly	Typical	Limiting
Decentration	5 µm	2 μm
Run-out	5 µm	2 μm
Fit of cell into housing**	10 µm	2 μm
Air space error	5 µm	2 μm

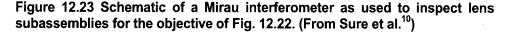
Table 12.1 Production tolerances for high performance UV microscope objectives.

Notes: * All λ at 633 nm. ** Across diameter.

Source: Adapted from Sure et al.¹⁰.

Testing of subassemblies for this objective is conducted using a Mirau interferometer as indicated in Fig. 12.23. The location of the flat surface of the mount is determined by placing an optical flat on the annular knife edge and moving the interferometer so its focus is on the flat surface. The flat is then removed and the lens subassembly to be measured is placed on the knife edge. The interferometer focus is then shifted to the vertex of the lens. Note that the interferometer can be tilted in two directions to ensure beam propagation along the normal to the lens surface. The distance labeled " Δh " in the figure is then measured to an accuracy of $\pm 0.200 \ \mu m$ and compared to the design requirement. Subassemblies within tolerance are accepted for use in production of the objectives.





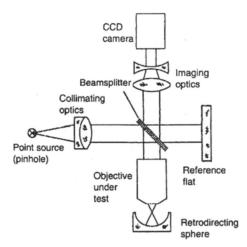


Figure 12.24 Schematic of the Twyman-Green interferometer used to measure performance of the objective shown in Fig. 12.22. (From Sure et $al.^{10}$)

Wavefront error is monitored during adjustment of one or more laterally adjustable elements in a lens such as that of Fig. 12.24 using a more elegant version of the visual star test described in conjunction with Fig. 12.19. Figure 12.25 shows this instrument schematically. With the reflection from the flat reference mirror obscured, the image formed by the imaging optics in a CCD camera can be observed directly and in real time on

a video monitor at about 20 frames per second by the optician while the lens adjustment is accomplished. Wavefront errors such as coma that is caused by the objective under test result in asymmetry of the image. Once the image appears to be good visually, the reference beam can be allowed to interfere with the beam from the spherical reference mirror and the point spread function (PSF) of the wavefront can be determined by fast Fourier transform methods from the fringe pattern.

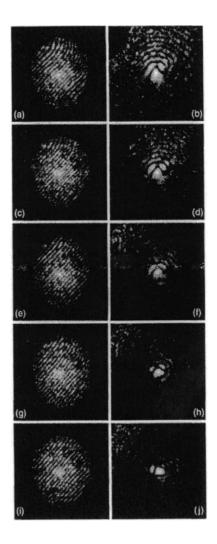


Figure 12.25 Interferograms (a), (c), (e), and (i) and their corresponding PSFs (b), (d), (f), and (j) for the objective of Fig. 12.22 at 266-nm wavelength recorded as the compensating element was adjusted. (From Sure et al.¹⁰)

The technique for doing this was summarized by Sure et al¹⁰ and is discussed in more detail by Heil et al.¹¹ In brief, a set of interferograms and the corresponding point spread functions (PSFs) such as those shown in Fig. 12.25 were interpreted as follows. The sequence of interferograms was recorded while the compensating element of the objective was moved to reduce coma. The initially observed ~ 2 fringes of coma [view (b)] were reduced to zero in view (j). Some residual higher order comatic wavefront error (trefoil) can

be seen in view (j) because the lens under test was not perfect. Sure et al.¹⁰ indicated that they chose this example to show in their paper because it demonstrates how use of both fringe patterns and PSFs provide insight into the behavior of the test sample that is hard to gain from interferograms alone.

12.2.3 Aligning multiple lenses on a precision spindle

A high precision technique for assembling a series of lens elements with nearly perfect centration was described by Carnell et al.¹² Figure 12.26 is a simplified sectional view of the assembly. It was to be used as a wide-field (110-deg.) objective lens for bubble-chamber photography. A large amount of optical distortion of a particular form was designed into the lens and was required to appear very precisely (i.e., within a few micrometers of the design values) over the entire image.

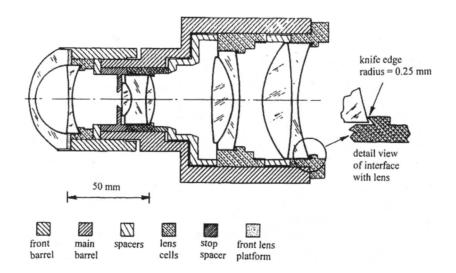


Figure 12.26 Optomechanical schematic (simplified) of a lens assembly requiring very precise alignment of many components. (Adapted from Carnell et al.¹²)

The technique was to mount each lens individually in a brass cell that had been machined true by SPDT machining. A rounded 0.25-mm (0.010-in.) radius "knife edge" seat (actually a toroidal interface) was turned inside each cell to contact the lens's spherical surface. All machining and assembly of the lenses were completed before removing the cell from the spindle. As shown in Fig. 12.27, centering was monitored by observing Fresnel interference fringes between the lens surfaces and a spherical test plate held close to the exposed surface of the lens. This test technique is essentially the same as discussed in Section 12.1.2. Centration was judged to be correct when the fringe pattern appeared (through a microscope) to remain stationary as the spindle was rotated slowly. This indicated that the rotating surface ran true to a centered sphere within a fraction of a wavelength of the laser beam used (typically, $\lambda = 0.63 \ \mu$ m). The lens was then bonded to the cell with a room temperature curing epoxy that remains slightly flexible when cured. An epoxy layer thickness of about 0.1 mm (0.004 in.) was reported to be satisfactory for the intended application. The individual cemented subassemblies were assembled without further alignment, in a precisely bored barrel. Upon evaluation, the authors reported that the system's centration errors did not exceed 1 μ m (4×10⁻⁵ in.) at any field angle.

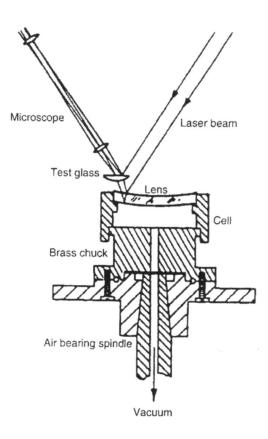


Figure 12.27 Instrumentation used to monitor centration of each optical surface in the assembly of Fig. 12.26. (Adapted from Carnell et al.¹²)

The doublets and the triplet of the design were cemented together with a variation of the same technique. One element was attached to a trued vacuum chuck on the air-bearing spindle with the concave surface up and centered as described earlier. The appropriate amount of optical cement was added and then the second element was lowered into place. The cement layer was squeezed until testing with the Fizeau interferometer indicated that the inner surfaces were parallel. The upper element was then positioned so the top surface ran true. Any wedge introduced in the cement layer was then removed by differential application of axial pressure and the top element's centering was refined as necessary. The referenced paper by Carnell et al.¹² mentioned that a sufficient index of refraction difference occurred in the glass-to-cement interface at the test wavelength to give about 1% Fresnel reflectance at each surface, allowing the fringes to be seen by the unaided eye. This process was used repeatedly for the triplet.

Carnell et al.¹² further reported that upon evaluation the system aberrations were essentially as expected. When the assembly was rotated in a vee-block, the axial image moved by no more than 1 μ m (4×10⁻⁵ in.), indicating that excellent rotational symmetry about the mechanical axis had been achieved.

Variations of this technique for mounting lenses within subcells include designs in which each lens is burnished or held in place with elastomer. In some cases, the lenses are aligned to premachined cells, while in other cases, the cylindrical outer surface of each cell is machined on a SPDT machine to be concentric with and parallel to the optical axis of the lens. The cell ODs are also machined to the proper OD for insertion along with similarly machined subcells into a common barrel. These techniques are as described in Section 12.1.2 for individual lens mounting and alignment.

12.2.4 Aberration compensation at final assembly

The assembly of lenses into individual cells as prealigned poker chips is enhanced if the design allows performance optimization of the assembled product by fine transverse or axial adjustment of one or more lenses during the final stage of assembly. Figure 12.28 (also Fig. 4.18) shows an example of such an assembly in which the third element can be transversely adjusted with three screws to allow modification of its aberration contributions to compensate for residual aberrations of the optical system. The moveable lens must have sufficient sensitivity to the specific aberration to be compensated for so that reasonable movement produces the desired effect. On the other hand, it must not be too sensitive to this and other aberrations, for that would make the adjustment too critical. The choice of which element or elements to move is made by the lens designer during the tolerance analysis. In some lens assemblies, multiple components are chosen as "compensators," each affecting one aberration more than others

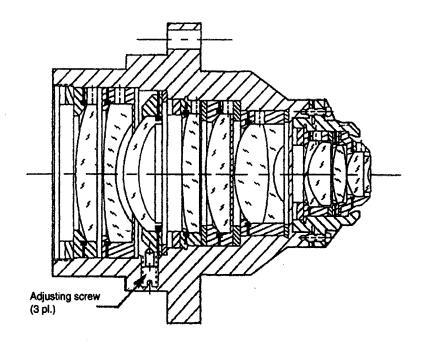


Figure 12.28 Sectional view of a lens assembly with one element acting as an aberration compensator for performance optimization at final assembly. (From Vukobratovich.¹³)

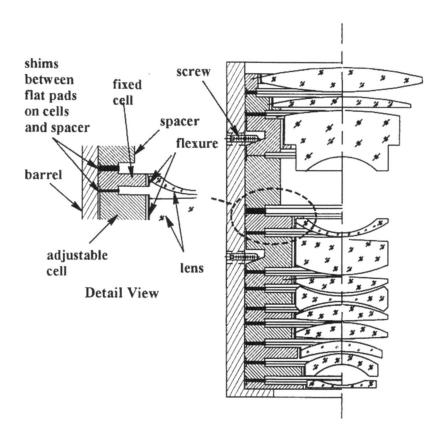


Figure 12.29 Schematic partial-section view of a lens assembly with two laterally adjustable cells. The lenses are assembled and aligned in their cells as poker chips and the cell ODs machined precisely to fit the barrel ID.

Figure 12.29 shows an even higher-precision lens assembly. This assembly is part of a microlithography mask optical projection system used in making computer chips. It is assembled and aligned in general accordance with alignment Technique No. 4 of Section 12.1.2. On the right side, one can clearly see the lenses. All are singlets made of fused silica. As shown in the detail view of the figure, they are each attached to flexures machined into the cell walls. The lens elements are attached to the flexures with epoxy. After checking alignment, the cells are diamond turned in situ. The ODs of all but two cells are only a few micrometers smaller than the ID of the barrel into which they are to be inserted. The two special cases are those for the third and fifth lenses from the top, which are transversely adjustable with radially directed screws to optimize performance at the final stage of alignment. These cell ODs are sufficiently undersized to allow the adjustments to be made.

A customized, thick annular spacer is provided between the third and fourth lenses from the top to set the corresponding air space. This annular plane parallel spacers of appropriate thicknesses (shown in black and exaggerated in thickness in the figure) are located between the poker chips to allow those air spaces to be set within tolerances. These typically would be made of the same material as the cells and barrel for thermal expansion compatibility.

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Figure 12.30 illustrates an interferometric setup that might be used to monitor alignment of the lens assembly during adjustment of the two sliding lenses of Fig. 12.29. Fringes, formed in a double pass through the lens assembly between a reference flat surface and the concave retrodirective mirror, are observed using a video camera located above the beamsplitter cube. The image quality is recorded after each iterative adjustment of the moveable lenses.

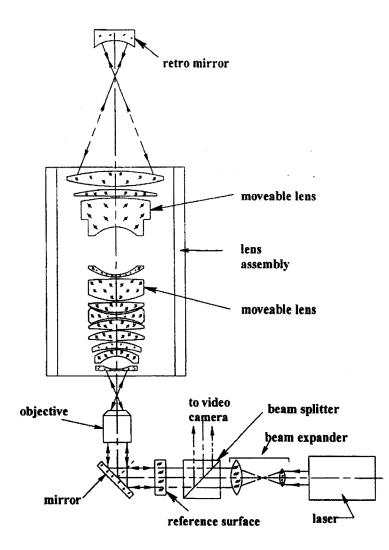


Figure 12.30 A test setup that might be used to evaluate the performance of the assembly of Fig. 12.29 during adjustment of the laterally moveable lenses.

Figure 12.31 is a schematic of an optomechanical setup for supporting and adjusting a lens barrel containing many lenses, three of which act as aberration compensators. These lenses are located at the axial locations indicated by the "access holes." Push rods driven by two micrometers penetrate such holes at each location to move each lens laterally by sliding its moveable poker-chip subassembly relative to all the others. A restoring force is exerted by a spring (not shown) attached to the fixture and acting through a third hole symmetrically located with respect to the motions of the micrometers in their plane of action. This allows the micrometers to operate essentially in a simple "push-pull" mode.

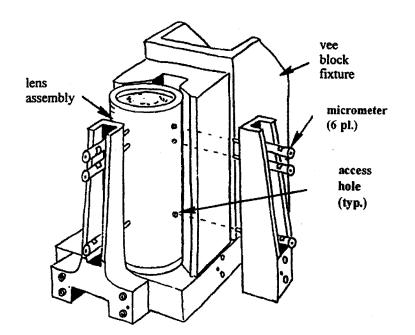


Figure 12.31 Schematic of a test fixture with a lens assembly in which three lenses are adjusted transversely for optical performance optimization.

In this test setup, the lens barrel is clamped to a "vee" block fixture, which is in turn installed in an interferometer, such as that shown in Fig. 12.30. The effects of the movement of each lens on overall performance can be assessed in near real time. Optimization is accomplished by successive approximations. When the system's performance is maximized, the moveable lenses are locked in place by some internal mechanisms (not shown) to preserve the alignment. The access holes are then sealed to keep moisture and dirt out. To optimize the performance of some lens assemblies, small movements of one or more poker chips in the axial direction are necessary in addition to lateral adjustments of some other poker chip(s). Customizing spacers or shims placed between cells as previously mentioned is the usual approach. The use of threaded cells turning in mating threads in the barrel (similar to some eyepiece focus adjustments) for axial positioning of the lens is not practical here because the threads are too coarse, even if they are designed to act differentially. Runout errors of threads also could affect centration of the lenses. Furthermore, the cells must not be turned about their axes during axial adjustment because minute residual wedge defects in the optical and/or mechanical components can then deviate the beam transmitted through the optics and may increase aberrations.

Bacich¹⁴ described some mechanisms for making fine axial adjustments that can be accessed from the outside of the lens barrel. Figure 12.32 shows two such mechanisms. In view (a), three balls distributed at 120-deg. intervals slide in vertical holes in the lens cell (poker chip) and rest on cone-point setscrews penetrating the wall of the cell. Another lens cell (not shown) rests on top of the three balls. By accessing the screws through holes in the barrel wall and turning them by equal amounts, one can increase or decrease the air space between adjacent lenses by small amounts. In view (b), the same result is obtained by setscrews driving into three wedged slots in the cell wall. Half-balls attached to the tops of the cantilevered wedges touch the adjacent cell (not shown). In either case, when the cells are locked together axially, the adjustments are secured. The use of these adjustment means to differentially tilt a lens is recommended only if the assembly design does not require the cell rims to be aligned to each other—as when the stack of poker chips are to be inserted into a close-fitting lens barrel ID.

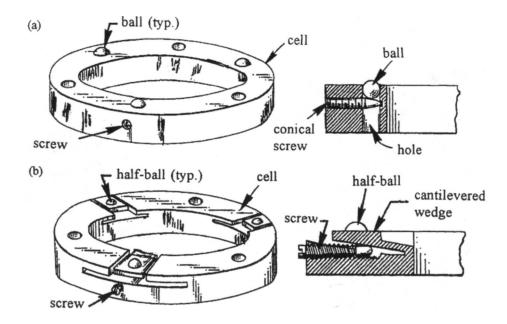


Figure 12.32 Concepts for mechanisms to adjust airspaces between poker chips. (From Bacich.¹⁴)

One way to clamp a stack of poker chips together axially after its performance has been maximized is shown in Fig. 12.33. Here three rods are passed through clearance holes in each cell and shim and thread into holes in the barrel's lower endplate. Nuts are attached to the upper threaded ends of the rods. By tightening the nuts, the rods are put in tension and clamp the cells together axially without disturbing centration because all interfacing pads on the cells are flat and perpendicular to the mechanical axis of the system.

Another concept for alignment and clamping together of poker chip lens subassemblies is shown in Fig. 12.34. Here, three cells with their lenses installed and aligned (but not shown in the figure) are shown in exploded view. These cells are part of a stack of poker chip modules that function in the manner of those shown in Fig.12.29. Each cell has multiple pads on its top and bottom surfaces. These pads are lapped flat, coplanar, and parallel. Shims are placed between the cell pads to set the axial spacings. Alternatively, the pads may be machined provide the specified air spaces when contacted together. The lenses are aligned in the cells so their optic axes are geometrically centered to the cell ODs and perpendicular to the pad's faces.

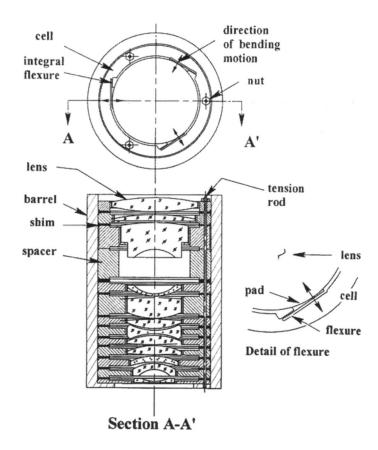


Figure 12.33 Schematic of a means for axially clamping the poker chips in the assembly of Fig. 12.29 with three tension rods at 120-deg intervals. Some details of the flexure mounts for the lenses also are shown.

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The lower cell in Fig. 12.34 has three rods pressed into or threaded into holes in the cell so as to be perpendicular to the plane defined by the pads on the cell's top surface. These rods protrude through clearance holes in the middle and top cells when they are brought together. During assembly, the lower cell acts as the reference. The middle cell is aligned by sliding it laterally until the optic axis of that middle cell is collinear with that of the lower cell. This adjustment would be made on an air-bearing spindle using interferometric sensing of errors. When the adjustment is completed, the spaces between the rods and the holes in the upper cell are filled with epoxy and cured. The third cell is then placed on top and aligned in the same manner as the second. It too is epoxied in place. The same process could be followed to build a stack of all the lens cells in the assembly.

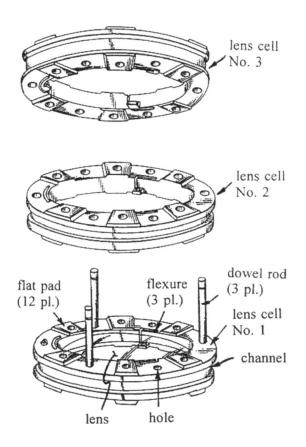
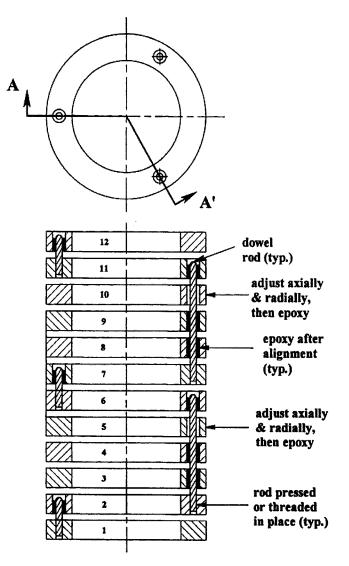


Figure 12.34 "Exploded" view of three poker chips that illustrate a technique for affixing one cell to another by liquid pinning the rods in clearance holes after alignment. (Adapted from Bacich.¹⁴)

Figure 12.35 shows a schematic sectional view through a stack of 12 poker chips. Once aligned to each other and pinned with epoxy, they form a complete optical assembly. Two cells (Nos. 5 and 10) are to be adjusted axially and radially to optimize performance of the whole optical system. A test setup of the type shown in Fig. 12.30 might be used. A fixture as shown in Fig. 12.31 could provide a means for making the lateral adjustments. Custom-ground shims (not shown) located between the cells would establish the proper air spaces.

In Fig. 12.35, all cells except those to be adjusted are shown epoxied to the rods that are firmly attached to cells 1, 2, 6, 7, and 11. At this stage of assembly, the nonadjustable components will have previously been aligned to a common axis and the adjacent air spaces established before the epoxy was inserted around the rods and cured. As shown, the assembly is ready for final positioning of the moveable cells and epoxying them to the rods. Once this is accomplished, the entire stack would be inserted into a housing to enclose and protect the assembly, as well as to provide interfaces for attaching the lens barrel to an external structure.

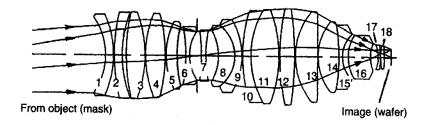


Section A-A'

Figure 12.35 Schematic of poker chips stacked, aligned, and epoxied to dowel rods in preparation for final alignment of two adjustable modules. After alignment, these cells are epoxied in place. (Adapted from Bacich.¹⁴)

12.2.5 Selecting aberration compensators

A technique for selecting the appropriate aberration compensator(s) in a high performance lens system was described by Williamson.¹⁵ He described and applied this technique to the microlithographic projection lens shown in Fig. 12.36. This lens, designed to be used at $5\times$ reduction, comprises 18 elements, has an image space numerical aperture of 0.42, and covers a 24-mm (0.944-in.) image field diameter. Compensation occurs in two stages. The first is the recomputation of air spaces between elements (and perhaps radii) to reduce the effects of minute measured residual errors on surface radii, element thicknesses, and refractive indices. Errors in surface figure and surface irregularity are also considered. The second stage is post-assembly alignment optimization of the selected compensating element adjustments while assessing the effects on performance of those changes.





Williamson¹⁵ indicated that the fifth- and higher-order aberrations that are so carefully minimized during the design process are not strongly affected by small perturbations of lens element alignments. On the other hand, the third-order aberrations are significantly affected by these adjustments. Hence, the compensator selection process involves determination of sensitivities of each element for third-order spherical aberration, coma, astigmatism, and distortion as functions of axial, lateral, and clocking motions. These aberrations are expressed in terms of Zernike polynomial coefficients. The following are typical constraints applied in the selection process. Each compensator is limited to either a small axial or a lateral displacement to reduce mechanical complexity. Preferably, the elements with largest diameters should not be selected as compensators as the mechanism to affect the adjustments would tend to increase overall assembly diameter and this parameter usually must be minimized in any practical assembly. To third order, tilts and decentrations produce equivalent results so only decentrations are considered. Finally, compensation must be accomplished without disassembly of the lens.

Figure 12.37(a) shows the results of the sensitivity analysis for individual element axial shifts of 25 μ m (0.001 in.). Elements 4, 7, 16, and 17 appear to be the best to correct coma, spherical aberration, astigmatism, and distortion respectively. Figure 12.37(b) shows similar results for element decentrations of 5 μ m (0.0002 in.). If elements 5 and 6 are moved as a pair, then the coma changes from each element add while the astigmatism

contributions cancel and the distortion changes are opposite. Doublet 8-9 shows good orthogonality in that only astigmatism changes significantly when that lens subassembly is decentered. Doublet 14-15 would serve well to compensate distortion because it causes relatively little coma or astigmatism when decentered. From these results, the appropriate aberration compensators would be axial adjustment of element 7 for spherical aberration and transverse motion of elements 5 and 6 moving together for coma, doublet 8-9 for astigmatism, and doublet 14-15 for distortion.

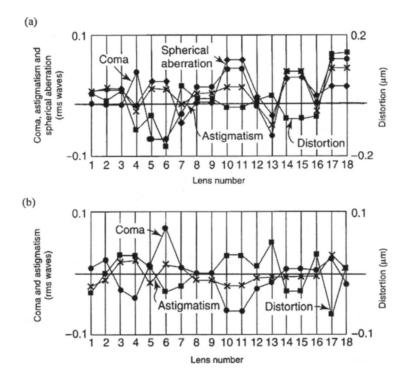


Figure 12.37 Effects on rms wavefront aberrations caused by (a) individual axial shifts of $25-\mu m$ magnitudes and (b) transverse shifts of $5-\mu m$ magnitudes for each element in the system of Fig. 12.36. (Adapted from Williamson.¹⁵)

In the reported case, successive iterations of the above-axial and transverse adjustments except for the distortion compensator resulted in significant improvement of image quality as compared to that measured before alignment. After completion of the wavefront compensation, the distortion was measured by lithographic tests, the system was put back in the interferometer, and the distortion compensator was utilized to minimize that aberration. Finally, all compensators were used to reoptimize the wavefront errors. The measured residual distortion (shown by the lengths of the vectors at various points within the field at the scale indicated) was then as indicated in Fig. 12.38(a). The improvement over measured distortion performance before compensation [Fig. 12.38(b)] is quite apparent.

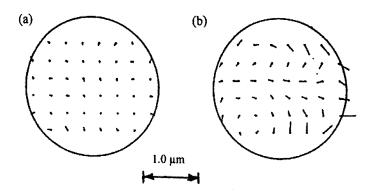


Figure 12.38 Measured residual distortion of the system of Fig. 12.36 (a) after compensation and (b) before compensation. The lengths of the vectors indicate magnitudes of the localized aberrations at the scale indicated. (From Williamson.¹⁵)

12.3 Aligning Reflecting Systems

In this section, we summarize some techniques for aligning a few optomechanical systems utilizing image-forming reflecting optics. Only geometric approaches are discussed. Techniques involving alignment optimization using interferometry, those requiring quantification of observed aberrations to define corrective measures, and those using real stars as objects are not included because of their complexity and space limitations. Wilson¹⁶ addresses the latter aspects of system alignment in detail. Each system discussed is considered as a camera objective to be used with photographic plates or a focal plane array, i.e., without an eyepiece. Details about mounting the optics and mechanisms for tracking objects are not considered.

12.3.1 Aligning a simple Newtonian telescope

Figure 12.39(a) shows a simple Newtonian telescope comprising a first-surface parabolic primary mirror and a right-angle prism mounted in a tubular housing. An eyepiece fits into the adapter shown at the traditional location near the entrance aperture. The primary mirror mount provides two-axis tilt capability, while the prism is adjustable along the axis, in rotation around that axis, and in rotation about an axis normal to the plane of the figure. The telescope optical and relative apertures are assumed characteristic of construction and use by an amateur astronomer. The following is a suggested alignment procedure for such a system based largely on ones given by McAdam¹⁷ and Eliason.¹⁸ Larger systems require the use of more sophisticated alignment techniques. See Wilson.¹⁶

Items A, B, and C of Fig. 12.39(b) are metal discs machined to fit snugly inside the telescope tube at the entrance and primary cell ends. Each disc has a \sim 3-mm diameter hole bored at its center. Light sources are placed just beyond the holes in C and A. Item D is a tube sized for a snug sliding fit into the eyepiece adapter bore. It has a crosshair crossing the inner aperture and a small sight hole of 1- to 2-mm diameter in the plate at its outer end.

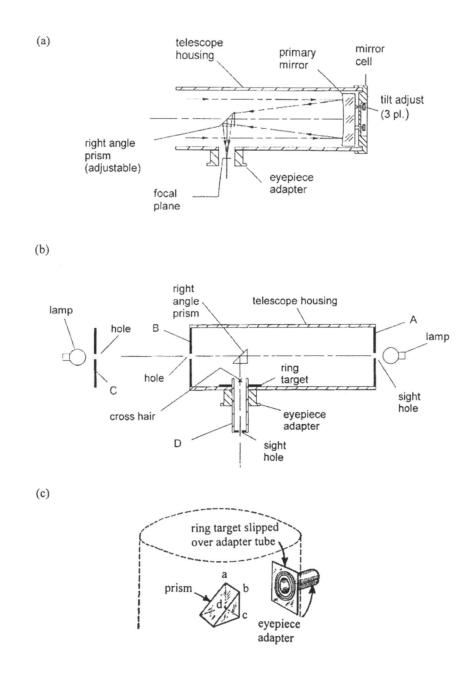


Figure 12.39 Sketches illustrating a suggested procedure for aligning a Newtonian telescope. (a) The completed telescope, (b) alignment setup, and (c) detail of the prism and its alignment target. [View (c) from McAdam.¹⁷ Copyright 1996 by Jeremy Graham Ingals and Wendy Margaret Brown.]

The first step is to install the prism and move it along the axis until it appears approximately centered when observed through the sight hole in D. A target made of white cardboard or similar material is prepared with a series of concentric black circles surrounding a hole that allows the target to slip snugly over the inner end of D. See Fig. 12.39(c). After installing this target, disc A is installed. A faint reflection of the ring target will then be seen reflected in the square face of the prism [a, b, c, d in Fig. 12.39(c)] when observing through the hole in D. The prism is tilted about the telescope axis and about an axis normal to the plane of the figure until this ring image appears centered to the prism face. The hole in A is then back-illuminated sufficiently for it to be seen reflected by the prism's hypotenuse when observing through the sight hole in D. The axial location of the prism and its tilts are then fine tuned until this reflection is centered on the crosshairs and the ring target reflection from the square face of the prism appears centered to that face. The prism adjustments are then secured and the ring target removed.

Disc B is installed and disc C is located approximately on the telescope axis and at a short distance inside the expected center of curvature of the primary. Its lateral position is set by observation through the holes in A and B. The latter discs and the ring target are then removed and the primary installed. The primary is then tilted until the reflected image of the illuminated hole in C is concentric with that hole. This works best with the image slightly larger than the hole. The mirror adjustments are now secured and the alignment is complete.

The alignment of the telescope can be verified visually during use by observation of a star image centered in the field of view and racking a high-powered eyepiece back and forth through best focus to see if the out-of-focus images are circular. If not, corrective alignment may be warranted.

12.3.2 Aligning a simple Cassegrain telescope

A typical Cassegrain telescope is shown schematically in Fig. 12.40(a). The technique for its alignment considered here is appropriate to an aperture and relative aperture corresponding to an amateur astronomer's equipment. A larger professional instrument will require techniques that are more advanced. See Ruda¹⁹ and Wilson.¹⁶

The technique, adapted from McAdam¹⁷ and Lower,²⁰ assumes that the primary is perforated centrally and the focal plane is accessible beyond that mirror. The primary mount has three tilt adjustments, as does the secondary mirror mounted in a cell supported by a spider. An eyepiece adapter, located at the rear of the telescope housing, is aligned mechanically to be concentric with the axis of the housing.

To initiate alignment, we install the primary and a disc A, which has a centrally located hole of \sim 3-mm diameter as shown in Fig. 12.40(b). A sighting tube with front crosshair and a sight hole of a 1- to 2-mm diameter is installed in the eyepiece adapter. Another disc, B, is located approximately on the telescope axis at a short distance inside the expected center of curvature of the primary. Its lateral position is set by observation through the holes in the sighting tube and disc B. The latter disc is then removed. Note that these alignment aids are similar to those mentioned in the preceding section.

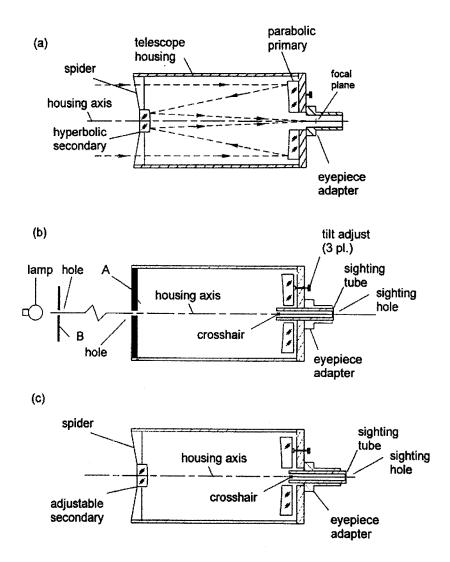


Figure 12.40 Aligning a simple Cassegrain telescope. (a) Schematic of the complete telescope, (b) alignment of the primary mirror to the housing axis, and (c) alignment of the secondary mirror.

The spider and the cell are now installed. During fabrication, the secondary cell should have been centered mechanically to the OD of the spider and a mark made at the cell center. This mark is to coincide with the axis of the telescope housing. One should be able to see this mark by looking through the sighting tube. It should appear to coincide with the crosshair in that tube. If not, the spider must be adjusted, modified, or shimmed until this error is corrected. Then, the secondary is installed. Again looking through the sighting tube, the aperture of the secondary, the reflected image of the end of the sighting tube, and the image of the crosshair should all appear centered to each other and to the actual crosshair. If this is not true, the secondary should be tilted until that alignment is achieved. The sighting tube is then removed.

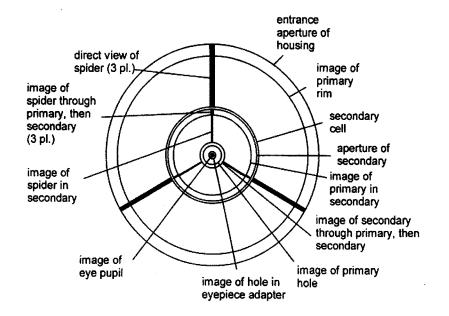


Figure 12.41 Schematic appearance of a generic Cassegrain system when viewed through the eyepiece adapter after alignment. The sketch is not necessarily to scale.

Upon looking through the eyepiece adapter, one should see a number of concentric circles. Figure 12.41 shows how these might look for a generic system. The sketch is not necessarily to scale. The rings represent, in sequence from outer to inner: the entrance aperture of the telescope housing, the reflected rim of the primary, the secondary cell, the aperture of the secondary, the reflection of the primary in the secondary, the reflection of the secondary by way of the primary and then the secondary, the reflection of the hole in the primary, the reflection of the hole in the eyepiece adapter, and the pupil of your eye. If these circles are not concentric, the tilts of the primary (and perhaps those of the secondary) should be fine adjusted. Once this desired condition is achieved, the alignment is completed and the mirror adjustments should be secured.

12.3.3 Aligning a simple Schmidt camera

A classical Schmidt telescope is shown schematically in Fig. 12.42. It comprises a spherical primary, an aspheric corrector plate (located at the primary center of curvature), and a film (or plate) holder located at the convex focal surface, as well as the usual housing, mounts for the optics, and the spider (not shown in the figure) that holds the film or plate. Adjustments include two tilts for the primary and centering means for the corrector plate. Tilt of the plate is generally not very significant so adjustment is not needed. Centration means for the plate may not be needed if the plate's optic axis coincides with its mechanical centerline as a result of careful fabrication.

Alignment of the optics usually occurs in three steps. The primary is installed and aligned to the housing axis as described in Section 12.3.1 for a Newtonian telescope. The spider is then installed and positioned mechanically. If centering alignment of the

corrector plate is needed, it is usually done photographically at the time of "squaring" the film surface to the image. These are a process of successive approximations using a real star field or a simulated one from a full-aperture collimator. Detailed instructions for this process are not included here as they obviously involve careful inspection of images for quality and consistency over the field and iterative adjustment to optimize the image.

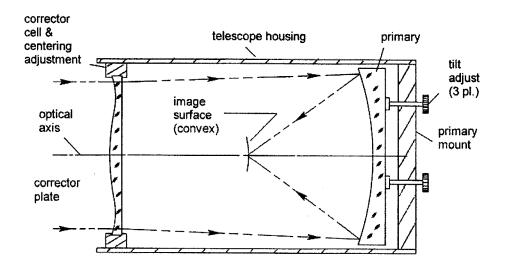


Figure 12.42 Optomechanical schematic of a Schmidt camera. The spider supporting the film holder is not shown.

A particularly interesting Schmidt camera was described by Paul.²¹ It was a 5-in. (12.7-cm) aperture, 4-in. (10.2-cm) focal length, f/0.8 system. The need for a spider was eliminated by attaching the film holder to the center of the corrector plate with a lightweight (hollow) stalk made of magnesium. This material had a CTE approximately twice that of stainless steel used in the housing. The camera automatically compensated for temperature changes because the stalk was essentially half the length of the housing. Absence of the spider removed diffraction effects from the image.

12.4 References

- 1. Smith, W. J., *Modern Optical Engineering*, 3 rd. ed., McGraw-Hill, New York, 2000.
- 2. Bayar, M. "Lens barrel optomechanical design principles," Opt. Eng. 20, 1981:181.
- 3. Parks, R.E., and Kuhn, W.P., "Optical alignment using the Point Source Microscope," *Proceedings of SPIE* **58770B-1**, 2005.
- 4. Parks, R.E., "Alignment of optical systems," Proceedings of SPIE 634204, 2006.
- 5. Parks, R.E., "Versatile autostigmatic microscope," *Proceedings of SPIE* **62890J**, 2006:
- 6. Parks, R.E., Precision centering of lenses," Proceedings of SPIE 6676, 2007:TBD.
- 7. Hopkins, R.E., "Some thoughts on lens mounting," Opt. Eng. 15 1976:428.
- 8. Yoder, P. R., Jr., Principles for Mounting Optics, SPIE Short Course SC447, 2007.
- 9. Benford, J.R., "Microscope Objectives," Chapter 4 in Applied Optical and Optical Engineering III, R. Kingslake, ed., Academic Press, New York, 1965.

- Sure, T., Heil, J., and Wesner, J., "Microscope objective production: On the way from the micrometer sale to the nanometer scale," *Proceedings of SPIE* 5180, 2003:283.
- 11. Heil, J., Wesner, J., Mueller, W., and Sure, T., Appl. Opt., Optical Technol. Biomed. Opt., 42, 2005:5073.
- Carnell, K. H., Kidger, M. J., Overill, A. J., Reader, R. W., Reavell, F. C., Welford, W. T., and Wynne, C. G. (1974). "Some experiments on precision lens centering and mounting," *Optica Acta* 21:615.
- Vukobratovich, D., "Optomechanical Systems Design," Chapt. 3 in *The Infrared & Electro-Optical Systems Handbook*, IV, ERIM, Ann Arbor and SPIE, Bellingham, WA, 1993.
- 14. Bacich, J.J., "Precision Lens Mounting," U.S. Patent 4,733,945, 1988.
- 15. Williamson, D.M., "Compensator selection in the tolerancing of a microlithography lens," *Proceedings of SPIE* **1049**, 1989:178.
- 16. Wilson, R.N., Reflecting Telescope Optics II, Springer-Verlag, Berlin, 1999.
- 17. McAdam, J.V., "Collimation," Sect. C.1.1 in *Amateur Telescope Making, Book 2,* Willmann-Bell, Richmond, 1996:363.
- 18. Eliason, C.W., "Collimation," Sect. C.1.1 in *Amateur Telescope Making, Book 2,* Willmann-Bell, Richmond, 1996:364
- 19. Ruda, M., *Introduction to Alignment Techniques*, SPIE Short Course SC010, SPIE Bellingham (2003).
- 20. Lower, H.A., "Collimating a Cassegrainian," Sect. C.1.3.1 in *Amateur Telescope Making, Book 2,* Willmann-Bell, Richmond, 1996:372.
- 21. Paul, H.E., "Schmidt camera notes," Chapt. E.4 in *Amateur Telescope Making, Book* 2, Willmann-Bell, Richmond, 1996:451.

CHAPTER 13 Estimation of Mounting Stresses

13.1 General Considerations

Contact stress is created whenever force is applied within small areas on the surface of an optical component. This stress depends on the magnitude of the force, the shapes of both surfaces in contact, the size of the contact area between the optical and mechanical bodies (both considered to be elastic), and the pertinent mechanical properties of the contacting materials. In this chapter, we summarize a theory based on equations of Roark¹ for estimating the magnitude of compressive contact stress for a variety of commonly used glass-to-metal interface types involving lenses, windows, mirrors, and prisms. A relationship from Timoshenko and Goodier² is then applied to estimate the tensile stress based on work published elsewhere is stated. The stress and surface deflections resulting from radially unsymmetrical application of axial clamping forces on opposite sides of rotationally symmetrical components are approximated for simple cases. The analytical models forming the bases for the stress equations given here are believed be conservative representations of real-life situations.

A compressive force exerted over a small area on an optical surface causes elastic deformation, i.e., strain, of the local region and hence proportional stress within that region. If the stress exceeds the damage threshold of the optical material, failure may occur. Rigorous calculations of damage thresholds for glass-type materials are complex and rely on statistics to determine the probability of failure under specified conditions.³⁻¹² We report key results of these studies and the basis for the generally accepted rule-of-thumb value of 1000 lb/in.² (6.9 MPa) for the tensile stress in glass that might cause damage. As a further approximation, we assume that the same tolerance applies to nonmetallic mirror materials and optical crystals. For simplicity, we refer to all optical materials as glass and all mechanical ones as metal. Stress also builds up within the mechanical members that compress the glass. This usually is compared with the yield strength of that metal (generally taken as that stress resulting in a dimensional change of two parts per thousand) to see if an adequate safety margin exists. In critical applications (such as those demanding extreme long-term stability), stress in mechanical components may be limited to the microyield stress value for the material.

Since operational environmental conditions invariably are less stringent than survival conditions, damage is not a concern, but detrimental effects on performance can occur. Mounting forces then may cause optical surfaces to deform, i.e., become strained. Such deformations may affect performance. No meaningful general tolerances for surface deformations can be given since they depend on the level of performance required and the location of the surface in the system (surface deformations are less significant near an image but are more significant near a pupil). Closed form equations for estimating surface deformations, as functions of force applied to optical components, are available only for a few cases. These evaluations are best done by finite-element analysis methods. Such methods are beyond the scope of this work.

The stress induced by operational levels of strain can affect the performance of optical components used in applications involving polarized light through the introduction of birefringence. This is a localized change in refractive index of the material that affects the phase of the orthogonal polarized components of the radiation transmitted through the stressed region. The magnitude of the effect depends on the stress level, the stress optic coefficient for the material, and the path length within that material. How this effect is estimated and applicable tolerances are considered in Section 13.3.

13.2 Statistical Prediction of Optic Failure

Earlier in this book, we considered many ways to constrain lenses, windows, prisms, and small mirrors using such means as threaded retaining rings, flanges, and springs. Each of these designs uses specifically shaped mechanical surfaces at the interfaces with the glass components. The actual glass-to-metal contacts are "points," "lines," or small areas of specific shapes through which forces are transferred to the optic. These forces deform (i.e., strain) the glass and the metal elastically. Strain is expressed as $\Delta d/d$, where Δd represents deflection and d represents the dimension that is strained. Strain is dimensionless. Strains develop both at the interface where the force is applied and at the opposing interface, where restoring force is provided in order to constrain the optic. Ideally, but not always necessarily, the interfaces are directly opposite each other so the force vectors applied normal to the respective surfaces pass through the opposing interfaces.

Hooke's Law requires that strains produce proportional stresses within and around the deformed regions. This stress is expressed as force per unit area with units of lb/in.² or Pa. For example, a prism pressed by a straddling spring bearing a cylindrical pad centered on the top surface against three small coplanar locating pads arranged in a triangular pattern on the baseplate would experience compressive and tensile stresses at all four interfaces with pads. In this case, the preload provided by the spring cannot be aimed towards all the pads. Typically, it would be aimed towards the centroid of the triangular pattern. Bending moments thus introduced would not be problematic if the prism and the baseplate carrying the three pads both are sufficiently stiff. Another example would be a lens preloaded against a cell shoulder by a retaining ring that presses against an annular area on the polished glass surface outside the clear aperture. Compressive and tensile stresses are generated in the glass within and adjacent to both contact areas.

In this section, we are concerned with the effects of stress buildup in optical components such as lenses, small mirrors, windows, and prisms. These stress values can then be compared with damage thresholds for the materials involved. Rigorous assessments of stress-related damage utilize statistics to determine the probability of immediate or delayed failure under specified conditions. Use of these methods requires knowledge of the quality of finish (i.e., existence of defects and the shapes, sizes, orientations, and locations of those defects) of the optical surfaces. Defects are produced during grinding and polishing operations as well as during handling, mounting, and use of the optic. Environmental exposure is also a factor here.

The strength of a glass component under stress can be estimated by testing it several times at, or above, the anticipated stress level. This gives confidence, but not proof, that the

item will not fail when exposed to that level of stress. Ideally, the test should simulate the conditions of use. If it is not possible to test the actual item or items, the next best course of action would be to test multiple identical coupons made of the same material as the component of concern that have been manufactured and handled in the same manner as that component. This data can be scaled to the size of the actual optic as an indication of the capability of that part to withstand a specified level of stress.⁹ Testing of the coupons can be accomplished by methods such as three-point, four-point, or ring-on-ring bending as illustrated schematically in Figs. 13.1 and 13.2. The latter method is preferred for optical glasses because the sample is a disk that can be polished to the same specification as the actual component. Coupon testing can also be accomplished by indenting the surface with the square pyramidal point of a crystalline diamond tool on a Vickers indenter (see Fig. 13.3) or the elongated rhomboid point on a Knoop indenter. The dimension a is the radius of the circle circumscribed around the deformed area while c is the depth of the characteristic crack that extends into the material. These tests have been described and illustrated by many authors including Adler and Mihora¹¹ and Harris.⁹ Lacking measured data on the actual optic or coupons, a less reliable lifetime prediction can be based on surface quality assumptions based on prior experience with typical manufacturing and handling methods for materials similar to the actual optic.

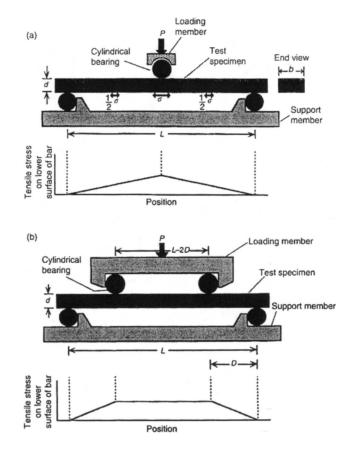


Figure 13.1 Equipment for flexure strength measurement by (a) three-point bending and (b) four-point bending of samples. The black circles are the ends of cylinders. (From Harris.⁹)

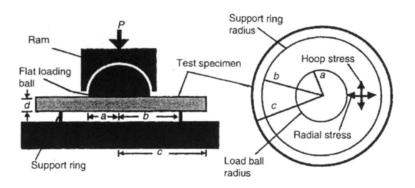


Figure 13.2 Equipment for flexure strength measurement by ring-on-ring test. Within the area of radius a, radial and tangential stresses are equal. (From Harris.⁹)

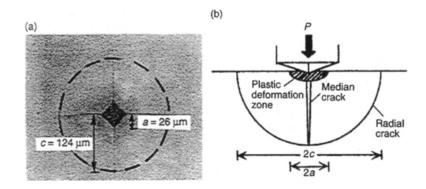


Figure 13.3 (a) Micrograph of a Vickers indentation pattern created by a load P in a coupon. The dashed line was added to show the extent of the radial crack. (b) An idealized cross section of the damage with a crack of depth c extending into the material. (Adapted from Harris.⁹)

The best way to predict whether a component will survive a given amount of stress is by way of fracture mechanics involving (1) the tendency for existing surface flaws or cracks in the optic to grow when stressed, (2) known or assumed characteristics of those defects, and (3) the anticipated level of stress. Growth of a flaw such as a microscopic crack in the surface results from the tendency for tensile stress to be concentrated at the tip of the crack. When this concentrated stress becomes large, failure would be expected to occur.

The velocity with which a crack propagates has been shown to correlate well with the intensity of the applied stress and with the size of the crack. In a dry environment (vacuum or cryogenic temperature), a flaw may not propagate until a threshold stress is reached. For a constant stress larger than that threshold, the flaw would grow until the part fails. The crack growth velocity increases in direct proportion to the relative humidity of the environment surrounding the optic and within the crack.¹² The flaw grows under constant

stress at a predictable rate until it reaches a critical dimension. Then, the optic probably fails. As might be expected, the velocities differ for different materials.

Fuller et al.⁸ wrote that "the processes that engender defects and cracks in glass can also give rise to localized residual stresses that influence the propagation of the ensuing cracks." This residual stress field significantly reduces the ability of the optic to survive stress applied for an extended time.

Weibul³ established a theoretical method that is particularly useful in defining the survivability of an optical component under stress. It involves the statistical probability of the occurrence of an event. The component may be thought of as a continuum of N elements, any of which can fail. The probability of failure P_F of an optic when the weakest element of that optic fails can be predicted from experimental data, such as the stress levels at which a set of coupons fail.

Pepi⁷ showed how to use Weibul's theory to estimate the time to failure of an optic subjected to a constant level of stress. His mathematical treatment resulted in the graph shown in Fig. 13.4. The material was BK7 glass, the relative humidity was high, the reliability of the prediction was 99%, and the confidence level was 95%. In the uppermost curve, the "as manufactured" surfaces were polished to 60-10 scratch and dig surface quality per U.S. military specification MIL-O-13830A.¹³ In the lower curves; the surface has intentionally been damaged from progressively more severe exposures. The next curve represents erosion by airborne dust traveling at 234 m/s (768 f/s) and impacting the surface at 15-deg., The third curve represents erosion by windblown sand traveling at 29 m/s (95 f/s) and impacting the surface at 90-deg., and the lower curve represents the presence of a single, centrally located scratch 50- to 100-µm wide by 20- to 25-µm long produced with a Vickers diamond. It is this author's opinion that, in the absence of corresponding information pertinent to other specific optics, these curves may also be applied, in general, to optics made with care from other optical glasses used in a typical natural (unfriendly) environment.

Doyle and Kahan¹⁰ applied similar statistical methods to predict the lifetime to failure of a 10-in. (25.4-cm) diameter, 0.75-in (1.90-cm) thick BK7 corrector plate for a Schmidt telescope that was mounted in an aluminum cell by means of six titanium tangential flexures bonded to the rim of the plate. The graphs of Fig. 13.5 were derived. For a desired 10-year lifetime (the vertical dashed line) and a desired probability of failure of 10^{-5} (lowest curve), a design tensile stress of 1700 lb/in.² (12.75 MPa) (the horizontal dashed line) was established for each of the bonds between the flexures and the plate rim. Note, this critical stress differs slightly from that specified by Doyle and Kahan¹⁰ and represents a refinement in the calculation as provided by Doyle.¹⁴

The investigations reported by Pepi,⁷ Fuller et al,⁸ Doyle and Kahan,¹⁰ and Doyle¹⁴ all dealt with optics carefully manufactured by a process designed to ensure that subsurface damage during each stage of grinding would be eliminated by the next stage using a finer grit size. Stoll et al¹⁵ first described this process, called "controlled grinding." The material removal at each stage is three times the average diameter of the preceding abrasive. A comparison of this process with the conventional process, in which significantly less material is removed with grit, is given in Table 13.1. For the maximum lifetime of an optic under stress, it should always be made by controlled grinding.

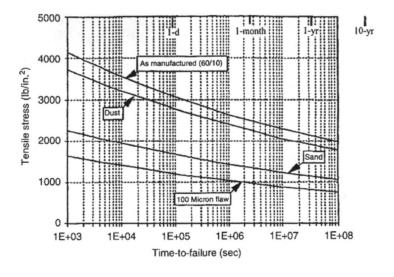


Figure 13.4 Plots of time-to-failure vs. applied tensile stress for BK7 glass with different degrees of surface damage for 99% reliability with 95% confidence. (From Pepi.⁷)

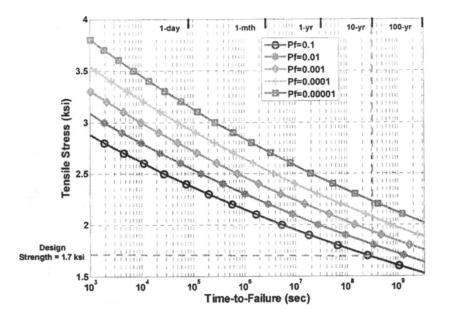


Figure 13.5 Time-to-failure curves for a 10-in. (25.4-cm) diameter BK7 Schmidt telescope corrector plate vs. applied tensile stress for five levels of failure probability. (From Doyle and Kahan¹⁰ as refined by Doyle.¹⁴)

		Average		Material removal			
		particle size		Conventional		Controlled	
Operation	Abrasive	(mm)	(in.)	(mm)	(in.)	(mm)	(in.)
Milling	150 grit diamond	0.102	0.004				
Fine grind	2F Al ₂ O ₃	0.0304	0.0012	0.0381	0.0015	0.3048	0.0120
Fine grind	3F Al ₂ O ₃	0.0203	0.0008	0.0177	0.0007	0.0914	0.0036
Fine grind	KH Al ₂ O ₃	0.0139	0.00055	0.0127	0.0005	0.0609	0.0024
Fine grind	KO Al ₂ O ₃	0.0119	0.00047	0.0076	0.0003	0.0406	0.0016
Polish	Barnsite rouge						

Table 13.1 Typical schedules for conventional and controlled grinding of optical materials.

Source: Stoll et al.¹⁵

13.3 Rule-of-Thumb Stress Tolerances

Because the actual qualities of an optic's surfaces are usually unknown in real life, statistical failure analysis methods cannot always be applied with confidence. We here will apply rule-of-thumb values for the stress levels in glass that are likely to cause problems. For many years, these were based on guidance from Shand¹⁶ as 50,000 lb/in.² (345 MPa) for compressive stress and 1000 lb/in.² (6.9 MPa) for tension. In the immediately preceding sections, we have shown that failure of brittle materials most often occurs in tension. It would then seem appropriate to base our design decisions on estimated tensile stress rather than compressive stress. As indicated earlier, Doyle and Kahan¹⁰ suggested a rule-of-thumb tensile stress tolerance for glass of 1000 to 1500 lb/in.² (6.9 MPa) value as a generic glass survival tensile stress tolerance.

Most of the stress calculations that follow apply equations adapted from Roark¹ to estimate compressive stress values at glass-to-metal interfaces. Timoshenko and Goodier² pointed out that compressive contact stress at these interfaces is accompanied by tensile stress in both materials. This stress occurs at the boundary of the region elastically compressed by the applied preload and is directed radially. Those authors gave the following equation for this tensile stress S_T in terms of Poisson's ratio for the glass v_G and the compressive stress S_C :

$$S_{T} = \frac{(1 - 2v_{G})(S_{C})}{3}.$$
 (13.1)

In Table 13.2, this equation is applied to selected optical glasses, crystals, and mirror materials. The glasses represent those from the 50 glasses listed in Table B1 having the smallest and largest v_G values as well as the ubiquitous BK7. The crystals are ones frequently used for IR optics (from Tables B1 and B4 through B7) whereas the mirror materials are the most common types of nonmetallics (from Table B8a). We see from the right-hand column of Table 13.2 that S_T is typically S_C divided by a number ranging from

a low of 4.54 to a high of 9.55 for these materials. Crompton¹⁷ indicated that some members of the optical industry use a S_T/S_C ratio of 0.167 or 1/6 as a rule-of-thumb value for stress analysis of optomechanical interfaces. Roark¹ and later versions of that work, such as Young,¹⁸ gave no equation for this relationship, but specified $S_T \approx 0.133S_C = S_C/7.52$ for mechanical structures. We use Eq. (13.1) for tensile contact stress estimation in this chapter. If v is not specified, we assume that a factor of 1/6 applies.

	Poisson's S _T /S _C		-					
Material	Ratio	per Eq. (13.1)	S_C/S_T					
Optical glasses:								
K10	0.192	0.205a	4.87a					
BK7	0.208	0.195	5.14					
LaSFN30	0.293	0.138a	7.25a					
IR Crystals:								
BaF ₂	0.343	0.105a	9.55a					
CaF ₂	0.290	0.140	7.14					
KBr	0.203	0.198	5.05					
KCl	0.216	0.189	5.28					
LiF	0.225	0.183	5.45					
MgF ₂	0.269	0.154	6.49					
ALON	0.240	0.173	5.77					
Al ₂ O ₃	0.270	0.153	6.52					
Fused silica	0.170	0.220a	4.54a					
Ge	0.278	0.148	6.76					
Si	0.279	0.147	6.79					
ZnS	0.290	0.140	7.14					
ZnSe	0.280	0.147	6.82					
Mirror materials:								
Pyrex	0.200	0.200	5.00					
Ohara E6	0.195	0.203	4.92					
ULE	0.170	0.220a	4.54a					
Zerodur	0.240	0.173	5.77					
Zerodur M	0.250	0.167a	6.00a					

Table 13.2 Ratio of tensile to compressive contact stress for selected optical materials.

Note: "a" indicates extreme high or low value for this group.

As mentioned at the outset of this chapter, we assume that the 1000 lb/in.² (6.9 MPa) tensile stress tolerance applies to nonmetallic mirror materials and optical crystals. For simplicity, we refer to all optical materials as glass and all mechanical ones as metal. In general, application of a safety factor of at least two is advisable.

The stresses induced by operational levels of strain can degrade the performance of optical components used in applications involving polarized light through the introduction of birefringence. This is a localized change in refractive index of the material that creates an optical path difference between two orthogonal polarized components of the radiation transmitted through the stressed region. Tolerances on birefringence are usually expressed in terms of the permitted optical path difference (OPD) for the parallel (||) and perpendicular (\perp) states of polarization of transmitted light at a specified wavelength. According to Kimmel and Parks,¹⁹ birefringence of components for various instrument applications should not exceed 2 nm/cm for polarimeters or interferometers, 5 nm/cm for precision applications such as photolithography optics and astronomical telescopes, 10 nm/cm for camera, visual telescope, and microscope objectives, and 20 nm/cm for eyepieces and viewfinders. Higher birefringence can be tolerated in condenser lenses and most illumination systems. In all cases, the material's stress optic coefficient K_S determines the relationship between the applied stress and the resulting OPD. Equation 1.3 applies. It is repeated here for easy reference.

$$OPD = \left(n_{\parallel} - n_{\perp}\right)t = K_s St, \qquad (1.3)$$

where t is the path length in the material in cm, K_S is expressed in mm²/N, and S is the stress in N/mm². Table 1.5 lists the values of K_S at a wavelength of 589.3 nm and a temperature of about 21°C for the optical glasses listed in Table B1. Knowing this factor and the path length, it is a simple task to establish a limit for stress in an optic.

It should be noted that surface deformations and related birefringence effects from mounting forces are seen primarily in the local regions where those forces are applied (see Sawyer²⁰). Typically, these regions are near, but outside the optic's clear aperture, so the effects may not be significant over most of that aperture.

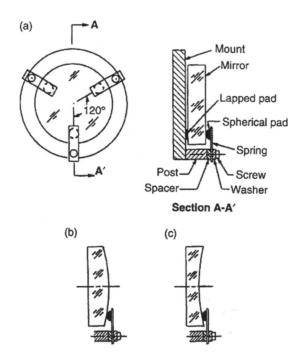


Figure 13.6 Spherical pad-to-optic interfaces with (a) a flat optic, (b) a convex optic, and (c) a concave optic.

13.4 Stress Generation at Point, Line, and Area Contacts

Point contacts occur when spherical mechanical pads interface with curved or flat optical surfaces. Figure 13.6(a) shows a concept for preloading a flat mirror against lapped pads by three cantilevered spring clips. Each spring has a spherical pad. Deflections of the springs provide preload, as discussed in Section 9.1. Pad interfaces with convex and concave optical surfaces are shown in views (b) and (c) of the figure. Any of these surface contours could exist on lenses and mirrors. Prisms typically have flat optical surfaces. Interfaces with flat surfaces are also found at flat bevels adjacent to the rims of some concave optical surfaces and at step bevels used with convex optical surfaces.

Three cases involving optical and mechanical (pad) surfaces in contact under preload P are shown in Fig. 13.7. In view (a), both the optical surface and the mechanical surface are convex and spherical. In view (b), both surfaces are spherical, but the convex surface is concave and the mechanical surface is convex. In view (c), the optical surface is flat

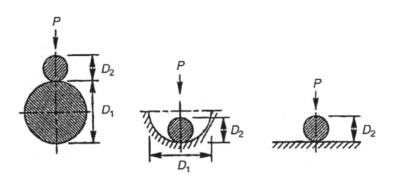


Fig. 13.7 Key dimensions related to point contacts between elastic bodies. Shown are convex spherical pads touching (a) a convex optic, (b) a concave optic, and (c) a flat optic. *P* represents the total applied preload.

while the mechanical surface is spherical and convex. In all cases, both contacting surfaces deform elastically creating circular contact regions of radius r_c . The areas of all these circles are given by

$$A_{CSPH} = \pi r_C^2. \tag{13.2}$$

These areas depend on the shapes and radii of both surfaces, the material characteristics, and the preload. The following equations, adapted from Roark,¹ apply:

$$r_{c} = 0.721 \left(\frac{P_{i}K_{2}}{K_{1}}\right)^{1/3}$$
(13.3)

^{*} The "SPH" in the subscript for A_C identifies a spherical contact. Terms used in other applications are "CYL," "SC," "TAN," and "TOR" representing cylindrical, sharp corner, tangent, and toroidal contacts, respectively.

$$K_{1} = \frac{\left(D_{1} \pm D_{2}\right)}{D_{1}D_{2}}$$
(13.4a)

use + for a convex and – for a concave surface

$$K_1 = \frac{1}{D_2}$$
 (13.4b)

for a flat optical surface

$$K_{2} = K_{G} + K_{M} = \left[\frac{\left(1 - \nu_{G}^{2}\right)}{E_{G}}\right] + \left[\frac{1 - \nu_{M}^{2}}{E_{M}}\right],$$
 (13.5)

where P_i is the preload per spring = P/N; N is the number of springs; D_1 is twice the radius of curvature of the optical surface; D_2 is twice the radius of curvature of the contacting pad; and E_G , E_M , v_G , and v_M are Young's modulus and Poisson's ratio values for the glass and metal respectively. In general, either body in Fig. 13.7 can be the optic (glass) and the other body the mechanical pad (metal). Usually, D_1 and D_2 are the larger and smaller body diameters respectively. Although theoretically possible in some cases, concave pads are usually not used.

For all the geometric cases shown in Fig. 13.7, the average compressive stress in the contact region is:

$$S_{CAVG} = \frac{P_i}{A_{CSPH}}.$$
(13.6)

In the case shown in Fig. 13.8(a), intimate contact between a flat pad and a flat optical surface is assumed to exist. The stressed area A_c is still given by Eq. (13.2), but r_c equals one-half the diameter of the pad (assumed to be circular). Equation (13.6) gives the average stress within this contact area. It obviously is a smaller value for a flat pad than would occur with a convex pad. As shown in Fig. 13.8(b), a misaligned (i.e., tilted) pad will contact the optical surface asymmetrically leading to stress concentration in a localized region. This is undesirable and is a reason why a curved pad is preferable.

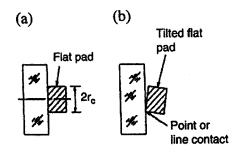


Figure 13.8 Contact between a flat pad and a flat optic: (a) Intimate contact between surfaces, and (b) nonsymmetrical contact resulting from a misaligned (tilted) pad causing stress concentration.

Example 13.1: Stresses at a spherical pad interface on various optic surfaces. (For design and analysis, use File 13.1 of the CD-ROM.)

A convex spherical pad with radius of 20.000 in. (508.000 mm) made of 6061 aluminum is attached to the end of a cantilevered spring. The pad presses against the polished surface of a large diameter N-BK7 lens. The optic surface radius is (a) 16.000-in. (406.400-mm) convex, (b) the same radius, but concave, and (c) infinite. The total preload is 1.800 lb (8.007 N). Estimate the peak tensile stress for each case.

From Tables B1 and B12:
$$E_M = 9,900 \times 10^{\circ} \text{Ib/m}^{\circ} (6.826 \times 10^{\circ} \text{MPa}), v_M = 0.332,$$

 $E_G = 1.200 \times 10^{7} \text{Ib/m}^{2} (8.274 \times 10^{4} \text{MPa}), v_G = 0.206.$
 $D_2 = (2)(20.000) = 40.000 \text{ in.} (1016.000 \text{ mm}),$
and $P_i = \frac{1.800}{1} = 1.800 \text{ Ib} (8.007 N).$
Per Eq. (13.5): $K_2 = \left[\frac{(1-0.206^2)}{1.200 \times 10^7}\right] + \left[\frac{(1-0.332^2)}{9.900 \times 10^6}\right]$
 $= 1.697 \times 10^{-7} \text{in.}^2/\text{Ib} (2.461 \times 10^{-5} \text{MPa}^{-1})$
Per Eq. (13.1): $\frac{S_{TSPH}}{S_{CSPH}} = \frac{[1-(2)(0.206)]}{3} = 0.1960$
(a) $D_1 = (2)(16.000) = 32.000 \text{ in.} (812.8 \text{ mm})$
Per Eq. (13.4a): $K_1 = \frac{(32.000 + 40.000)}{(32.000)(40.000)} = 0.0563 / \text{in.} (0.0022 / \text{mm})$
Per Eq. (13.7): $S_{CSPH} = 0.918 \left[\frac{(0.0563^2)(1.800)}{(1.697 \times 10^{-7})^2}\right]^{1/3} = 5347.0 \text{ Ib/in.}^2$
Then, $S_{TSPH} = (5347.0)(0.1960) = 1048.3 \text{ Ib/in}^2 (7.2 \text{ MPa})$
(b) $D_1 = (2)(16.000) = 32.000 \text{ in.} (812.8 \text{ mm})$
Per Eq. (13.4a): $K_1 = \frac{(32.000 - 40.000)}{(32.000)(40.000)} = -0.0063 / \text{in.} (-0.0002 / \text{mm}) *$
Per Eq. (13.4a): $K_1 = \frac{(32.000 - 40.000)}{(32.000)(40.000)} = -0.0063 / \text{in.} (-0.0002 / \text{mm}) *$
Per Eq. (13.7): $S_{CSPH} = 0.918 \left[\frac{(-0.0063^2)(1.800)}{(1.697 \times 10^{-7})^2}\right]^{1/3} = 1241.8 \text{Ib/in.}^2$
Then, $S_{TSPH} = (1241.8)(0.1960) = 243.4 \text{ Ib/in.}^2 (1.7 \text{ MPa})$
(c) $D_1 = \text{infinity so, per Eq. (13.4b):}$
 $K_1 = \frac{1}{D_2} = \frac{1}{40.000} = 0.025 / \text{in.} (9.8 \times 10^{-4} / \text{mm})$
Per Eq. (13.7): $S_{CSPH} = 0.918 \left[\frac{(0.025^2)(1.800)}{(1.697 \times 10^{-7})^2}\right]^{1/3} = 3112.3 \text{ Ib/in.}^2$
Then, $S_{TSPH} = (3112.3)(0.1960) = 610.0 \text{ Ib/in.}^2 (4.2MPa).$

* The negative sign is ignored

The compressive stress created by a spherical pad touching an optic actually is not uniform across the contact area. The peak contact stress occurs at the center of the area; it decreases towards the edge of that area. Equation 13.7 from Roark¹ is used to estimate the peak compressive stress value. We will call this S_{CSPH} :

$$S_{CSPH} = 0.918 \left(\frac{K_1^2 P_i}{K_2^2}\right)^{1/3}$$
(13.7)

Example 13.1 illustrates the use of Eqs. (13.2) through (13.7) for a spherical pad interface with a convex optic.

If a calculation such as this results in a tensile stress that is too large in comparison with the survival tolerance of 1000 lb/in.² (6.9 MPa) suggested earlier, possible design modifications would be to increase the pad radius and/or (better) to increase the number of springs.

Instead of using spherical pads on springs to hold an optic in place, we could provide convex cylindrical pads as the mechanical interface. Typically, such a pad would be oriented crosswise on the end of the spring and its axial length would equal the width b of the spring. Alternatively, the cylindrical axis could be oriented at any convenient angle to the cantilevered length of the spring. A cylindrical pad cannot be used with a concave optical surface. Point contact between the cylinder and a convex spherical optical surface would occur under light axial loading. With greater preload, the elastic bodies would deform and contact would occur over a small area.

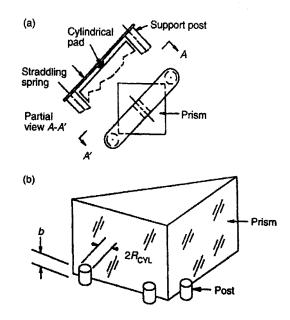


Figure 13.9 (a) A prism preloaded by a straddling spring bearing a cylindrical pad at its center. (b) A prism located on a baseplate by three cylindrical pins. The preloading spring is not shown, but presses against the prism hypotenuse near the base.

From the viewpoint of contact stress, the advantage of a cylindrical pad over a spherical one is slight when the optical surface is convex. This advantage increases when that surface is flat, so the primary uses of the cylindrical pad are to provide preload against flat surfaces of lenses, mirrors, or windows; against flat or step bevels on curved lens or mirror surfaces; or against prism surfaces. Figure 13.9 shows two examples of the latter case: (a) a cylindrical pad on a straddling spring preloading a prism and (b) interfaces at posts or pins that locate a prism on a baseplate.

The geometry of a cylindrical pad or pin interface with a flat optical surface can be modeled as shown in Fig. 13.10. The pad length equals the spring width b and its radius is R_{CYL} . D_2 is twice R_{CYL} . The surfaces are pressed together by the total preload per spring P_i so the preload per unit length p_i is P_i/b . The peak compressive stress $S_{C CYL}$ that occurs along the short line contact is given by the following equation:

$$S_{C \text{ CYL}} = 0.564 \left[\frac{p_i}{(R_{\text{CYL}} K_2)} \right]^{\frac{1}{2}},$$
 (13.8)

where K_2 is given by Eq. (13.5). Equation 13.1 provides the corresponding tensile stress.

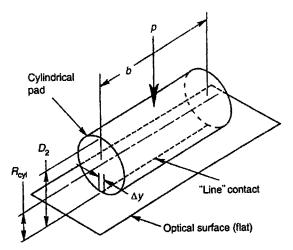


Figure 13.10 Analytical model of the interface between a cylindrical pad and a flat optical surface. The parameter p is the preload per unit length of contact.

The average stress within the deformed area is determined by dividing the total preload per pad by the area. The width is given by the equation:

$$\Delta y = 1.600 \left(\frac{K_2 p_i}{K_1}\right)^{\frac{1}{2}},$$
 (13.9a)

where K_1 is $1/D_2$ [from Eq. (13.4b)] because D_1 is infinite for a flat optical surface. Since $D_2 = 2R_{CYL}$, we can rewrite Eq. (13.9a) as

$$\Delta y = 2.263 \left(K_2 p_i R_{\rm CYL} \right)^{\frac{1}{2}}$$
(13.9b)

The area of the deformed region is then:

$$A_{C \text{ CYL}} = b\Delta y = 2.263b \left(K_2 p_i R_{\text{CYL}}\right)^{\frac{1}{2}}$$
(13.10)

The average stress in this area is then:

$$S_{C \text{ AVG}} = \frac{P_i}{A_{C \text{ CYL}}}$$
(13.11)

Example 13.2 illustrates estimation of peak and average tensile stresses with a cylindrical pad spring loaded against a flat optic surface.

The contact stress at a locating pin for an optic, such as the prism shown in Fig. 13.9(b) can also be estimated. The actual height of contact on each pin b and the pin diameter $2R_{CYI}$ must be known. Generally, the stress in the surface touching a single pin will be higher than that for a surface touching two pins. A typical case is illustrated in Fig. 13.11. The prism face width is A. The locating pins are positioned just outside the clear apertures (CA) of the entrance and exit faces. The preload is directed against a surface opposite the locating pins at a height approximating the midpoint of the pins. In Fig. 13.11(a), the preload is applied with a straddling spring normal to the base of the penta prism. In Fig. 13.8(b), it is directed normal to the hypotenuse surface. Yoder ²¹ explains how to do an analysis of this type and how to choose the angle with which the preload is applied to optimize the distribution of forces directed against the pins. Techniques for reducing the stress at the pins also include increasing the contact lengths b and the pin radii R_{CYL} . These changes are limited by the need to not introduce vignetting of the prism aperture. Adding more pins is not a viable change because the design would depart from the desired semikinematic approach. Reducing the required preload would help, but requires a reduction in the acceleration specification.

In all mounting arrangements wherein a prism is located against cylindrical pins, it is essential that the pin surface be parallel to the prism face. Otherwise, if the pin is slightly tilted due to manufacturing tolerances, localized stress concentration could occur at point contacts with the glass surfaces. Yoder²¹ suggested configuring the locating pins as shown in Fig. 13.12. The pin in view (a) is a commercially available "ball pin" while that in view (b) is a customized device comprising a cylinder attached to the top of a conventional cylindrical pin. One end of the upper cylinder has a long convex spherical radius that touches the prism face. This design has the advantage that the radius of the contacting surface can be made very long, thereby reducing contact stress under preload. The radius of the ball pin, on the other hand, is relatively short. Previously given equations for spherical contact on a flat surface allow estimation of stresses at the

interfaces for either design. Either of these pins can be pressed into a locating hole in the baseplate of the mount as would be done with a conventional cylindrical locating pin. The tolerances on angular orientation of the pin should result in contact with the prism at or near the center of the pad surface.

Example 13.2: Peak and average stresses at a cylindrical pad interface with a flat optic. (For design and analysis, use File 13.2 of the CD-ROM.)

A cylindrical pad made of 6061 aluminum has a radius of 12.000 in (304.800 mm) and length b of 0.125 in. (3.175 mm). It is pressed by a spring against a flat surface on an N-BK7 prism with a preload of 4.167 lb (18.534N). What are (a) the peak and (b) the average tensile stresses in the glass?

From Tables B1 and B12: $E_M = 9.900 \times 10^6$ lb/in.² (6.826×10⁴ MPa) and $v_M = 0.332$ $E_G = 1.200 \times 10^7$ lb/in.² (8.274×10⁴ MPa) and $v_G = 0.206$

The linear preload
$$p_1 = \frac{P_1}{b} = \frac{4.167}{0.125} = 33.336 \text{ lb/in} (5.837N / mm)$$

Per Eq. (13.4b): $K_1 = \frac{1}{D_2} = \frac{1}{(2)(12.000)} = 0.0417 / \text{in.} (0.0016 / \text{mm})$
Per Eq. (13.5): $K_2 = \left[\frac{(1-0.206^2)}{1.200 \times 10^7}\right] + \left[\frac{(1-0.332^2)}{9.900 \times 10^6}\right]$
 $= 1.697 \times 10^{-7} \text{ in.}^2 / \text{ lb} (2.461 \times 10^{-5} \text{ MPa}^{-1})$
(a) Per Eq. (13.8): $S_{C \text{ CYL}} = 0.564 \left[\frac{33.336}{(12.000)(1.697 \times 10^{-7})}\right]^{\frac{1}{2}}$
 $= 2282.0 \text{ lb/in.}^2 (15.7 \text{ MPa})$
Per Eq. (3.1): $S_{T \text{ CYL}} = \frac{[1-(2)(0.206)](2282.0)}{3} = 447.3 \text{ lb/in.}^2 (3.1 \text{ MPa})$
(b) Per Eq. 13.9a: $\Delta y = 2.263 \left[\frac{(1.697 \times 10^{-7})(33.336)}{0.0417}\right]^{\frac{1}{2}} = 0.0264 \text{ in.} (0.671 \text{ mm})$
Per Eq. 13.10: $A_{C \text{ CYL}} = (0.0264)(0.125)$
 $= 0.0033 \text{ in.}^2 (2.129 \text{ mm}^2)$
Per Eq. 13.11: $S_{C \text{ AVG}} = \frac{4.167}{0.0033} = 1263 \text{ lb/in.}^2 (8.7 \text{MPa})$
Per Eq. 13.11: $S_{T \text{ AVG}} = \frac{[1-(2)(0.206)](1263)}{3} = 248 \text{ lb/in.}^2 (1.7 \text{ MPa})$
These stresses are smaller than 1000 lb/in.² (6.9 MPa), therefore acceptable.

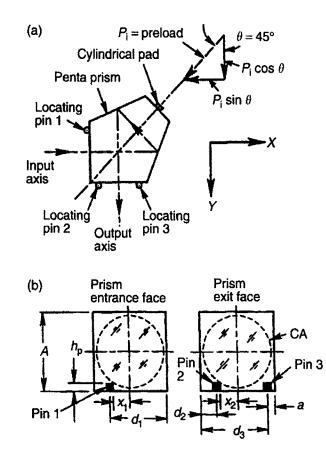


Figure 13.11 (a) Preload applied to the base of a penta prism pushes it against the locating pins opposite. (b) Arrangement of locating pins outside the apertures of the entrance and exit faces of the prism. (From Yoder.²¹)

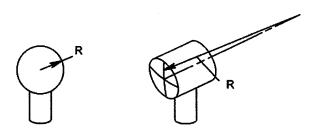


Figure 13.12 Noncylindrical locating pins. (a) A conventional ball pin and (b) a pin with a cylindrical top element bearing a long spherical radius at one end to contact the optic. (From Yoder.²¹)

13.5 Peak Contact Stress in an Annular Interface

The contact stress developed within a circular optic in a surface contact mounting from an axial force applied around the edge of the polished surface by means such as a threaded retainer or a flange depends on the preload, the radius of the optical surface, the geometric shapes of the contacting surfaces, and the physical properties of the materials involved. The force generally varies with temperature. The consequences of this variation are considered in Chapter 14.

Because the materials in the lens and the mount are both elastic, axial stress has a peak value S_C along the centerline of the narrow annular deformed area of contact between the metal and glass around the outer edge of the optical surface. This centerline is at a radius of y_C from the axis. The stress decreases at points within the lens progressively farther away radially from this centerline; i.e., toward and away from the lens axis. Figure 13.13 shows an analytical model of the interface. The larger cylinder of diameter D_1 represents the optical surface while the smaller cylinder of diameter D_2 represents the mount interface. Both cylinders have lengths equal to $2\pi y_C$, which is the perimeter of a circle of radius y_C . The cylinders are pressed against each other by the linear preload force p, i.e., the preload per unit length of contact, as indicated in the figure. The annular width of the elastically deformed region is indicated as Δy ; it is given by Eq. (13.9a). The optical surface in Fig. 13.13 is shown as convex. If the surface were to be concave, the smaller cylinder of the figure would contact the inside surface of the larger cylinder. Otherwise, the geometry would be unchanged.

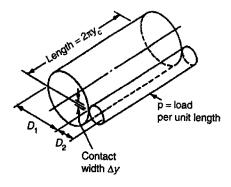


Figure 13.13 Analytical model of the annular surface contact between a convex mechanical constraint (smaller cylinder) and a convex optical surface (larger cylinder).

The average compressive contact stress within this rectangular deformed area can be calculated using Eq. (13.12) while Eq. (13.13) gives the peak compressive stress. The parameter p is the linear preload (or the preload per unit length of the contact). In Eq. (13.12), Δy is obtained from Eq. (13.9a):

$$S_{C \text{ AVG}} = \frac{P}{A_c} = \frac{P}{(2\pi y_C \Delta y)} = \frac{P}{\Delta y}$$
(13.12)

$$S_C = 0.798 \left(\frac{K_1 p}{K_2}\right)^{1/2}$$
 (13.13)

Here, K_1 is derived from Eq. (13.4a) with the algebraic sign depending on whether the optical surface is convex or concave. If that surface is flat, K_1 is derived from Eq. (13.4b). K_2 is determined from Eq. (13.5). The parameter K_1 is discussed in the following subsections in conjunction with various possible mechanical interface shapes of the mechanical interface.

If one divides Eq. (13.13) by Eq. (13.12) it can easily be shown that $S_C / S_{AVG} = (0.798)(1.600)$, so the peak axial contact stress for a lens, window, or mirror surface constrained axially near its rim is always 1.277 times the average value. Once again, the tensile stress S_T at the interface is obtained by applying Eq. (13.1).

13.5.1 Stress with a sharp corner interface

The sharp corner interface was described earlier as one in which the intersection of flat and cylindrical machined surfaces on the metal part has been burnished to a radius on the order of 0.002 in. (0.051 mm) (See Delgado and Hallinan²²). This small radius mechanical edge contacts the glass at height y_c , as shown schematically in Fig. 13.14. The angle between the intersecting machined surfaces can be 90-deg. (as shown) or preferably >90-deg. An obtuse angle edge between the machined surfaces usually can be made smoother and with fewer stress concentrating defects (pits or burrs) than an edge with a smaller included angle. The analytical model of Fig. 13.13 applies.

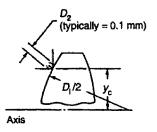


Figure 13.14 Sectional view of the sharp corner mechanical interface with a convex optical surface.

Assuming that D_2 for the sharp corner interface is always ~0.004 in. (~0.102 mm), substitution of this value into Eq. (13.4a) gives $K_{1SC} = (D_1 \pm 0.004)/0.004D_1$ (in USC units). For a convex or concave optical surface radius larger than 0.200 in. (5.080 mm), D_2 can be ignored and the value of K_1 is constant at 250/in. (10/mm). The error from this approximation does not exceed 2%.

Example 13.3 shows a typical calculation for a design with sharp corner interfaces.

Example 13.3: Peak and average contact stress in a lens with sharp corner mechanical interfaces. (For design and analysis, use File 13.3 of the CD-ROM.)

Consider a biconvex germanium lens with the following dimensions: $D_G = 3.100$ in. (78.7450 mm), $R_1 = 18.000$ in. (457.200 mm), and $R_2 = 72.000$ in. (1828.800 mm). The lens is mounted in a 6061 aluminum cell with sharp corner interfaces at $y_C = 1.500$ in. (38.100 mm) on both surfaces. (a) What peak tensile contact stresses are developed at each interface if the axial preload is 20.000 lb (88.964 N)? (b) What are the average tensile contact stresses at the interfaces?

(a) From Table B6: $E_G = 1.504 \times 10^7$ lb/in.² (1.037×10⁵ MPa) and $v_G = 0.278$ From Table B12: $E_M = 9.900 \times 10^6$ lb/in.² (6.820×10⁴ MPa) and $v_G = 0.332$

As previously defined:
$$p = \frac{20.000}{(2\pi)(1.500)} = 2.122 \text{ lb/in.}^2 (0.372 \text{ N / mm})$$

From Eq. (13.5): $K_2 = \left[\frac{(1-0.278^2)}{1.504 \times 10^7}\right] + \left[\frac{(1-0.332^2)}{9.9 \times 10^6}\right]$
 $= (6.135 \times 10^{-8}) + (8.988 \times 10^{-8})$
 $= 1.512 \times 10^{-7} \text{ in.}^2/\text{lb}(2.193 \times 10^{-5} \text{ MPa}^{-1})$

Because both radii exceed 0.200 in. (5.080 mm), $K_{1 \text{ SC}} = 250/\text{in.}$ (10/mm) for each surface. Hence:

From Eq. (13.13):
$$S_{C \text{ sc}} = 0.798 \left[\frac{(250)(2.122)}{1.512 \times 10^{-7}} \right]^{\frac{1}{2}}$$

= 47,268 lb/in.² (325.910 MPa) at each surface
From Eq. (13.1): $S_{T \text{ sc}} = \frac{\left[1 - (2)(0.278)\right] [47268]}{3}$
= 6996 lb/in.² (48.2MPa) at each surface.

(b) As stated in the text, the average axial tensile stress is (peak value/1.277) or, in this example, 6996/1.277 = 5478 lb/in.² (37.8 MPa) at each surface.

These stresses greatly exceed the suggested 1000 lb/in.² (6.9 MPa) tolerance so are not acceptable.

13.5.2 Stress with a tangential interface

A sectional view of the tangential interface and its analytical model are shown in Fig. 13.15. This interface was described earlier as an interface in which a convex spherical lens surface contacts a conical mechanical surface. This interface type cannot be used with a concave optical surface. Equation (13.13) is used to calculate $S_{C TAN}$. According to Eq. (13.5b), K_1 is $1/D_1$ where D_1 is twice the optical surface radius while p and K_2 are the same as for the

sharp corner case. Example 13.4 illustrates the calculation for the same design as evaluated in Example 13.3, but with tangential interfaces.

Example 13.4: Peak contact stress in a lens with tangential mechanical interfaces. (For design and analysis, use File 13.4 of the CD-ROM.)

Consider a biconvex germanium lens with the following dimensions: $D_G = 3.100$ in. (78.7450 mm), $R_1 = 18.000$ in. (457.200 mm), and $R_2 = 72.000$ in. (1828.800 mm). The lens is mounted in a 6061 aluminum cell with tangential interfaces at $y_C = 1.500$ in. (38.100 mm) on both surfaces. What peak tensile contact stresses are developed at each interface if the axial preload is 20.000 lb (88.964 N)?

From Table B6: $E_G = 1.504 \times 10^7$ lb/in.² (1.037×10⁵ MPa) and $v_G = 0.278$ From Table B12: $E_M = 9.900 \times 10^6$ lb/in.² (6.820×10⁴ MPa) and $v_G = 0.332$

As previously defined:
$$p = \frac{20.000}{(2\pi)(1.500)} = 2.122 \text{ lb/in.}(0.372 \text{ N/mm})$$

From Eq. (13.5): $K_2 = \left[\frac{(1-0.278^2)}{1.504 \times 10^7}\right] + \left[\frac{(1-0.3322)}{9.9 \times 10^6}\right]$
 $= 6.135 \times 10^{-8} + 8.988 \times 10^{-8} = 1.512 \times 10^{-7} \text{ in.}^2/\text{lb}(2.193 \times 10^{-5} \text{ MPa}^{-1})$
(a) For the tangential interface at R_1 , $K_{1,\text{TAN}} = \frac{1}{D_1} = \frac{1}{(2)(18.000)} = 0.0278 \text{ in.}$
From Eq. (13.13): $S_{C,\text{TAN}} = 0.798 \left[\frac{(0.0278)(2.122)}{1.512 \times 10^{-7}}\right]^{\frac{1}{2}} = 498.4 \text{ lb/in.}^2 (3.44 \text{ MPa})$
From Eq. (13.1): $S_{T,\text{TAN}} = \frac{\left[1-(2)(0.278)\right](498.4)}{3} = 73.8 \text{ lb/in.}^2 (0.51 \text{ MPa})$
(b) For the tangential interface at R_2 , $K_{1,\text{TAN}} = \frac{1}{D_1} = \frac{1}{(2)(72.000)} = 0.0069/\text{in.}$
From Eq. (13.13): $S_{C,\text{TAN}} = 0.798 \left[\frac{(0.0069)(2.122)}{1.512 \times 10^{-7}}\right]^{\frac{1}{2}} = 249.1 \text{ lb/in.}^2 (1.72 \text{ MPa})$
From Eq. (13.1): $S_{T,\text{TAN}} = \frac{\left[1-(2)(0.278)\right](249.1)}{3} = 36.9 \text{ lb/in.}^2 (0.25 \text{ MPa})$
These stresses are far below the 1000 lb/in.² (6.9 MPa) tolerance so are acceptable.

If we compare the results from Example 13.4 to those from Example 13.3, we see the advantage of the tangential interface over the sharp corner interface from the viewpoint of peak contact stress. The simple change in the mechanical interface is well justified because it reduces the unacceptable stress of the sharp corner interface to a very acceptable stress and adds only a very small amount to the cost of the hardware.

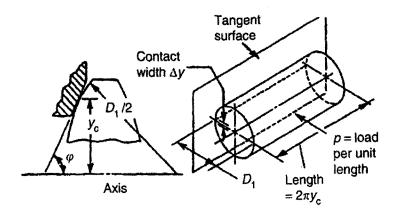


Figure 13.15 (a) Sectional view through the interface between a tangential (conical) metal surface and a convex optical surface. (b) An analytical model of that interface.

13.5.3 Stress with a toroidal interface

In Section 3.8.3, toroidal (or donut-shaped) mechanical surfaces contacting spherical lens surfaces were described. Figure 13.13 again applies and K_1 values for interfaces on convex or concave surfaces are given by Eq. (13.4a) with D_1 set equal to twice the optical surface radius and D_2 set equal to twice the sectional radius (R_T) of the toroid. Toroidal mechanical surfaces contacting optical surfaces are usually convex. The limiting case for small values of R_T would be equivalent to a sharp corner. If R_T increases to infinity and the lens surface is convex, the limiting case is the same as with a tangential interface. Only a convex toroid can contact a concave lens surface. The limiting case for a large R_T is then a radius equaling that of the optical surface. This is equivalent to a spherical interface (see Section 3.8.4).

Example 13.5 shows peak contact stress calculations similar to those in Examples 13.3 and 13.4, but now with toroidal interfaces on each side of the lens. The lens analyzed is meniscus shaped. For reasons given in Section 13.5.6, we assume $R_T = 10R_1$ at the convex surface, and $R_T = 0.5R_2$ at the concave surface.

Note that the peak tensile contact stresses at both surfaces in this example using toroidal interfaces are reduced significantly from those with a sharp corner interface, and that, for the convex surface (R_1) , they are almost the same as those with the tangential interface. The average stresses also would be very nearly the same as with the tangential interface. Since a toroid also works well with a concave optical surface, it is seen to be a favorable type of interface for the optomechanical interface in any surface contact mounting design.

Example 13.5: Peak contact stress in a lens with toroidal mechanical interfaces. (For design and analysis, use File 13.5 of the CD-ROM.)

Consider a meniscus germanium lens with the following dimensions: $D_G = 3.100$ in. (78.7450 mm), $R_1 = 18.000$ in. (457.200 mm), and $R_2 = 72.000$ in. (1828.800 mm). The lens is mounted in a 6061 aluminum cell with toroidal interfaces at $y_C = 1.500$ in. (38.100 mm) on both surfaces. Let R_T at $R_1 = (10)(R_1) = 180.000$ in. (4572.000 mm) and R_T at $R_2 = (0.5)(R_2) = 36.000$ in. (914.400 mm). What peak tensile contact stresses are developed at each interface if the axial preload is 20.000 lb (88.964 N)?

 $E_G = 1.504 \times 10^7$ lb/in.² (1.037×10⁵ MPa) and $v_G = 0.278$. From Table B6: $E_M = 9.900 \times 10^6 \text{ lb/in.}^2 (6.820 \times 10^4 \text{ MPa}) \text{ and } v_M = 0.332.$ From Table B12: $p = \frac{20.000}{(2\pi)(1.500)} = 2.122 \text{ lb/in.}(3.72 \text{ N/mm}).$ As previously defined: $K_2 = \left\lceil \frac{\left(1 - 0.278^2\right)}{1.504 \times 10^7} \right\rceil + \left\lceil \frac{\left(1 - 0.332^2\right)}{9.9 \times 10^6} \right\rceil$ From Eq. (13.5): $= 6.135 \times 10^{-8} + 8.988 \times 10^{-8}$ $= 1.512 \times 10^{-7} \text{ in.}^{2}/\text{lb} (2.193 \times 10^{-5} \text{ MPa}^{-1})$ (a) For the toroidal interface at R_1 , $D_1 = (2)(18.000) = 36.000$ in. (914.400 mm). $D_2 = (2)(180.000) = 360.000$ in. (9140.000 mm). From Eq. (13.4a): $K_{1 \text{ TOR}} = \frac{(36.000 + 360.000)}{(36.000)(360.000)} = 0.0306 / \text{ in.}$ From Eq. (13.13): $S_{C \text{ TOR}} = 0.798 \left[\frac{(0.0306)(2.122)}{1.512 \times 10^{-7}} \right]^{\frac{1}{2}}$ $= 526.4 \text{ lb/in.}^2 (3.63 \text{MPa}).$ From Eq. (13.1): $S_{T \text{ TOR}} = \frac{\left[1 - (2)(0.278)\right](526.4)}{2} = 77.9 \text{ lb/in.}^2 (0.54 \text{ MPa}).$ (b) For the toroidal interface at R_2 , $D_1 = (2)(72.000) = 144.000$ in. (3657.600 mm). $D_2 = (2)(36.000) = 72.000$ in. (1828.800 mm). From Eq. (13.4a): $K_{1 \text{ TOR}} = \frac{(144.000 - 72.000)}{(144.000)(72.000)} = 0.0069 / \text{in.}.$ From Eq. (13.13): $S_{C \text{ TOR}} = 0.798 \left[\frac{(0.0069)(2.122)}{1.512 \times 10^{-7}} \right]^{\frac{1}{2}}$ $= 248.3 \text{ lb/in.}^2 (1.71 \text{ MPa}).$ From Eq. (13.1): $S_{T \text{ TOR}} = \frac{\left[1 - (2)(0.278)\right](248.3)}{3} = 36.7 \text{ lb/in.}^2 (0.25 \text{ MPa}).$

These stresses are far below the 1000 lb/in.² (6.9 MPa) tolerance so are acceptable.

13.5.4 Stress with a spherical interface

Spherical mechanical contact on a convex or concave lens surface (discussed in Section 3.8.4) distributes axial preloads over large annular areas and hence can be nearly stress free. If the surfaces match closely (i.e., within a few wavelengths of light) in radius, the contact stress equals the total preload divided by the annular area of contact. Since the area is relatively large, the stress is small and may be ignored. If the surfaces do not match closely, the contact can degenerate into a narrow annular area or even a line (i.e., a sharp corner interface). Either of these alternatives would be unfavorable because of the high potential for stress generation. Because other shapes of interfaces are available, easily created, and less expensive, the spherical interface is not often used.

13.5.5 Stress with a flat bevel interface

In Section 3.8.5 we considered flat-bevel interfaces. If used as a mechanical reference surface and precisely perpendicular to the optic axis of the lens, the bevel can be in close contact with the mechanical surface. As in the case of the spherical interface, the contact stresses that are due to axial preloads (total preload/contact area) are inherently small because the area of contact is large. Therefore, these stresses may be ignored. However, if the contacting surfaces are not truly flat and parallel, the area of contact will decrease and the stress will increase. In the limit, line contact (i.e., a sharp-corner interface) occurs. This could lead to high localized stress.

13.5.6 Parametric comparisons of interface types

Figure 13.16(a) shows the variation of axial tensile contact stress with radius of the mechanical surface contacting the lens surface for a particular design having a convex lens surface with R = 10.000 in. (254.000 mm), a lens diameter of 1.500 in. (38.100 mm), and a linear preload *p* of 1.000 lb/in. (0.175 *N*/mm) on an annular area near the lens rim. The lens is made of BK7 glass and the mount is made of 6061 aluminum. Both the stress and the mechanical surface radius are plotted logarithmically to cover large ranges of variability. At the left is the short radius characteristic of the sharp corner interface (at the dashed vertical line) while at the right; the tangential interface case is approached asymptotically. Between these extremes are an infinite number of toroidal interface designs. The small circle indicates a "recommended" minimum toroidal radius ($R_T = 10R$) for which the stress is within 5% of the value that would exist with a tangential interface. See Yoder.²³

Figure 13.16(b) shows a similar relationship for a concave lens surface example. All other parameters are the same as in view (a). The dashed vertical line at the left again represents the sharp corner case. As the toroidal corner radius increases toward the matching radius limit (which is equivalent to a spherical interface), the stress decreases. The circle represents an arbitrarily chosen "recommended" minimum toroidal radius of 0.5*R* at which the stress will approximate the value that would prevail at the same preload on a convex surface of the same radius using a $R_T = 10R$ toroidal interface. The relationships indicated in these figures demonstrate conclusively that the axial contact stress is always significantly higher with a sharp corner interface than with any other type of interface.

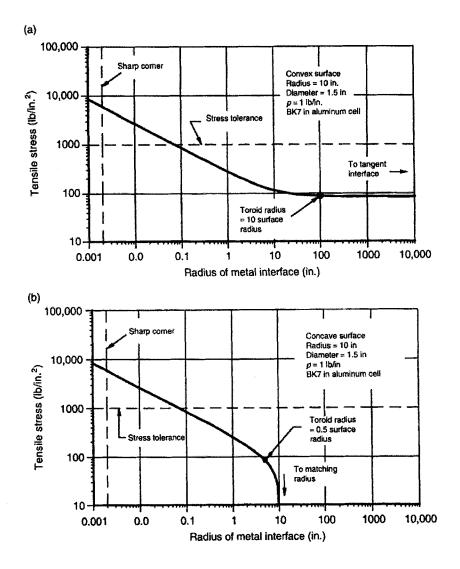


Figure 13.16 Variation of tensile contact stress as a function of the sectional radius of the mechanical contacting surface for (a) a typical preloaded convex lens surface and (b) a typical preloaded concave lens surface. The design dimensions are as indicated. (From Yoder.²⁴ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc, Reprinted with permission.)

Figure 13.17(a) shows what happens to the axial contact stress for the same lens as in the last figure if the linear preload p (and, hence, the total preload) is changed by factors of 10 from 0.001 to 10 lb/in. $(1.75 \times 10^{-4} \text{ to } 1.75 \text{ N/mm})$. View (b) shows a similar relationship for a concave surface with all other parameters unchanged. In general, if the total axial preload P on an optic with any type of interface and any surface radius increases from P_1 to P_2 while all other parameters remain fixed, the resulting axial contact stress changes by a factor of $(P_2/P_1)^{1/2}$. A tenfold increase in preload therefore increases the stress by a factor of $10^{1/2} = 3.162$.

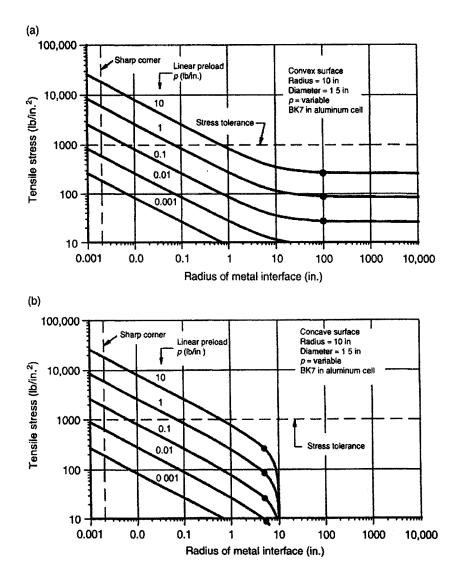


Figure 13.17 Variation of tensile contact stress as functions of the sectional radius of the mechanical contacting surface and linear preload for (a) a typical preloaded convex lens surface and (b) a typical preloaded concave lens surface. The design dimensions are as indicated. (From Yoder.²⁴ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc, Reprinted with permission.)

Figures 13.18(a) and (b) show how the axial tensile contact stress for the same example as in Fig. 13.17 varies as the surface radius of the lens is changed by successive factors of 10 for convex and concave surface cases, respectively. The preload is held constant [p = 1.0lb/in. (0.175 N/mm)]. The stress is seen to be independent of the surface radius or its algebraic sign (i.e., convex or concave) for a sharp corner interface (vertical

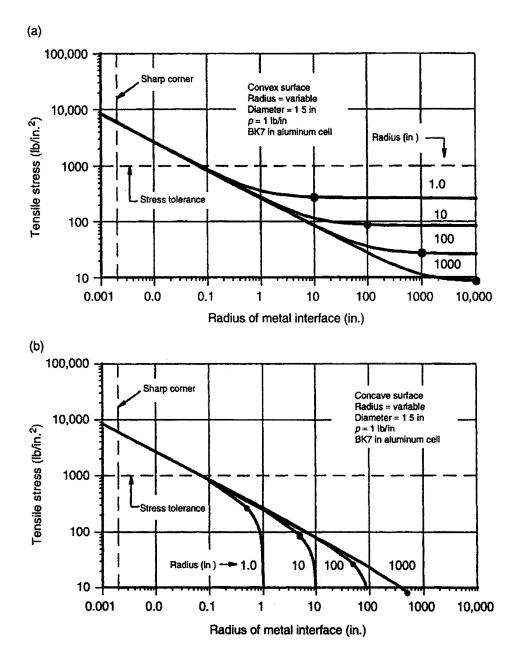


Figure 13.18 Variation of tensile contact stress as functions of the sectional radius of the mechanical contacting surface and lens surface radius for (a) a typical preloaded convex lens surface and (b) a typical preloaded concave lens surface. The design dimensions are as indicated. (From Yoder.²⁴ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc, Reprinted with permission.)

dashed line at the left side of each graph). The greatest changes occur for long radii toroids on either type of surface. The limits are the tangential interface and the matching radii interface for the convex surface and the concave surface cases, respectively. Once again, the toroids indicated by the circles on each curve (toroid radius = 10*R* for convex surfaces and 0.5R for concave surfaces) are the recommended minimum sectional radii for the mechanical component. Use of toroids with longer radii than these recommended minimums would, of course, cause the contact stresses to decrease. The positive tolerance on R_T can therefore be quite loose. The variation of stress with decreasing toroid radius causes an increase in stress. Because the stress for R_T values near the circles on the graphs are generally very low, a small increase may be acceptable. The negative tolerance on R_T can, in most cases, also be loose. A $\pm 100\%$ change in R_T may well be acceptable. This simplifies inspection of such parts.

Because the tangential (conical) interface is slightly easier and, therefore, less costly to manufacture than a toroid, it is recommended that tangential interfaces be used on all convex lens surfaces. Further, we recommend that toroidal interfaces of radius R_T approximately 0.5*R* be used on all concave surfaces of radius *R*. Both these interface shapes will significantly reduce axial contact stresses as compared to those resulting from use of sharp corner interfaces.

If the surface radius changes from R_i to R_j with all other parameters unchanged, the corresponding contact stress with long radius toroidal interfaces changes by $(R_i / R_j)^{1/2}$. Hence, for the 10:1 step increases in surface radius depicted in Figs. 13.17(a) and (b), the stress decreases by a factor of $(1/10)^{1/2} = 0.316$ between steps.

13.6 Bending Effects in Asymmetrically Clamped Optics

When mounting a circular optic (lens, window, or small mirror) with axial force from a retainer or flange holding that optic to a shoulder or another constraint in the mount, the force and the constraint should be directly opposite (i.e., at the same height from the axis on both sides). If this is not the case, a bending moment is exerted on the optic around the contact zone. This moment tends to deform the optic so that one surface becomes more convex and the other surface becomes more concave as illustrated schematically in Fig. 13.19. These deformations of the optical surfaces may adversely affect the performance of the optic. The same general effect occurs in noncircular optics clamped asymmetrically by springs or flanges.

When bent, the surface that becomes more convex is placed in tension. The other surface is compressed. Since optical materials break much more easily in tension than in compression (especially if the surface is scratched or has subsurface damage) catastrophic failure may occur if the moment is large. The "rule-of-thumb" tolerance for tensile stress given earlier [1000 lb/in.² (68.9 MPa)] applies here.

13.6.1 Bending stress in the optic

Bayar²⁵ indicated that an analytical model that uses an equation from Roark¹ based on a thin plane parallel plate (as shown in Fig. 13.19) applies also to simple lenses with long radii. We here extend the analogy to include circular windows and small circular unperforated mirrors. The degree of approximation depends, in part, on the curvatures of the surfaces.

Greater curvature tends to change the accuracy of the calculation because the component is more stiff than the flat plate.

The tensile stress S_T in a surface made more convex by bending is given approximately by these equations:

$$S_T = \frac{K_6 K_7}{t_E^2},\tag{13.14}$$

$$K_6 = \frac{3P}{2\pi m},$$
 (13.15)

$$K_{7} = 0.5(m-1) + (m+1)\ln\left(\frac{y_{2}}{y_{1}}\right) - (m-1)\left(\frac{y_{1}^{2}}{2y_{2}^{2}}\right), \qquad (13.16)$$

where P is the total axial preload, m is (1/Poisson's ratio) for the glass, t_E is the edge or axial thickness of the optic (whichever is the smaller), y_1 is the smaller contact height, and y_2 is the larger contact height. To decrease the probability of breakage from this bending moment, the contact heights should be made as equal as possible. Increasing the optic's thickness also tends to reduce this danger. The following example illustrates how to estimate the bending stress in the optic.

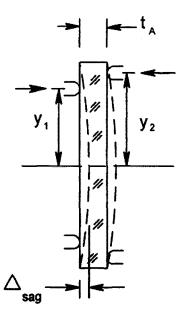


Figure 13.19 Geometry for estimating the effects of a bending moment from preload and constraining force applied at different heights on an optic.

Example 13.6: Bending stress in an optic clamped asymmetrically. (For design and analysis, use File 13.6 of the CD-ROM.)

A 20.000 in. (508.000 mm) diameter fused silica mirror of plane parallel shape and thickness 2.500 in. (63.500 mm) is contacted on one side by a toroidal shoulder at y_1 = 9.500 in. (241.300 mm) and on the opposite side by a toroidal clamping flange at y_2 = 9.880 in. (250.952 mm). The applied preload at low temperature is 2000 lb (8.9×10³ N). What is the bending stress?

From Table B5, Poisson's ratio (v_G) for fused silica is 0.17, so m = 1/0.17 = 5.882.

From Eq. (13.15):
$$K_6 = \frac{(3)(2000)}{(2)(\pi)(5.882)} = 162.348 \text{ lb.}$$

From Eq. (13.16): $K_7 = \left[0.5(5.882 - 1)\right] + \left[(5.882 + 1)\ln\left(\frac{9.880}{9.500}\right)\right]$
 $-\left[\frac{(5.882 - 1)(9.500^2)}{(2)(9.880^2)}\right] = 2.441 + 0.270 - 2.257 = 0.454.$
From Eq. (13.14): $S_T = \frac{(162.348)(0.454)}{2.500^2} = 11.7 \text{ lb/in.}^2 (0.08 \text{ MPa}).$
This stress is far below the suggested tolerance of 1000 lb/in.² (6.900 MPa) and therefore not a problem.

13.6.2 Change in surface sagittal depth of a bent optic

The following equation for the change in sagittal depth Δ_{SAG} at the center of a plane parallel plate such as that shown in Fig. 13.19 resulting from the bending moment exerted by unsymmetrical annular mounting interfaces was given by Roark:¹

$$\Delta_{\rm SAG} = \frac{K_8 K_9}{t_E^3},\tag{13.17}$$

$$K_8 = 3P \frac{\left(m^2 - 1\right)}{2\pi E_G m^2},$$
(13.18)

$$K_{9} = \frac{\left[(3m+1)y_{2}^{2} - (m-1)y_{1}^{2} \right]}{(2)(m+1)}.$$
(13.19)

All terms are as defined earlier. To see if this surface deformation is acceptable, it can be compared with the tolerance (such as $\lambda/2$ or $\lambda/20$) on surface figure error for the optic. Example 13.7 illustrates the use of these equations.

Example 13.7: Surface deflection in a plane parallel optic clamped asymmetrically. (For design and analysis, use File 13.7 of the CD-ROM.)

A 20.000 in. (508.000 mm) diameter fused silica mirror of plane parallel shape and thickness 2.500 in. (63.500 mm) is contacted on one side by a toroidal shoulder at $y_1 = 9.500$ in. (241.300 mm) and on the opposite side by a toroidal clamping flange at $y_2 = 9.880$ in. (250.952 mm). The applied preload at low temperature is 2000 lb (8.9×10³ N). What is the deflection of the surface at its center in in. (mm) and waves at 633 nm?

From Table B5, $E_G = 1.060 \times 10^7$ lb/in.² (7.300 × 10⁴ MPa) and $v_G = 0.170$. Thus, m = 1/0.170 = 5.882.

From Eqs. (13.18), (13.19), and (13.17):

$$K_{8} = \frac{(3)(2000)(5.882^{2} - 1)}{(2\pi)(1.06 \times 10^{7})(5.882^{2})} = 8.748 \times 10^{-5} \text{ in.}^{2}.$$

$$K_{9} = \frac{\left[(3)(5.882) + 1\right](9.880^{2}) - \left[(5.882 - 1)(9.500^{2})\right]}{(2)(5.882 + 1)}$$

$$-\left\{9.500^{2}\left[\ln\left(\frac{9.880}{9.500}\right) + 1\right]\right\} = 6.437 \text{ in.}^{2}.$$

$$\Delta_{\text{SAG}} = \frac{\left(8.748 \times 10^{-5}\right)(6.437)}{2.500^{3}} = 3.604 \times 10^{-5} \text{ in.}(9.115 \times 10^{-4} \text{ mm})$$

$$= 1.45\lambda \text{ for } \lambda = 0.633 \mu\text{m}.$$

The mirror mounting design of Example 13.7 is probably unsatisfactory for any practical application even though the stress level (from Example 13.6) is quite low. The design could be improved significantly by making y_1 and y_2 equal.

13.7 References

- 1. Roark, R.J., Formulas for Stress and Strain, 3rd ed., McGraw-Hill, New York, 1954.
- Timoshenko, S.P. & Goodier, J.N., *Theory of Elasticity*, 3rd ed., McGraw-Hill, New York, 1970.
- 3. Weibull, W.A., "A statistical distribution function of wide applicability," J. Appl. Mech. 13, 1951:293.
- 4. Wiederhorn, S.M., "Influence of water vapor on crack propagation in soda-lime glass," J. Am. Ceram. Soc. 50, 1967:407.
- 5. Wiederhorn, S.M., Freiman, S.W., Fuller, E.R., Jr., and Simmons, C.J., "Effects of water and other dielectrics on crack growth, *J. Mater. Sci.* 17, 1982:3460.

- 6. Vukobratovich, D., "Optomechanical Design," Chapter 3 in *The Infrared and Electro-Optical S ystems Ha ndbook* 4, ERIM, Ann Arbor and SPIE Press, Bellingham, 1993.
- 7. Pepi, J.W., "Failsafe design of an all BK7 glass aircraft window," *Proceedings of* SPIE 2286, 1994:431.
- Fuller, E.R., Jr., Freiman, S.W., Quin, J.B., Quinn, G.D., and Carter, W.C., "Fracture mechanics approach to the design of aircraft windows: a case study," *Proceedings of* SPIE 2286, 1994:419.
- 9. Harris, D.C., *Materials for Infrared Windows and Domes*, SPIE Press, Bellingham, 1999.
- Doyle, K.B. and Kahan, M., "Design strength of optical glass," *Proceedings of SPIE* 5176, 2003:14.
- 11. Adler, W.F., and Mihora, D.J., "Biaxial flexure testing: analysis and experimental results," *Fracture Mechanics of Ceramics* 10, Plenum, New York, 1992.
- 12. Freiman, S., Stress Corr osion Cr acking, ASM International, Materials Park, OH, 1992.
- 13. MIL-O-13830A, Optical C omponents f or Fi re C ontrol I nstruments: G eneral Specification Governing the Manufacture, Assembly and Inspection of, U.S. Army, 1975.
- 14. Doyle, K.B., private communication, 2008.
- 15. Stoll, R., Forman, P.F., and Edelman, J., "The effect of different grinding procedures on the strength of scratched and unscratched fused silica," *Proceedings of Symposium on the St rength of Gl ass and Ways t oImprove lit*, Union Scientifique Continentale du Verr, Florence, 1961.
- 16. Shand, E.B., Glass Engineering Handbook, 2nd ed., McGraw-Hill, New York, 1958.
- 17. Crompton, D., private communication, 2004.
- 18. Young, W.C., *Roark's Formulas for Stress and Strain*, 6th ed., McGraw-Hill, New York, 1989.
- 19. Kimmel, R.K. and Parks, R.E., ISO 10110 Optics and Optical Instruments— Preparation of Drawings for Optical Elements and Systems, 2nd ed., Optical Society of America, Washington, 2004.
- 20. Sawyer, K.A., "Contact stresses and their optical effects in biconvex optical elements," *Proceedings of SPIE* 2542, 1995:58.
- 21. Yoder, P.R., Jr., "Improved semikinematic mounting for prisms," *Proceedings of* SPIE 4771, 2002:173.
- 22. Delgado, R.F. and Hallinhan, M., "Mounting of optical elements," Opt. Eng. 14; 1975:S-11.
- 23. Yoder, P.R., Jr., "Axial stresses with toroidal lens-to-mount interfaces," *Proceedings* of SPIE 1533, 1991:2.
- 24. Yoder, P.R., Jr., Opto-Mechanical Systems Design, 3rd. ed., CRC Press, Boca Raton, 2005.
- 25. Bayar, M., "Lens barrel optomechanical design principles," Opt. Eng. 20, 1981:181.

CHAPTER 14 Effects of Temperature Changes

Temperature changes cause a myriad of corresponding changes in optical components and systems. These include changes in surface radii, air spaces and lens thicknesses, in the refractive indices of optical materials and of the surrounding air, and in the physical dimensions of structural members. These effects tend to defocus and/or misalign the system. Passive and active techniques for athermalizing optical instruments to reduce these effects are considered here. Dimensional changes of optical and mechanical parts forming assemblies can cause changes in clamping forces (preloads). These changes affect contact stresses at optomechanical interfaces. Optical component misalignment caused by loss of contact with the mount at higher temperatures, as well as axial and radial stress buildup at lower temperatures, are considered. Although these problems may be serious if they are not attended to, most can be eliminated or drastically reduced in magnitude by careful optomechanical design. We briefly consider how temperature gradients, axial and/or radial, can affect the system performance. Finally, shear stresses in bonded joints caused by temperature changes are discussed.

14.1 Athermalization Techniques for Reflective Systems

Athermalization is the process of stabilizing an instrument's optical performance by designing the optics, mounts, and structures to compensate for the effects of temperature changes. In this section, we limit our discussion to axial defocus effects that can be approached passively or actively by choices of configuration, materials, and dimensions.

14.1.1 Same material designs

Reflective systems with all optical and mechanical components made from the same material offer an advantage over systems that include refractive optics. An example is the telescope for the Infrared Astronomical Satellite (IRAS).^{1,2} This was a cryogenically cooled Ritchey-Chretien system orbited by NASA in 1983. All structural and optical components of the telescope were beryllium. This telescope was described in Section 10.3 in the context of flexure mountings for metal mirrors. Figure 14.1 shows the optomechanical system schematically.^{*} Because all parts of the telescope that affect imagery have the same CTE, changes from the fabrication and assembly temperature on Earth to the cryogenic temperature in space will change all component and spacing dimensions equally. Such a system is called "athermal" because temperature changes do not affect focus or image quality. Small changes do occur in scale of the image. The all-aluminum telescope shown in Fig. 10.33 offers similar athermal characteristics.³

In the more common configuration for reflective systems, such as the Cassegrain or Gregorian telescope with two mirrors separated by an axial distance, the mirrors are typically made of ULE or Zerodur with low CTEs and the structure is made of a higher CTE material such as aluminum. A change in temperature would generally cause the focus of such a system to move inward or outward. If the mirrors and structure have different

^{*} This figure duplicates Figs. 4.35 and 10.24.

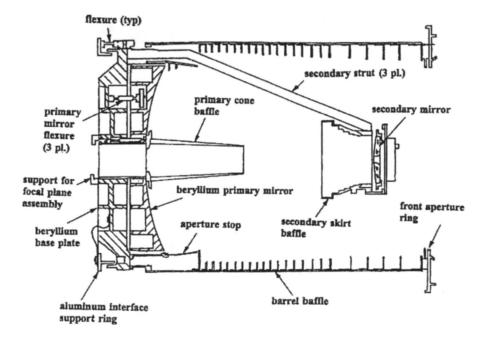


Figure 14.1 Optomechanical configuration of the 24-in. (61-cm) aperture, allberyllium telescope for the IRAS telescope. (Adapted from Schreibman and Young.¹)

CTEs, more flexibility is available and an athermal design is theoretically possible. This is especially true if the structure is made of different lengths of different materials.

In many cases, materials are chosen for reasons other than thermal ones (such as manufacturability, cost, and density) so other means must be employed to reduce the effects of temperature changes. A possible means is active control of the location(s) of one or more mirrors in the system. Temperature distribution within the system might be measured and motor driven mechanisms used to drive the mirror separation and/or final image distance to optimum values. A better, but more complex, means would be to sense focus or quality of the image and actively control mirror location(s) to optimize system performance. Both techniques require an expenditure of energy that may not be available. Passive athermalization then might be an attractive approach.

14.1.2 Metering rods and trusses

The 12.5-in. (31.1-cm) aperture Cassegrain telescope of the geostationary operational environmental satellite (GOES) uses "metering rods" to passively control the axial air space between the two mirrors. As shown in Fig. 14.2, six Invar tubes connect the spider supporting the secondary mirror to the cell holding the primary mirror. The primary mount and the secondary spider are made of aluminum. The mount for the secondary mirror is described in Section 9.1. The instrument design is axially athermal because the lengths of the dissimilar structural metals have been chosen so that the axial separation between the

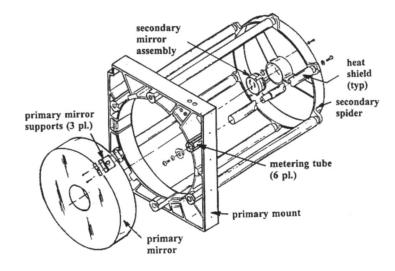


Figure 14.2 Diagram of the passively athermalized structure of the GOES telescope. (Adapted from Zermehly and Hookman.⁵)

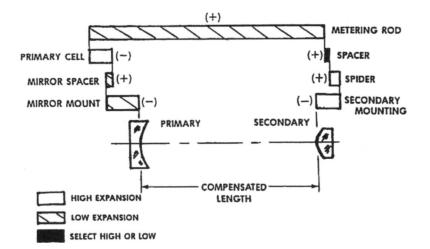


Figure 14.3 Model of the passively compensated structure of the GOES Telescope. (Adapted from Zermehly and Hookman.⁵)

mirrors remains constant when the temperature ranges from $1^{\circ}C$ (34°F) to 54°C (129°F) as the satellite orbits through the Earth's shadow.^{4,5}

The way this is accomplished is shown schematically in Fig. 14.3. The materials involved have low or high CTEs, as indicated by the legend. The mirror vertices are located at the points indicated in the sketch. The plus and minus signs indicate how an increase in

temperature affects the central air space between these mirrors. The direction of an individual change is determined by which end of the component is attached to its neighboring components. The algebraic summation of contributions, each consisting of individual component length times CTE times temperature change, for the various structural members defines the mirror separation. The material for one spacer in the secondary mount is selected at assembly to accommodate minor variations in component parameters. The net result is that the air space automatically remains fixed throughout the temperature excursion.

A uniform distribution of temperature is assumed in such a design. To help regulate temperature, aluminum heat shields painted black on the outside surfaces to maximize thermal emissivity and gold plated on the inside surfaces to minimize emissivity are placed over the major mechanical components, including the metering tubes. These shields are not structural members so they do not enter directly into the temperature compensation mechanism.

In large reflective systems, athermalized trusses are often used to maintain the separations of mirrors. An example is the truss used to support the secondary mirror of the Hubble Telescope. That truss was constructed of 48 tubes and three rings made of graphite fiber reinforced epoxy. The tubes were 2.13-m (84-in.) long and 6.17-cm (2.43-in.) diameter. They had the required $0.25 \times 10^{6/9}$ F CTE within the specified tolerance of $\pm 0.10 \times 10^{6/9}$ F.⁶ McCarthy and Facey⁷ described how the CTEs of the as-manufactured tubes were measured and sorted into different groups to be used at specific locations in the truss. See Fig. 14.4. For example, ones with higher CTEs were placed in the bay adjacent to the primary mirror where the operational temperature changes would be the least.

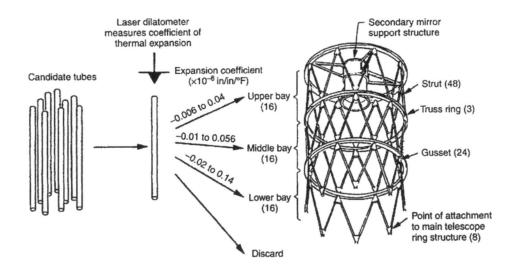


Figure 14.4 Selection of tubes based on measured CTEs for use in the Hubble Space Telescope metering truss structure. (From McCarthy and Facey.⁷)

14.2 Athermalization Techniques for Refractive Systems

Refractive and catadioptric systems present more complex athermalization problems than all-reflecting systems because of refractive index variations that occur along with dimensional variations within structural and transmissive materials as the temperature changes. When significant temperature effects are expected, a common approach to designing a refractive optomechanical system is to create a lens design with minimal adverse effects from those changes and then to design a mechanical structure that compensates for the residual thermal effects. The discussion here is a first-order approximation to this athermalization design task. It shows an approach and not a detailed design procedure.

Key parameters here are the CTEs of all materials and the optical material's refractive indices n_G , as well as the rates of change of those indices with temperature. Unless the surrounding medium is a vacuum, the variation with temperature of the refractive index of that medium (usually air) must be considered. To separate these refractive variations, the absolute index for glass $n_{G, ABS}$ is obtained from the $n_{G, REL}$ value relative to air (as listed in glass catalogs at a given temperature and wavelength) using the following equation:

$$n_{\rm GABS} = (n_{\rm GREL})(n_{\rm AIR}), \qquad (14.1)$$

where n_{AIR} at 15°C is calculated using Eq. 14.2, which is from Edlen:⁸

$$(n_{AIR15}) \times 10^8 = 6432.8 + \left[\frac{2949.810}{146 - \left(\frac{1}{\lambda}\right)^2}\right] + \left[\frac{25,540}{41 - \left(\frac{1}{\lambda}\right)^2}\right].$$
 (14.2)

Here, λ is in micrometers.

The index of air varies with temperature at a rate obtained from the following equation for n_{AIR} from Penndorf:⁹

$$\frac{dn_{\text{AIR}}}{dT} = \frac{(-0.003861)(n_{\text{AIR}\ 15} - 1)}{(1 + 0.00366T)^2},\tag{14.3}$$

where T is expressed in degrees Celsius. At 20°C, dn_{AIR}/dT and $(n_{AIR} - 1)$ have the values at selected wavelengths shown in Table 14.1.

Table 14.1 Values for dn_{AIR}/dT and $(n_{AIR} - 1)$ for selected λ at 20°C.

Wavelength (nm)	$dn_{\rm AIR}/dT$ (°C ⁻¹)	$(n_{AIR}-1)$
400	-9.478×10 ⁻⁷	2.780×10 ⁻⁴
550	-9.313×10 ⁻⁷	2.732×10 ⁻⁴
700	-9.245×10 ⁻⁷	2.712×10 ⁻⁴
850	-9.211×10 ⁻⁷	2.701×10 ⁻⁴
1000	-9.190×10^{-7}	2.696×10 ⁻⁴

Jamieson¹⁰ has defined the following expression for the change in focal length f at a given wavelength and temperature of a single element thin lens with a change in temperature ΔT :

$$\Delta f = -\delta_G f \Delta T. \tag{14.4}$$

In this equation, δ_G is a glass "coefficient of thermal defocus" given by:

$$\delta_G = \left[\frac{\beta_G}{(n_{GABS} - 1)}\right] - \alpha_G.$$
(14.5)

Note that β_G is the same as dn/dT for the glass and can be obtained from glass catalogs.

Equation (14.4) has the same form as the temperature variation of a length L of a material with a CTE of α , which is $\Delta L = \alpha L \Delta T$. The parameter δ_G depends only on physical properties and wavelength. Some authors refer to it as the "thermo-optic coefficient" for the glass. The value of α_G is positive for all refracting materials. For optical glasses, it ranges from $\sim 3.2 \times 10^{-5}$ to $\sim 2.2 \times 10^{-5}$. Those glasses with small δ_G values are those for which the increase in focal length, due to rising temperature and the resultant expansion of surface radii, is nearly balanced by a corresponding decrease because of a reduced index of refraction. The δ_G values for optical plastics and infrared materials are more extreme than those of optical glasses. Jamieson¹⁰ listed δ_G for 185 Schott glasses, 14 infrared crystals, 4 plastics, and 4 index-matching liquids.

Consider a thin singlet lens having a positive value for δ_G and focal length *f* mounted as shown in Fig. 14.5 in a simple (uncompensated) barrel made of a metal with a CTE = α_M and length L = f. A change in temperature $+\Delta T$ will lengthen the barrel by $\alpha_M L\Delta T$. At the same time, the lens focal length will lengthen by $\delta_G f \Delta T$. If the materials could be chosen so $\alpha_M = \delta_G$, the system would be athermal and the image would remain at the end of the barrel at all temperatures. If $\alpha_M \neq \delta_G$, temperature changes will cause defocus. Choosing materials for this system that have nearly the same CTEs does not necessarily make them athermal.

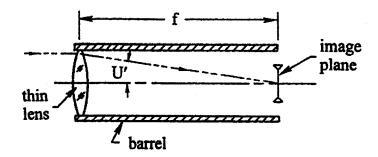


Figure 14.5 Schematic of a simple thermally uncompensated lens and mount system.

The defocus that occurs in a simple uncompensated thin lens system can be estimated as follows. Assume a thin BK7 lens with f = 100 mm (3.937 in.) is mounted in a 6061 aluminum barrel of length 100 mm (3.937 in.) configured as shown in Fig. 14.5. The image is then at the end of the barrel. Let the temperature change by +40°C (+72°F). From Table B12, $\alpha_{A1} = 23.6 \times 10^{-6}$ /°C (13.1×10⁻⁶/°F). Then, $\Delta L_{A1} = (23.6 \times 10^{-6})(100)(40) = 0.0944$ mm (0.0037 in.). Assuming δ_G for BK7 glass = 2.41×10⁻⁶/°F (4.33×10⁻⁶/°C),¹⁰ then, from Eq. (14.4), $\Delta f = (4.33 \times 10^{-6})(100)(40) = 0.0174$ mm (0.0006 in.). The defocus of the image relative to the end of the barrel is 0.0944 – 0.0174 = 0.0770 mm (0.0030 in.). This defocus might well be significant in many applications.

14.2.1 Passive athermalization

One means for athermalizing real, i.e., thick, lens systems is to design the lenses to provide the required image quality and, through proper choice of glasses, make them as independent of temperature as possible. We then design a mount from multiple materials combining different CTEs so as to make the change in key dimensions of the mount with temperature change equal the change in back focal length (i.e., image distance) for that same ΔT . Structures based on the design principles of Fig. 14.6 can create positive or negative changes in overall length from specific lengths of different materials such as Invar, aluminum, titanium, stainless steel, composites (typically graphite epoxy), fiberglass, or plastics (such as Teflon, Nylon, or Delrin).

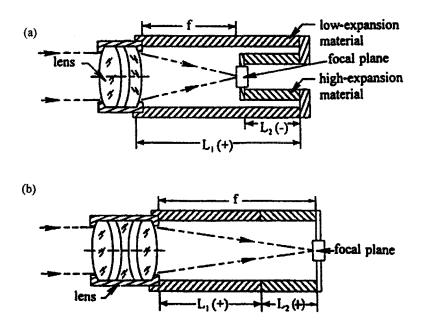


Figure 14.6 Schematics of lens mountings using two materials to athermalize the barrel with respect to the image distance of a lens. (a) Re-entrant case, (b) series case. (Adapted from Vukobratovich.¹¹)

Vukobratovich¹¹ gave equations (here slightly rewritten) for the design of these dualmaterial structures to thermally compensate an optical system with a coefficient of thermal defocus δ_G , CTEs α_1 and α_2 , and focal length *f*.:

$$\delta_{G} f = \alpha_{1} L_{1} + \alpha_{2} L_{2}, \qquad (14.6)$$

$$L_{1} = f - L_{2}, \qquad L_{2} = f \frac{(\alpha_{1} - \delta_{G})}{(\alpha_{1} - \alpha_{2})}.$$

where

and

All the geometric parameters are as defined in Fig. 14.6.

Concepts for passive mechanisms that have been used to varying degrees of success in providing component motions for athermalization purposes in infrared systems were listed by Povey.¹² The mechanisms provided a variety of members connecting a fixed lens to an axially moveable sensor. The CTE and length(s) of the connecting member(s) were selected to maintain focus over a range of temperatures. Included were (1) a solid rod or tube of metal or other material with a particular CTE, (2) two or more members with different CTEs and different lengths connected in series, (3) two or more members with different CTEs connected in opposition, i.e., a reentrant configuration, (4) a bipod arrangement of materials with different CTEs used in the legs and base, (5) a configuration of three (or more) concentric split rings wrapped around the optic cell, attached in series at their split ends, fixed at one end, and attached to a ring gear that drives a focus mechanism at the other end, (6) a cylinder filled with wax or fluid of selected CTE connected to a piston attached to the moveable element, and (7) a shape memory actuator. A concept for an active sensor also was mentioned by Povey in Ref. [12]. Mechanisms (2) and (3) are illustrated conceptually in Fig. 14.6.

Ford et al.¹³ gave a comprehensive description of an interesting thermal compensator mechanism used with each of a series of nine lenses in NASA's Multiangle Imaging Spectro-Radiometer (MISR). The science goals were to monitor global atmospheric particulates, cloud movements, surface BRDF,^{*} and vegetative changes on the day-lit side of Earth during a nominal six-year mission in polar orbit and in four wavelengths from the Terra Satellite, operational since 2000. One of these lenses is shown in Fig. 14.7. The mechanism for focus compensation that keeps the digital focal plane assembly in the plane of best focus under temperature changes is shown in Fig. 14.8. The flanges on each lens and its compensator are attached in series (see Fig. 14.9).

Optimization of the detector location relative to the image was achieved in this design with a set of concentric tubes made of different materials connected at alternate ends so their length variations with temperature add and subtract in a predictable manner.

BRDF means bidirectional reflectance distribution function.

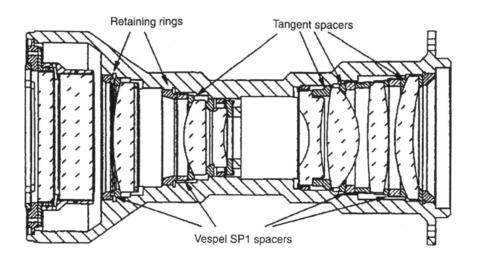


Figure 14.7 Sectional view through a typical MISR lens assembly. (From Ford et al.¹³ Reprinted by courtesy of NASA/JPL/Caltech.)

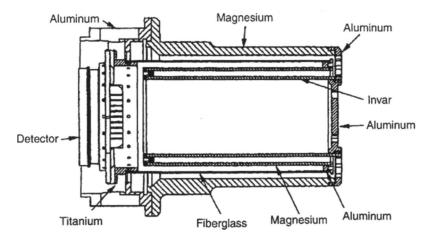


Figure 14.8 Schematic of the temperature compensator used with the MISR lenses. (From Ford et al.¹³ Reprinted by courtesy of NASA/JPL/Caltech.)

In this design, the materials used in the components tending to reduce the total length with increasing temperature were made of low CTE materials (Invar and fiberglass), while those tending to increase that length were made of high CTE materials (aluminum and magnesium). It was found that identical compensator designs would not provide complete compensation at all temperatures for all four types of lenses (EFLs of 59.3, 73.4, 95.3, and 123.8 mm), but that the performance degradation resulting from the use of one compromise design would be acceptable.

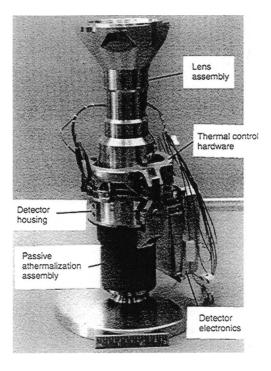


Figure 14.9 Photograph of an engineering model of a MISR camera. (From Ford et al.¹³ Reprinted by courtesy of NASA/JPL/Caltech.)

The detectors of the MISR cameras were cooled with thermoelectric coolers to $-5.0 \pm 0.1^{\circ}$ C. They were thermally insulated from the surrounding structure. The detector housings were gold coated to reduce emissivity, the cold structures were mounted on a thin fiberglass tube (selected for its low thermal conductivity), and the fiberglass tube was covered with low emissivity aluminized Mylar. The other tubes in the assembly were metallic to obtain high thermal conductivity.

It was recognized that thermal gradients between the camera and the lens assembly could cause the temperature compensator subsystem to correct the focus for the wrong temperature. Of particular concern was the joint between the lens housing and the camera assembly, which was connected through a spacer that did not have good thermal conductivity. Additionally, heat had to be removed from the thermal electric cooler and the detector preamplifier. To stabilize the temperature, special thermal control hardware (indicated in Fig. 14.9) was designed and added to the system. This hardware, made of highly conductive type 7073 aluminum alloy, bridged the joint between the lens and camera housings and clamped onto a conductive finger that removed heat from the electronics and the cooler. Soft pure aluminum shims were provided in all joints of this hardware to maximize conduction of heat through those joints. Heat was then conducted from the lens housing to structure of the instrument where it was radiated away.

Modern lens design programs, such as Code V, are capable of athermalization with little intervention by the designer. The programs include thermal modeling features and

stored thermal/mechanical properties of a variety of commonly used optical and mechanical materials (including mirror materials). The nominal design typically is known at a specific design temperature such as 20°C. The athermalization design process generally involves (1) calculation of the refractive index of air at the desired extreme high and low temperatures, (2) conversion of the glass catalog values for refractive indices relative to air into absolute values by multiplying them by the air index, (3) calculation of the glass refractive indices at the extreme temperatures using the dn/dT values from the manufacturer's data, (4) calculation of the surface radii at the extreme temperatures using the known CTEs of the optical materials, (5) calculation of the air spaces and component thicknesses at the extreme temperatures using the given CTEs of the mechanical and optical materials, (6) evaluation of the system's performance at the extreme temperatures and the best focus locations for those temperatures, (7) design of the mechanical structure and mechanisms as needed to adjust component spacings and/or bring the image to the proper location at each extreme temperature, and, finally, (8) assessment of the system's performance at the nominal and extreme temperatures with the mechanical compensation means adjusting axial spacings. If the optomechanical design is proper, the performance after the specified temperature changes will be acceptable. Many steps in this process are taken care of automatically by the design software, but the designer needs to participate in key decisions, such as configurations, materials, and dimensions of the metallic components.

To illustrate a simple (manual) application of this design process, we summarize here an analysis by Friedman¹⁴ in which a passive mechanical system was devised to correct the final image distance of a 24-in. (60.96-cm) focal length, f/5 aerial camera lens to optimize performance over the range 20°C to 60°C. Figure 14.10 shows the lens system schematically. It was assumed that the temperature of the camera would stabilize at each temperature considered. The specification required that the performance not be degraded by more than 10% over the full temperature range.

Friedman's analysis indicated that the back focal length (BFL) of the lens increased monotonically from 365.646 mm (14.395 in.) at 20°C to 365.947 mm (14.407 in.) at 60°C for a change Δ BFL of 0.303 mm (0.012 in.). To ensure proper performance at all temperatures between these extremes, it would be necessary to adjust the film location. A bimetallic mechanical design of the additive type shown in Fig. 14.6(a) was created using aluminum 6061 and type 416 stainless steel in series as materials for spacers between the lens mount and the image (film) plane. The configuration is shown schematically in Fig. 14.11. The fixed dimensions and other parameters are listed in Table 14.2.

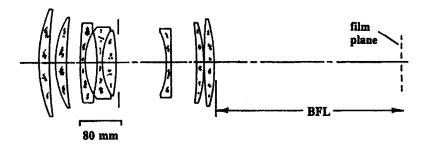


Figure 14.10 Optical schematic diagram of a 24-in. (60.96-cm) focal length, f/3.5 aerial camera lens. (Adapted from Friedman.¹⁴)

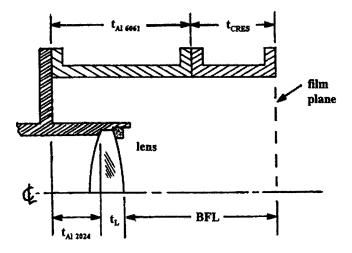


Figure 14.11 Mechanical schematic of the athermalizing structure. (From Friedman.¹⁴)

Friedman's analysis indicated that the back focal length (BFL) of the lens increased monotonically from 365.646 mm (14.395 in.) at 20°C to 365.947 mm (14.407 in.) at 60°C for a change Δ BFL of 0.303 mm (0.012 in.). To ensure proper performance at all temperatures between these extremes, it would be necessary to adjust the film location. A bimetallic mechanical design of the additive type shown in Fig. 14.6(a) was created using aluminum 6061 and type 416 stainless steel in series as materials for spacers between the lens mount and the image (film) plane. The configuration is shown schematically in Fig. 14.11. The fixed dimensions and other parameters are listed in Table 14.2.

Table 14.2 Parameters used in design of a passive compensation system.

Material at 20°C	CTE (°C)	Length (mm) @ 20°C	ΔT (°C)
Al 6061	23.6	154.102	40
CRES	9.9	unknown	:
Al 2024	23.2*	154.102	
Glass	6.3	15.252	
	at 20°C Al 6061 CRES Al 2024	at 20°C (°C) Al 6061 23.6 CRES 9.9 Al 2024 23.2	at 20°C (°C) (mm) @ 20°C Al 6061 23.6 154.102 CRES 9.9 unknown Al 2024 23.2* 154.102

From Friedman.¹⁴

From the geometry of Fig. 14.11, we obtain the following two relationships:

 $(\alpha_{A1\,6061}\,t_{A1\,6061} + \alpha_{CRES}\,t_{CRES} - \alpha_{A1\,2024}\,t_{A1\,2024} - \alpha_G\,t_L)(\Delta T) = \Delta_{BFL},$

 $t_{\rm A1\,6061} + t_{\rm CRES} = t_{\rm A1\,2024} + t_L = 535.00.$

Note: This value differs slightly from that given in Table B12 for the material.

Substituting data from Table 14.2 into the first relationship, we obtain:

$$236t_{A16061} + 99t_{CRES} = 112462.54.$$

Solving this equation simultaneously with the second relationship, Friedman obtained $t_{A1 6061} = 434.289$ mm and $t_{CRES} = 100.711$ mm. These dimensions were used in the camera's mechanical design.

Evaluation of the system's performance at its best focus and at the lowest and highest temperatures predicted that the polychromatic optical transfer function (OTF) as a function of spatial frequency in the image in line pairs per millimeter (lp/mm) in daylight with a minus-blue filter would be as indicated in Figs. 14.12(a) and 14.12(b). The response of a particular film type (Panatomic-X, Type 136) is also indicated in each figure. The intersections of the latter curve with the OTF curves for different points in the image (on-axis, 6-deg off-axis radial and tangential) represent the resolution capability of the lens and film system at the respective temperatures. These resolution predictions are summarized in Table 14.3. The results indicate that the design meets the specification over the temperature range. The system therefore is considered athermal.

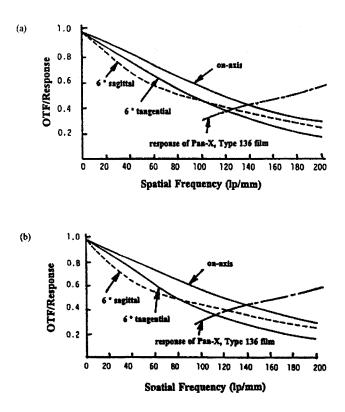


Figure 14.12 Polychromatic OTF for the temperature-compensated lens system at (a) minimum temperature (20°C) and (b) maximum temperature (60°C). (From Friedman.¹⁴)

System Resolution (lp/mm)				
Semi-field angle	At 20°C	At 60°C	Percent change	
On axis	140	140	0	
6° sagittal	126	123	-2	
6° tangential	122	113	_9	

Table 14.3 Resolution capability of the 24-in. focal length temperature compensated lens and film system.

From Friedman¹⁴

14.2.2 Active compensation

A possible means for athermalizing focus of optical systems is active control of the location(s) of one or more optic. In such systems, the temperature distribution within the system is measured, and motor-driven mechanisms are used to drive the mirror or lens separation and/or BFL to optimum values in accordance with pre-established algorithms.

From the systems viewpoint, a better but more complex approach would be to sense sharpness of focus or overall quality of the image and actively control the location(s) of one or more components to optimize performance. Both techniques require an expenditure of energy that may not be easily available.

Table 14.4 Requirements for an actively athermalized afocal zoom attachment.

Parameter	Requirement at Temperature		
	Ambient	0 and 90°C	
Magnification range	0.9× to 4.5×		
Relative aperture	<i>f</i> /2.6		
Spectral range	8.0 to 11.7 μm		
Elapsed time full range change	≤2 sec		
MTF (relative to diffraction limit)			
On axis	≥85%	≥77%	
0.5 field	≥75%	≥68%	
0.9 field	≥50%	≥45%	
Length	5.19 in.		
Diameter	5.50 in.		
Weight goal	≤5 lb		
Athermalization	Focus maintained 0 to 50°C		
Distortion	≤5%		
Target range	500 ft to infinity		
Vignetting	none		

From Fischer and Kampe.¹⁵

An active temperature compensation system described by Fischer and Kampe¹⁵ was a 5:1 afocal zoom attachment for a military forward looking infrared (FLIR) sensor operating in the spectral range of 8 to 12 μ m. Requirements for the system are listed in Table 14.4. The optical system developed to meet these requirements is shown in Fig. 14.13. The first element is fixed, as are the smaller lenses that are so indicated. The moveable lenses are designated Groups 1 (air spaced doublet) and 2 (singlet). All of these lenses are made of germanium as is the second small fixed lens. The other small fixed lens is zinc selenide. Its purpose is primarily to correct chromatic aberration. There are four aspherics in the design. Image quality of this design met all requirements over the specified temperature and target distance ranges when the locations of the moveable lens groups were optimized.

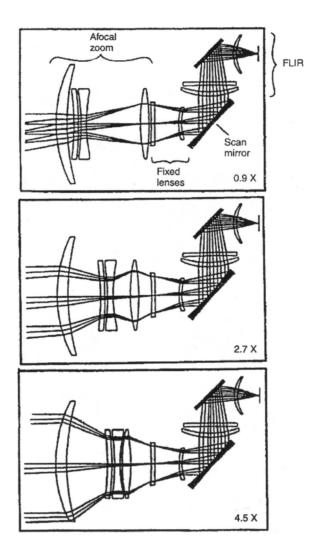


Figure 14.13 Optical system configurations for an afocal zoom lens at three magnifications. (From Fischer and Kampe.¹⁵)

Achievement of athermalization was accomplished by mounting the moveable groups on guide rods through linear bearings (see Fig. 14.14) and driving them independently with two stepper motors acting through appropriate spur gear trains as shown schematically in Fig. 14.15. The motors were controlled either by a local microprocessor (during operation) or an external personal computer (during test). The operator commanded the magnification to be provided and the target range. The system electronics then referred to a look-up table stored in a built-in erasable programmable read only memory (EPROM) to determine the appropriate settings for the moveable lenses at room temperature. Two thermistors attached to the lens housing sensed the temperature of the assembly. Signals from these sensors were used by the electronics to select, from a second look-up table stored in the EPROM, the required changes to the lens locations to correct for temperature effects on system focus. The corrected signals were then used to drive the motors to position the lenses to achieve best imagery at the measured temperature. The lens group motions varied as functions of magnification, target range, and temperature as indicated in Fig. 14.16.

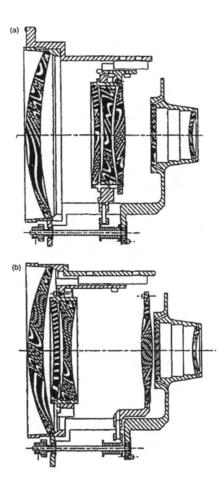


Figure 14.14 Section views of the zoom lens optomechanical configuration at (a) high magnification (4.53 \times) and (b) low magnification (0.93 \times). (From Fischer and Kampe.¹⁵)

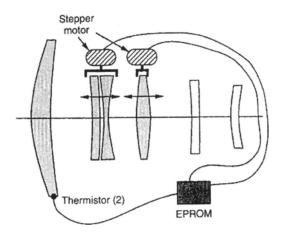


Figure 14.15 Schematic of the temperature sensing and motor drive system used to athermalize the zoom lens of Fig. 14.14. (From Fischer and Kampe.¹⁵)

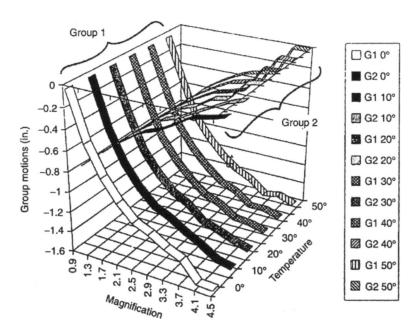


Figure 14.16 Motions of zoom lens groups as functions of magnification and temperature over the 0 to 50°C range. (From Fischer and Kampe.¹⁵)

14.3 Effects of Temperature Changes on Axial Preload

Optical and mechanical materials usually have dissimilar CTEs so changes in temperature cause proportional changes in the axial preload. Yoder¹⁶ quantified this relationship as

$$\Delta P = K_3 \Delta T, \tag{14.7}$$

where K_3 is the rate of change of preload with temperature for the design. This factor might well be called the design's "temperature sensitivity factor." It normally is a negative number. Knowledge of the value of K_3 for a given optomechanical design would be advantageous because it would allow the estimation of actual preload at any temperature by adding ΔP to or subtracting ΔP from the assembly preload depending on the direction of the temperature change. In the absence of friction, this preload is the same at all surfaces of all lenses clamped against a shoulder by a single retaining ring or flange.

Because we know the materials used in a given design, we can determine the applicable value of K_2 at each of the glass/metal interfaces using Eq. (13.5). In multiple lens assemblies, the elastic and/or thermal properties of the various lenses may differ, thereby giving different values for K_2 for each. Knowing the type of mechanical interface, the optical surface radius, and the preload at each interface, the value for K_1 can be calculated using the appropriate form of Eq. (13.4). The axial contact stress S_C at that surface can then be estimated through use of Eq. (13.13) and the corresponding tensile stress estimated using Eq. (13.1). In general, the stresses at the two surfaces of a given lens element will differ if they have different radii and/or different shapes or if the mechanical interface shapes are different. This is because those variables are used to determine K_1 .

14.3.1 Axial dimension changes

If α_M exceeds α_G (as usually is the case), the metal in the mount expands more than the optic for a given temperature increase ΔT . The axial preload P_A existing at assembly temperature T_A [typically 20°C (68°F)] will then decrease. If the temperature rises sufficiently, that preload will disappear. If the lens is not otherwise constrained (such as with an elastomeric sealant), it will be free to move within the mount in response to externally applied acceleration forces. We define the elevated temperature at which the axial preload goes to zero as T_C . This temperature is

$$T_C = T_A - \left(\frac{P_A}{K_3}\right). \tag{14.8}$$

The mount maintains contact with the lens until the temperature rises to T_c . A further temperature increase introduces an axial gap between the mount and lens. This gap should not exceed the design tolerance for despacing of this lens.

The increase in axial gap $\Delta_{GAP A}$ created in a general multilens subassembly as the temperature rises *above* T_C to T can be approximated as:

$$\Delta_{\text{GAP}A} = \Sigma_1^n \left(\alpha_M - \alpha_i \right) (t_i) (T - T_C).$$
(14.9)

A single element lens subassembly, a cemented doublet subassembly, and an air-spaced-doublet subassembly are shown schematically in Fig. 14.17. For these cases, Eq. (14.9) becomes:

$$\Delta_{\text{GAP}A} = (\alpha_M - \alpha_G)(t_E)(T - T_C), \qquad (14.10)$$

$$\Delta_{\text{GAP}A} = \left[\left(\alpha_M - \alpha_{G1} \right) \left(t_{E1} \right) + \left(\alpha_M - \alpha_{G2} \right) \left(t_{E2} \right) \right] \left(T - T_C \right), \tag{14.11}$$

$$\Delta_{\text{GAP}A} = \left[\left(\alpha_M - \alpha_{G1} \right) \left(t_{E1} \right) + \left(\alpha_M - \alpha_S \right) \left(t_S \right) + \left(\alpha_M - \alpha_{G2} \right) \left(t_{E2} \right) \right] \left(T - T_C \right), \quad (14.12)$$

where subscript "S" refers to the spacer and all other terms are as defined earlier.

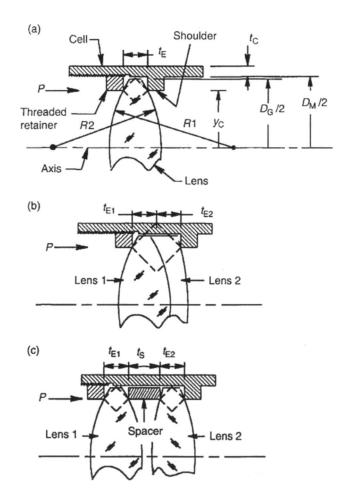


Figure 14.17 Schematics of lens mountings for (a) a singlet lens, (b) a cemented doublet, and (c) an air spaced doublet. (From Yoder.¹⁷ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

If the preload applied to a lens or lenses at assembly is very large or K_3 very small, the value for T_C calculated with Eq. (14.8) may exceed T_{MAX} . In such cases, ΔGAP_A will be negative, indicating that glass-to-metal contact is never lost within the range $T_A \leq T \leq T_{MAX}$.

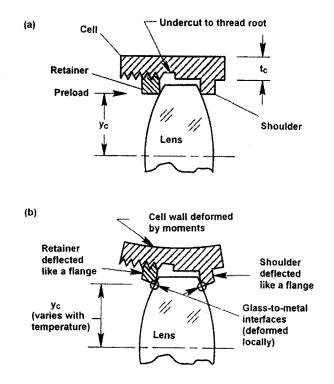


Fig. 14.18 Schematic of a simply-mounted biconvex lens (a) nominal design (b) some of the effects of a temperature change (decrease) that affect the factor K_3 . (From Fischer et al.¹⁹)

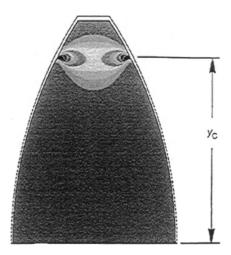


Figure 14.19 FEA representation of the stress distribution within a lens, as in Fig. 14.18(a), when preloaded. (Adapted from Genberg.²⁰)

In nearly all applications, small changes in position and orientation of a lens within axial and radial gaps created by differential expansion are tolerable. However, high accelerations (vibration or shock) applied to the lens assembly when clearance exists between the lens and its mechanical reference surfaces may cause damage to the lens from glass-to-metal impacts. Such damage (called "fretting") of glass surfaces under sustained vibrational loading has been reported by Lecuyer¹⁸ and has been experienced on many other occasions. To minimize this threat, it is advisable to design the lens assembly to have sufficient residual preload at T_{MAX} to hold the lens against the mechanical interface under the maximum expected acceleration. As indicated earlier, the preload (in pounds) needed to constrain a lens of weight W under axial acceleration a_G is simply W times a_G . In the SI system, this preload (in newtons) is $9.807Wa_G$ where W is in kilograms. In order for adequate preload to exist at T_{MAX} , the preload at assembly should be the sum of the needed minimum preload and the preload decrease that is caused by the temperature increase.

For example, a lens weighing 0.25 lb (0.013 kg) is to be held in contact with its mount by a flange at a maximum temperature of 160°F (71.1°C) under acceleration of 15-times gravity in the axial direction. Assume that K_3 is -0.200 lb/°F (-1.599 N/°C) and assembly takes place at 68°F (28°C). The temperature change ΔT is 160 - 68 = 92°F (51.1°C). The preload needed to overcome axial acceleration at T_{MAX} is (0.25)(15) = 3.750 lb (16.681 N). The preload dissipated from T_A to T_{MAX} is (-0.200)(92) = -18.400 lb (-81.847 N). The total preload P_A needed at assembly is then 3.750 - (-18.400) = 22.150 lb (98.528 N). It is safe to conclude that the lens will not move under axial acceleration at the maximum temperature because it maintains contact with the shoulder. By similar reasoning, the increased preload at minimum temperature $T_{MIN} = -80°F$ (-62°C) would equal $P_A + (K_3)(T_{MIN} - T_A) = 22.150$ + (-0.200)(-80 - 68) = 51.750 lb (230.194 N). The tensile stress at this low temperature should be checked to make sure it does not exceed the suggested tolerance.

14.3.2 Quantifying K₃

The factor K_3 that makes these types of estimations possible depends on the optomechanical design of the subassembly and the pertinent material characteristics. It is difficult to quantify, even for a simple lens/mount configuration. For example, consider the design shown schematically in Fig. 14.18(a). Here, a biconvex lens is clamped axially in a cell between a shoulder and a retainer with a nominal preload. The glass-to-metal interfaces are shown as sharp corners, but conical (tangent) interfaces would be more appropriate in an actual design. The contact stress developed within the lens by the preload is distributed approximately as indicated in Fig. 14.19. This is a FEA representation from Genberg.²⁰

Yoder and Hatheway²¹ identified the following mechanical changes that can occur in this design when the temperature changes by some ΔT and that contribute to the magnitude of the design's unique value for K_3 (assuming $\alpha_M > \alpha_G$):

- change in bulk compression of the glass at height y_C ,
- change in bulk elongation of the cell wall of thickness t_c at the lens rim,
- change in elongation of the (weaker) threaded and undercut regions of the cell,
- changes in local deformations of the glass surfaces R_1 and R_2 at the interfaces,
- changes in local deformations of the retainer and shoulder surfaces at the interfaces,
- changes in flange-like deflections of the retainer and of the shoulder,
- change in "pincushion" deformation of the cell wall at the lens rim from the axiallysymmetric moment imposed by the preload,

- · flexibility within the threaded joint,
- unequal radial dimension changes of the lens and of the mechanical parts,
- · uncertainties caused by asperities on mechanical and glass surfaces, and
- frictional effects.

Some of these effects are represented schematically in Fig. 14.18(b).

Designs, such as those for cemented doublet lenses [see Fig. 14.17(b)] or multiple air spaced lenses with intermediate spacers [see Fig. 14.17(c)] would increase complexity and provide additional elastic variables. In prior discussions of K_3 , the present author considered only the first two of these contributing factors (bulk glass compression and bulk cell wall stretching).^{16,17,22-24} That theory, which may be considered a first approximation of K_3 , is summarized in the following section.

14.3.2.1 Considering bulk effects only

 $K_{3 BULK}$ for any lens surface mounted with axial preload is given approximately as:

$$K_{3BULK} = \frac{-\sum_{i}^{n} \left(\alpha_{M} - \alpha_{i}\right) t_{i}}{\sum_{i}^{n} C_{i}},$$
(14.13)

where C_i is the compliance of one of the elastic components in the subassembly. For a lens, compliance is approximated as $[2t_E/(E_G A_G)]_i$, that of a cell as $[t_E/(E_M A_M)]_i$, and that of a spacer as $[t_S/(E_S A_S)]_i$. The terms in these equations should all be obvious, except for the cross sectional areas A_i for the stressed regions in the glass and metal components. Figures 14.20 through 14.22 illustrate the geometry for these terms. Equations (14.14) through (14.15b) define A_M and A_G for the mount and lens areas, respectively:

$$A_{M} = 2\pi t_{C} \left[\left(\frac{D_{M}}{2} \right) + \left(\frac{t_{C}}{2} \right) \right] = \pi t_{C} \left(D_{M} + t_{C} \right), \qquad (14.14)$$

where D_G is the OD of the lens, D_M is the ID of the mount at the lens rim, and t_C is the thickness of the mount wall adjacent to the lens rim.

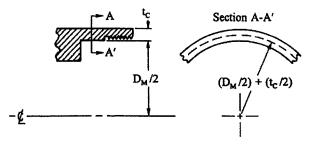


Figure 14.20 Geometric relationships used to approximate the cross sectional area of the stressed region in a lens mount. (From Yoder.¹⁶)

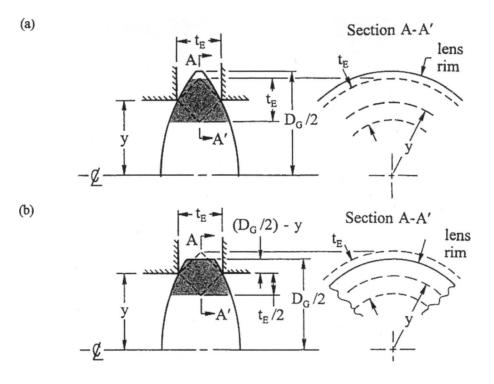


Figure 14.21 Geometric relationships used to determine cross sectional area of the stressed region within a lens: (a) when completely within the lens rim and (b) when truncated by the rim. (From Yoder.¹⁶)

For the lens, either of the following two cases can apply. If $(2y_C + t_E) \leq D_G$, the stressed region (the diamond shaped region in Fig. 14.21) lies entirely within the lens rim and Eq. (14.15a) applies. The lens thickness t_E is measured at the height of contact y_C . This height is assumed the same on both sides of the lens. If $(2y_C + t_E) \geq D_G$, the stressed region is truncated by the rim and Eq. (14.15b) is used:

$$A_G = 2\pi \, y_C \, t_E, \tag{14.15a}$$

$$A_G = (\pi/4)(D_G - t_E + 2y_C)(D_G + t_E - 2y_C).$$
(14.15b)

The expression for K_3 [Eq. (14.13)] becomes the following for the single lens element:

$$K_{\rm 3BULK} = \frac{-(\alpha_M - \alpha_G)t_E}{\left(\frac{2t_E}{E_G A_G}\right) + \left(\frac{t_E}{E_M A_M}\right)}.$$
(14.16)

As indicated earlier, the terms in the denominator are the compliances for the lens and the cell.

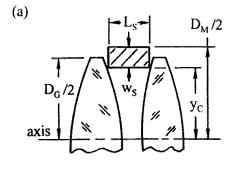
For a cemented doublet mounted in a cell per Fig. 14.17(b), we would rewrite Eq. (14.13) as follows:

$$K_{3 \text{ BULK}} = \frac{-(\alpha_M - \alpha_{G1})t_{E1} - (\alpha_M - \alpha_{G2})t_{E2}}{\left(\frac{2t_{E1}}{E_{G1}A_{G1}}\right) + \left(\frac{2t_{E2}}{E_{G2}A_{G2}}\right) + \left[\frac{(t_{E1} + t_{E2})}{E_M A_M}\right]}.$$
(14.17)

For an air spaced doublet mounted in a cell with a spacer per Fig. 14.17(c), we would rewrite Eq. (14.13) in this form:

$$K_{3 \text{ BULK}} = \frac{-(\alpha_M - \alpha_{G1})t_{E1} - (\alpha_M - \alpha_S)t_S - (\alpha_M - \alpha_{G2})t_{E2}}{\left(\frac{2t_{E1}}{E_{G1}A_{G1}}\right) + \left(\frac{t_S}{E_SA_S}\right) + \left(\frac{2t_{E2}}{E_{G2}A_{G2}}\right) + \left[\frac{(t_{E1} + t_S + t_{E2})}{E_MA_M}\right]},$$
 (14.18)

where A_S is the cross sectional area of the spacer and t_S is its axial length.



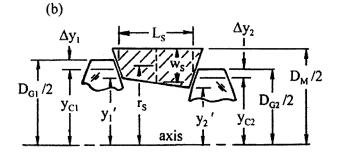


Figure 14.22 Schematic of typical lens spacers. (a) cylindrical type with sharp-corner interfaces. (b) Tapered type with tangential interfaces. (From Yoder.¹⁶)

EFFECTS OF TEMPERATURE CHANGES

Figure 14.22 shows two versions of spacers: (a) one with a rectangular cross section and wall thickness w_s and (b) one with a tapered cross section whose wall thickness is approximated as its average annular thickness. Equations (14.19) through (14.24) are used to determine the annular areas to be used in Eq. (14.18). Appropriate modifications to these equations should be made if the interface types are changed or if one or more lens surface(s) is/are concave.

For a simple cylindrical spacer:

$$w_{S \text{ CYL}} = \left(\frac{D_M}{2}\right) - (y_C)_i. \tag{14.19}$$

For a simple tapered spacer:

$$\Delta y_i = \left[\frac{\left(D_G\right)_i}{2}\right] - \left(y_C\right)_i, \qquad (14.20)$$

$$y'_{i} = (y_{c})_{i} - (\Delta y)_{i},$$
 (14.21)

$$w_{S \text{ TAPER}} = \left(\frac{D_{M}}{2}\right) - \left[\frac{(y'_{1} + y'_{2})}{2}\right].$$
 (14.22)

In both cases:

$$r_{s} = \left(\frac{D_{M}}{2}\right) - \left(\frac{w_{s}}{2}\right), \tag{14.23}$$

$$A_s = 2\pi r_s w_s. \tag{14.24}$$

Similar equations can be written for other spacer designs.

14.3.2.2 Considering other contributing factors

At the beginning of this section, we listed several effects other than bulk effects that can occur in a lens mounting and that may contribute to the value of K_3 for that mounting. Here, we show how certain of these effects can be included in the analysis of a design.

Local deformations of the glass and metal surfaces: Young²⁵ gave the following equation for the reduction Δx in the distance between the centers of two parallel cylinders of cross sectional diameters D_1 and D_2 when forced together by a linear preload p. Figure 13.13 shows the geometric model for this case. This dimensional change accounts for the local elastic deformations of both the glass and metal surfaces:

$$\Delta x = \left\lfloor \frac{2p(1-v^2)}{\pi E} \right\rfloor \left[\left(\frac{2}{3} \right) + \ln \left(\frac{2D_1}{\Delta y} \right) + \ln \left(\frac{2D_2}{\Delta y} \right) \right], \quad (14.25)$$

This equation assumes that Young's modulus E and Poisson's ratio v of the two materials are equal. This generally is not the case in mounting optics. To the level of accuracy required here, however, it is appropriate to use the average of the material values. The linear preload p was defined earlier as $P/(2\pi y_C)$ where P is the total preload and y_C is the height of metal contact with the lens surface. The width Δy of the deformed areas in the interface is given by Eq. (13.9a), repeated here for convenience:

$$\Delta y = 1.600 \left(\frac{K_2 p_i}{K_1} \right)^{\frac{1}{2}}.$$
 (13.9a)

The surface deformations occur at the interfaces on both sides of the lens. They act as two springs in series with the compliance of each spring equal to:

$$C_D = \frac{\Delta x}{P}.$$
 (14.26)

For a single lens element in a cell, the compliances would be added to the denominator of Eq. (14.16) along with the compliances corresponding to bulk effects in the glass and metal to derive a better approximation for K_3 .

Retainer and shoulder deflection effects: If we assume that a threaded retainer is rigidly attached to the cell wall by its thread, as is the shoulder, they each can deflect under preload by Δx in the manner of a continuous circular flange as described in Section 3.6.2. We repeat the applicable Eqs. (3.38) through (3.40) for convenience:

$$\Delta x = \left(K_{A} - K_{B}\right) \left(\frac{P}{t^{3}}\right), \qquad (3.38)$$

where

$$K_{A} = \frac{3(m^{2}-1)\left[a^{4}-b^{4}-4a^{2}b^{2}\ln\left(\frac{a}{b}\right)\right]}{4\pi m^{2}E_{M}a^{2}},$$
(3.39)

$$K_{B} = \frac{3(m^{2}-1)(m+1)\left[2\ln\left(\frac{a}{b}\right) + \left(\frac{b^{2}}{a^{2}}\right) - 1\right]\left[b^{4} + 2a^{2}b^{2}\ln\left(\frac{a}{b}\right) - a^{2}b^{2}\right]}{(4\pi m^{2}E_{M})\left[b^{2}(m+1) + a^{2}(m-1)\right]}, \quad (3.40)$$

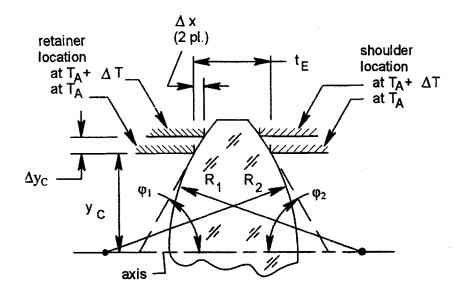
P is the total preload, *t* is the axial thickness of the retainer or shoulder, *a* and *b* are the outer and inner radii of the cantilevered sections, *m* is the reciprocal of Poisson's ratio (v_M) and E_M is the Young's modulus of the flange material.

The compliances C_R and C_S of the retainer and shoulder acting as flanges are:

$$C_{i} = \frac{\left(\Delta x\right)_{i}}{P} = \frac{\left(K_{A} - K_{B}\right)}{t_{i}}.$$
(14.27)

Both values obtained from Eq. (14.27) for the retainer and shoulder would be added to the denominator of Eq. (14.16) for K_3 to obtain a better approximation of K_3 . Because the $(t)_i$ are relatively large, C_R and C_S undoubtedly are small values so do not change K_3 very much. Similarly, the bending stress changes in these components resulting from ΔT are small so need not be of concern.

Interface radial dimension change effects: Figure 14.23 shows schematically how the interfaces between the retainer and the lens and between the shoulder and the lens move when the temperature rises by some ΔT . The locations of the interfaces move radially outward by Δy_C because of differential expansion with $\alpha_M > \alpha_G$. Because the lens surfaces are inclined by the angles φ_i at the interfaces, those contact interfaces move axially toward each other by Δx .





The following relationships apply:

$$\varphi = 90 \operatorname{deg} - \arcsin\left(y_C/R\right), \qquad (14.28)$$

$$\Delta y_C = (\alpha_M - \alpha_G) y_C, \tag{14.29}$$

$$\Delta x = -\Delta y_C / \tan \varphi. \tag{14.30}$$

These dimensional changes occur at each surface-to-mount interface so, for a biconvex lens, Δx must be calculated for each surface and both values inserted into the *numerator* of Eq. (14.16) for K_3 because they represent axial dimension changes due to differential expansion under a temperature change ΔT , just as do the terms already in that numerator. For a given ΔT , a convex surface and a concave surface would have Δx values of opposite sign. Δx is zero for a flat lens surface so it then has no effect on K_3 . The same is true for a concentric meniscus lens, because the effect at one surface would cancel that at the other surface.

A methodology for K_3 estimation: The approximate theory described above for estimating K_3 for a single lens/cell combination has been used in the literature to predict about how much preload is required at assembly to ensure sufficient residual preload at T_{MAX} for alignment of a symmetrical biconvex lens to be preserved.²¹ The magnitudes of some effects that influence K_3 and that are considered above and in the referenced publication depend on the preload P_A applied at assembly. There is no direct method for finding the right P_A , so an iterative procedure has been employed. An initial value is assumed for P_A and K_3 is calculated using appropriate algebraic signs for each effect. The corresponding residual preload is then $P'_A = P_A - (K_3)(T_{MAX} - T_A)$. This residual will undoubtedly not equal the desired value so the positive or negative preload discrepancy is added to the prior preload value to obtain a new P_A for a second iteration. After a few cycles, the error should be small and the computation is complete. A computer spreadsheet is useful in performing these repetitive calculations.

The computational procedure for analyzing a simple single lens and mount design described here and in the referenced publication could be applied to multiple element designs, but that process would certainly be quite complex and the mathematics arduous. Not knowing how to estimate K_3 and/or to avoid mathematical difficulties in its determination, designers and engineers frequently incorporate one or more features in the mechanical design of an optomechanical subassembly to make K_3 for that subassembly so small that it causes only minor effects as the temperature changes. In the next section, we describe a few designs of this type and show examples to illustrate the design principles.

14.3.3 Advantages of athermalization and axial compliance

One technique for reducing K_3 of an optical subassembly to insignificance is to make its design axially athermal so dimensional changes due to temperature changes are internally compensated in passive fashion. An example is the air spaced triplet shown in Fig. 14.24. In view (a) of this figure, we see the design as it might be created without concern about temperature changes. View (b) describes the spacers and the threaded retainer. The materials used are quite common. Pertinent dimensions are given in the figure. The subassembly is to survive temperature increases from 68°F to 160°F ($\Delta T = +92°F$) as well as temperatures decreases to T_{MIN} of -80°F ($\Delta T = -148°F$).

The axial length from point A to point B of Fig. 14.24(a) can be expressed as $L_{AB} = t_{E1} + t_{S1} + t_{E2} + t_{S2} + t_{E3}$ where the thicknesses t_E are the edge thicknesses of the lenses at the height of contact with the metal y_C . The thicknesses t_S are the thicknesses of the spacers at the same height y_C . At assembly, these dimensions are as indicated in the third column of Table 14.5. Note that showing the dimensions to six decimal places does not

mean that each must be controlled to that accuracy. This number of significant figures is provided to reduce round-off errors in computations involving subtraction of small numbers to illustrate the principle of the theoretical design. The total length L_{AB} for the original design at the assembly temperature T_A , is 1.064000 in. When the temperature increases by $\Delta T = 92^{\circ}$ F, each component lengthens by $(t_i)(\alpha_i)(\Delta T)$ where the α terms are as listed in the fourth column of the table. We note that L_{AB} grows by 0.000523 in. The portion of the cell wall extending from A to B also grows as the temperature rises. At T_{MAX} , its length is $(L_{AB})(\alpha_{CELL})(\Delta T) = 1.065282$ in., so it grows by 0.001282 in. Differential expansion then causes an axial gap of +0.000759 in. to occur. This gap can exist at any single interface in the assembly or it can be distributed among the various elements. This distribution cannot be controlled by design. When such an axial gap exists, the preload imposed at assembly no longer constrains the lenses so they are free to move under acceleration.

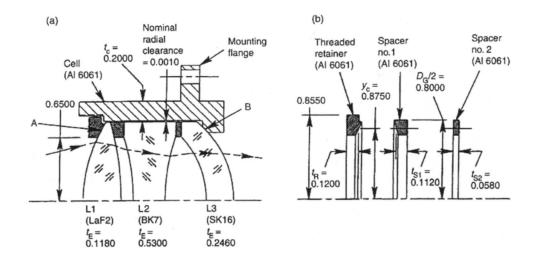


Figure 14.24 (a) Layout of an air-spaced triplet lens subassembly. (b) Details regarding the retainer and spacers. Dimensions are inches. (From Yoder.¹⁷ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

When the temperature drops to $T_{\rm MIN}$, the components in the assembly of Fig. 14.24(a) contract as indicated in the last column of Table 14.5. A difference in length L_{AB} of -0.001222 in., with the path through the cell changing length more than the path through the lenses and spacers, occurs from differential contraction. This would be expected to result in increases in the compression of the lenses and spacers and in the stretching effect in the cell wall.

One way in which the design of Fig. 14.24 could be improved would be to make it axially athermal. This is shown in Fig. 14.25. Here, the metals used in some components are selected so that their CTEs more closely match those of the glasses, thereby reducing the tendency to create axial gaps in the assembly at high temperatures. These changes are

to substitute CRES 416 for the cell material and CRES 303 for the second spacer material. For the nominal length L_{AB} be the same for the cell path as for the stack of lenses and spacers (optic path), the axial length of the second spacer must be increased from 0.058000 in. to 0.122737 in. The added length of the second spacer is accommodated without changing the optical design, i.e., the axial thicknesses of the lenses or the axial separations thereof, by grinding a step bevel into the right side of lens two. Alternatively, smaller step bevels could have been placed on both lens two and three to achieve the same results, but this would increase cost and have little benefit. We would not want to provide the full step bevel on lens three, as that would excessively reduce the edge thickness of that lens.

Table 14.5 Axial dimensions at T_A , T_{MAX} , and T_{MIN} of the lenses, spacers, and cell wall of the subassembly shown in Fig. 14.24.

Element	Material	Element length <i>t_i</i>	Element CTE	Element length <i>t_i</i>	Element length <i>t_i</i>
		(<i>a</i>) T_A (in.)	(in./in.×10 ⁻⁶)	@ T _{MAX} (in.)	@ T _{MIN} (in.)
Lens L ₁	LaF2	0.118000	4.5	0.118049	0.117921
Spacer S ₁	Al 6061	0.112000	13.1	0.112135	0.111783
Lens L ₂	BK7	0.530000	3.9	0.530190	0.529694
Spacer S ₂	Al 6061	0.058000	13.1	0.058070	0.057888
Lens L ₃	SK16	0.246000	3.5	0.246079	0.245873
L_{AB} (optic path)		1.064000		1.064523	1.063158
L_{AB} (cell path)	Al 6061	1.064000	13.1	1.065282	1.061937
ΔL (optic path)				+0.000523	-0.000842
ΔL (cell path)				+0.001282	-0.002063
$\Delta(\Delta L)$				+0.000759	-0.001221

Note: $T_A = 68^{\circ}F$, $T_{MAX} = 160^{\circ}F$, $T_{MIN} = -80^{\circ}F$. Dimensions are inches.

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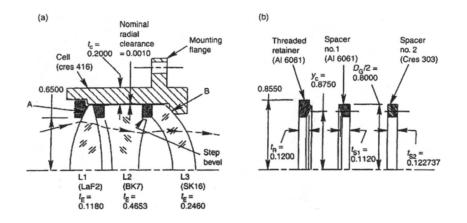


Figure 14.25 Layout of the modified design for the air-spaced triplet lens subassembly of Fig. 14.24. (b) Details for the retainer and spacers. Dimensions are in inches. (From Yoder.¹⁷ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

The design dimensions and their changes with temperature are listed in Table 14.6. As indicated in the last two columns, the $\Delta(\Delta L)$ parameter is zero at both T_{MAX} and T_{MIN} .

Table 14.6 Axial dimensions at T_A , T_{MAX} , and T_{MIN} of the lenses, spacers, and cell wall of the subassembly shown in Fig. 14.25.

Element	Material	Element length t_i @ T_A (in.)	Element CTE (in./in.×10 ⁻⁶)	Element length <i>t_i</i> @ T _{MAX} (in.)	Element length <i>t_i</i> @ T _{MIN} (in.)
Lens L ₁	LaF2	0.118000	4.5	0.118049	0.117921
Spacer S ₁	Al 6061	0.112000	13.1	0.112135	0.111783
Lens L ₂	BK7	0.465263	3.9	0.465430	0.464994
Spacer S ₂	CRES 303	0.122737	9.6	0.122845	0.122563
Lens L ₃	SK16	0.246000	3.5	0.246079	0.245873
L_{AB} (optic	CRES 416	1.064000		1.064538	1.063134
path)					
L_{AB} (cell path)		1.064000	5.5	1.064538	1.063134
ΔL (optic path)				+0.000538	-0.000866
ΔL (cell path)				+0.000538	-0.000866
$\Delta(\Delta L)$				+0.000000	-0.000000

Note: $T_A = 68^{\circ}$ F, $T_{MAX} = 160^{\circ}$ F, $T_{MIN} = -80^{\circ}$ F From Yoder.¹⁷ (Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

The basis for this conclusion is slightly inaccurate because a few effects of the temperature changes have been ignored. These include the radial dimension changes for the height of contact y_C in the glass-to-metal interfaces (see Fig. 14.23) and possible flexure of the retaining ring under preload. Neither of these factors is believed to be very significant for the design under consideration.

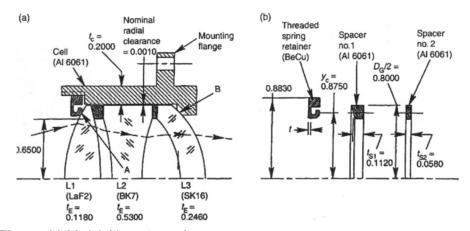


Figure 14.26 (a) Variation of the optomechanical design of Fig. 14.24 to provide increased axial compliance at the retaining ring. (b) Details of the retainer and spacers. Dimensions are in inches. (From Yoder. ¹⁷Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

Another way to improve the lens assembly of Fig. 14.24 is to increase the axial compliance. This approach is shown in Fig. 14.26. The cell and spacer materials are the same as in the original design, but the axially stiff retaining ring is replaced with a compliant flange so assembly preload can be applied by deflecting the portion of the ring that extends towards lens L_1 . The thickness t of the compliant portion of the retainer and the annular dimensions of that portion are chosen to give the required axial force with a reasonable deflection while not introducing excessive stress into the bent flange.

To see how this revised design might work, let us assume that the design requires a deflection of the compliant retainer equaling 0.0200 in. to provide the required preload. As noted from Table 14.5, a temperature change from T_A to T_{MAX} produces a differential expansion for the optic path of + 0.0008 in. as compared to the path through the cell. This change in length is 4% of the retainer deflection. Because of the linear relationship between flange deflection and preload [see Eq. (3.38)], the assembly preload will be reduced approximately by this same amount at T_{MAX} . Similarly, at T_{MIN} , differential expansion will decrease the retainer deflection by 0.0012 in. as compared to the path through the cell. This change in length is 6% of the flange deflection so the assembly preload will be increased by about this amount at T_{MIN} . These changes would probably not be considered significant for the intended application of the lens assembly.

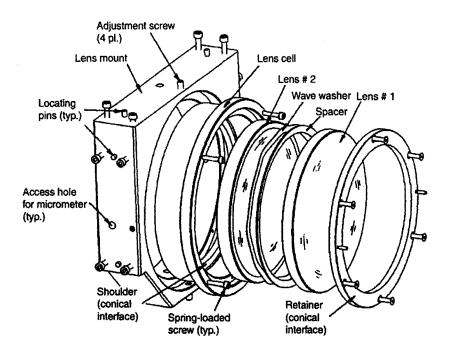


Figure 14.27 Exploded view of a lens assembly with an axially compliant lens constraint and adjustable mechanical centration. (From Stevanovic and Hart.²⁶)

Another lens mounting that provides axial compliance was described by Stevanovic and Hart.²⁶ See the exploded view of Fig. 14.27. It was designed at the Research School of Astronomy and Astrophysics, Australian National University in Canberra, Australia

and is used in the Gemini South Adaptive Optics Imager (GSAOI). This is a near infrared camera that serves as the main science instrument in the Multi-Conjugate Adaptive Optics (MCAO) system in Chile. A stiff flange-type retaining ring with a conical interface to the outermost lens clamps the two lens elements axially against a conical shoulder in the lens cell. A wave washer located along with a conventional spacer between the lenses provides predetermined axial compliance. The lenses in this assembly are Infrasil and CaF_2 and have diameters of 170 mm (6.69 in.). These are the largest elements in the camera. The mountings for the smaller lenses are similar to this configuration. Theoretically, the lenses have the correct radial clearance at the time of assembly at room temperature to provide zero clearance at the operating temperature of 70K. This should center the lenses mechanically to the axis of the cell ID.

A useful feature of the assembly design of Fig. 14.27 is that the alignment of the lens cell is adjustable within the lens mount after assembly. Four setscrews are provided to allow the lateral position of the cell to be adjusted along two axes at assembly. Holes are provided in the walls of the mount for micrometer measurements of cell location to be made. The authors do not mention just how the centered cell is held in its aligned position.

Stevanovic and Hart²⁶ also give a detailed analysis of the effects of transient dimensional changes within the lens mount as the system cools to operating temperature. All components do not cool at the same rate, so differential dimensional effects occur because of temporal and spatial temperature gradients. The published analysis shows the design to be conservatively adequate to prevent damage to the optics during the expected temperature excursions. Alignment is restored after the temperature stabilizes.

14.4 Radial Effects in Rim Contact Mountings

When the temperature increases, radial clearances around the lenses and spacers tend to increase while, as the temperature decreases, those radial clearances shrink. If the clearances are small at assembly, they may disappear completely at some lower temperature and radial stresses, as well as hoop stresses in the cell, might then develop. When the temperature rises, the radial gap provided at assembly will increase. If axial preload is insufficient to hold the lenses in place, they may change decenter or tilt within these spaces. It would be helpful if we knew the radial dimensional changes for any given design so the magnitude of the potential problems could be assessed.

We can determine these facts for the design of Fig. 14.24(a) as indicated in Table 14.7. It lists the radial dimensions and the changes in those dimensions at the three temperatures of interest. The materials and CTEs are as listed in Table 14.5. In the fourth column of Table 14.7, we find the differences between (element diameters)/2 and the (cell ID)/2 at T_{MAX} . These differences are the radial clearances. The component can decenter by this amount if not constrained by axial preload. The nominal clearance of 0.001000 in. at assembly is increased significantly at all lenses. The radial clearances for the two spacers do not change because they are made of the same material as the cell (Al 6061). We learned earlier that axial preload does not exist in this design at T_{MAX} because of differential expansion. Hence, lens decentrations would be expected. Damage, called fretting, to the lens surfaces also may occur under vibration. When the temperature returns to the operating range, the preload will return and the lenses may be constrained

while decentered. The optical performance of the system may be degraded if these decentrations exceed the allowable limits for the design.

Element	Element OD/2 @ T _A (in.)	Element OD/2 @ T _{MAX} (in.)	Possible Decentration @ T _{MAX} (in.)	Element OD/2 @ T _{MIN} (in.)	Possible Decentration @ T _{MIN} (in.)
Lens L ₁	0.800000	0.800331	0.001634	0.799467	-0.000020
Spacer S ₁	0.800000	0.800964	0.00100	0.798449	0.001000
Lens L ₂	0.800000	0.800287	0.001678	0.799538	-0.000091
Spacer S ₂	0.800000	0.800964	0.001000	0.798449	0.000100
Lens L ₃	0.800000	0.800258	0.001707	0.799586	-0.000139
Cell ID	0.80100	0.801965		0.799447	

Table 14.7 Radial dimensions at T_A , T_{MAX} , and T_{MIN} of the lenses, spacers, and cell wall of the subassembly shown in Fig. 14.24.

Note: $T_A = 68^{\circ}\text{F}$, $T_{MAX} = 160^{\circ}\text{F}$, $T_{MIN} = -80^{\circ}\text{F}$ From Yoder.¹⁷ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.

Table 14.7 also indicates, in the sixth column, that the assembly's lenses do not decenter when the temperature drops to T_{MIN} . The negative signs mean that the cell compresses the glasses. Once again, the clearances for the spacers are unchanged because they and the cell are made of the same material, so they shrink equally. It would be expected that these changes would be less significant with the modified design described in Fig. 14.23 because the differences in CTEs for the CRES cell and the glasses are reduced.

14.4.1 Radial stress in the optic

A rim contact mounting for a conventional lens, mirror, window, etc. can subject the optic to radial stress at reduced temperatures. The magnitude of the radial stress, S_R , in this optic under a given temperature drop, ΔT , can be estimated as:

$$S_R = -K_4 K_5 \Delta T, \tag{14.31}$$

where:

$$K_{4} = \frac{\left(\alpha_{M} - \alpha_{G}\right)}{\left(\frac{1}{E_{G}}\right) + \left[\frac{D_{G}}{\left(2E_{M}t_{C}\right)}\right]},$$
(14.32)

$$K_{5} = 1 + \left\{ \frac{2\Delta r}{\left[D_{G}\Delta T \left(\alpha_{M} - \alpha_{G} \right) \right]} \right\}.$$
 (14.33)

Here, D_G is the optic OD, t_C is the mount wall thickness outside the rim of the optic, and Δr is the radial clearance at assembly. Note that ΔT is negative for a temperature decrease. Also, $0 < K_5 < 1$. If Δr exceeds $[D_G \Delta T(\alpha_M - \alpha_G)/2]$, the optic will not be constrained by the mount ID and radial stress will not develop within the temperature decrease ΔT as a result of rim contact. Examples 14.1 and 14.2 demonstrate the use of these equations.

Example 14.1: Estimation of radial stress in an optic and hoop stress in its mount. (For design and analysis, use File 14.1 of the CD-ROM.)

An SF2 lens of diameter 2.384 in. (60.554 mm) is mounted in a 416 CRES cell with 0.0002-in. (5.08×10^{-3} -mm) radial clearance Assembly is at 68°F (20°C). The cell wall is 0.062-in. (1.575-mm) thick at the lens rim. What radial stress is developed within the lens and what hoop stress is felt in the cell wall at -80° F (-62° C)?

From Tables B1 and B12:

 $E_G = 7.98 \times 10^6$ lb/in.² (5.50×10⁴ MPa), $\alpha_G = 4.7 \times 10^{-6}$ /°F (8.4×10⁻⁶ /°C), $E_M = 2.9 \times 10^7$ lb/in.² (2.00×10⁵ MPa), $\alpha_M = 5.5 \times 10^{-6}$ /°F (9.9×10⁻⁶ /°C),

$$\Delta T = -80^{\circ}\text{F} - 68^{\circ}\text{F} = -148^{\circ}\text{F} (-82.2^{\circ}\text{C}).$$

From Eqs. (14.32) and (14.33):

$$K_{4} = \frac{5.5 \times 10^{-6} - 4.7 \times 10^{-6}}{\left(\frac{1}{7.98 \times 10^{6}}\right) + \left[\frac{2.384}{(2)(2.9 \times 10^{7})(0.062)}\right]} = 1.015 \text{lb/(in.}^{2} - {}^{\circ}\text{F}),$$

$$K_{5} = 1 + \frac{(2)(0.0002)}{(2.384)(-148)(5.5 \times 10^{-6} - 4.7 \times 10^{-6})} = -0.417.$$

From Eq. (14.31):

$$S_R = -(1.015)(-0.417)(-148) = -62.6 \text{ lb/in.}^2 (-0.43 \text{ MPa})$$

The negative sign on K_5 indicates that no radial stress can develop in the lens because the rim does not touch the cell. The stress calculation also is negative, confirming this result.

The hoop stress in the cell is given by Eq. (14.34):

$$S_{M} = \frac{(-62.6)\left(\frac{2.384}{2}\right)}{0.062}.$$

This value is negative, therefore there is no hoop stress.

Example 14.2: Estimation of radial stress in a radially constrained mirror and hoop stress in its mount. (For design and analysis, use File 14.2 of the CD-ROM.)

A mirror of diameter 20.000 in. (508.000 mm) and made of Ohara E6 glass is mounted in a 6061-T6 aluminum cell with 00002-in. (5.08×10^{-3} -mm) radial clearance Assembly is at 68°F (20°C). The cell wall is 0.250-in. (5.080-mm) thick at the mirror rim. What radial stress is developed within the mirror and what hoop stress is felt in the cell wall?

From Tables B8a and B12:

 $E_G = 8.5 \times 10^6 \text{ lb/in.}^2 (5.86 \times 10^4 \text{ MPa}), \alpha_G = 1.5 \times 10^{-6} \text{ /°F} (2.7 \times 10^{-6} \text{ /°C}),$ $E_M = 9.9 \times 10^7 \text{ lb/in.}^2 (6.83 \times 10^5 \text{ MPa}), \alpha_M = 13.1 \times 10^{-6} \text{ /°F} (23.6 \times 10^{-6} \text{ /°C}),$

 $\Delta T = -80^{\circ} F - 68^{\circ} F = -148^{\circ} F (-82.2^{\circ} C).$

From Eqs. (14.32) and (14.33):

$$K_{4} = \frac{13.1 \times 10^{-6} - 1.5 \times 10^{-6}}{\left(\frac{1}{8.500 \times 10^{6}}\right) + \left[\frac{20.000}{(2)(9.9 \times 10^{6})(0.250)}\right]} = 2.790 \text{lb/(in.}^{2} - \text{°F)}$$
$$K_{5} = 1 + \frac{(2)(0.0002)}{(20.000)(-148)(13.1 \times 10^{-6} - 1.5 \times 10^{-6})} = 0.988.$$

From Eq. (14.31): $S_R = -(2.790)(0.988)(-148) = 408 \text{ lb/in.}^2 (2.81 \text{ MPa}).$

This stress poses no danger to the mirror.

The hoop stress in the cell is given by Eq. (14.34):

,

$$S_{M} = \frac{(408)\left(\frac{20.000}{2}\right)}{0.250} = 16,320 \text{ lb/in.}^{2} (112.53\text{MPa}).$$

From Table B12, S_Y for 6061-T6 aluminum is 38,000 lb/in.² (262.0 MPa) so a safety factor of 38,000/16,320 = 2.3 exists. This is acceptable.

14.4.2 Tangential (hoop) stress in the mount wall

Another consequence of differential contraction of the mount relative to the rim contact optic is that stress is built up within the mount in accordance with the following equation:

$$S_M = \frac{S_R\left(\frac{D_G}{2}\right)}{t_C},\tag{14.34}$$

where all terms are as defined earlier.

With this expression, we can determine if the mount is strong enough to withstand the force exerted on the optic without exceeding its elastic limit. If the yield strength of the mount material exceeds S_M , a safety factor exists. Typical calculations are included in Examples 14.1 and 14.2.

14.4.3 Growth of radial clearance at high temperatures

The nominal value for the radial clearance between an optic and its mount at assembly can be defined as Gap_R . This dimension will increase by ΔGap_R due to a positive temperature increase of ΔT . The magnitude of this change can be estimated by:

$$\Delta \text{Gap}_{R} = \left(\alpha_{M} - \alpha_{G}\right) \left(\frac{D_{G}\Delta T}{2}\right).$$
(14.35)

If there is no axial constraint (as might happen at high temperature), whatever total radial clearance Gap_R exists between the optic OD and mount ID allows the optic to roll (i.e., tilt about a transverse axis) until its rim touches the mount ID at diametrically opposite points of the edge thickness t_A . This roll angle can be estimated by the equation:

$$\operatorname{Roll} = \arctan\left(\frac{2\operatorname{Gap}_{R}}{t_{E}}\right). \tag{14.36}$$

Calculations of a radial gap increase and possible roll of an optic are illustrated in Example 14.3.

Example 14.3: Growth in radial clearance around an optic at high temperature and possible roll (tilt) of that optic within this expanded clearance. (For design and analysis, use File 14.3 of the CD-ROM.)

What increase in radial clearance exists in the 20.000-in. (508.000-mm) diameter mirror assembly described in Example 14.2 at $T_{MAX} = 160^{\circ}$ F (71.1°C)? The radial clearance at assembly is 0.0002 in. (5.08×10⁻³ mm). The mirror is Ohara E6 glass and the cell is 6061 aluminum. The mirror thickness is 2.500 in. (63.500 mm), $\Delta T = 160 - 68 = 92^{\circ}$ F (51.1°C), $\alpha_G = 1.5 \times 10^{-6}$ /°F (2.7×10⁻⁶/°C), and $\alpha_M = 13.1 \times 10^{-6}$ /°F (23.6×10⁻⁶/°C).

By Eq. (14.35):

$$\Delta \text{Gap}_{R} = \left(13.1 \times 10^{-6} - 1.5 \times 10^{-6}\right) \left| \frac{(20.000)(92)}{2} \right| = 0.0107 \text{in.} (0.271 \text{ mm}).$$

The nominal radial gap at T_{MAX} is then 0.0002 + 0.0107 = 0.0109 in. (0.2769 mm).

By Eq. (14.36): Roll = $\frac{(2)(0.0109)}{2.500} = 8.72 \times 10^{-3}$ rad = 0.500 deg.

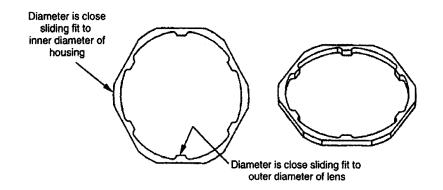


Figure 14.28 Vespel SP-1 spacer ring configuration as used to ensure centration of each lens in the MISR lens assemblies. (From Ford et al.¹³ Reprinted by courtesy of NASA/JPL/Caltech.)

14.4.4 Adding radial compliance to maintain lens centration

In Section 14.2.1, we described the optomechanical design of the multiangle imaging spectro-radiometer (MISR) lens assemblies. They have Vespel SP-1 spacers between each threaded retainer and the lens it constrains (see Fig. 14.7). The thicknesses of these high CTE spacers were determined so that, at extreme temperatures, the total axial lengths through the lenses were essentially equal to the corresponding lengths through the housing. This design feature rendered those assemblies axially athermal in the manner just described for the subassembly of Fig. 14.24. The MISR lenses also employ compliant annular spacer rings around all lens elements. These spacers have the configuration shown in Fig. 14.28. They also are made of Vespel SP-1. The ODs of these spacers were dimensioned for slight interference fits inside the lens housing IDs while their IDs gave slight interference fits around the lens ODs. The configuration of each spacer provides flexures between the six external and six internal lands. Radial force was applied to the lenses symmetrically at all temperatures by the spacers, thus keeping them well centered.

Figure 14.29 shows another hardware implementation of a compliant axial constraint for a single lens that also provides compliance in the radial direction. This design, from Barkhouser et al.,²⁷ is used in a high-resolution infrared camera for the Wisconsin, Indiana, Yale, National Optical Astronomical Observatory (WIYN) 3.5-m (138-in.) diameter telescope on Kitt Peak. Axial differential expansion effects are compensated for by flexure of a disk spring (similar to the continuous flange of Fig. 3.21). Six screws secure this spring. The floating ring acts as a spacer between the spring and the lens. Its thickness determines the spring deflection that provides axial preload.

Radial differential expansion effects in this design are compensated by a series of six "roll-pin" flexures as detailed in view (b) of the figure. These flexures are machined into the ID of the aluminum centering ring by an electrical discharge machining (EDM) process in much the same manner as the radial flexures shown in Fig. 3.43. In this case, however, the lens rim is not bonded to the flexures, but is constrained symmetrically with predetermined radial preload applied to the lens rim. The magnitude of this preload is determined by dimensional control during machining.

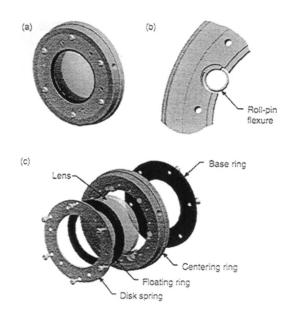


Figure 14.29 A lens mount with both axial and radial compliance. (a) The assembly, (b) detail of the radial flexure, and (c) exploded view. (From Barkhouser, et al.²⁷)

Another technique for maintaining centration of a lens is described in Section 15.20. This design uses a very sophisticated set of 16 springs that will center one of several crystalline lenses in the NIRCam to be used with the James Webb Space Telescope.

14.5 Effects of Temperature Gradients

Temperature gradients exist in optical instruments when all points within an optical instrument are not at the same temperature (spatial gradients) or when the temperature of any given part of the instrument is changing with time (temporal gradients). Spatial gradients may be axial or radial; both types may occur simultaneously within the same component or assembly. Gradients result from changes in ambient conditions, movement of the instrument from one temperature environment to another, varying heat load from the sun or from more local heat sources, etc. If an optical instrument is held in a constant temperature environment for a long time (a process called "soaking"), the temperatures tend to equalize and all gradients reduce in severity. Experience based on analysis and tests with various types of optical equipment indicates that a moderate-sized instrument may need several hours at constant temperature to stabilize. Under some conditions, the instrument never really reaches equilibrium. This usually happens when the instrument is exposed to a time-varying temperature environment.

Some optical instruments are exposed to rapidly changing temperatures as part of their intended application. This exposure is called thermal shock. In most cases, the instrument must perform to specification after being cooled or heated rapidly from one temperature to another temperature. One such assembly was described by Stubbs and Hsu.²⁸ This was an infrared sensor objective designed to cool ~150°C from room temperature to <120*K* within

five minutes. It contained a 26-mm (1.02-in.) aperture germanium singlet that was cooled by conduction of heat through annular interfaces with the mount. Figure 14.30 is a schematic sectional view of the objective while Fig. 14.31 is an exploded view of that device.

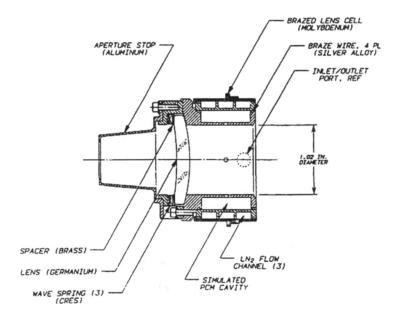


Figure 14.30 Sectional schematic of a lens assembly designed to withstand rapid cooling by conduction through annular contacts near the rim. (From Stubbs and Hsu.²⁸)

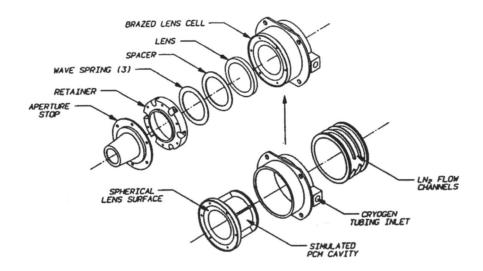


Figure 14.31 Exploded view of the lens assembly of Fig. 14.30. (From Stubbs and Hsu. 28)

The mount was made of molybdenum TZM (CTE = $5.5 \times 10^{-6}/K$) to match that of germanium ($4.9 \times 10^{-6}/K$). Heat transfer was maximized by establishing intimate contact between a flat bevel on the front (concave) lens surface and a flat brass spacer and also between the spherical rear (convex) lens surface and a matching concave spherical mechanical interface. The latter surface was ground and polished using optical test plates made to match the radius of the lens when at 120K within ~11 fringes at 0.633 µm wavelength. An assembly preload of 55 lb (245 N) was provided by three stainless steel wave spring washers in series with a flange-type retainer held in place by screws. The authors indicated that the room-temperature axial preload was 113 lb/in.² (0.78 MPa), so the contact area probably was about 0.5 in.² (322 mm²). With such large surface contacts, stress within the lens from the preload would be minimal and the spatial temperature gradient would be minimized.

Three flow channels were machined into the housing's outer cylindrical surface and a cylindrical plenum cover was brazed over these exposed channels. Fill and vent tubes were then brazed radially onto the cover. The chamber labeled "simulated PCM cavity" is a region reserved for a phase-change material to be used to stabilize the temperature of the assembly for about 25 minutes during operation after cool down with liquid nitrogen flow through the three channels. Coolant lines were epoxied to the radial tubes using Epibond epoxy type 1210A/9615-10 supplied by CIBA-Geigy Furane Aerospace Products.

Interferometric tests of a model of the lens assembly showed that the lens would survive the imposed thermal shock and gradients or compressive forces would not excessively distort its surfaces during operation. Laboratory tests of thermal behavior of the assembly showed that temperatures measured as a function of time after cryogen flow was initiated followed predictions reasonably well. Further, it was determined that the lens's temperature could be stabilized at about 100K for the desired 25 minute time period.

Another situation involving rapid temperature change and the possibility of thermal shock is an aerial camera moved from a warm environment on a flight tarmac to the frigid environment of high altitude above the Earth. Proper operation of the camera's mechanisms and full optical performance may not be realized in the severe operational environment for a significant time, if ever. Orbiting scientific optical payloads typically pose severe thermal design problems. In many such cases, excess heat may be radiated into outer space during part of a mission.

The temperatures of refracting components such as lenses, windows, filters, and prisms as well as large mirrors for astronomical telescopes can sometimes be temperature stabilized by blowing conditioned air across their surfaces or through cavities within the substrates, by flowing current through electrically conducting coatings on one or more surface(s), or by conduction from the mount. Heated window and filter examples were described in Chapter 5. Typically, small and moderate-sized mirrors are cooled (or heated) by conduction through their mounts or by heat-transfer devices attached to their back surfaces. Mirrors used in high-energy laser applications generally are temperature controlled by flowing coolant through heat-exchange channels within their substrates. Large ground-based astronomical telescope mirrors usually are temperature stabilized with attached heaters or coolers or by airflow across the back surfaces. An example of the latter approach is the new MMT telescope primary described in Section 11.3.3. Components whose temperatures are controlled by heat flow through their mounts around the peripheries of their apertures tend to suffer from radial gradients and these may be nonsymmetrical.

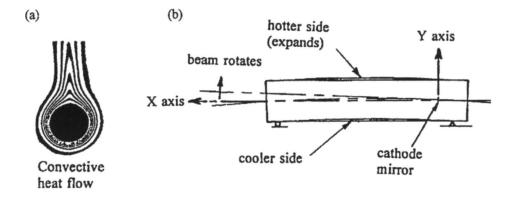


Figure 14.32 Effect of natural heat convection (a) on beam direction (b) in a gas laser with axis horizontal and inadequate control of temperature gradients. (Adapted from Hatheway.²⁹)

Hatheway²⁹ described how an argon ion laser was cooled by flowing air through a heat exchanger contacting the OD of the laser cavity wall. Figure 14.32(a) shows the natural air convection flow around a hot horizontal laser cavity. A vertical temperature gradient then develops, the structure warps, and the end mirrors of the cavity tilt thereby causing the beam to deflect in the vertical plane as indicated in Fig. 14.32(b). For the laser to be stable and usable in any orientation relative to gravity, the temperature gradient must be minimized. Cooling the assembly with flowing air accomplishes this goal and maintains the integrity of the brazed and frit bonded seals in the cavity.

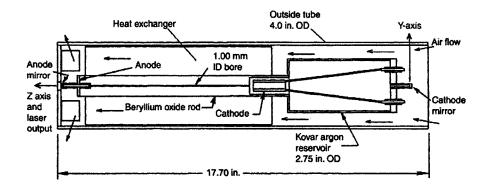


Figure 14.33 Schematic configuration of the gas laser modified with forced air cooling. (Adapted from Hatheway.²⁹)

Figure 14.33 shows schematically the configuration of the laser. It is built around a beryllium oxide rod that has a 1.0-mm diameter hole bored through along the centerline. Argon gas from the reservoir fills the cavity between the end mirrors and lases when

electrically excited between the cathode and anode. The laser design called for dissipation of about 2500 watts along the cavity bore plus about 100 watts of cathode heater power. From the outset, it was known that the laser would not function without cooling. To remedy the problem, a high efficiency aluminum heat exchanger was designed to fit within the available space around the rod and inside the outside tube ID. A centrifugal fan was installed to provide airflow equivalent to about 0.5 in. water pressure head. Performance was then acceptable.

14.5.1 Radial temperature gradients

A generic radial gradient in a simple lens is illustrated in Fig. 14.34. The lens is in air and is subjected to a condition in which the glass near the rim is warmer than that near the axis by an amount ΔT . The temperature, lens thickness, and refractive index of the axial region remain essentially constant at T_A , t_A , and n_A , while those parameters at the rim increase as indicated to $T_A + \Delta T$, $t_A + \Delta t$, and $n_A + \Delta n$. Jamieson¹⁰ indicated that, neglecting temperature gradients along the axis, the optical path difference (OPD) between the arbitrary ray shown passing between the points A and B compared with the corresponding ray along the axis is approximated by the expression: OPD = $[(n - 1) + \Delta n (t_A + \Delta t) - (n - 1)](t_A)$. Since $\Delta n = \beta_G \Delta T$ and $\Delta t = \alpha_G t \Delta T$, we obtain OPD = $[(n - 1)(\alpha_G) + \beta] t_A \Delta T$. Note: $\beta_G = dn/dT$ from the glass data sheet (see, for example, Fig. 1.7). The working equations are:

$$OPD = \left[(n-1)(a_G) + \beta_G \right] t_Z \Delta T = (n-1)(\gamma_G t_A \Delta T), \qquad (14.37)$$

$$\gamma_G = \alpha_G + \left[\frac{\beta}{(n-1)}\right]. \tag{14.38}$$

Example 14.4 illustrates the use of these equations.

The parameter γ_G is the thermo-optical coefficient for the glass that describes its sensitivity to spatial temperature variations. Jamieson¹⁰ indicated that γ_G for most optical glasses lies between 5×10⁻⁶/°C and 25×10⁻⁶/°C. Exceptions are fluor crown (FK) and phosphate crown (PK) glasses from Schott and Ohara, and some glasses from Hoya. Jamieson¹⁰ lists γ_G values for a variety of refractive materials. A few glasses with small or negative values are available. These reduce the sensitivity of lens systems to temperature

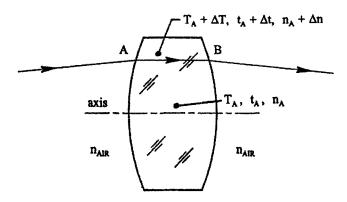


Figure 14.34 A generic radial temperature gradient in a lens element.

gradients. Optical plastics and some infrared-transmitting materials (notably germanium) have larger values of γ_G than glasses. The thermal conductivities and heat capacities of plastics are low, so these materials tend to be quite sensitive to spatial temperature gradients. Germanium has a high conductivity and heat capacity, so lenses made from that material might not suffer much when the temperature is not uniform. Germanium is, however, subject to thermal runaway. This means that the hotter it gets, the more it absorbs light. Pronounced transmission degradation starts at about 100°C and degrades rapidly between 200°C and 300°C. Absorption may result in catastrophic failure of the optic. Combinations of materials with high and low γ_G tend to reduce gradient sensitivity. Liquids are sometimes used to fill the air spaces between lenses to make the system more athermal.^{10,30}

Example 14.4: Estimation of effects of radial gradients in typical thin lenses. (For design and analysis, use File 14.4 of the CD-ROM.)

Thin lenses made of (a) BK-7 glass, SF11 glass, and germanium are all 3.500 mm (0.137 in.) thick and have radial temperature gradients causing the rim to be 2°C hotter than at the axis. Assume $\lambda = 0.546 \ \mu m$ for the glass lenses and 10.6 μm for the Ge lens. Let $n_{\lambda BK7} = 1.5187$, $n_{\lambda SF11} = 1.7919$, and $n_{\lambda Ge} = 4.0000$. What OPDs are created in each case?

From Jamieson¹⁰, $\gamma_{G BK7} = 9.87 \times 10^{-6} / ^{\circ}\text{C}$, $\gamma_{G SF11} = 20.21 \times 10^{-6} / ^{\circ}\text{C}$, and $\gamma_{G Ge} = 136.3 \times 10^{-6} / ^{\circ}\text{C}$ From Eq. (14.37), (a) $(1.5187 - 1)(9.87 \times 10^{-6})(3.500)(2) = 3.584 \times 10^{-5} \text{ mm} = 0.07 \lambda @ 0.546 \mu\text{m}$, (b) $(1.7919 - 1)(20.21 \times 10^{-6})(3.500)(2) = 1.12 \times 10^{-4} \text{ mm} = 0.20 \lambda @ 0.546 \mu\text{m}$, (c) $(4.000 - 1)(136.3 \times 10^{-6})(3.500)(2) = 2.86 \times 10^{-3} \text{ mm} = 0.27 \lambda @ 10.6 \mu\text{m}$. These OPDs are large enough to be of concern in many applications.

Jamieson¹⁰ further indicated that Eq. (14.37) and thin lens approximations are quite helpful in making general choices of optical materials for a preliminary design or in estimating the significances of anticipated temperature gradients, but is not sufficiently accurate for final design purposes. Final design requires that ray traces be conducted using realistic input temperature distributions to predict the index and thickness values as functions of zonal locations within the optic and their effects on image quality.

Because there is no refraction, given radial temperature gradients will affect reflecting optical components by changing the radii of optical surfaces and surface sagittal depths, i.e., optical figure, as functions of height from the axis. Lens design programs may evaluate the effects of these changes by considering the surfaces aspheric. The resulting effects on the image are easily determined using such programs.

14.5.2 Axial temperature gradients

An axial temperature gradient can be created in an optical component such as a window, lens, or prism by absorption of an incident heat flux, such as solar or laser radiation. That gradient will cause changes in bending of the optic. Barnes³¹ gave a classic treatment of thermal effects on space optics. With uniform axial irradiation, a plane parallel window becomes a shallow, concentric meniscus. Its mean curvature $C = 1/R = \alpha q/k$, where α is the material's linear thermal expansion coefficient, q the heat flux per unit area, and k the material's thermal conductivity. If the thickness t is small compared to R, the optical power P of this bowed window is given by:

$$P = \left[\frac{(n-1)}{n}\right] \left[t\frac{q}{k}\right]^2.$$
 (14.39)

With this equation, Barnes³¹ showed that, for an optical system at 300K in low earth orbit, the axial thermal gradient in a 2.5-cm (1.0-in.) thick crown glass window caused by the approximately 15% of incident solar radiation absorbed may be negligible for apertures smaller than 2.9 m (9.5 ft) since the focal shift introduced would be smaller than the Rayleigh $\lambda/4$ tolerance. This critical aperture varies inversely as the square root of the window thickness for a given absorbed heat flux.

Temperature gradients introduced through the edge mounting for a window or telescope corrector plate introduce differences in optical path length at various radial zones. This is due to changes in mechanical thickness of the optic as well as changes in the refractive index of the optical material. In general, stresses are built up in the glass and birefringence is introduced. These effects are small.

To illustrate the use of the analytical tools discussed in his paper, Barnes³¹ gave an example of an edge-insulated, single-glazed, nominally plane parallel crown glass window 3.0-cm (1.2-in.) thick and 61-cm (24-in.) in aperture. When used in an earth-oriented satellite at 960-km (600-mi) altitude, this window was found to become a shallow negative lens and to have an optical path difference distribution of zero on axis and at a zonal radius of 0.9, but a zonal aberration peaking at a zonal radius of 0.6 to 0.7 that was equivalent to approximately 0.5 wave p-v at visible wavelengths during operation. If used in an f/5 optical system of 55-cm (21.7-in.) aperture, this deformation would cause the system focus to shift about 42 μ m. This shift is nearly twice the Rayleigh guarter-wave tolerance for that system. The zonal aberration would reduce the system's performance even if it were to be refocused. Barnes concluded that the use of a window with an aperture significantly (about 25%) larger than that of this particular optical system would reduce the error to a much more tolerable magnitude without resorting to complex on-board thermal controls. Reducing the window thickness or decreasing the thermal coupling between the mount and the window by increasing the degree of thermal insulation provided would tend to reduce the effects of such a thermal gradient.

Vukobratovich¹¹ indicated that the change in curvature of a mirror if exposed to a steady-state linear axial thermal gradient is given by

$$\left(\frac{1}{R_0}\right) - \left(\frac{1}{R}\right) = \left(\frac{\alpha}{k}\right)q.$$
(14.40)

Here, R_0 and R are the original and new radii of curvature, respectively; α is the mirror material CTE, k is its thermal conductivity, and q is the heat flux absorbed per unit surface area. The ratio of (α/k) in Eq. (14.40) is the steady-state thermal distortion coefficient listed in Table B9 for a variety of mirror materials. A preferred material from the viewpoint of resistance to a thermal gradient has a low value for this coefficient.

14.6 Temperature Change-Induced Stresses in Bonded Optics

In Sections 7.5 and 9.2, we considered techniques for bonding prisms and small mirrors to mounts. There are three major sources of stress in the bonded joints between such optics and their mounts. These are shrinkage of the adhesive during curing, acceleration in a direction that tends to pull the optic from the mount or shear the joint, and differential expansion and contraction at high and low temperatures. The latter effect occurs in the joint between optical surfaces in cemented doublets and in multi-component (cemented) prisms. We will consider each of these factors briefly.

Shrinkage during curing typically amounts to a few percent of each dimension of the adhesive layer and may persist throughout the life of the device. Assuming that the material adheres well to both the optic and mount surfaces throughout the contact area, the adhesive layer and the adjacent surfaces of the optic and mount are somewhat stressed. This stress is usually small, but will tend to bend the optic. If the optic is too thin, this may change the figures of optical surfaces sufficiently to degrade performance. Corrective actions include making sure that the thickness of the optic is appropriately large, choosing an adhesive with minimal curing shrinkage, and minimizing the lateral dimensions of the bond. Using optical materials with high stiffness (i.e., large Young's modulus) also will help in the case of mirrors. In cemented optics, the size of the bond is usually determined by aperture requirements.

High acceleration directed normal to the bond joint and in a direction that places the joint in tension can cause sufficient force to break something. The strength of the adhesive joint (~2000 to ~2500 lb/in.²) often is greater than the tensile strength of the optical material (~1000 lb/in.²), so fracture of the latter can occur under high tensile stress. The worst situation would be when this happens at an extreme temperature so differential contraction or expansion of the materials and the effect of acceleration act together.

Temperature change-induced effects in joints between materials with CTEs α_1 and α_2 bonded with adhesive of CTE α_e result from a mismatch of component CTEs. In the common case with $\alpha_e \gg \alpha_1 > \alpha_2$, differential dimension changes in the two components introduces stress in all components. Fracture of bonded optical parts has, on occasion, been attributed to excessive shear forces exerted in the joint. Finite-element analysis methods can be used to predict thermally induced stress in the optic caused by this effect, but they are beyond the scope of this book.

Vukobratovich³² called this author's attention to an analytical method, developed by Chen and Nelson,³³ for estimating the shear stress developed in a thin bonded joint between two plates of dissimilar materials as a result of differential dimensional changes at

temperatures other than that at assembly. This theory can be applied to glass-to-metal or glass-to-glass bonds. The pertinent equations^a are as follows:

$$S_{s} = \frac{2(\alpha_{1} - \alpha_{2})(\Delta T)(S_{e})[I_{1}(x)]}{t_{e}\beta(C_{1} + C_{2})},$$
(14.41)

where

$$S_e = \frac{E_e}{(2)(1 + v_e)},$$
 (3.63)

$$\beta = \left\{ \left(\frac{S_e}{t_e} \right) \left[\frac{\left(1 - v_1^2 \right)}{E_1 t_1} + \frac{\left(1 - v_2^2 \right)}{E_2 t_2} \right] \right\}^{\frac{1}{2}}, \qquad (14.42)$$

ı

$$x = \beta R, \tag{14.43}$$

$$C_{1} = -\left[\frac{2}{(1+v_{1})}\right] \left\{ \left[\frac{(1-v_{1})I_{1}(x)}{x}\right] - I_{0}(x) \right\},$$
 (14.44)

$$C_{2} = -\left[\frac{2}{(1+v_{2})}\right] \left\{ \left[\frac{(1-v_{2})I_{1}(x)}{x}\right] - I_{0}(x) \right\}.$$
 (14.45)

Here, S_S is the shear stress in the joint, α_1 and α_2 are the CTEs of the two bonded components, ΔT is the temperature change from assembly temperature, S_e is the shear modulus of the adhesive, R is one half the lateral dimension of the bond (here assumed circular), t_e is the thickness of the adhesive layer, E_1 , v_1 , E_2 , v_2 , E_e , and v_e are the Young's modulus and Poisson's ratio values for the three materials, t_1 and t_2 are the thicknesses of the components, and $I_0(x)$ and $I_1(x)$ are modified Bessel functions of the first kind. The latter functions are plotted in Fig. 14.35 over the range 0 < x < 5.0. In Examples 14.5 through 14.7, $I_0(x)$ and $I_1(x)$ are estimated from this figure. For larger values of x, and in the CD-ROM that accompanies this book, $I_0(x)$ and $I_1(x)$ are calculated from the following polynomials:

$$I_0(x) = a_0 + b_0 x^2 + c_0 x^4 + d_0 x^6 + e_0 x^8 + f_0 x^{10}, \qquad (14.46)$$

$$I_1(x) = a_1 x + b_1 x^3 + c_1 x^5 + d_1 x^7 + e_1 x^9 + f_1 x^{11}, \qquad (14.47)$$

The constants in each equation are as listed in Table 14.8.

^a In the first edition of this book, the treatment of this subject was based on Chen and Nelson's theory for shear stress along a single axis. We here apply their equations for the maximum axisymmetric stress in two circular plates bonded together with a thin adhesive layer filling the space between the plates. This stress is zero on axis and peaks at the rim. The plates are assumed not to bend.

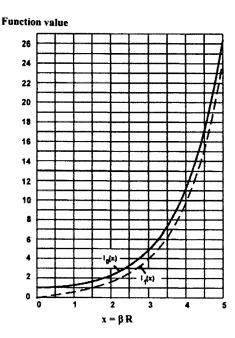


Figure 14.35 Variations of the modified Bessel functions of the first kind $I_0(x)$ and $I_1(x)$ over the range 0 < x < 5. Data from this graph is utilized in Example 14.5, which illustrates an estimation of shear stress in the bonded joint between a baseplate of metal and a glass prism that have widely differing CTEs.

Table 14.8 Values of the constants $a_0 \dots f_0$ and $a_1 \dots f_1$ used in Eqs. (14.46) and (14.47) for $l_0(x)$ and $l_1(x)$.

a_0	1.00000E-00	a_1	5.00000E01
b_0	2.50000E-01	b_1	6.25000E-02
<i>C</i> ₀	1.56250E-02	c_1	2.60417E-03
d_0	4.27350E-04	d_1	5.42535E05
e_0	6.78168E-06	e_1	6.78168E-07
f_0	1.17738E-10	f_1	5.65140E-09

The stress in the failed bond for the prism/mount subassembly of Fig. 7.21 is estimated from Equations 14.41 through 14.45 in part (a) of Example 14.5 as ~1205 lb/in². This exceeds the tensile stress tolerance for glass (1000 lb/in.²) as defined in Chapter 13. Hence, there is some risk that the prism might break at extreme temperatures—especially if the prism surface has not been processed by controlled grinding to remove subsurface damage. As indicated in the prior discussion of the prism mounting design, the glass prism cracked during low temperature testing. Corrective action was to reduce the size of the bond, dividing it into three spots arranged in an equilateral triangle plus a single spot in the center of that pattern. These spots were 0.250-in. (6.350-mm) in diameter. In part (b) of Example 14.5, the stress in a bond of this size at T_{MIN} would be reduced to ~324 lb/in.². This stress should not cause failure. Low temperature tests of bonded subassemblies with these smaller bonds showed the new design to be successful.

Example 14.5: Stress in a bonded prism joint caused by differential thermal expansion. (For design and analysis, use File 14.5 of the CD-ROM.)

The cube-shaped prism shown in Fig. 7.21 is made of fused silica and is bonded with 3M 2216 epoxy to a titanium base. The face width of the prism is 1.378 in. (35.000 mm). The base is 1.051-in. (26.695-mm) thick. The bond is circular, 0.004-in. (0.102-mm) thick, and has a diameter 2R of 1.378 in. (35.000 mm). (a) Assuming that the shear stress in the bond equals the stress applied to the prism, what is that stress as a result of a temperature change ΔT of -90° F? (b) Let this bond be replaced by four equal areas of diameters 2R = 0.250 in. (6.350 mm) arranged in an equilateral triangle plus one in the center. Estimate the stresses in these smaller bonds for ΔT of -90° F.

From Tables B1, B12, and B14:
$$\alpha_M = 4.90 \times 10^{-6} / {}^{\circ}F$$
 $\alpha_G = 0.32 \times 10^{-6} / {}^{\circ}F$ $E_M = 16.5 \times 10^6 \, lb/in.^2$ $E_G = 10.6 \times 10^6 \, lb/in.^2$ $E_e = 1.00 \times 10^5 \, lb/in.^2$ $\upsilon_M = 0.310$ $\upsilon_G = 0.170$ $\upsilon_e = 0.430$

From Eq. (3.63):
$$S_e = \frac{1.00 \times 10^3}{(2)(1+0.43)} = 3.497 \times 10^4 \, \text{lb/in.}^2$$

From Eq. (14.42):

$$\beta = \left\{ \left(\frac{3.497 \times 10^4}{0.004} \right) \left[\frac{1 - 0.310^2}{\left(16.5 \times 10^6 \right) \left(1.051 \right)} + \frac{\left(1 - 0.170^2 \right)}{\left(10.6 \times 10^6 \right) \left(1.378 \right)} \right] \right\}^{\frac{1}{2}} = 1.018 \text{ in.}^{-1}$$

(a) From Fig. 14.35 at $x = \beta R = (1.018)(1.378/2) = 0.701$, $I_0(x) = 1.10$, and $I_1(x) = 0.40$, From Eqs. (14.44) and (14.45):

1

$$C_{M} = -\left[\frac{2}{1+0.310}\right] \left[\frac{(1-0.310)(0.4)}{0.701} - 1.1\right] = -1.078 \text{ in.}^{-1}$$
$$C_{G} = -\left[\frac{2}{1+0.170}\right] \left[\frac{(1-0.170)(0.4)}{0.701} - 1.1\right] = -1.071 \text{ in.}^{-1}$$

From Eq. (14.41):

$$S_{S} = \frac{(2)(4.90 \times 10^{-6} - 0.32 \times 10^{-6})(-90)(3.497 \times 10^{4})(0.4)}{[-1.178 + (-1.171)](0.004)(1.018)} = -1205 \text{ lb/in.}^{2}(-8.31\text{MPa})$$

(b) From Fig. 14.35 at
$$x = \beta R = (1.018)(0.250/2) = 0.127$$
, $I_0(x) = 1.01$ and $I_1(x) = 0.08$,
From Eqs. (14.44) and (14.45):
 $\begin{bmatrix} 2 \\ -2 \end{bmatrix} \begin{bmatrix} (1-0.310)(0.08) \end{bmatrix}$

$$C_{M} = -\left[\frac{2}{(1+0.310)}\right] \left[\frac{(1-0.170)(0.08)}{0.127} - 1.01\right] = -0.894 \text{ in.}^{-1}$$
$$C_{G} = -\left[\frac{2}{1+0.170}\right] \left[\frac{(1-0.170)(0.08)}{0.127} - 1.01\right] = -0.853 \text{ in.}^{-1}$$

From Eq. (14.41):

$$S_{s} = \frac{(2)(4.90 \times 10^{-6} - 0.32 \times 10^{-6})(-90)(3.497 \times 10^{4})(0.08)}{[-0.894 + (-0.853)](0.004)(1.018)} = 324 \text{ lb/in.}^{2} (2.23 \text{MPa})$$

A common application of Chen and Nelson's theory³³ is for cemented doublet lenses involving glasses with significantly different CTEs. A case in point is the 90-mm diameter doublet shown schematically in Fig. 14.36(a). The lens design called for the use of Schott FK51 crown glass and KzFS7 flint glass for optical performance reasons. Concern was expressed, however, that there would be a problem at the specified low survival temperature of -80° F because the CTEs of $7.389 \times 10^{-6}/^{\circ}$ F and $2.722 \times 10^{-6}/^{\circ}$ F differ considerably. At that time (1970), no analytical method was available to check the design. Rather than to risk making the required number of doublets and having them fail during testing, a less expensive model was made. Two plane-parallel plates of the chosen glasses with thicknesses representative of the lens elements were made and cemented together. See Figure 14.36(b). Figure 14.37 shows what happened during cooling towards the lowest temperature. Both plates were damaged during the test.

Rather than redesign the optical system to use glasses with more nearly equal CTEs, a search was initiated to identify a more flexible adhesive that might solve the problem. Sylgard XR-63-489 (a conformal coating for electronic circuit boards formerly made by Dow Corning) was chosen because it was sufficiently transparent for the application, it cured to a softer bond than conventional optical cement, and it could be used as a thicker layer. Low-temperature tests with additional bonded plates indicated that this approach would be satisfactory so production of the doublets proceeded without further delay.

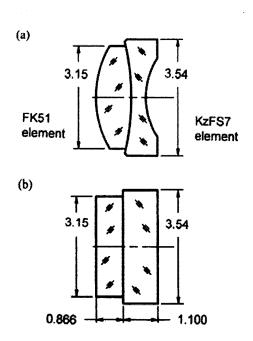


Figure 14.36 Schematic diagrams of (a) the configuration of a cemented doublet with widely differing glass CTEs, and (b) the two-plate model created for low-temperature tests.

EFFECTS OF TEMPERATURE CHANGES

Example 14.6 shows how shear stress in the original design (with optical cement) can now be estimated using Chen and Nelson's theory. Assumptions are made as to the Young's modulus, Poisson's ratio, and bond thickness for the optical cement because that data is not readily available. Dimensions are taken from Fig. 14.36(b). The stress for a ΔT of -148° F from assembly temperature is estimated as ~8531 lb/in.² (58.8 MPa). The lens would be expected to fail long before it reaches -80°F. In this example, we interpolate the values for $I_0(x)$ and $I_1(x)$ from Figure 14.33. Using File 14.6 on the CD-ROM, we estimate this stress at minimum temperature as ~8400 lb/in.² These results agree well. It is further a simple exercise to find the temperature change that reduces the stress to 1000 lb/in.². With a few manual successive approximations using File 14.6 on the CD-ROM, we find that this ΔT is approximately -18° F.



Figure 14.37 Photograph of a pair of thick glass plates optically cemented together to simulate a cemented doublet lens and then subjected to low temperature test. Fractures occurred in both plates because of the widely differing CTEs of the glasses. (From Yoder.¹⁷ Copyright 2005, Taylor and Francis Group, LLC, a division of Informa, plc. Reprinted with permission.)

This same procedure can be used to analyze other cemented-doublet lenses. Let us consider the design shown in Figure 14.38. It is typical of what might be employed as a 7×50 binocular objective. Its dimensions and the glass types are indicated in the figure. Although little, if any, consideration would normally be given during design to differential expansion and contraction effects, these glasses have nearly equal CTEs of 4.3×10^{-6} lb/in.² and 4.6×10^{-6} lb/in.². Example 14.7 estimates the shear stress in the bond at -80°F to be 279 lb/in.² (1.93 MPa). This stress would generally be considered acceptable for the application.

Example 14.6: Stress in a cemented doublet caused by differential thermal expansion with widely differing glass CTEs. (For design and analysis, use File 14.6 of the CD-ROM.)

The two-plate thermal test model of a cemented doublet shown in Fig. 14.36(b) is made of FK51 and KzFS7 glasses. It is cemented with optical cement. The diameter of the bond (2*R*) is 3.150 in. (80.0 mm). The thicknesses of the crown and flint plates are 0.866 in. (21.996 mm) and 1.100 in. (27.940 mm) respectively. What shear stress develops in the bond because of a temperature change ΔT of -148° F?

From the 1992 Schott catalog:

 $\begin{array}{ll} \alpha_{G1} = 7.39 \times 10^{-6} / \,^{\circ}\text{F}, & \alpha_{G2} = 2.72 \times 10^{-6} / \,^{\circ}\text{F}, \\ E_{G1} = 1.175 \times 10^{7} \, \text{lb/in.}^{2}, & E_{G2} = 9.86 \times 10^{6} \, \text{lb/in.}^{2}, \\ \upsilon_{G1} = 0.274, & \upsilon_{G2} = 0.293. \end{array}$

Assume: $E_e = 1.6 \times 10^5 \text{ lb/in.}^2$ $v_e = 0.430$, and $t_e = 0.001 \text{ in.} (0.025 \text{ mm})$. From Eq. (3.63): $S_e = \frac{1.6 \times 10^5}{(2)(1+0.430)} = 5.594 \times 10^4 \text{ lb/in.}^2$.

From Eq. (14.42):

$$\beta = \left\{ \left(\frac{5.594 \times 10^4}{0.001} \right) \left[\frac{\left(1 - 0.274^2 \right)}{\left(1.175 \times 10^7 \right) \left(0.866 \right)} + \frac{\left(1 - 0.293^2 \right)}{\left(9.860 \times 10^6 \right) \left(1.100 \right)} \right] \right\}^{\frac{1}{2}} = 3.130 \text{ in.}^{-1}.$$

From Fig. 14.35 at $x = \beta R = (3.130)(3.150/2) = 4.930$, $I_0(x) = -24.5$ and $I_1(x) = -22.8$.

From Eqs. (14.44) and (14.45):

$$C_{G1} = -\left[\frac{2}{(1+0.274)}\right] \left[\frac{(1-0.274)(22.8)}{4.930} - 24.5\right] = 33.191 \text{ in.}^{-1},$$

$$C_{G2} = -\left[\frac{2}{(1+0.293)}\right] \left[\frac{(1-0.293)(22.8)}{4.930} - 24.5\right] = 32.839 \text{ in.}^{-1}.$$
Eq. (14.41):

$$S_{s} = \frac{(2)(7.390 \times 10^{-6} - 2.72 \times 10^{-6})(-148)(5.594 \times 10^{4})(22.8)}{(33.191 + 32.839)(0.001)(3.130)} = 8531 \text{ lb/in.}^{2} (58.8 \text{ MPa})$$

It might be interesting to see how shear stress depends on bond size. An approach would be to scale the dimensions of a given lens design by various factors and calculate S_S for each case. The materials, bond thickness, and ΔT remain constant. The CD-ROM accompanying this book is particularly useful for this type of parametric analysis. Using the nominal design of the binocular objective from Fig. 14.38 and Example 14.7 as the starting points, the shear stress S_S in the bond has been determined for scale factors

ranging from 0.5 to 2.0. The results are plotted in Figure 14.39. We see that the stress generally increases with bond diameter and the variation is nonlinear with scale factor. This, of course, is only one specific design. Others may behave differently.

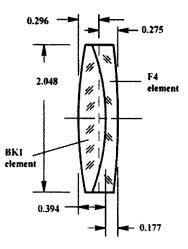


Fig. 14.38 Schematic of a generic cemented doublet lens that might be used in a 7×50 binocular objective. The glasses have nearly equal CTEs. The lowtemperature stress in the bond is estimated in Example 14.7.

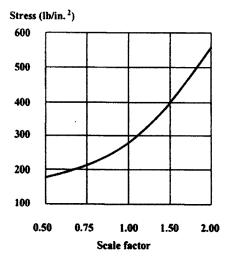


Figure 14.39 The variation of shear stress in the bond of a family of lenses per Fig. 14.38 scaled down and up from the nominal design without change in materials, bond thickness, or ΔT .

A detail that warrants explanation is how the thicknesses $[t_1 \text{ and } t_2 \text{ in Eq. (14.42)}]$ of the two glass components in a cemented doublet are determined. For lack of a technical basis for this choice, this author suggests that these dimensions be the axial distances from either vertex to the midpoint of the sagittal depth of the buried surface. In Figure 14.38, those dimensions are 0.296 and 0.275 in. and are measured to the dashed line. They enter into the calculation of β .

Example 14.7: Stress in the bond of a cemented doublet caused by differential thermal expansion with nearly equal glass CTEs. (For design and analysis, use File 14.7 of the CD-ROM.) From the 1992 Schott catalog for BK1 and F4 glasses: $\alpha_{G1} = 4.3 \times 10^{-6} / {}^{\circ}\text{F},$ $\alpha_{G2} = 4.6 \times 10^{-6} / {}^{\circ}\text{F},$ $E_{G1} = 1.07 \times 10^{7} \text{ lb/in.}^{2},$ $E_{G2} = 7.98 \times 10^{6} \text{ lb/in.}^{2},$ $v_M = 0.210$. $E_e = 1.6 \times 10^5 \text{ lb/in.}^2$ $v_e = 0.430$, and $t_e = 0.001$ in. (0.025 mm), Assume: From Eq. (3.63): $S_e = \frac{1.6 \times 10^5}{(2)(1+0.430)} = 5.594 \times 10^4 \, \text{lb/in.}^2$. From Eq. (14.42): $\beta = \left\{ \left(\frac{5.594 \times 10^4}{0.001} \right) \left[\frac{\left(1 - 0.210^2 \right)}{\left(1.07 \times 10^7 \right) \left(0.296 \right)} + \frac{\left(1 - 0.225^2 \right)}{\left(7.98 \times 10^6 \right) \left(0.275 \right)} \right] \right\}^{\frac{1}{2}} = 6.410 \text{ in.}^{-1}.$ From Fig. 14.35 at $x = \beta R = (6.410)(2.048/2) = 6.563$, $I_0(x) = 98.66$ and $I_1(x) = 102.38$. From Eqs. (14.44) and (14.45): $C_{G1} = -\left[\frac{2}{(1+0.210)}\right]\left[\frac{(1-0.210)(102.38)}{6.563} - 98.66\right] = 142.70 \text{ in.}^{-1},$ $C_{G2} = -\left[\frac{2}{(1+0.225)}\right] \left[\frac{(1-0.225)(102.38)}{6.563} - 98.66\right] = 141.34 \text{ in.}^{-1}.$ From Eq. (14.41): $S_{s} = \frac{(2)(4.3 \times 10^{-6} - 4.6 \times 10^{-6})(-148)(5.594 \times 10^{4})(102.38)}{(0.001)(6.410)[142.70 + 141.34]} = 279.3 \text{ lb/in.}^{2}(1.93\text{MPa})$

A few words of caution are appropriate before we leave this subject. Chen and Nelson's method³³ has recently been found to be incomplete and is not rigorous.³⁴ Furthermore, the shear stress estimated by this method does *not* refer to the tensile stress in the optic. Application of the "rule-of-thumb" 1000 lb/in.² (6.9 MPa) tolerance from Chapter 13 to the results obtained by the equations presented here is therefore not strictly

appropriate. There is evidence based upon FEA modeling³⁵ that the tensile stress in the glass may, in some cases, be significantly different from the shear stress in the bonded glass interface.

Although approximate and not verified by alternative analytical methods or controlled experiments, the analytical methods presented in this section are potentially useful tools for preliminary evaluation of proposed designs of subassemblies with glass-to-glass and glass-to-metal joints involving materials with widely differing CTEs. To provide confidence in such new designs, subassemblies should be thoroughly tested at the specified extreme low and high temperatures. This is particularly important for hardware that is to be exposed to harsh military or aerospace environments. In most cases, testing can be done at minimal cost early in the design phase using surrogate components, such as those illustrated in Fig. 14.36(b).

14.7 References

- 1. Schreibman, M., and Young, P., "Design of Infrared Astronomical Satellite (IRAS) primary mirror mounts," *Proceedings of SPIE* **250**, 1980:50.
- Young, P., and Schreibman, M., "Alignment design for a cryogenic telescope," Proceedings of SPIE 251, 1980:171.
- 3. Erickson, D.J., Johnston, R.A., and Hull, A.B., "Optimization of the opto-mechanical interface employing diamond machining in a concurrent engineering environment," *Proceedings of SPIE* CR43, 1992:329.
- 4. Hookman, R., "Design of the GOES telescope secondary mirror mounting," *Proceedings of SPIE* **1167**, 1989:368.
- Zurmehly, G.E., and Hookman, R., "Thermal/optical test setup for the Geostationary Operational Environmental Satellite Telescope," *Proceedings of SPIE* 1167, 1989:360.
- 6. Golden, C.T. and Speare, E.E., "requirements and design of the graphite/epoxy structural elements for the Optical Telescope Assembly of the Space Telescope," *Proceedings of AIAA/SPIE/OSA Technology Space Astronautics Conference: The Next 30 Years*, Danbury, CT, 1982:144.
- 7. McCarthy, D.J. and Facey, T.A., "Design and fabrication of the NASA 2.4-Meter Space Telescope, *Proceedings of SPIE* **330**, 1982:139.
- 8. Edlen, B., "The Dispersion of Standard Air," J. Opt. Soc. Am., 43, 339, 1953.
- Penndorf, R., "Tables of the Refractive Index for Standard Air and the Rayleigh Scattering Coefficient for the Spectral Region between 0.2 and 20 μ and their Application to Atmospheric Optics," J. Opt. Soc. Am. 47, 1957:176.
- 10. Jamieson, T.H., "Athermalization of optical instruments from the optomechanical viewpoint," *Proceedings of SPIE* CR43, 1992:131.
- Vukobratovich, D., "Optomechanical Systems Design," Chapter 3 in *The Infrared & Electro-Optical Systems Handbook*, 4, ERIM, Ann Arbor and SPIE, Bellingham, WA, 1993.
- 12. Povey, V. "Athermalization techniques in infrared systems," *Proceedings of SPIE* 655, 1986:563.
- Ford, V.G., White, M.L., Hochberg, E., and McGown, J., "Optomechanical design of nine cameras for the Earth Observing System Multi-Angle Imaging Spectro-Radiometer, TERRA Platform," *Proceedings of SPIE* 3786, 1999:264.

- 14. Friedman, I., "Thermo-optical analysis of two long-focal-length aerial reconnaissance lenses," *Opt. Eng.* 20, 1981:161.
- 15. Fischer, R.E., and Kampe, T.U., "Actively controlled 5:1 afocal zoom attachment for common module FLIR," *Proceedings of SPIE* **1690**, 1992:137.
- 16. Yoder, P.R., Jr., "Advanced considerations of the lens-to-mount interface," *Proceedings of SPIE* CR43, 1992:305.
- 17. Yoder, P.R., Jr., *Opto-Mechanical Systems Design*, 3rd ed., CRC Press, Boca Raton, 2005.
- 18. Lecuyer, J.G., "Maintaining optical integrity in a high-shock environment," *Proceedings of SPIE* **250**, 1980:45.
- Fischer, R.E., Tadic-Galeb, B., and Yoder, P.R., Jr., *Optical System Design*, 2nd ed., McGraw-Hill, New York and SPIE Press, Bellingham, WA, 2008.
- 20. Genberg, V.L., private communication, 2004.
- 21. Yoder, P.R. Jr., and Hatheway, A.E., "Further considerations of axial preload variations with temperature and the resultant effects on contact stresses in simple lens mountings, *Proceedings of SPIE* **5877**, 2005.
- 22. Yoder, P.R., Jr., "Parametric investigations of mounting-induced contact stresses in individual lenses," *Proceedings of SPIE* **1998**, 1993:8.
- 23. Yoder, P.R., Jr., "Estimation of mounting-induced axial contact stresses in multielement lens assemblies," *Proceedings of SPIE* **2263**, 1994:332.
- 24. Yoder, P.R., Jr., *Mounting Optics in Optical Instruments*, SPIE Press, Bellingham, WA, 2002.
- 25. Young, W.C., *Roark's Formulas for Stress and Strain*, 6th. ed., McGraw-Hill, New York, 1989.
- 26. Stevanovic, D. and Hart, J., "Cryogenic mechanical design of the Gemini south adaptive optics imager (GCAOI)," *Proceedings of SPIE* **5495**, 2004:305.
- 27. Barkhouser, R.H., Smee, S.A., and Meixner, M., "optical and optomechanical design of the WIYN high resolution infrared camera, *Proceedings of SPIE* **5492**, 2004::921.
- 28. Stubbs, D.M. and Hsu, I.C., "Rapid cooled lens cell," *Proceedings of SPIE* **1533**, 1991:36.
- 29. Hatheway, A.E., "Thermo-elastic stability of an argon laser cavity, *Proceedings of SPIE* **4198**, 2000:141.
- 30. Andersen, T.B., "Multiple-temperature lens design optimization," *Proceedings of SPIE* **2000**, 1993:2.
- 31. Barnes, W.P., Jr., "Some effects of aerospace thermal environments on high-acuity optical systems, *Appl. Opt.* 5, 1996:701.
- 32. Vukobratovich, D, private communication, 2001.
- 33. Chen, W.T. and Nelson, C.W., "Thermal stress in bonded joints," *IBM J. Res. Develop.* 23, 1979:179.
- 34. Hatheway, A.E., Alson E. Hatheway, Inc., private communication, 2008.
- 35. Barney, S., Lockheed-Martin Missiles and Fire Control, private communication, 2008.

CHAPTER 15 Hardware Examples

In this chapter are found descriptions and illustrations of twenty examples of optical hardware involving a variety of mountings for simple and complex lenses, catadioptric systems, and prisms, as well as for mirrors and gratings. Concepts and design features described earlier in this book are frequently revisited here. Many of these examples are described in more detail and in context with their applications in other publications. References are provided so the reader can explore those resources for items of particular interest.

15.1 Infrared Sensor Lens Assembly

The optomechanical configuration of a 2.717-in. (69-mm) focal length, f/0.87 objective assembly is illustrated in Fig. 15.1. The singlet lens is silicon, while the first element of the cemented doublet has silicon and sapphire elements. Wedge angles of the flat bevels on the concave faces of the lenses are held to 10 arcsec for the singlet and 30 arcsec for the doublet to ensure good centration to the assembly's mechanical axis.

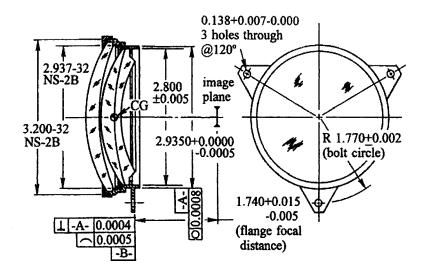


Figure 15.1 Sectional and frontal views of a triplet infrared sensor assembly. Dimensions are in inches. (Courtesy of Goodrich Corporation, Danbury, CT.)

The lens cell is made of Invar 36 and is stabilized after rough machining by repeated cycling between 320°F (160°C) and room temperature. The registering OD (-A-) and flange-ear mounting interface surfaces (-B-) are closely toleranced for diameter and perpendicularity to the optical axis, respectively, to ensure precise alignment with related components of the optical system. The lenses are lathe assembled with 0.0002 in. (0.0051 mm) maximum clearances in the cell and constrained axially by threaded 303 CRES retainers. Prior to tightening the retainers, the lenses are differentially rotated (phased) about the axis to maximize symmetry of the axial image and to minimize decentration of

that image relative to the OD (-A-). Final image quality is measured in terms of percent energy concentration of radiation from a collimated infrared source passing through a specific (small) on-axis aperture located at the lens system focus relative to the total energy received at that focus.

15.2 A Family of Commercial Mid-Infrared Lenses

Figure 15.2 is a photograph of a set of four f/2.3 lens assemblies intended for use with standard commercial infrared cameras in a variety of applications.¹ They are designed to operate to near the diffraction limit in the 3- to 5- μ m spectral region and have the optical and mechanical characteristics listed in Table 15.1.



Figure 15.2 Photograph of four *f*/2.3 commercial lens assemblies with focal lengths of 13 to 100 mm designed for the mid-infrared region. (Courtesy of Janos Technology, Inc., Keene, NH.)

Table 15.1 Characteristics of the f/2.3 commercial mid-IR lenses shown in Fig. 15.2.

Focal length (mm)	Field of View (deg)	Length (mm)	Diameter (mm)	Weight (oz)
13	±38.9	46.8	57.1	< 8
25	±22.8	46.8	57.1	< 8
50	±11.8	46.8	61.9	< 7.5
100	±6.0	107.6	117.3	< 31

Data courtesy of Janos Technology, Inc., Keene, NH.

Typical construction of the assemblies is indicated by the sectional view shown in Fig. 15.3. The mechanical parts are 6061-T6 aluminum and the lenses are silicon and germanium. An IR-pass filter, cold stop, window, and detector array are contained in a separate dewar furnished by the user. Each lens is held in place with GE RTV Type 655 sealant. During assembly, the applicable areas of the cell and the rims of each lens are primed with GE primer SS4155 to facilitate adhesion. Special care is exercised in applying the primer to the lenses since it can damage the polished surfaces if they are accidentally contacted. The lenses are then installed with their flat bevels contacting the shoulders provided in the cell and shimmed to center them mechanically within ± 30 µm relative to the mount axis. Once aligned, the RTV is applied with a fluid dispensing system and cured according to the manufacturer's directions. A retaining ring is then installed. It does not apply significant preload to the lens, but serves as a convenient location for lens identification information.

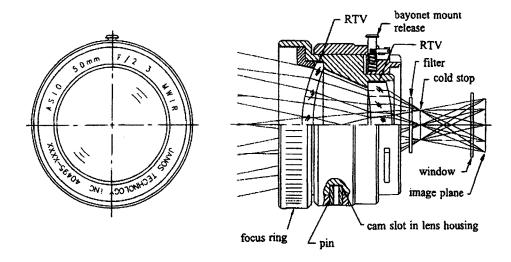


Figure 15.3 Section view through one of the lenses shown in Fig. 15.2. (Courtesy of Janos Technology, Inc., Keene, NH.)

The lens housing attaches to the camera through a bayonet connection. The release mechanism for that connection is shown in the figure. Rotating the knurled ring at the front focuses the lens. This rotates the lens housing within the fixed mount body and drives the lenses axially by virtue of a helical cam slot in the lens cell that engages a brass pin fixed in the body. The object space range, as determined by image quality considerations, is from infinity to 50, 150, 425, and 1750 mm respectively for the 13-, 25-, 50-, and 100-mm EFL types. Focus can be clamped with a soft-tip setscrew (not shown).

15.3 Using SPDT to Mount and Align Poker Chip Subassemblies

Extremely high dimensional and alignment precision can be achieved by applying singlepoint diamond turning (SPDT) methods when mounting a lens. In the SPDT process,*

See Section 10.1.

extremely fine cuts are taken on the surface in work using a specially prepared and oriented diamond crystal as the cutting tool. The work piece is supported and rotated on a highly precise spindle (air or hydrostatic bearing). The tool is moved slowly across the surface on highly precise linear or rotary stages. Real time interferometric control systems are used to ensure location and orientation accuracy of the cutting tool at all times. Useful details regarding the SPDT process as employed for assembling a lens into its mount are provided in Erickson et al.,² Rhorer and Evans,³ and Arriola.⁴

As an example of the use of the SPDT technology in assembling, aligning, and finishmachining of a typical lens/cell subassembly, consider the case of creating a "poker chip" module^{*} using a meniscus-shaped BK7 lens that has a minimum clear aperture diameter of 3.000 in. (76.200 mm), axial thickness of 0.667 ± 0.004 in. (16.942 ± 0.102 mm), and radii of 6.375 ± 0.001 in. (161.925 ± 0.025 mm) and 10.200 ± 0.002 in. (259.080 ± 0.050 mm). The lens is to be mounted athermally in a 6061 aluminum cell by the elastomeric "potting" method described in Section 3.9. The cell is to be machined so its OD is concentric with the optical axis within 0.0005 in. (0.012 mm) and parallel to that axis within 10 arc seconds. The axial thickness of the cell is to be 1.1510 ± 0.0002 in. (29.2354 ± 0.0051 mm) and the front and back surfaces of the cell are to be parallel within 10 arc seconds. The OD of the cell is to be 4.0000 ± 0.0002 in. (101.6000 ± 0.0051 mm) and concentric to the optical axis within 0.0050 in. (0.0125 mm). The desired modular subassembly is illustrated in Fig. 15.4.

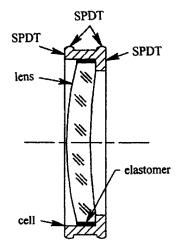


Figure 15.4 The poker chip module described in the text. Surfaces to be machined by SPDT techniques are indicated.

Key to success in the process described here is the use of a centering chuck such as that shown in Fig. 15.5. This device is usually made of brass because it can be SPDT machined very easily to precise dimensions. The surfaces to be made in this manner are indicated in the figure. They can all be cut in the same set-up, thus ensuring high accuracy relative to each other. Other surfaces require only conventional machining. The conical interface is cut to the proper angle to interface well with the convex surface of the lens.

^{*} Applications and alignment of the "poker chip" module are discussed in Sections 4.5 and 12.2.

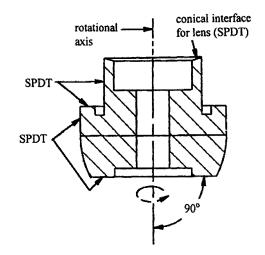


Figure 15.5 A centering chuck tool designed to interface a lens to the SPDT machine. Surfaces indicated by SPDT are machined in the same setup for maximum precision.

The centering chuck is made to fit snugly into a receptacle on a base plate attached to the SPDT machine spindle. It seats against a surface having air path recesses for applying a vacuum to secure the chuck in place. The finished lens is attached to the chuck with blocking wax (see Fig. 15.6). The lens is moved laterally to center its axis to the spindle axis before the wax solidifies (or the adhesive is cured). Initial alignment can be accomplished mechanically using precision indicators and then finalized using interferometric means, such as the Fizeau technique described in Section 12.1.2. Measurement of the vertex distance indicated in the figure provides information to be used in establishing the axial location of a machined surface on the cell in a later step.

The cell into which the lens is to be mounted is placed over the rim of the lens as indicated in Fig. 15.7. This cell has finished dimensions on all surfaces except those to be finished by SPDT. The cell is aligned mechanically to the spindle axis and waxed to the lens as shown.

The next steps in the process are to remove the chuck-and-lens subassembly from the spindle, invert it onto a horizontal surface, as shown in Fig. 15.8, and inject elastomer (typically RTV) into the annular cavity between the lens OD and the cell ID. Four radially directed holes are used for this purpose to ensure complete filling of the cavity. Note that this operation cannot be accomplished without inverting the subassembly because the elastomer must be constrained by gravity while curing. The use of the removable chuck allows the SPDT machine to be used for other purposes while the elastomer cures.

After the elastomer has completely cured, the subassembly is returned to the spindle base plate and the exposed cell surfaces are turned to final dimensions (see Fig. 15.9). It would be advisable at the time to verify the centration of the lens interferometrically before the subassembly is removed from the chuck. Removal is accomplished by heating gently to melt the wax. Finally, the subassembly is cleaned, inspected, and bagged for future use.

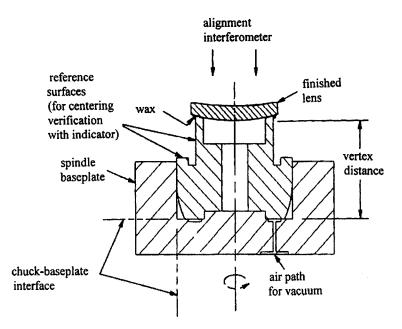


Figure 15.6 The centering chuck installed on the SPDT machine spindle with the lens waxed in place and centered interferometrically.

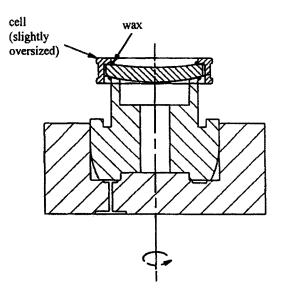


Figure 15.7 A partially machined cell centered to spindle axis on top of the lens and attached to the lens with wax.

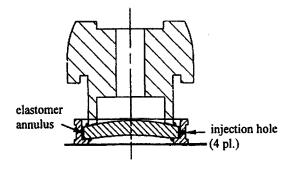
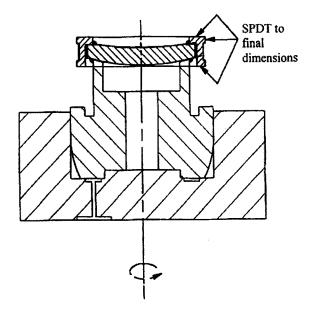
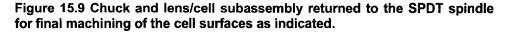


Figure 15.8 Chuck and lens removed from spindle and inverted onto a horizontal surface for injection of elastomer.





If the lens material itself is compatible with single-point diamond turning (SPDT) processing, the finished surfaces can be processed by SPDT methods. These materials are most generally crystals used in infrared applications and a few plastics. See Table 15.2. Very important advantages of the process as compared with traditional grinding and polishing methods for optical component surface finishing are (1) a rough lens blank can be mounted on the bell of the SPDT machine, shaped, and finished; (2) the first optical surface can be registered to the spindle axis on the bell so the second refracting surface, the rim, and bevels can be machined accurately with minimal wedge and centering error; (3) the ability to create aspheric surfaces on one or both lens surfaces; and (4) the speed with which optical and non-optical surfaces can be created.

Cadmium telluride	КТР	Strontium fluoride
Calcium fluoride	Magnesium fluoride	Zinc selenide
Gallium arsenide	Silicon*	Zinc sulfide
Germanium	Sodium chloride	
KDP	Sodium fluoride	

Table 15.2 Crystalline materials	that can	be machined	to optical finish by
SPDT techniques.			

* This material causes rapid diamond tool wear

To illustrate the basic steps of this process, Figure 15.10 shows a cylindrical blank of a suitable optical material attached with blocking wax to the bell on the spindle of a SPDT machine. The outline of the desired finished lens is indicated in the figure. The convex surface of the lens (dashed line) is machined to the proper radius and surface finish. The blank is then removed from the spindle and reattached with wax to the bell as indicated in Fig. 15.11. The concave surface, the lens rim, and both bevels can then be shaped and finished.[†] The lens is then inspected for the required centration and surface quality. The completed lens is then removed from the bell, cleaned, inspected again, and bagged.

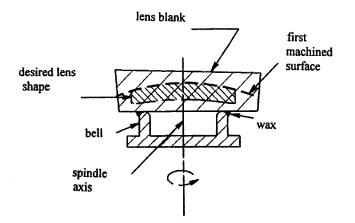


Figure 15.10 A crystalline lens blank mounted on the bell of a SPDT machine spindle for shaping and finishing the first lens surface.

[†] Using a multiple-axis SPDT instrument, such as that shown in Fig. 10.7, aspheric surfaces can be produced with high precision.

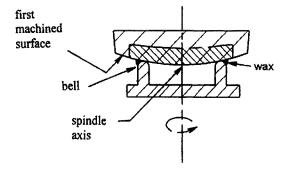


Figure 15.11 The partially machined lens from Fig. 15.10 mounted on the bell of a SPDT machine for shaping and finishing the second lens surface, the rim, and bevels.

15.4 A Dual-Field IR Tracker Assembly

Guyer et al.⁶ described the use of SPDT-made crystalline lenses in a cryogenically cooled dual-field infrared-imaging missile tracker operating in the 4- to 5- μ m spectral range. The optical system of the tracker is illustrated in Fig. 15.12. It was designed to be nearly athermal by distributing optical power to utilize the temperature variations of material properties and dimensions favorably and keep optical performance above the diffraction limit. The system included a magnification changer subsystem that rotated about a transverse axis to change the field of view between the acquisition and tracking modes. A rotary solenoid drove the changer from one position to the other.

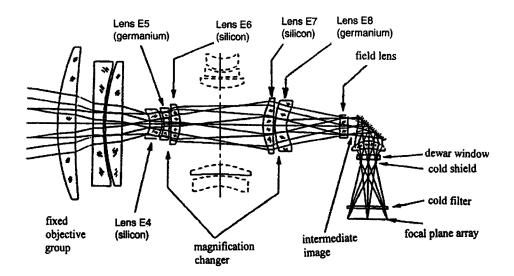
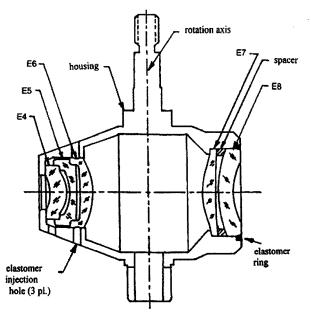


Figure 15.12 Optical system schematic for a dual-field IR tracker assembly. (From Guyer et al.⁶)

Each lens in the system had diamond-turned surfaces. The lenses in the magnification changer subsystem each had one aspheric surface. Table 15.3 shows the tolerances assigned as results of a sensitivity analysis for tilt and axial and radial displacements of the magnification changer portion of this design. The tolerance budget allowed the larger lenses (E7 and E8) in the magnification changer to be mounted conventionally with a metal spacer as indicated in Fig. 15.13. A fillet of elastomer constrained these lenses axially. The air spaced triplet at the other end of the subassembly (lenses E4 to E6) needed more precise centering so the lens's mechanical mounting interfaces were diamond turned in the same machine setup as the optical surfaces. These lenses were nested together and mounted as a group to minimize centration errors. In addition, the mechanical interfaces for those lenses in the aluminum housing were diamond turned to accept the lenses. Radial clearance at lens E6 was ~2.5 μ m. The group was held in place with elastomer injected into the three access holes indicated in Fig. 15.13. The special precautions taken during design and fabrication resulted in successful production and use of the sensors.

Table 15.3 Results of a mounting sensitivity analysis for the magnification changer of Fig. 15.12.

Element	Tilt	Axial Displacement		Transverse Displacement	
	(arcsec)	(µm)	(in.)	(µm)	(in.)
E4	6	10	0.0004	15	0.0006
E5	6	5	0.0002	10	0.0004
E6	6	10	0.0004	10	0.0004
E7	10	15	0.0006	20	0.0008
E8	10	15	0.0006	20	0.0008



From Guyer et al.⁶

Figure 15.13 Optomechanical layout of the magnification changer subsystem for the IR tracker. (From Guyer et al.⁶)

15.5 A Dual-Field IR Camera Lens Assembly

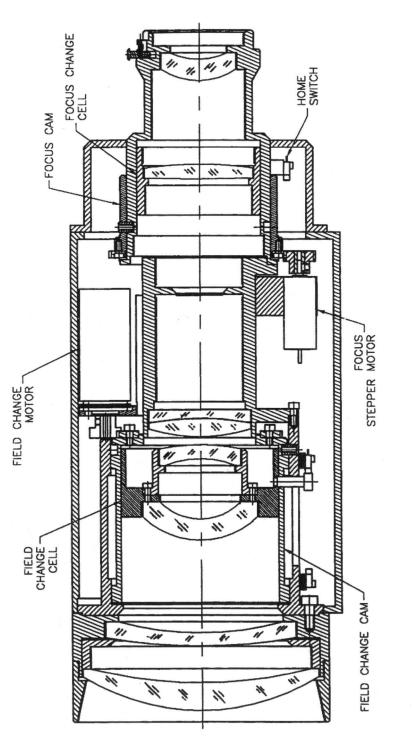
Figure 15.14 is a photograph of another dual-field lens assembly, this one for use with a 3to 5- μ m infrared camera. It was described by Palmer and Murray.¹ This lens system has focal lengths of 50 and 250 mm and is nearly diffraction-limited at both settings. It is 321.3mm (12.65-in.) long and generally cylindrical in configuration. The maximum lateral dimensions are 126.8-mm (4.99-in.) high and 133.0-mm (5.24-in.) wide. The assembly weighs 3.75 kg (8.23 lb).



Figure 15.14 Photograph of a dual-field IR camera objective assembly. (Courtesy of Janos Technology, Inc., Keene, NH.)

Figure 15.15 is the optomechanical schematic of this objective. Switching from one focal length/field size to the other is accomplished by sliding a cell containing two lenses axially inside the assembly. Focus is established at either setting by an axial movement of another cell containing one lens. A dc motor drives the focal length switching mechanism, while a stepper motor is used to drive the focus adjustment. Each mechanism contains a spur gear that rotates a ring gear on a cylindrical cam. Helical slots in the cams engage pins affixed to the lens cell and to the focus cell and drive those cells axially as the cams rotate. The pins also engage slots in the fixed portion of the housing to prevent rotation of the lenses, thereby maintaining constant boresight alignment. Sliding surfaces have 16-microinch finishes, are hard anodized, and not lubricated. Radial clearances between these surfaces are typically $\pm 12 \ \mu m$ (~0.0005 in.). The interface with the IR camera is a bayonet mount.

The main housing of the assembly is 6061-T6 aluminum; the lenses are silicon and germanium. The larger lenses are constrained by a threaded retaining ring and located by seating them against shoulders. The spacer between these lenses also provides the seat for the outermost lens. The remaining lenses are held in place by GE RTV-655 seals around their rims.





15.6 A Passively Stabilized 10:1 Zoom Lens Objective

The Bistovar 15- to 150-mm (0.59- to 5.90-in.) focal length, f/2.8 zoom lens assembly shown in Fig. 15.16 measures about 172-mm (6.77-in.) in length and 155-mm (4.53-in.) in diameter. At the camera end of the housing is a standard C-mount interface for a visiblelight video camera with an 11.0-mm (0.433-in.) format diagonal. The lens has fixed (infinity) focus; its focal length and relative aperture are electrically variable over a 10:1 range. The relative aperture can be varied from f/2.8 to f/16 by driving its iris. As designed, the lens assembly weighs approximately 1600 g (3.57 lb). Weight reduction has not been attempted.

The optical system contains, in sequence, a four-element passive stabilization system with ± 5 -deg. dynamic range at the entrance aperture, a seven-element 5:1 zoom system, a five-element, dual position 2:1 focal length extender system, and a Schott GG475 (minus blue) filter. The average polychromatic MTF performance over the zoom range at 20 lp/mm on-axis and at a 0.9 field is 69% and 25% respectively, including diffraction effects.

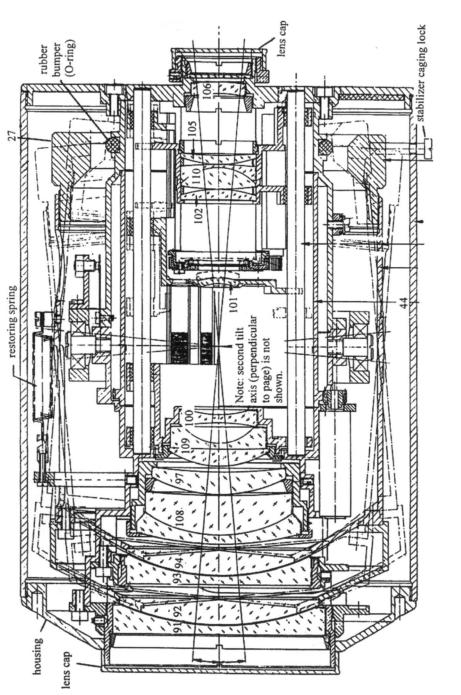
Movements of the two zoom lens groups (Items 109 and 100, and Item 101) in the 5:1 system are synchronized by a motor driven cylindrical cam (Item 44) carrying slots custom machined for the specific set of lenses used in that assembly. A third lens group (Items 102, 110, and 105) moves axially under the control of a separate slot in the same cam so as to switch the 2:1 extender system whenever the main zoom system reaches its limits. Lenses 108, 97, and 106 remain stationary.

Two air-spaced doublets, each consisting of a plano concave element and a plano convex element, make up the stabilization subsystem. The concentric curved surfaces of each doublet are closely adjacent. The positive singlets (Items 92 and 94) are attached to a lightweight tubular structure (Item 23) that pivots on ball bearings about either of two orthogonal gimbal axes. A counterweight (Item 27) at the camera end of this tube statically balances the lenses in both transverse directions.

Most of the lenses used in this assembly are conventionally mounted in aluminum alloy cells and held in place by threaded retainers. A few glass-to-metal interfaces are spherical, but the rest are of the sharp corner type. The beveled edge of one lens (Item 100) directly contacts a flat bevel on the adjacent doublet (Item 109). The pivoted lenses (Items 92 and 94) and one fixed lens (Item 101) are secured in place with adhesive (Ciba-Geigy Araldite 1118 epoxy) since there is no room for retainers.

15.7 A 90-mm, f/2 Projection Lens Assembly

Figure 15.17 is a sectional view through a 90-mm (3.54-in.) focal length, f/2 objective assembly designed for motion picture projection. Only single-element lenses are used here since the assembly is typically subjected to very high temperatures during operation and cemented components would be damaged. Large physical apertures are used in the lenses so that geometric vignetting is minimized and illumination remains high at the corners of the format. The MTF at 50 lp/mm is specified as over 70% on-axis, with the average radial and tangential MTF falling to about 30% at the extreme corners of the image. The field





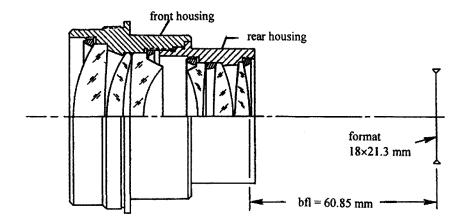


Figure 15.17 A 90-mm, f/2 motion picture projection lens assembly. (Courtesy of Schneider Optics, Inc., Hauppauge, NY.)

curvature of the design is compatible with the natural cylindrical curvature of the film as it passes through the film gate and helps to maintain image sharpness at the horizontal edges of that image. For the intended application of this lens, an iris is not needed; the lens operates at a fixed relative aperture.

As may be seen from the figure, mechanical construction of the assembly is conventional. All metal parts are anodized aluminum alloy. The barrel is made in two parts joined at the center by a piloted and threaded interface. Starting at the larger diameter end, the first lens is seated against a shoulder in the barrel and held by a threaded retainer. The second and third lenses are inserted from the right side of this barrel and clamped together without a spacer and against a shoulder in the barrel by a threaded retainer. In this case, the retainer bears against a flat bevel formed at the base of a step ground into the lens rim. The sixth (outermost) lens in the smaller end of the assembly is held against a shoulder by a threaded retainer. The fourth and fifth lenses are held against a shoulder in series with an intermediate spacer of conventional design by another retainer, which also fits into a step on the rim of the fourth lens.

15.8 A Solid Catadioptric Lens Assembly

The optical system of a "solid" catadioptric lens (see Fig. 15.18) was designed as a compact, durable, environmentally stable, long-focal length objective for 35-mm single lens reflex camera use.⁷ Essentially, it fills the space between the primary and the secondary mirrors of a Cassegrain objective with useful glass. Image quality is maximized while the optical surfaces are closely coupled for mechanical stability. The relatively wide rims of the larger components provide long contacts with the IDs of the lens barrel. Owing to the telephoto effect of the mirrors, the overall system length is considerably shorter than the focal length. The small size of this long focal length lens is apparent from Fig. 15.19.

Several versions of this lens have been manufactured and used in aerospace and consumer applications. The one shown here has a focal length of 1200 mm (47.244 in.), a

relative aperture of f/11.8 at infinity focus, and covers a 24×36 mm format (semifield of 1.03-deg.). The fifth through tenth elements serve as field lenses for aberration correction and lengthen the focal length in the manner of a Barlow lens.^{*} The system does not have an iris, so variations in lighting conditions are compensated for by exposure variations or by filtering. The filter located following the last lens is easily interchanged for this purpose.

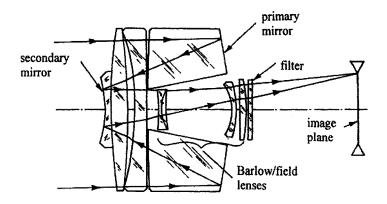


Figure 15.18 Optical schematic of a 1200-mm (47.244-in.) focal length, f/11.8 solid catadioptric lens assembly. (Courtesy of Goodrich Corporation, Danbury, CT.)



Figure 15.19 Photograph of an early model of the Solid catadioptric Lens being tested on a 35-mm camera by its designer in 1975. (Courtesy of Juan L. Rayces.)

^{*} The Barlow lens is defined as a "lens system used in telescopes, in which one or more strongly negative powered lens elements are used to increase the effective focal length and thereby increase the magnification."⁸

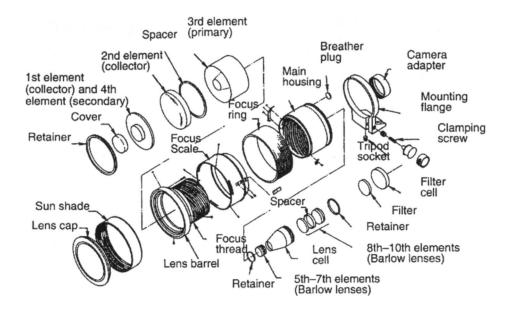


Figure 15.20 Exploded view of the solid catadioptric lens assembly. (Courtesy of Goodrich Corporation, Danbury, CT.)

The mechanical construction of the assembly is illustrated in the exploded view of Fig. 15.20 All metal parts are aluminum. The mounting flange mounts to a tripod and the camera attaches to the adapter (shown here without detail). The larger optics are mounted in a barrel with the primary resting against an internal shoulder; the two lenses and primary are secured by a single retainer. The Barlow/field lenses are mounted in a cell that attaches to a threaded central hole in the rear plate portion of the main housing. The focus ring drives the lens barrel on a 14-start Acme thread to focus the lens as close as 23 ft (7 m) with convenient rotation of the focus ring of about one-third turn. Glass-to-metal interfaces are the sharp corner type. Tolerances are controlled so the lenses and the primary can be installed without lathe assembly.

15.9 An All-Aluminum Catadioptric Lens Assembly

Figure 15.21 is a photograph of a 557-mm (21.9-in.) focal length, 242.174-mm (9.534-in.) aperture, f/2.3 infrared catadioptric lens¹ The lens, which is designed to operate in the 8- to 12-µm spectral region, is shown mounted on a tripod. The rectangular object at the rear of the assembly is an infrared camera.

Figure 15.22 shows front and sectional side views of the assembly. The mirrors and mechanical parts are constructed of 6061-T6 aluminum, while the field lenses are germanium. All optical surfaces and the optomechanical interfaces of this assembly are single-point diamond turned for highest alignment accuracy. Pockets are milled into the back of the primary mirror to reduce weight. The reflecting surfaces are coated with silicon monoxide to protect them during cleaning. Their reflectivities are greater than 98% in the 8-to 14-µm spectral range and the surfaces are adequately smooth for the application.

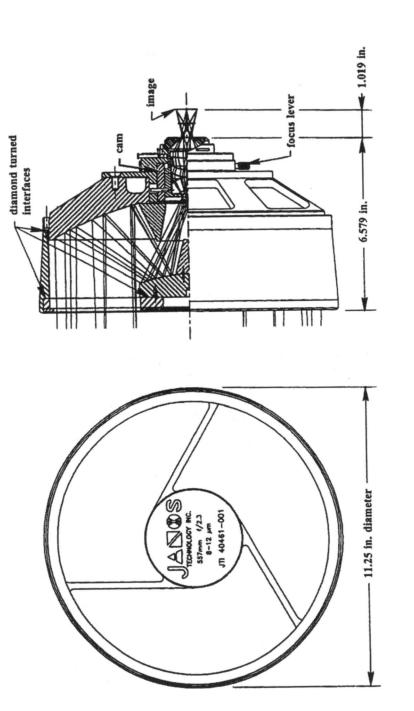


Figure 15.21 Photograph of a 557-mm (21.93-in.) focal length, f/2.3 catadioptric lens assembly with aluminum mirrors and structure. (Courtesy of Janos Technology, Inc., Keene, NH.)

No adjustments for axial locations or optical component tilts are needed at assembly because of the precision achieved with SPDT machining. Centering of the secondary mirror is accomplished in an interferometer (before installing the refractive components). Wave front error is measured in the same test setup to ensure that the mirrors are not distorted. The lenses are held in place with RTV655. Internal baffles suppress stray light. The lens assembly is focused for various object distances by manual rotation of a focus ring with the focus lever indicated in the figure. Turning this ring actuates a helical cam that drives the lenses axially without rotation.

15.10 A Catadioptric Star Mapping Objective Assembly

A catadioptric lens assembly developed for use as a star-field mapping sensor in a spacecraft attitude monitoring application⁹ is illustrated schematically in Fig. 15.23. Figure 15.24 is a photograph of the assembly. This system had a focal length of 10.0 in. (25.4 cm), a relative aperture of f/1.5, a field of view of ± 2.8 -deg., and a charge transfer device as detector.





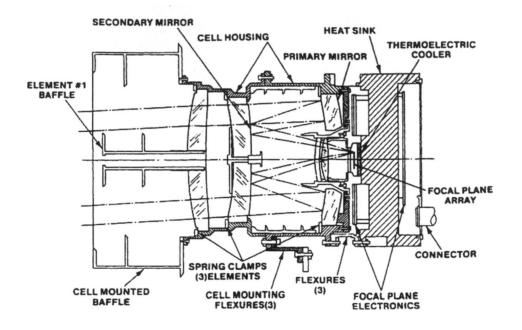


Figure 15.23 Sectional schematic view of the star-mapper lens assembly. (Adapted from Cassidy.⁹)

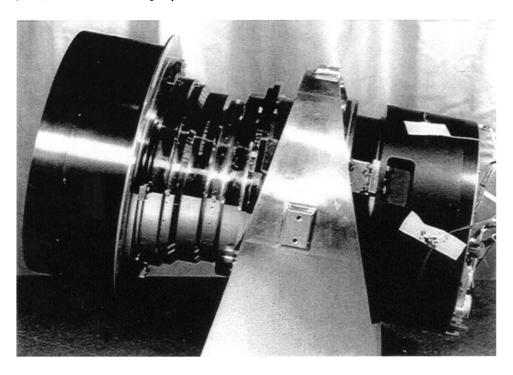


Figure 15.24 Photograph of the star-mapper lens assembly. (From Cassidy.⁹)

The image quality of the Cassegrain telescope formed by the spherical primary and secondary mirrors was optimized for the application by two full-aperture corrector lenses. One lens had an aspheric surface designed to make the images of stars to have the same spot sizes over the entire field (see Bystricky and Yoder).¹⁰ An air-spaced doublet field lens group helped control the spherical, chromatic, and off-axis aberrations.

The secondary mirror was coated onto the second surface of the inner corrector lens. The image surface was located about 1.4 in. (36 mm) beyond the second vertex of the primary mirror to provide space for a thermoelectrically cooled detector array and its adjacent heat sink structure.

Invar was used as the material for the main barrel to minimize the effects of thermal expansion and contraction. The exposed surfaces of this barrel were chrome plated to prevent corrosion. This barrel was flexure mounted to the aluminum structure of the spacecraft so temperature changes would not affect image quality or alignment between the sensor and the spacecraft attitude control system.

The two corrector lenses were provided with flat bevels accurately aligned to the opposite spherical surface. These lenses were held in place by individual cantilevered flat spring clips. Radially-directed screws passing through the barrel wall and bearing against the lens rims were used to center the lenses. After alignment, RTV60 elastomer was injected through several radially directed holes into the space between the lens rims and the barrel ID. After curing of the elastomer, the alignment screws were removed and their holes plugged with elastomer.

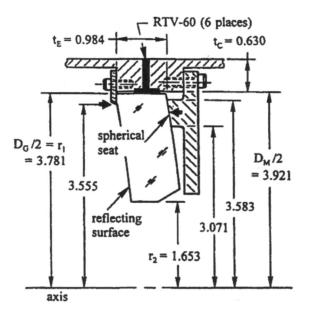


Figure 15.25 The mounting for the primary mirror in the star mapper. Dimensions are in inches.

The concentric meniscus first surface primary mirror was edge-clamped with spring clips against three spherical seats machined and lapped into the rear mounting plate attached to the main lens barrel. The mirror was further secured by RTV60 locally injected at six places around the mirror rim as shown in Fig. 15.25. Note that the axial preload was applied at a different height from the axis than the support provided by the seats. The tensile stress developed in the mirror because of this mismatch in contact heights and the resulting bending moment was not sufficient to threaten the survivability of the mirror nor did it create significant surface deformation under operational conditions.

The field lenses were lathe assembled in their cell as explained in Section 4.3. The axial location of this subassembly was adjusted by custom grinding a spacer (not shown) located between the cell flange and the rear housing. Once aligned, that subassembly was pinned in place. The focal plane array, heat sink, thermoelectric cooler, and local electronics were mounted on the lens assembly with flexures as indicated in Fig. 15.23. The axial position of the array was adjusted by customizing the thickness of spacers located at each flexure attachment point.

15.11 A 150-in., f/10 Catadioptric Camera Objective

A relatively simple catadioptric assembly is shown in section in Figs. 15.26 and 15.27. This lens had a focal length of 150 in. (3.8 m) and operated at f/10. The figures show, respectively, the front and rear (camera) portions of the assembly. The system was intended for use with a 70-mm format Mitchell motion picture camera to photograph missiles during launch. It was designed to be mounted on an antiaircraft gun mount to provide the required azimuth and elevation motions for tracking the target. Weight had to be limited in order to facilitate rapid angular acceleration of the line of sight during operation.

The system was of Cassegrain form with two full-aperture correcting lenses located near the primary mirror's center of curvature and an air-spaced triplet field lens group near the image plane to correct off axis aberrations. The lens had a flat ± 0.6 -deg field.

The mechanical construction of the lens assembly had a front aluminum cell holding the corrector lenses and secondary mirror and a rear housing holding the primary mirror and field lenses, respectively. The rear housing was an aluminum casting that attached to the gun mount and supported the camera. The front housing was supported from the rear housing through a dual-walled aluminum tube with internal thermal insulation. A light weight tubular lens shade projected forward from the front aperture. The assembly was painted white to reflect sunlight.

In the front cell, a flange retainer clamped the two lenses against an internal shoulder with an internal spacer to provide axial separation. The glass-to-metal interfaces at the lens rims were padded locally with single or multiple layers of 0.001-in. (0.025-mm) thick Mylar tape [see Fig. 15.28(a)]. The required thicknesses of the tape shims were determined by supporting the housing with its axis vertical on a precision rotary table and measuring the runout as the table was slowly rotated. Once centered adequately and shimmed, the retainer was installed and tightened to hold the alignment. The secondary mirror was then mounted in its cell, which had previously been attached through a perforation at the center of the second lens. The interface for this cell to the lens and that for the mirror to the cell

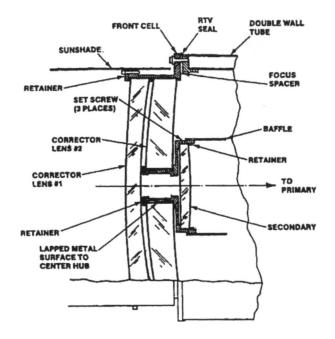


Figure 15.26 Sectional view of the front portion of a 150-in (3.8-m) focal length, f/10 catadioptric objective lens assembly.

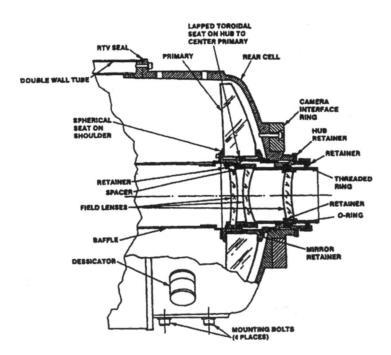


Figure 15.27 Sectional view of the rear (camera) portion of the catadioptric lens assembly.

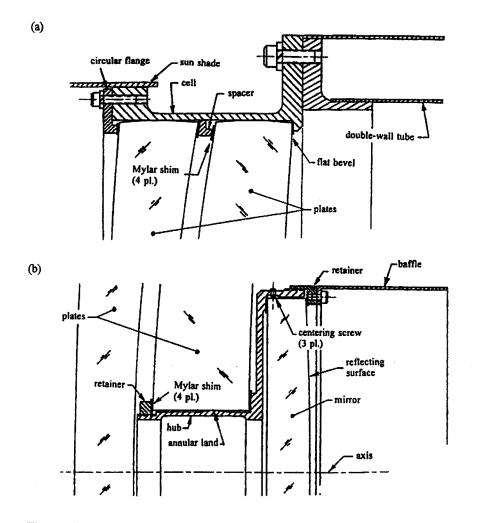
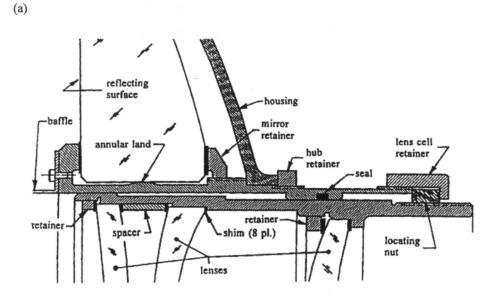


Figure 15.28 Detail views showing the use of Mylar shims to pad the glass-tometal interfaces in the front assembly of the camera objective: (a) at the corrector lens rims and (b) at the secondary mirror mounting.

were shimmed with Mylar [see Fig. 15.28(b)]. Three setscrews were used to center the secondary mirror to the rotation axis of the table and hence to the lens axis. The retainer was secured in place after alignment was completed and the setscrews were removed.

In the rear housing, the primary mirror was hub mounted and clamped between a spherically lapped flange shoulder and a threaded retainer. A convex toroidal seat on the cylindrical hub was lapped to fit closely inside the ID of the primary mirror's central perforation to center that mirror. The hub was in turn clamped axially within the rear housing by another threaded retainer. With the exception of the hub-to-mirror interface, all contacts between the optics and the mount were padded with Mylar shims as shown in Fig. 15.29(a).



(b)

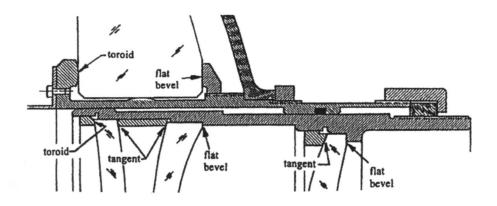


Figure 15.29 (a) Detail view of the rear (camera) portion of the objective showing the use of Mylar shims in the interfaces. (b) An alternative configuration without the shims.

The front portion of the assembly was focused by inserting metal shims of varying thickness between the tube and the front housing until the central air space was correct and the corrector lenses were squared to the axis. The lateral adjustment was made by successive approximations using motions of the front housing relative to the end of the tube assembly in slightly oversize holes for attachment bolts. After alignment (as measured by viewing the image of an artificial star image under magnification) was accepted, the shims were replaced by custom ground permanent spacers.

The field lens assembly was lathe assembled as described in Sect. 4.3 and installed in the hub of the rear housing. Axial location was measured mechanically and adjusted to the design value by rotating the threaded ring located at the rear end of the hub. The camera interface was then installed and adjusted for the proper back focus distance using photographic tests to measure residual errors during the process.

An alternative design for the primary mirror and field lens optic-to-mount interfaces is shown in Fig. 15.29(b). A similar optomechanical interface design could be used to advantage in mounting the optics (corrector lenses and secondary mirror) in the front portion of the assembly. In these cases, no Mylar shims would be used; the mounts would be configured with tangential, toroidal, or flat surfaces as now known to be appropriate for direct contacts on the optical surfaces and discussed in Chapter 3. Mounting stresses would then be acceptable throughout the assembly without use of the Mylar padding.

In Fig. 15.27, one may note the desiccator built into the rear housing of this assembly. This device is provided to allow the instrument to breathe as the internal air pressure varies with temperature. The structural tube connecting the front and rear assemblies is not stiff enough to withstand significant pressure differential. Making it stronger and stiffer would have raised the weight too much for the weight goal to be met. Be allowing air to flow freely between the housing and the outside world, the need for a pressure resistant housing was eliminated. The desiccator's function is to prevent moisture from entering the housing as the internal pressure subsides as the temperature falls at night. A dust filter is also incorporated into the desiccator to prevent dust and other contaminants from entering.

15.12 The Camera Assembly for the DEIMOS Spectrograph

Mast et al.,¹¹ described the DEIMOS (Deep Imaging Multiobject Spectrograph) as a large spectrograph with an imaging mode that is part of the Keck 2 telescope on Mauna Kea in Hawaii. It is capable of measuring spectra in the 0.39- to 1.10- μ m range of as many as 100 objects simultaneously. The object field of view of the system is 16.7 arcmin (slit length), which is equivalent to a 730-mm image at the focus of the 10-m aperture telescope. The detector used is a 2×4 mosaic of eight charge coupled devices (CCDs), each with 2048×4096 pixels at 15 μ m per pixel.¹² A series of gratings with 600 to 1200 lines per millimeter provide the necessary dispersion.

The optical system is diagrammed in Fig. 15.30. It has five lens groups with nine lenses; the largest being 330-mm (12.99-in.) diameter. Its EFL is 381 mm, so the plate scale is 125 μ m per arcsec in object space. The system includes three aspheric surfaces and the materials indicated in the figure. The combinations of CTEs and the fragility of the three CaF₂ lenses posed especially complex mounting problems. The survival temperature range was specified as -4 to 6°C (24.8 to 42.8°F). Part of the solution was to fill the spaces inside the multiplets (groups 1, 3, and 4) with optical coupling fluid. The thicknesses of these cavities were small [0.003 to 0.006 in. (0.076 to 0.152 mm)] and created with shims. The cavities had to be vented to bladders to accommodate the specified survival temperature excursion. Experiments reported by Hilyard et al.¹³ formed the basis for choice of Cargille LL1074 fluid, ether-based polyethylene film for the bladders, Viton VO763-60 or VO834-70 for O-ring fluid constraints, GE RTV 560 for seals, and Mylar for the shims. These materials proved to be compatible with each other and with the glasses, the CaF₂, and the mounting materials. The bladders were heat sealed to eliminate the need for adhesives.

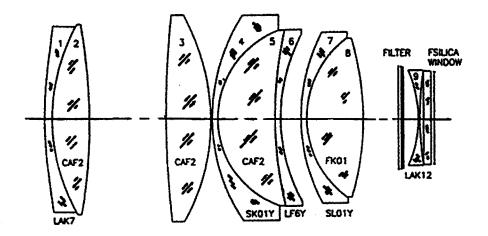


Figure 15.30 Optical systems for the DEIMOS spectrograph camera. (Adapted from Mast et al. 11)

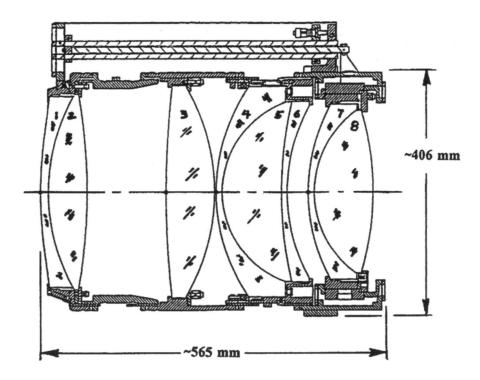


Figure 15.31 Optomechanical schematic of the DEIMOS camera assembly. (Adapted from Mast et al. 11)

The optomechanical layout of the camera optics is shown in Fig. 15.31. The barrel contains several 303 CRES segments with spacers to establish the required air spaces between lens groups. The lenses are mounted in ring-shaped CRES cells. The field flattener (lens 9) and the fused silica window shown in Fig. 15.30 are part of the detector assembly, which is mounted in a separate vacuum vessel. The filter, also shown in Fig. 15.30, is contained in a separately mounted filter wheel. A shutter (not shown) is located in the optical path near the filter.

The CaF₂ lenses are mounted in annular RTV rings inside thin aluminum rings that are, in turn, inside 303 CRES cells. Those cells have a slight interference fit (~75 μ m) over the rings. This construction effectively keeps the crystalline material under compression at all temperatures. It prevents the material going into tension at the lowest expected temperature since that could cause fractures along intercrystalline boundaries. The mathematical basis for this design is explained by Mast et al.¹¹ The stress in the lenses at assembly (~20°C) is 0.13 MPa. This decreases to 0.04 MPa at the lowest temperature of -20°C.

An error analysis and tolerance budget study conducted by Optical Research Associates indicated that the lens groups should be tilted by no more than 50 arcsec, decentered by no more than 75 μ m, and despaced by no more than 150 μ m (all 2 σ values). It was known that the aspheric curves on three lenses could be decentered with respect to the mechanical axes by as much as 350 μ m. To achieve proper performance in spite of these errors and the residual errors in centration elsewhere in the system during assembly, lens group four was designed to be transversely adjustable. Four flexures were built into the mount for this lens group. Two orthogonal adjusting screws and preload springs were provided to make the adjustment. The maximum transverse motion possible is about 500 μ m (0.020 in.) in any direction. Since the flexures can develop high stress points, the cell for this group is 17-4SH CRES.

Lenses within each multiple-lens group are held axially between Mylar shims on flat pads machined locally into shoulders in the lens cells and a spring loaded Delrin retaining ring. The outer lenses of each multiple-lens group are sealed to adjacent metal parts with GE-560 RTV elastomer. These seals support the lenses radially and serve as dams to constrain the fluid. In most cases, thicknesses of the annular elastomer layers were chosen to render radially athermal designs as developed by Mast et al.¹⁴ The exception is the final doublet, which has a thicker layer to better accommodate the different stainless steel (Type 17-4SH) used in the cell.

In order to provide a constant scale factor in the dispersed image throughout the operating temperature range of -4 to 6°C, it was found necessary to move the final doublet lens group axially as a function of temperature. This was achieved by attaching the lens cell to a bimetallic compensator consisting of a Delrin tube concentric with an Invar rod as indicated in Fig. 15.31. The cell is mounted on flexures to allow this axial motion. The resultant CTE of the compensator is 0.036 mm/°C.

15.13 Mountings for Prisms in a Military Articulated Telescope

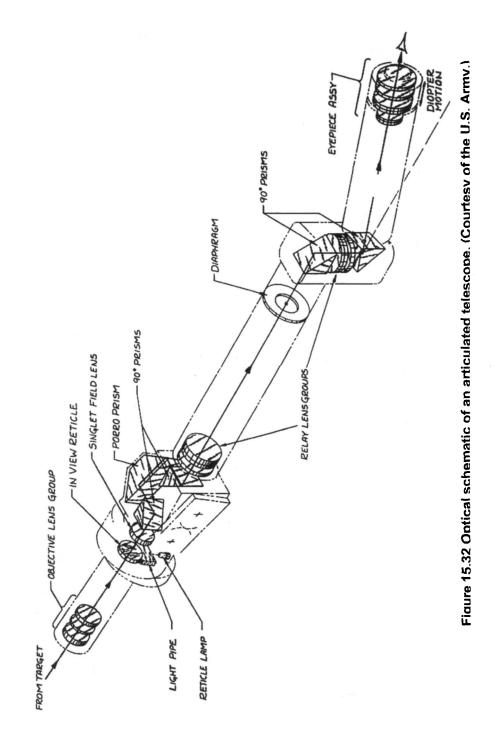
The main weapon of an armored vehicle (tank) is usually operated by a gunner who uses one of two optical instruments to acquire and fire at hostile targets. The primary fire control sight is a periscope protruding through the turret roof while the secondary sight is a telescope protruding through the front of the turret alongside and attached by mechanical links to the weapon. Key design features of a typical embodiment of the latter type of instrument are discussed here. The specific telescope considered is of the articulated configuration, i.e., it is hinged near its midsection so the front end can swing in elevation with the gun, while the rear section is essentially fixed in place so the gunner has access to the eyepiece at all times without significantly moving his head. The latter requirement is vital to success because the location of the eye behind the eyepiece must be accurate within a few millimeters for the target to be seen and the gunner's ability to move his head is limited, especially in the vertical direction.

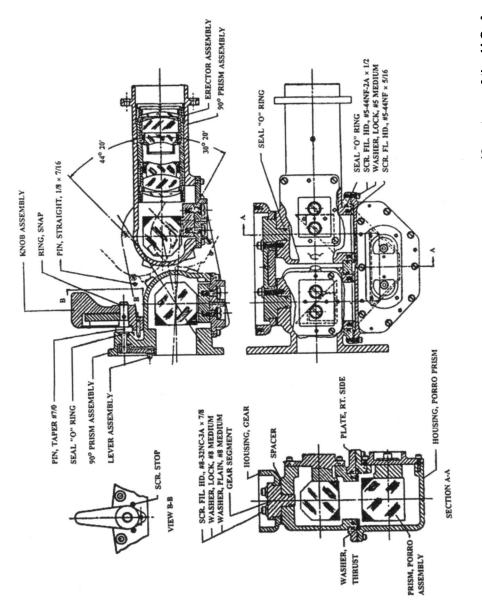
Figure 15.32 shows the optical system schematically. It has a fixed magnification of 8 power and a total field of view in object space of about 8-deg. The exit pupil diameter is 5 mm, so the entrance pupil diameter is 40 mm (1.575 in.). The diameter of the telescope housing throughout its length is about 2.5 in. (63.5 mm); the prism housings naturally are somewhat larger. Widely separated relay lenses erect the image and transfer the image from the objective focal plane to the eyepiece focal plane. Two prism assemblies are shown in the figure. The first contains three prisms, two 90° prisms and a Porro prism, that function within the mechanical hinge to keep the image erect at all gun elevation angles. In the second prism assembly, two 90-deg prisms offset the axis vertically and turn that axis 20 deg in a horizontal plane to bring the eyepiece to a convenient location at the gunner's eye.

The articulated joint mechanism is shown in Fig. 15.33. The first right-angle prism is mounted in "Housing, 90° Prism" (see Fig. 15.34). The prism is bonded to a bracket that is attached with two screws and two pins to a plate that is in turn attached with four screws to the housing. After assembly and alignment, a cover is installed over the screws and sealed in place. Surface "W" of that housing attaches to the exit end of the reticle housing.

The second right angle prism is mounted in "Housing, Erector" as indicated in Fig. 15.35. It also is bonded to a bracket that is attached with two screws and two pins to a plate that is screwed fast to the housing. The note in this figure indicates the alignment requirements for the prism. Surface "W" mentioned there is shown in Fig. 15.34. After alignment, a cover is sealed over the screws.

As shown in Fig. 15.33, the Porro prism is contained within a separate housing and, together with a gear housing on the opposite side of the telescope, forms the mechanical link between the telescope's front and rear portions. The action of the gear train keeps this prism oriented angularly midway between the front and rear portions of the telescope. This angular relationship keeps the image erect. The housing for the Porro prism is made of hardened stainless steel since it acts as a bearing for the angular motion. The rotary joints in the assembly are sealed with lubricated O-rings that seat in grooves in the mating parts. The prism is bonded to a bracket that is attached to a cover by two screws riding in two slots. After installation of the bonded prism assembly in the housing, the prism is moved in the slots to adjust the optical path through the assembly. The screws are then secured and the plate pinned in place. A protective cover is then installed.







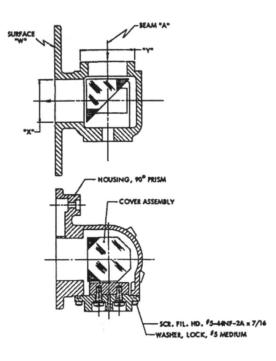


Figure 15.34 The first right-angle prism assembly. (Courtesy of the U.S. Army.)

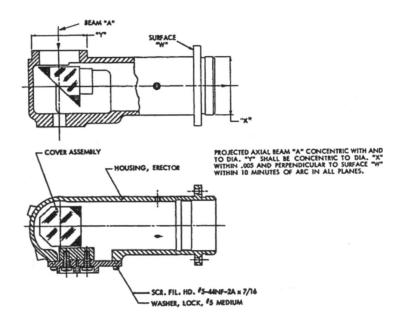


Figure 15.35 The second right angle prism assembly. (Courtesy of the U.S. Army.

15.14 A Modular Porro Prism Erecting System for a Binocular

The optomechanical layout of the U.S. Army's 7×50 Binocular M19, developed in the 1950s as a replacement for the modified commercial binoculars used in World War II and the Korean conflict is shown in Figs. 4.30 and 4.31. This binocular was a totally new design featuring improved optical imagery with significantly reduced weight and size, large quantity producibility, and improved reliability and maintainability compared with all prior designs. These advantages were achieved by making the device modular, with only five optomechanical parts, all of which were interchangeable in any reasonably clean location without adjustment and without special tools.^{15,16} In the hope that similar advantages might be gained through modular construction in future applications, we describe here in considerable detail how the prealigned Porro prism image erecting system was assembled and incorporated into the body housing. Success in such an undertaking depends largely on the detailed optomechanical design, the availability of optically based tooling, and special care exercised during manufacture.

A drawing and photographs of the M19 Porro prism cluster are shown in Figs. 15.36 and 15.37. The prisms were made of high index (Type 649338) glass to ensure total internal reflection and were tapered to have minimal volume and weight without vignetting.

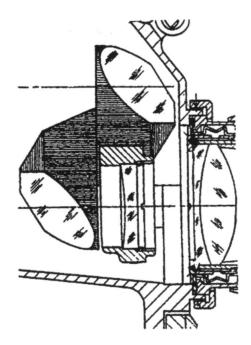


Figure 15.36 A section of the optomechanical layout of the Binocular M19 (from Fig. 4.31), showing the erecting prism assembly, its mounting bracket, and the lens/reticle mounting. (Adapted from a U.S. Army drawing.)

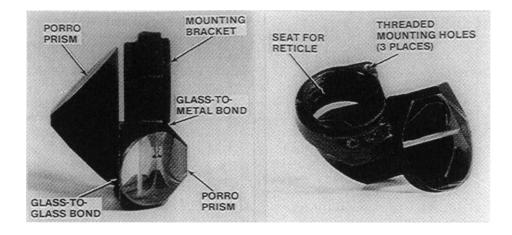


Figure 15.37 Photographs of the bonded Porro prism subassembly.

The first step in assembly was to bond one Porro prism to a die cast aluminum bracket with adhesive per MIL-A-4866 (such as Summers Milbond) in a fixture built to exacting tolerances and carefully maintained throughout use. This fixture ensured that the prism was properly located and oriented with regard to the interface to the mounting surface inside the binocular housing. After that bond was cured, the prism and bracket subassembly was mounted in a second precision fixture. Ultraviolet curing optical cement (Norland 61) was applied to the appropriate portion of the prism's hypotenuse surface. The second prism was positioned with respect to the first prism so that the input and output axes were parallel and displaced by the proper distance. Optical test equipment was used in making this adjustment. In addition, the second prism was rotated in the plane of the cement joint to correct rotation (tilt) of the image around the optical axis. A video camera and monitor were used to obtain and display to the operator the pointing and tilt relationships for the prism assembly. The operator first adjusted the second prism laterally until the image of a reticle projected through the system was positioned within a prescribed rectangular tolerance envelope on the monitor screen. Then, while the image was maintained inside this envelope, the prism was rotated slightly to align tilt reference indicators also displayed on the monitor screen. Once adjusted, the prism was clamped in position in the fixture. Curing of the cement took place under a bank of ultraviolet lamps adjacent to the setting station. Multiple setting and curing fixtures were necessary to support the required production rate. After curing, the same optical alignment apparatus was used as a test device to ensure that the desired prism setting had been retained throughout the curing process.

Both housings started out as identical thin-walled, vinyl-clad aluminum investment castings; they were machined differently to form their unique left and right shapes. The wall thickness was nominally 1.524 mm (0.060 in.). Over this, a 0.38-mm (0.015-in.) thick coating of soft vinyl was applied prior to machining of the critical mounting seats for the eyepiece, the prism assembly, and the objective. The locations of the eyepiece and prism assembly seats were established mechanically during the machining process. Normally, with a rigid, stable part, these would not have presented unusual problems despite the very demanding tolerances required. However, the structural flexibility of the thin walled housing was a serious handicap. In addition, owing to the soft vinyl, it was not possible to reliably locate the housings from any of the vinyl-clad surfaces or to clamp on them without

causing cosmetic damage. Elaborate fixtures relying on a few previously machined surfaces that were not vinyl clad had to be developed before acceptable production yields and rates could be attained.

Machining of the housing with the prism assembly installed was the critical step in obtaining the module precision required to permit interchangeability. Horizontal and vertical collimation requirements (divergence and dipvergence, respectively) for the monocular's optical axis with respect to the hinge pin centerline were such that the bore for the objective had to be properly located radially within 0.0127 mm (0.0005 in.). The requirement for perpendicularity between the objective seat and the optical axis was 0.0051 mm (0.0002 in.) measured across the objective seat. In addition, the objective seat had to be located axially to obtain the proper flange focal distance. To obtain these accuracies, it was necessary to use optical alignment techniques to position the housing for machining.

The housing was mounted directly on a CNC lathe with a hollow spindle that permitted passage of a light beam for monitoring alignment. Alignment in place proved to be very difficult and time consuming. This resulted in inefficient use of the machine. Only a fraction of its available time was actually being used for machining; most of the time was devoted to aligning the housing. This was unacceptable for high volume production, so this approach was abandoned.

The production approach that was finally developed was to hold the housing in a transferable setting and machining fixture. The housing was positioned using optical alignment instrumentation and locked in place on the fixture at an offline setting station. Then the fixture was transferred to the spindle of a CNC lathe for final machining. Multiple transferable fixtures were provided so that setting and machining could proceed in parallel.

The fixture and the optical alignment technique used at the setting station are shown schematically in Fig. 15.38. The fixture base was designed to mate precisely with the lathe spindle so that the fixture centerline was coincident with the rotational axis of the spindle during machining. In this way, the mounting seat for the objective could be machined concentric with the fixture centerline. Atop the fixture base was a sliding plate that could be translated laterally. This plate carried a post simulating a binocular hinge pin. The post centerline was always parallel to the fixture centerline.

An optical system in the setting station (not shown in Fig. 15.38) provided an image of a target at infinity along the input optical axis, which was coincident with the fixture axis. A master objective was mounted at a fixed location in the setting station and centered on this axis. This objective formed an image of the target at an image plane inside the housing. This image was then viewed through a master eyepiece (temporarily attached to the housing) by a video camera, with the output being displayed on a video monitor. The proper flange focus position for machining the objective lens seat in the housing was obtained by moving the housing vertically along the hinge post until the best focus was seen on the video monitor. The housing was then clamped to the post and sliding plate. Axial positioning of the housing on the fixture had been completed, but lateral adjustment to obtain collimation was still needed.

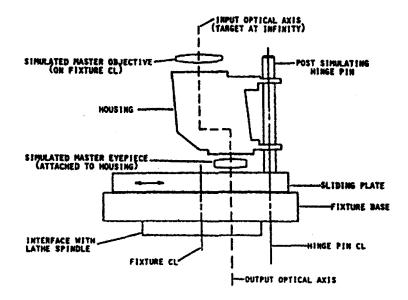


Figure 15.38 Schematic of the prism adjusting and holding fixture used to machine the binocular body housing with a prealigned prism installed. (From Trsar et al.¹⁵)

The collimation requirements for the housing/prism module were that the output optical axis be parallel to the hinge pin centerline within ± 5 arcmin in the dipvergence plane (normal to the plane of Fig. 15.38) and be diverging by 5 to 17 arcmin in the plane of the figure. Since the hinge pin and fixture centerlines were then parallel, the collimation requirement was referenced to the fixture centerline. After focus adjustment, the housing/sliding plate assembly was adjusted laterally (in two directions) with respect to the fixture base and the master objective until the required collimation conditions were achieved. This was indicated by a positioning of the target image within allowable limits marked on the video monitor. The sliding plate was then locked to the fixture base and the assembly transferred to the CNC lathe for machining of the objective mounting seat.

A similar procedure was used to orient the housing for machining the eyepiece interface. The result was a body housing with a prealigned prism assembly that would mate properly with any objective module and any eyepiece module to form one half of the binocular. The left and right housing assemblies also were then properly aligned to fit together at the hinge without adjustment.

15.15 Mounting Large Dispersing Prisms in a Spectrograph Imager

As described by Sheinis et al.,¹⁷ the Echellette Spectrograph/Imager (ESI) developed for use at the Cassegrain focus of the Keck II 10 m telescope employs two large (approx. 25 kg each) prisms for cross dispersion. The spectrograph optical system is described by Epps and Miller¹⁸ and by Sutin.¹⁹

In order to maintain optical stability in the operational modes, these prisms must maintain a fixed angle relative to the nominal spectrograph optical axis under a variety of flexural and thermal loads. These include gravity and thermally induced motions of the optical elements, stress induced deformations of the optical surfaces, and thermally induced changes in the refractive index of the materials making up the components. The major components of the ESI are shown in Fig. 15.39. The ESI has three scientific modes: medium-resolution echellette mode, low resolution prismatic mode, and imaging mode. To switch from one mode to another, one prism must be moved out of the beam as shown in Fig. 15.40. That prism is mounted on a single-axis stage. A mirror is moved into the beam to switch to the direct imaging mode.

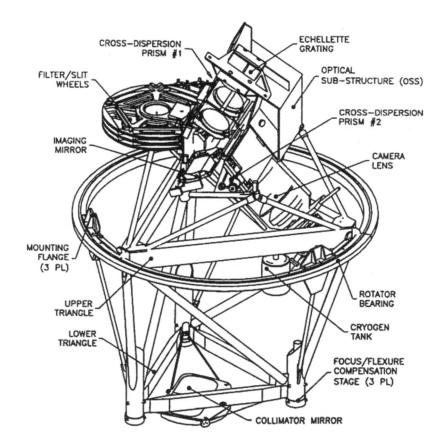


Figure 15.39 Major components of the echellette spectrograph and imager used in the Keck II Telescope. (From Sheinis et al.¹⁷)

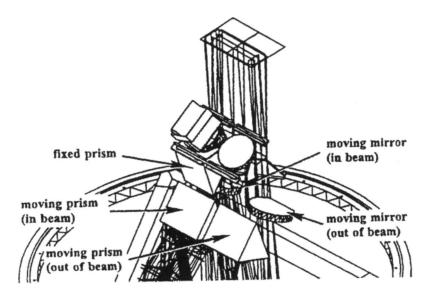


Figure 15.40 Another view of the ESI showing the fixed and moveable dispersing prisms. (From Sutin.¹⁹)

The design philosophy for the ESI is characterized by the use of determinate structures or space frames wherever possible. A determinate structure is one that constrains the six degrees of freedom of a solid body by six structural elements (here, struts) connected to the outside world at six points or nodes. Up to three pairs of the nodes may be degenerate. Struts are used in compression and tension only. Thus, deflections of the struts are linear with length, as opposed to struts or plates used in bending where the deflections are proportional to the third power of the part length. Other examples can be found in the description by Radovan et al.²⁰ of the active collimator used for tilt correction of the ESI and in the description by Bigelow and Nelson²¹ of the space frame that provides the backbone for the entire instrument. The desirable features of this type of mounting occur because no moments can be imparted at the strut connections. This has the advantage that distortions of one structural member will introduce displacements without inducing stresses in a second member (i.e., an optical component) mounted on the first member.

In the ESI, the cross-dispersing prisms are in collimated light; therefore, to a first order, small translations of the prisms will produce pupil motion only and no corresponding image motion. Tilts of the prisms will, however, produce combinations of the following: image motion, change of the cross-dispersion direction, change in the amount of cross-dispersion, change in the anamorphic magnification factor, and increased distortion. The most important stability criterion for the prisms is control of tip and tilt, with a very loose tolerance on displacement stability. The sensitivities for ESI are ± 0.013 , ± 0.0045 , and ± 0.014 arcsec of image motion for ± 1 arcsec of prism tilt about the X-, Y-, and Z-axes, respectively. The desired spectrograph performance is ± 0.06 arcsec of image motion without flexure control and ± 0.03 arcsec of image motion with flexure control during a two hour integration time period. For a reasonable choice of the allowable percentage of the

total error allotted to the prism motion, these sensitivities give a requirement of less than \pm 1.0, \pm 2.0, and \pm 1.0 arcsec rotation about the X-, Y-, and Z-axes. The normal operating temperature range for the Keck instruments is $2 \pm 4^{\circ}$ C (3.6 \pm 7.2°F), and the total range seen at the summit of Mauna Kea is –15 to 20°C.

The instrument must maintain all the above translational and rotational specifications over the entire working temperature range. Therefore the prism mounts need to be designed to be athermal with respect to tilts over this working temperature range and must keep stresses below the acceptable limits over the complete temperature range of the site as well as extremes experienced during shipping. In addition, attention must be paid to the stresses induced in the prisms. Not only is the potential for fracture of the bond joint or of the glass a cause for concern, but also stress induced in the glass will induce a corresponding local change in the index of refraction of the glass, i.e., birefringence, causing a possible wavefront distortion. To ensure against glass breakage, the estimated stress should be limited to tolerable limits during shipping, in earthquakes, with drive errors, and during collisions of the telescope with other objects in the dome (e.g., cranes). Equally important requirements of the mounting design were (1) minimization of measurable hysteresis (which limits the accuracy of the open loop flexure control system); (2) the ability to make a one-time alignment adjustment of the prism tilts over a 30 arcmin range during the initial assembly; and (3) the ability to remove the prisms for recoating, with repeatable alignment position on reassembly.

The ESI design had all of the optical elements and assemblies (except the collimator mirror) mounted to a plate as an optical substructure (OSS). The prisms were attached to the OSS through six struts. The actual attachment to the prism consisted of two parts: a pad that was permanently bonded to the prism and a mating, detachable part that was permanently attached to the ends of the struts. This allowed the prism to be readily and repeatably installed and removed from its support system.

The fixed and movable prism mounting designs are shown in Figs. 15.41 and 15.42, respectively. Joined pairs of struts are connected to each prism in one point on each of the three nonilluminated faces via a bonded tantalum pad. The CTE of tantalum $(6.5 \times 10^{-6} / ^{\circ}C)$ closely matches that of the BK7 prisms $(7.1 \times 10^{-6} / ^{\circ}C)$. The struts attach either directly to the OSS (in the case of the fixed prism) or to the translation stage (in the case of the movable prism), which in turn is bolted to the OSS. The largest refracting faces of the fixed and movable prisms measure 36.0 by 22.8 cm (14.17 by 8.98 in.) and 30.6 by 28.9 cm (12.04 by 11.38 in.), respectively. The glass path is >80 cm (31.50 in.), so it was necessary for the refractive index to be unusually uniform throughout the prisms. The prisms were made of Ohara BSL7Y glass having a measured index homogeneity better than $\pm 2 \times 10^{-6}$. This science was believed to represent the then-available state of the art for prisms of this size.

Each pair of struts was milled from a single piece of ground steel stock. Since each strut should constrain only one degree of freedom of the prism, crossed flexures were cut into each end of each strut to remove four degrees of freedom (one rotational and one translational per flexure pair). The fifth degree of freedom, axial rotation, was removed by the low torsional stiffness of the strut and flexure combination.

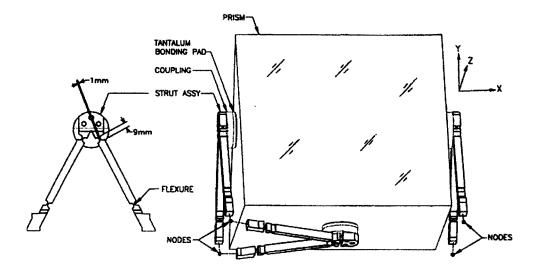


Figure 15.41 Sketch of the fixed prism assembly showing its six-strut mounting configuration. (From Sheinis et al.¹⁷)

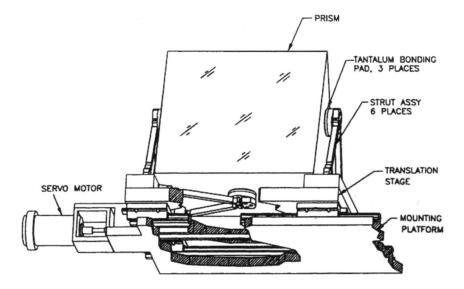


Figure 15.42 Sketch of the movable prism assembly showing its six-strut mounting configuration and translation stage. (From Sheinis et al.¹⁷)

Flexure thicknesses and lengths were designed to impart less stress than the selfweight loading of the prism into the prism pad connection and to be below the flexure material's elastic limit over the full range of adjustment, while keeping the strut as stiff as possible. Pad areas were chosen to give a self-weight-induced stress of 0.125 MPa. If we consider the tensile strength of glass to be 7 MPa,²¹ this gives a safety factor of 50. The glass-to-metal bond adhesive was Hysol 9313 and the thickness was chosen to be the same (0.25 mm) as that developed by Iraninejad et al.²² during development of the bonded connections in the Keck primary mirror segments. To confirm these choices, extensive stress testing over various temperature ranges was performed for BK7-to-tantalum and BK7-to-steel bonds. Several samples of BK7 were fabricated with the same surface finish as specified on the prisms. These were bonded to tantalum and steel pads mechanically similar to the actual bonding pads for the prism mounts. These assemblies were subjected to tensile and shear loads up to 10 times the expected loading in the instrument. The test jigs were then cycled 20 to 30 times through the expected temperature excursion range on the Mauna Kea summit. None failed. The joints were then examined for stress birefringence between crossed polarizers. The level of wavefront error was calculated to be less than the limit prescribed in the error budget in the case of the tantalum pad, but not for the steel pad. The tantalum material was then designated for use in all the bonding pads. Note that the CTE difference between tantalum and BK7 glass is 0.6×10⁻⁶/°C, whereas a good match to BK7 reported elsewhere²³ is 6Al-4V titanium with a CTE difference of 1.7×10^{-6} /°C.

15.16 Mounting Gratings in the FUSE Spectrograph

The Far Ultraviolet Spectroscopic Explorer (FUSE) is a low-earth orbiting astrophysical observatory designed to provide high spectral resolution observations across a 905 to 1195 μ m spectral band. Figure 15.43 shows schematically the optical arrangement of the spectrograph.²⁴ Light is collected by four off axis parabolic telescope mirrors (not shown) and focused onto four slit mirrors that act both as movable entrance slits for the spectrometer and as mirrors that direct the visible star field to fine pointing error sensors (also not shown). The diverging light passing through the slits is diffracted and reimaged by four holographic grating mount assemblies (GMAs). The spectra are collected on two microchannel plate detectors. The orbital temperature operating range is 15 to 25°C (59 to 77°F) while the survival range is -10 to 40°C (14 to 104°F). During an observation, the temperature is stabilized within 1°C (1.8°F).

The four gratings are identical in size at 266 by 275 by 68.1 mm (10.42 by 10.83 by 2.68 in.). They are made of Corning 7940 fused silica, class 0, grade F. This material was chosen for its low CTE and availability to accommodate the process of adding holographic gratings. Machining the rib pattern shown in Fig. 15.44 into its back surface reduced the weight of each blank. Two corners were removed to accommodate anticipated envelope interference. Strength and fracture control requirements dictated that the blank be acid etched after the ruled surfaces were coated with LiF or SiC to optimize performance in these bands.

The Invar mounting brackets were bonded in place with Hysol EA9396 epoxy. Tests of bonded samples showed that the bond strength consistently was >4000 lb/in.² (21.6 MPa), with some samples exceeding 5000 lb/in.² (34.5 MPa). This bond strength is more than adequate for the application.

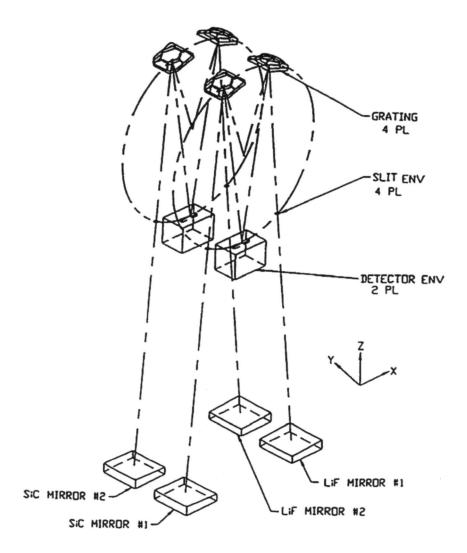


Figure 15.43 Optical configuration of the FUSE spectrograph. (From Shipley et al.²⁴)

Calculations of fracture probability using Gaussian and Weibull statistical methods were inconclusive.²⁵ The fact that the nonoptical surfaces of the gratings were not polished contributes significantly to this problem. In order to ensure success, a conservative mechanical interface design was employed; this was thoroughly evaluated by finite element

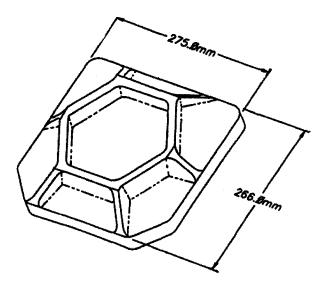


Figure 15.44 View of the back side of the grating blank showing its machined rib structure. (From Shipley et al.²⁴)

means throughout the evolution of the design. One major improvement was to add the flex pivots shown in Figs. 15.45 and 15.46 to allow compliance in the directions perpendicular to the radial flexures. The radial flexure blades were reduced in length to accommodate this addition while maintaining the height of the mechanism.

Figure 15.46 shows details of the flex-pivot installation. Each pivot consists of outer and inner pivot housings, two 0.625-in. (15.875-mm) diameter welded cantilever flex pivots, and eight cone-point setscrews. The location of each pivot is maintained by the setscrews, which are driven into shallow conical divots machined at two places in each cantilevered end of the flexure. Rigorous vibration testing of prototype and flight model grating mounts confirmed the success of the design.

The wedge-shaped optical angle mount seen in Fig. 15.45 between the radial flexures and the bottom of the outer tubular central structure serves to orient the grating at the proper angle relative to the coordinate system of the device. A spacer (Z-shim) is used between the outer tube and the optical bench for axial adjustment. The outer tube interfaces with spherical seats at top and bottom. This allows fine adjustment of the angular orientation by external motorized screws in an alignment fixture. This adjustment is clamped by torquing the nut at the top of the assembly. Optical cubes attached to the backs of the gratings are used with multiple theodolites as the metrology means during alignment.

Titanium was used extensively in the grating mount because of its high strength and relatively low CTE. All titanium parts except the flexures are Tiodized ²⁶ to reduce friction between mating surfaces during alignment and to facilitate cleaning at assembly. The convex sphere and the spherical washer of Fig. 15.45 are made of Type 17-4-PH stainless steel, the nut is type-303 stainless steel, and the Z-shim is a type-400 stainless steel.

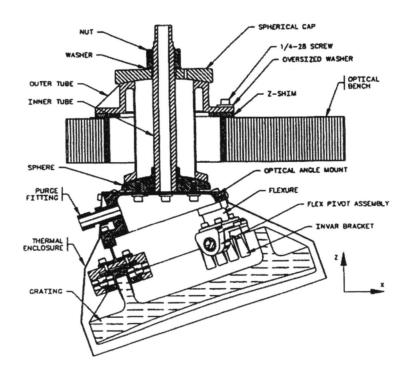
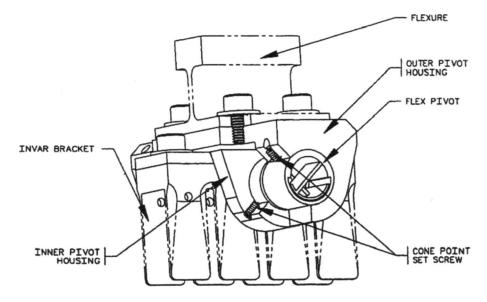


Figure 15.45 Sectional view of the grating mount assembly showing its adjustment provisions. (From Shipley et al. 25)





15.17 The Spitzer Space Telescope

A space infrared observatory featuring a telescope designed from a single material is the Spitzer Space Telescope [formerly known as the Space Infrared Telescope Facility (SIRTF)].^{27,28} It has an 85-cm (33.5-in.) diameter, f/12 lightweight all beryllium telescope as shown in Fig. 15.47. The primary mirror is a hub mounted, single-arch design. It is made of I-70H beryllium powder made by an impact grinding process to have grain size ~7.2 µm. It was compacted to 99.96% of theoretical density by hot isostatic processing. Aerial density of the completed mirror is 26.6 kg/m². Its mounting configuration is shown schematically in Fig. 15.48.²⁹

Optical parameters of the telescope at cryogenic temperature are listed in Table 15.4.³⁰ Schwenker et al.³¹ and Schwenker et al.³² describe how the telescope's optical performance was tested and the results of that testing respectively. Tests at 28K indicated that the mounted primary mirror had been cryofigured to 0.067- μ m rms surface figure. Other tests with beryllium mirrors have indicated that its CTE does not change appreciably at temperatures below 28K so optical performance should not degrade significantly in going to the operating temperature of ~5K. The mirror surface was not coated inasmuch as the reflectivity of bare beryllium is high in the infrared region of interest here.

NASA launched this observatory in August 2003. It operates in a one AU heliocentric orbit trailing the Earth where it benefits from reduced thermal inputs from the Earth, from shielding of solar heat inputs by the observatory's solar panels, and from its ability to radiate heat from most of the instrument's surface into space. We here concentrate on thermal design aspects of the system because they are unique. Publications dealing with the thermal design and performance verification include Lee et al.,³³ Hopkins et al.,³⁴ and Finley et al.³⁵

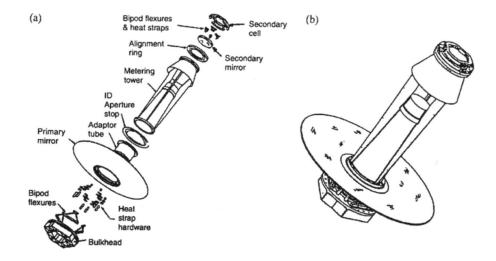


Figure 15.47 (a) Exploded view of the optical components of the Spitzer Space telescope. (b) The assembled telescope optics. (From Chaney et $al.^{30}$)

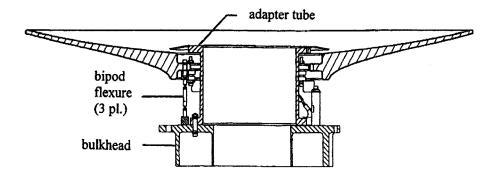


Figure 15.48 Schematic representation of the primary mirror hub mounting configuration. (From Coulter et al.²⁹)

Table	15.4	The	Spitzer	Telescope's	optical	parameters	at	cryogenic
tempe	rature	•						

Parameter	Units	Value @ ~5K					
System							
Focal length	cm	1020.0					
Relative aperture		<i>f</i> /12					
Back focal length	cm	43.700					
Field diameter	arcmin	32.0					
Spectral band pass	μm	3 to 180					
Aperture stop location		at primary rim					
Aperture stop OD	cm	85.000					
Aperture stop ID	cm	32.000					
Obscuration		37.6%					
Primary mirror							
Shape		hyperbolic					
Radius of curvature (concave)	cm	204.000					
Conic constant		-1.00355					
Clear aperture	cm	85.000					
Relative aperture		<i>f</i> /1.2					
Secondary mirror							
Shape		hyperbolic					
Radius of curvature (convex)	cm	29.434					
Conic constant		-1.5311					
Clear aperture	cm	12.000					

From Chaney et al.³⁰

In the Spitzer Observatory, only the science instruments are enclosed in the cryostat. Earlier infrared observatories, such as IRAS, had both the telescope and its instruments enclosed in vacuum cryostats and required a large volume of cryogen to be carried. In this new observatory, the majority of the instrument remained at ambient temperature and pressure until reaching orbit where it rapidly cooled passively to about 40K and enjoyed a

vacuum environment. These advantages greatly reduced the need for cryogen (superfluid He). The cryogen supply at launch of 360 liters is expected to further cool the instruments to \sim 5.5 K and the focal plane detectors to \sim 1.5 K for at least 2.5 years.

Figure 15.49 shows a cutaway view of the system with the telescope, scientific instruments, cryostat, cryogen supply, spacecraft bus, solar panels, shields, and associated equipment indicated. The multiple instrument chamber (MIC) contains the cold portions of four instruments (see Fig. 15.50). According to Lee et al.,³³ this chamber has a diameter of 84 cm (33.1 in.) and a height of 21 cm (8.3 in.). It is attached to the forward dome of the helium tank. Inside the MIC, pickoff mirrors send the light beam to the respective detector arrays. Signals from the detectors are preprocessed inside the cold region and then transferred through miniature ribbon cables to the electronics packages within the spacecraft bus. The helium tank is supported from the spacecraft bus by a truss made of alumina/epoxy for low thermal conductivity.

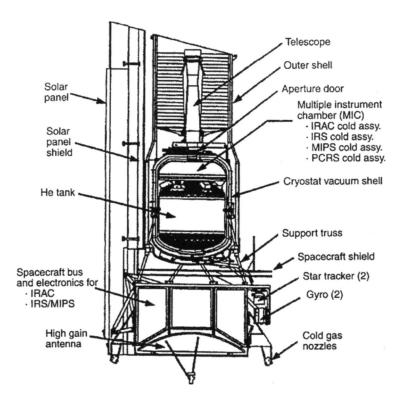


Figure 15.49 Cutaway view of the Spitzer Observatory. (From Fanson et al.²⁷ Reproduced through courtesy of NASA/JPL/Caltech.)

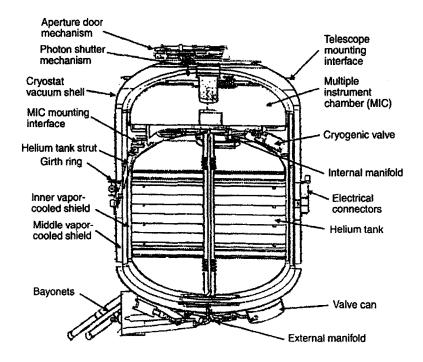


Figure 15.50 Cutaway view of the cryostat for the Spitzer Observatory. (From Lee et al.³³)

The scientific instruments in the observatory are as follows:

- (a) The infrared array camera (IRAC), which provides wide field imaging over two adjacent 5×5 arcmin fields. These fields are divided by beamsplitters into separate images at 3.6- and 5.8-μm wavelengths and 4.5- and 8.0-μm wavelengths. The arrays all have 256×256 pixels. The detectors for the 3.6- and 4.5-μm channels are indium antimonide while those for the 5.8- and 8.0-μm channels are arsenic doped silicon.
- (b) The infrared spectrograph (IRS), which has four separate spectrograph modules: two low resolution channels operating over the 5.3- to 14- μ m band the 14- to 40- μ m band respectively with resolving power $\lambda/\Delta\lambda$ of 60 to 120 and two high resolution channels operating over the 10- to 19.5- μ m and 19.5- to 37- μ m band respectively with resolving power $\lambda/\Delta\lambda$ of 600. The sensors are arsenic doped silicon for the shorter wavelengths and antimonide doped silicon for the longer wavelengths.
- (c) The multiband imaging photometer for SIRTF (MIPS) provides imagery and photometry centered at 24-, 70-, and 160- μ m wavelengths. The detector used at 24 μ m is a 128×128 pixel array of arsenic doped silicon. That used at 70 μ m is a 32×32 pixel array of gallium doped germanium, and that used at 160 μ m is a 2×20 pixel array of stressed gallium doped germanium. All three sensors observe the sky simultaneously.

A pointing calibration and reference sensor (PCRS) is provided within the MIC to calibrate thermomechanical drift errors between the telescope, the star trackers, and the gyroscopes with a radial 1- σ accuracy of 0.14 arcsec; to link the Observatory's coordinate system to the absolute J2000 Astrometric Reference Frame; and to define starting attitudes for high accuracy absolute offset maneuvers (see Mainzer et al.³⁶). By simultaneously observing a reference star from the Tyco Star Catalog and the externally mounted star tracker,³⁷ the relative alignment of these systems is established. An offset is then accomplished to center a target of interest in the selected science instrument's field of view.

All these cold assemblies are located as indicated schematically in Fig. 15.51 on an aluminum baseplate that serves as a stable optical bench. A thin-ribbed aluminum dome forms a light tight cover for the MIC. A photon shutter is attached to the top center of the cover. High purity copper thermal straps were attached between the instruments and the top of the helium tank to carry heat away from those temperature sensitive units. These straps pass through light tight seals.

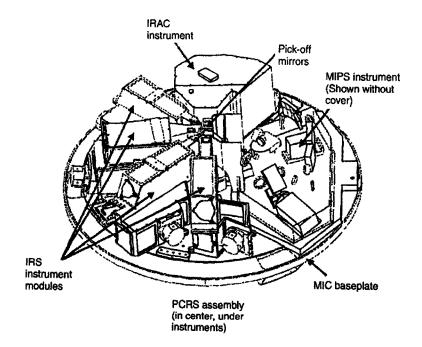


Figure 15.51 Arrangement of scientific instruments in the multiple instrument chamber of the Spitzer Observatory. (From Fanson et al.²⁷ Reproduced through courtesy of NASA/JPL/Caltech.)

15.18 A Modular Dual Collimator Assembly

A simple optical instrument that provided high performance was designed by Stubbs et al.^{38,39} This device was a compact and stable refractive dual collimator that accepted laser light from two fiber optic cables and generated two parallel 5.6-mm (0.220-in.) diameter

beams of collimated light separated laterally by 36.27 mm (1.428 in.). One beam was to serve as a reference while the other was used as a measurement beam in a high precision heterodyne metrology system. Figure 15.52 shows the finished instrument. It was designed to be aligned in fixtures so it would become an interchangeable module–identical in configuration, optomechanical interfaces, and performance with similar modules made in the same manner. Multiple units (or modules) could then be stacked together to form a one- or two-dimensional matrix of collimators. Simplicity, minimum number of parts, ease of assembly and alignment, and long term thermal stability were driving features of the design.

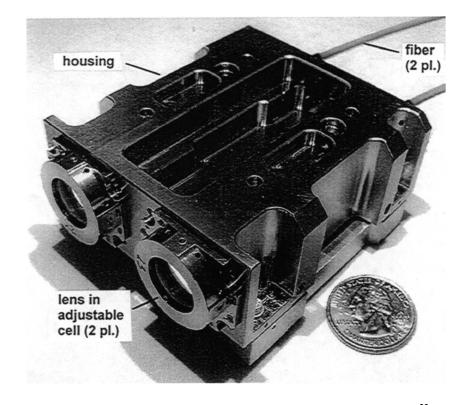
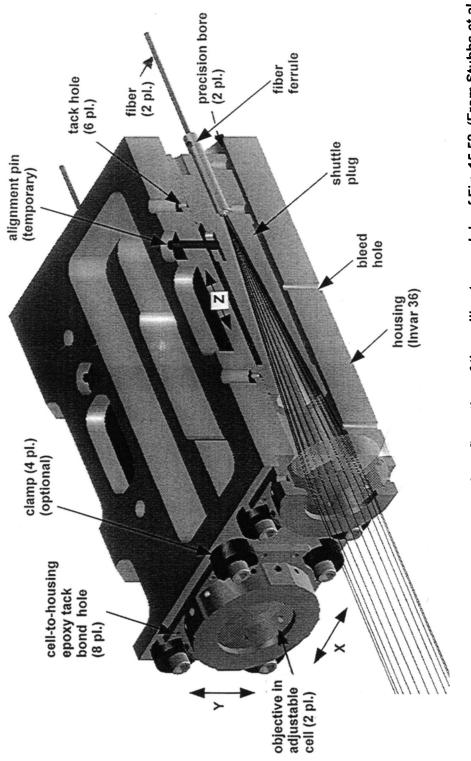


Figure 15.52 A dual refractive collimator. (From Stubbs et al.³⁸)

Overall dimensions of the assembly were 53-mm (2.087-in.) width, 38-mm (1.496in.) height, and 74-mm (2.913-in.) length. Total weight was 0.74 kg (1.63 lb). It was not required by the application that the weight be minimized so conventional machining techniques were employed, wall thicknesses were generous, and materials were selected for low CTEs rather than low densities. Figure 15.53 shows a partially cut-away view of the assembly. The metal parts were Invar 36. The lenses were commercially available cemented doublets. Analysis indicated that the optical performance of ≤ 0.010 wave p-v OPD of the projected wavefront over the operating temperature range of $20 \pm 1^{\circ}$ C would be acceptable.





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The lenses were mounted into cells and tack bonded in place with Dow Corning 6-1104 silicone sealant through eight access holes in the cell walls. These holes were inclined slightly with respect to the lens axes so that shrinkage of the sealant would draw the lenses towards the axial registration surface (see Fig. 15.54). No retainers were needed to secure the lenses. The lens cells were held against the housing with spring clamps during alignment then tack bonded with epoxy at eight places.

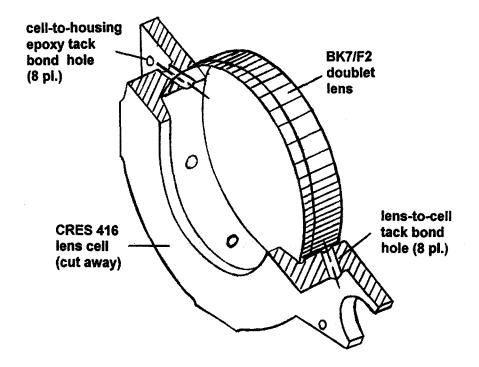


Figure 15.54 Cut away schematic of the lens mounting. (Adapted from Stubbs et al. 38)

The output end of each fiber bundle carried a ceramic ferrule that was tack bonded with epoxy through six access holes into a shuttle plug. The ferrules were rotated during insertion into the shuttle plugs to provide correct polarization plane orientation relative to slots in the tops of the shuttle plugs. These slots would engage alignment pins inserted temporarily in the top of the housing while the tack bonds securing the plugs were curing. The plugs slipped into two precision bores in the housing and were tack bonded in place after focusing. Epibond 1210 A/9861 epoxy was used for the tack bonds holding the lens cells and the shuttle plugs.

An optical alignment fixture was used during adjustment of the lens cells and in focusing the shuttle plugs in the housing. This fixture included precision stages and goniometers to move the optical parts transversely and angularly respectively and clamps to hold those parts while the adhesive joints cured. Figures 15.55 through 15.57 show the fixture and the stages used for these adjustments.

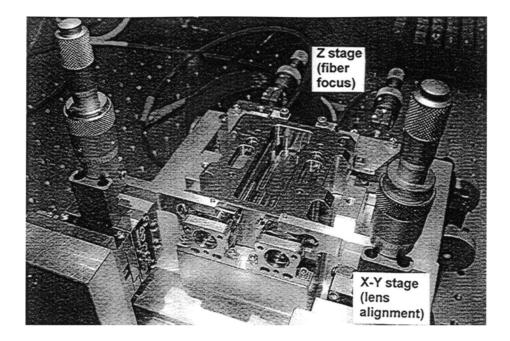


Figure 15.55 Front view of the alignment fixture showing stages used to align the lens cells for optimum beam pointing and quality. (From Stubbs et al. 38)

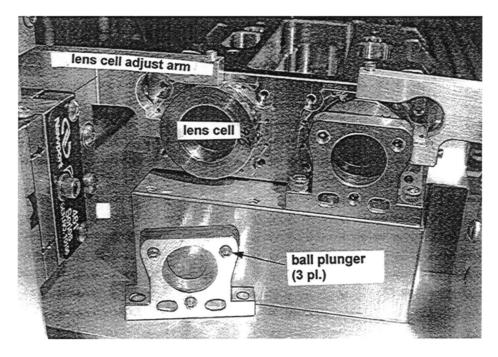


Figure 15.56 Close-up view of the lens cell alignment provisions for the collimator module. (From Stubbs et al. 38)

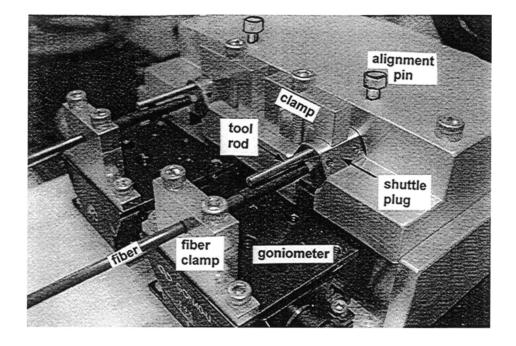


Figure 15.57 Rear view of the alignment fixture showing the goniometers used to align the input beams to the lenses. (From Stubbs et al.³⁸)

Alignment of the instrument was accomplished on an optical table in a clean room. The lens cells were supported by X-Y stages in front of the housing as indicated in Fig. 15.55. Each cell was pressed lightly against the flat datum surface on the housing by three ball plungers on a bracket attached to the fixture baseplate through a block. Figure 15.56 shows a close-up of this arrangement with one bracket removed for clarity. The X and Y stage motions were used to make the output laser beams parallel and at the proper horizontal separation. Then the cells were clamped to the housing and tack bonded in place. After the epoxy had cured, the clamps were removed.

At the back of the housing, the output beams from the fibers were tilted with the goniometers shown in Fig. 15.57 to center the beams in the lens apertures. The shuttle plugs were fine adjusted for focus using the Z stage mechanisms shown in Fig. 15.55. Beam quality was evaluated during final alignment with a laser beam imaging camera and a wavefront sensor during this adjustment. Diffraction limited performance was routinely achieved from the collimator module.

15.19 Lens Mountings for the JWST's NIRCam

A near infrared camera (NIRCam) is to be used with the James Webb Space Telescope (JWST) to observe very distant galaxies in the universe. This space-borne system is expected to provide distinct advantages over ground-based telescopes—even those with adaptive optics.⁴⁰ Proper function of the camera's optomechanical design depends largely on providing low-stress mountings for lenses made of LiF, BaF₂, and ZnSe that would accommodate temperature change from ~300K at launch to cryogenic operation at ~35K.

These crystals transmit at the appropriate IR wavelengths (0.6 to 5 μ m) for this application, but have low mechanical strengths. This makes them less desirable from the mechanical viewpoint, but they are superior to other crystals from a lens design viewpoint when used together to form a broad-band apochromatic design.⁴¹ In this section, we describe the mounting configuration for the LiF element, it being the most sensitive to mounting forces.⁴² We do not address the lens system design for the application other than to state that lens-to-lens alignment must be maintained at operating temperature to target values within 50 μ m and that lens diameters of 70- to 94-mm are involved.

Characteristics of the LiF material that must be taken into account are its CTE of \sim 37ppm/K (at 300K), its change in CTE of \sim 0.5% on cooling to 35K, its low apparent elastic limit of \sim 11 MPa (\sim 1600 lb/in.²), and the significant variability of its Young's modulus and Poisson's ratio values in various directions with respect to the cubic nature of the single-crystal structure. Under mechanical load, the crystalline lattice can suffer slippage in specific directions so the orientations of the applied forces must be optimized and the magnitudes of those forces controlled. Research into induced deformation of the crystal leads to an upper limit on stress of \sim 5 MPa.⁴³ It was decided early in the design phase that the LiF lens should not experience stress exceeding 2 MPa (290 lb/in.²).

15.19.1 Concept for axial constraint of the LIF lens

Conventional techniques for constraining axial motion of a single lens are discussed in Chapter 3. They include threaded retaining rings (see Fig. 3.17), continuous (flange) rings (see Fig. 3.21), and multiple cantilevered spring clips (see Fig. 3.23). A sophisticated form of the latter concept, created as a prototype design for mounting the LiF optic and described by Kvamme et al.,⁴⁴ is shown in Fig. 15.58. Twelve 6Al4V axially compliant titanium springs, symmetrically arranged around the edge of the lens surface, press against the lens through 0.500-mm (0.020-in.) thick strips of slightly compliant material (Neoflon fluro-polymer) to prevent direct contact between the metal and the crystal. The opposite surface of the lens rests on the inner edge of the flat (lapped) top surface of the base plate. From the JWST launch environmental specification,⁴⁵ it was determined that the total preload should be based on a static load of 54 times gravity. Assuming conservatively that only 5 of the 12 springs actually hold the lens in place during launch, analysis showed that a preload of 16.24 N (3.65 lb) would be needed from each spring and that this would produce a resolved stress in the lens' critical crystal plane of only 0.25 MPa (36 lb/in.²). The preload at each spring was set by controlling the thickness of a spacer between the bottom surface of the pad under the spring and the top surface of the central ring to produce the required spring deflection.

15.19.2 Concept for radial constraint of the LIF lens

Experimentation with a simple mount in which the NIRCam's LiF lens was spring-loaded radially against two fixed pads on the cell ID in a manner similar to that shown in Fig. 9.5 for radial constraint of a small mirror revealed excessive optical surface deformation when preloaded sufficiently to maintain centration of the LiF lens. The radially compliant centering ring shown in Fig. 15.59 was then developed. It was designed to be placed between the lens rim and the cell ID in the same manner as the Vespel centering ring shown in Fig. 14.28.

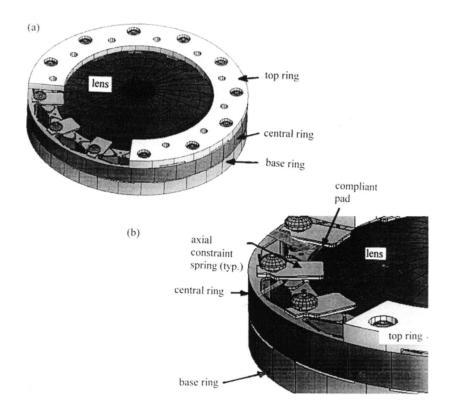


Figure 15.58 The prototype axial constraint used in the prototype mounting design. (a) Cutaway view, (b) close-up view. (From Kvamme et al.⁴⁴)

This ring is made of 6Al4V titanium and machined by EDM to provide twelve radially compliant diamond-shaped springs that press against the lens rim. This radial constraint ring forms the central ring designated in Fig. 15.61. A thin annular ring of Neoflan around the lens rim isolates the crystal from direct contact with the metal. Once again, the preload per spring was set at 16.24 N (3.65 lb) to hold the lens during launch. Because of symmetry, the lens should remain well centered to the mount during operation. Tests that confirm this are described in Section 15.19.4.

15.19.3 Analytical and experimental verification of the prototype lens mount

FEA analysis of the lens mounting design indicated a worst-case stress of ~0.5 MPa (~73 $lb/in.^2$) in the lens due to the applied axial and radial preloads. Similar analyses of stress buildup in a typical diamond-shaped spring and in a typical axial spring under preload also were conducted. The maximum stress in each spring during launch was determined to be acceptable.

The suitability of the axial and radial mounting design for the LiF lens was also checked by random vibration testing of the mounted lens to the applicable qualification vibration specification.⁴⁵ The reflected wavefront from the lens surface was evaluated before and after vibration and showed no significant change. This indicated that no dislocation of the crystal lattice had occurred.

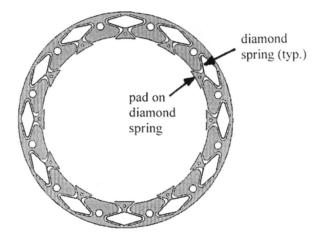


Fig. 15.59 Frontal view of the 12-pad diamond-element radial-constraint ring used between the lens rim and the cell ID in the prototype mounting design. This is the central ring of Fig. 15.58. (From Kvamme et al.⁴⁴)

15.19.4 Design and initial testing of flight hardware

In 2007, Kvamme et al.⁴⁶ reported on an improved, second generation LiF lens mounting design intended for use in the flight hardware with a 94-mm (3.700-in.) diameter lens. It has sixteen diamond-shaped radial springs and sixteen axial cantilevered springs. Figure 15.60 shows the radial constraint ring while Fig. 15.61 shows the configuration of one axial constraint.

During the assembly of the lens into its new mount, the lens cracked. Examination of the design revealed that the cell baseplate and the ring containing the diamond springs were deformed by the force exerted by the screws that held them together. This was caused by a small air gap in the load path of the screw. With a design change to eliminate the gap at each screw, the problem was alleviated.

Finite element analysis of this design, assuming only seven of the sixteen springs would carry the load, was conducted and proved to develop a maximum stress in the optic of ~0.69 MPa(~100 lb/in.²). This was acceptable in view of the predetermined design limit stress of 2 MPa (290 lb/in.²). The corresponding worst-case stress of 357 MPa (51,800 lb/in.²) in one of the diamond springs gave a quite acceptable safety factor with respect to the yield stress of the titanium.

Thermal analysis of the lens mounting design concentrated on cooling post launch, which could cause a thermal gradient across the lens. This posed a question as to whether the lens might fracture from thermal shock. The analysis indicated a worst-case gradient of ~1K might exist. This was considered not to be sufficient to cause damage. Cryogenic testing of a mounted lens through a range of 300K to 60K showed no damage and, surprisingly, a small improvement in surface figure at the low temperature. This improvement, which remained after testing, was attributed to the lens finding a more favorable and permanent orientation in its mount due to cooling.⁴⁷



Figure 15.60 Isometric view of the 16-pad diamond-element radialconstraint ring used in the second-generation mounting design. (From Kvamme et al.⁴⁶)

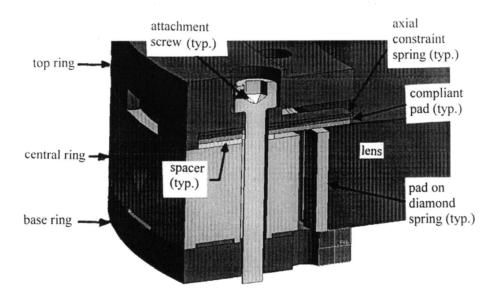


Figure 15.61 Isometric view of one axial constraint in the second-generation design. (From Kvamme et al.⁴⁶)

Tests were conducted to determine if the lens would remain centered after vibration and after cryogenic cycling. The test set-up is shown in Fig. 15.62. A dummy aluminum lens was substituted for the LiF because the density of that metal is practically the same as that of LiF, making it a good mass simulator. Aluminum posts with square cross-

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sections were attached to each surface of the dummy lens at their centers. KAMAN differential sensors were mounted as indicated in the figure to measure lateral motions of the lens during testing. Accelerometers were attached to the fixture to measure the actual 3-dimensional accelerations achieved during the test. The test fixture was attached to a vibration machine and subjected to the specified vibration spectrum. During the first run, a 5-µm lateral motion of the lens in one direction was measured. Subsequent runs showed the lens location to have stabilized and no further displacements were noted. This indicates that the lens settled in on the pads and then always returned to the same centered location after additional vibration cycles.

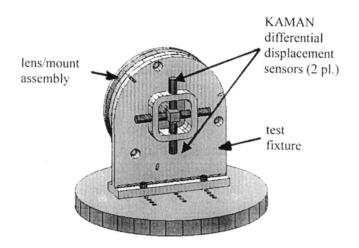


Figure 15.62 Test fixture used to measure lens decentration from specified vibration exposure. (From Kvamme et al.⁴⁴)

15.19.5 Long-term stability tests

To verify the long-term stability of the mounting configuration, 70-mm diameter and 94mm diameter lenses were mounted into mounts of the prototype configuration as described above and the surfaces evaluated interferometrically. They were retested periodically over several months. The surface figure errors remained essentially constant throughout these tests.

15.19.6 Further developments

A review of the NIRCam engineering test unit's mechanical system design, the process used in its assembly, the results of testing and evaluation of its mechanical aspects, and its optomechanical performance are described by Kvamme and Jacoby.⁴⁸

15.20 A Double-Arch Mirror Featuring Silicon-Foam-Core Technology

Silicon is compatible with forming very smooth optical surfaces, its CTE is low and its thermal conductivity is high. All of these attributes are favorable for making mirrors with

high optical performance and low sensitivity to temperature changes. In Section 8.6.3.6, we summarized the development during the 1999-2005 period of silicon foam cores used with single-crystal silicon facesheets, and later with CVD silicon cladding, to make lightweight mirrors.⁴⁹⁻⁵⁶ The use of low density Si foam as the core reduces mirror weight and increases specific stiffness significantly as compared with solid substrates, even when the latter are shaped to such lightweight forms as the double arch configuration. A specific example of a mirror made with current silicon foam core technology is described here.

View (a) of Fig. 15.63 shows a 21.65-in. (55-cm) diameter f/1 parabolic mirror, which is made in the classical double arch form. Goodman and Jacoby⁵⁷ describe it as an epoxy-bonded assembly comprising the three parts shown in view (b) of the figure. The SLMS mirror^a is meniscus shaped, 1.250-in. (3.175-cm) thick, and has a 5.0-in. (12.7-cm) diameter central perforation. Its optical surface has a roughness ≤ 1.0 nm rms and surface quality 40/20 and corresponds to the true parabola within 0.035-wave rms at 633-nm wavelength. It is coated to reflect 99.92%, 99.00%, and 90.00% at 1.315-1.319 µm, 1.06-1.08 µm, and 90.00% at 633-nm wavelengths, respectively. The core has a density of 10% to 12% relative to a solid. The Si cladding is ~0.050-in. (~1.27-mm) thick over the entire core.

The convex spherical back surface of the mirror is bonded with epoxy to a conformal concave spherical surface on the carbon/silicon carbide (C/SiC) mount to form an assembly 3.43-in. (8.71-cm) thick that weighs 27.6 lb (12.5 kg). For comparison, a solid Zerodur mirror of the same dimensions and configuration, such as that shown in half-sectional view in Fig. 15.64, would weigh ~46 lb (~20.86 kg). This represents a weight reduction to 60% of the solid version.

The physical properties of this lightweighted mirror are of importance in terms of its potential application as the primary mirror of a telescope for expanding and directing the beam from a high-energy laser.⁵⁸⁻⁶¹ While the coating is particularly efficient at the laser wavelength, some energy is absorbed and the mirror is heated. Because the type laser involved here does not deliver a uniform intensity distribution over its beam area, the temperature distribution at the mirror's surface will also be nonuniform. Figure 15.65 shows a representation of the beam intensity for a typical annular beam from such a laser. Because of the mirror's low CTE, the nonuniformly distributed absorbed heat will have only a small effect on the mirror's optical performance so we may define this design as athermal. Figure 15.66(a) shows an idealized prediction (i.e., one without heat losses) of the temperature distribution in the mirror's surface after a 50 second exposure to a laser beam in which 50 watts of energy is absorbed by the surface. Figure 15.66(b) shows the predicted p-v surface distortion while Table 15.5 lists the Zernike polynomial terms for that deformed surface. Note that piston, tilt, and focus are the dominant errors. The p-v error is seen to be 0.81 wave at 633 nm while the rms figure error is 0.17 waves at the same wavelength.

A feature of this new mirror design that can be used to improve hardware performance for this type application, but was not utilized in the above performance prediction is provision of a heat exchange manifold within the CSiC mount for flow of a

^a SLMS means "silicon lightweight mirrors" and is a trademark of Schafer Corporation.

coolant to assist in controlling temperature rise during exposure to the laser beam as might be needed for the application. The channel is divided into three arcuate regions that are individually provided with inlet and outlet connectors. The manifold is formed by bonding the CSiC ring of Fig. 15.63 into an annular recess in the back of the mount. See the section view of Fig. 15.66.

Mechanically, the bonded mirror/mount/coolant channel assembly has a fundamental frequency of ~1027 Hz. Peng et al.⁶² showed that the silicon foam has good vibration dampening characteristics which would reduce "ringing" following a disturbance. The assembly of Fig. 15.63(a) is designed to be attached to structure through Invar bosses (not shown) bonded with epoxy to the annular mounting interface on the back of the CSiC mount.

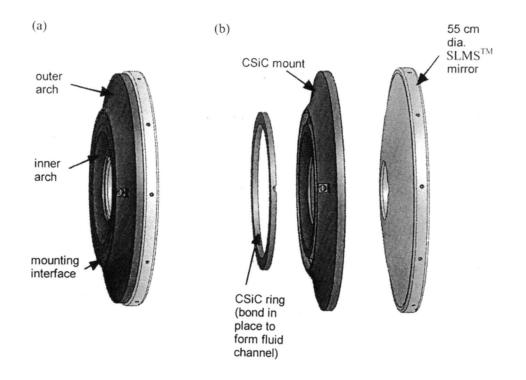


Figure 15.63 Lightweight silicon foam core mirror of double-arch configuration. (a) Complete assembly, and (b) three components that are bonded together to create the mirror assembly. (From Goodman and Jacoby.⁵⁷)

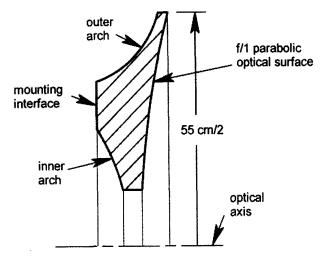


Figure 15.64 Half-section view of a solid double-arch mirror of the same dimensions as that shown in Fig. 15.63(a). (Adapted from Goodman and Jacoby.⁵⁷)

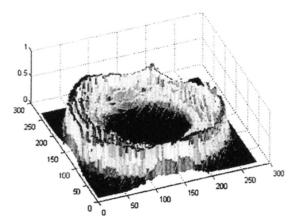


Figure 15.65 Three-dimensional representation of the annular nonuniform intensity distribution of a high-energy laser beam assumed to be incident on the mirror of Fig. 15.63(a). (From Goodman and Jacoby.⁵⁷)

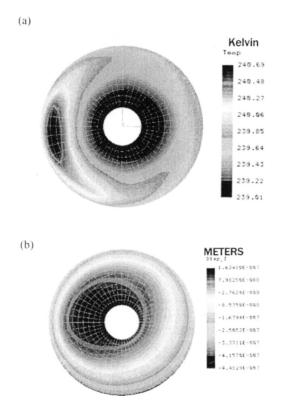


Figure 15.66 (a) Predicted temperature rise of the silicon mirror surface from the incident laser energy distribution per Fig. 15.65. (b) Predicted p-v surface distortion from temperature rise. (From Goodman and Jacoby.⁵⁷)

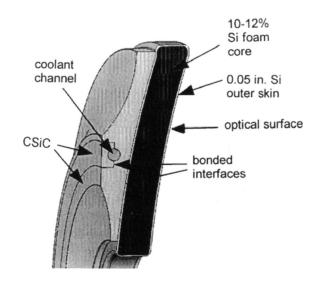


Fig. 15.67 Half-section view of the Si/Si foam mirror assembly of Fig. 15.63(a). (From Goodman and Jacoby.⁵⁷)

Table 15.5 Zernike decomposition of the predicted surface deformation of the Si/Si foam mirror assembly of Fig. 15.63(a) when irradiated by the laser energy distribution of Fig. 15.65.

Description	Term	Description	Term
piston	-3.73E-02	2 coma X	8.56E-03
tilt X	-9.80E-02	2 coma Y	3.55E-03
tilt Y	1.08E-01	2 trefoil Y	2.07E-03
focus	-2.60E-01	pentafoil Y	-3.19E-03
astigmatism X	2.99E-02	hexafoil X	1.58E-03
astigmatism Y	1.24e-02	2 tetrafoil X	1.06E02
trefoil X	-1.10E-02	2 astigmatism X	-3.38E -03
coma X	2.81E-02	2 spherical	-2.98E-02
coma Y	9.20E-03	integrated function value	-4.20E-08
trefoil Y	-1.28E-03	peak error in HeNe waves	2.37E-01
tetrafoil X	1.11E-02	valley error in HeNe waves	-5.78E-01
2 astigmatism X	-4.86E-03	p-v error in HeNe waves	8.15E-01
spherical	-4.42E-02	avg. error in HeNe waves	-2.17E-02
2 astigmatism Y	-1.13E-03	number of points	3853
tetrafoil Y	-1.13E-03	sum	-8.36E+01
pentafoil X	-2.72E-03	rms error in HeNe waves	1.74E-01
2 trefoil X	3.04E-03	sum of squares	7.36E-04

From Goodman and Jacoby.⁵⁷

15.21 References

- 1. Palmer, T.A. and D.A. Murray, Private communication, 2001.
- Erickson, D.J., Johnston, R.A., and Hull, A.B., "Optimization of the optomechanical interface employing diamond machining in a concurrent engineering environment," *Proceedings of SPIE* CR43, 1992:329.
- 3. Rhorer, R.L. and Evans, C.J., "Fabrication of optics by diamond turning," Chapter 41, in *Handbook of Optics*, 2nd ed., II, 1995.
- 4. Arriola, E.W., "Diamond turning assisted fabrication of a high numerical aperture lens assembly for 157 nm microlithography," *Proceedings of SPIE* **5176**, 2003:36.
- 5. Sanger, G.M., "The precision machining of optics," Chapter 6 in *Applied Optics and Optical Engineering* 10, 1987.
- Guyer, R.C., Evans, C.E. and Ross, B.D., "Diamond-turned optics aid alignment and assembly of a dual field infrared imaging sensor," *Proceedings of SPIE* 3430, 1998:109.
- Rayces, J.L., Foster, F., and Casas, R.E., "Catadioptric system," U.S. Patent 3,547,525, 1970.
- 8. Dictionary of Science and T echnology, C. Morris, ed., Academic Press, San Diego, 1992.
- Cassidy, L.W., "Advanced stellar sensors a new generation," in Digest of Papers, AIAA/SPIE/OSA Symposi um, Technol ogy for Space Ast rophysics C onference: The Next 30 Years, American Institute of Aeronautics and Astronautics, Reston, VA, 1982:164.

- 10. Bystricky, K.M., and Yoder, P.R., Jr., "Catadioptric lens with aberrations balanced with an aspheric surface," *Appl. Opt.* 24, 1983:1206.
- 11. Mast, T., Faber, S.M., Wallace, V., Lewis, J., and Hilyard, D., "DEIMOS camera assembly," *Proceedings of SPIE* 3786, 1999:499.
- 12. Mast, T., Brown, W., Gilmore, K., and Pfister, T., "DEIMOS detector mosaic assembly," *Proceedings of SPIE* 3786, 1999::493.
- 13. Hilyard, D.F., Laopodis, G.K., and Faber, S.M., "Chemical reactivity testing of optical fluids and materials in the DEIMOS Spectrographic Camera for the Keck II Telescope," *Proceedings of SPIE* **3786**, 1999::482.
- 14. Mast, T., Choi, P.I. Cowley, D., Faber, S.M., James, E., and Shambrook, A., "Elastomeric lens mounts," *Proceedings of SPIE* 3355, 1998::144.
- 15. Trsar, W.J., Benjamin, R.J., and Casper, J.F., "Production engineering and implementation of a modular military binocular," *Opt. Eng.*, 20, 1981:201.
- 16. Yoder, P.R., Jr., "Two new lightweight military binoculars," J. Opt. Soc. Am. , 50, 1960:491.
- 17. Sheinis, A.I., Nelson, J.E., and Radovan, M.V., "Large prism mounting to minimize rotation in Cassegrain instruments," *Proceedings of SPIE* **3355**, 1998:59.
- 18. Epps, H.W., and Miller, J.S., "Echellette spectrograph and imager (ESI) for Keck Observatory," *Proceedings of SPIE* **3355**, 1998:48.
- 19. Sutin, B.M., "What an optical designer can do for you AFTER you get the design," *Proceedings of SPIE* **3355**, 1998:134.
- Radovan, M.V., Nelson, J.E., Bigelow, B.C., and Sheinis, A.I., "Design of a collimator support to provide flexure control on Cassegrain instruments," *Proceedings of SPI E* 3355, 1998.
- 21. Bigelow, B.C., and Nelson, J.E., "Determinate space-frame structure for the Keck II Echellete Spectrograph and Imager (ESI)," *Proceedings of SPIE* **3355**, 1998:164.
- 22. Iraninejad, B., Lubliner, J., Mast, T., and Nelson, J., "Mirror deformations due to thermal expansion of inserts bonded to glass," *Proceedings of SPIE* **748**, 1987:206.
- 23. Yoder, P.R., Jr., Opto-Mechanical Systems Design, 3rd ed., CRC Press, Boca Raton, 2005.
- 24. Shipley, A., Green, J.C., and Andrews, J.P., "The design and mounting of the gratings for the Far Ultraviolet Spectroscopic Explorer," *Proceedings of SPIE* **2542**, 1995:185.
- Shipley, A., Green, J., Andrews, J., Wilkinson, E., and Osterman, S., "Final flight grating mount design for the Far Ultraviolet Spectroscopic Explorer," *Proceedings of* SPIE 3132, 1997:98.
- 26. Tiodize Process Literature, Tiodize Co., Inc., Huntington Beach, CA.
- 27. Fanson, J., Fazio, G., Houck, J., Kelly, T., Rieke, G., Tenerelli, D., and Whitten, M., "The Space Infrared Telescope Facility (SIRTF), *Proceedings of SPIE* 3356, 1998:478.
- Gallagher, D.B., Irace, W.R., and Werner, M.W., "Development of the Space Infrared Telescope Facility (SIRTF)," *Proceedings of SPIE* 4850, 2003:17.
- 29. Coulter, D.R., Macenka, S.A., Stier, M.T., and Paquin, R.A., "ITTT: a state-of-the-art ultra-lightweight all-Be telescope," *Proceedings of SPIE* CR67, 1997:277.
- Chaney, D., Brown, R.J., and Shelton, T., "SIRTF prototype telescope," *Proceedings* of SPIE 3785, 1999:48.
- Schwenker, J.P., Brandl, B.R., Burmester, W.L., Hora, J.L., Mainzer, A.K., Quigley, P.C., and Van Cleve, J.E., "SIRTF-CTA optical performance test," *Proceedings of* SPIE 4850, 2003:304.

- Schwenker, J.P., Brandl, B.R., Hoffman, W.F., Burmester, W.L., Hora, J.L., Mainzer, A.K., Mentzell, and Van Cleve, J.E., "SIRTF-CTA optical performance test results," *Proceedings of SPIE* 4850, 2003:30.
- Lee, J.H., Blalock, W., Brown, R.J., Volz, S., Yarnell, T., and Hopkins, R.A., "Design and development of the SIRTF cryogenic telescope assembly (CTA)," *Proceedings of* SPIE 3435, 1998:172.
- Hopkins, R.A., Finley, P.T., Schweickart, R.B., and Volz, S.M., "Cryogenic/thermal system for the SIRTF cryogenic telescope assembly," *Proceedings of SPIE* 4850, 2003:42.
- 35. Finley, P.T., Oonk, R.L., and Schweickart, R.B., "Thermal performance verification of the SIRTF cryogenic telescope assembly," *Proceedings of SPIE* **4850**, 2003:72.
- Mainzer, A.K., Young, E.T., Greene, T.P., Acu, J., Jamieson, T., Mora, H., Sarfati, S., and VanBezooijen, R., "The pointing calibration & reference sensor for the Space Infrared Telescope Facility, "*Proceedings of SPIE* 3356, 1998:1095.
- 37. van Bezooijen, R.W.H., "SIRTF autonomous star tracker," *Proceedings of SPIE* **4850**, 2003:108.
- 38. Stubbs, D., Smith, E., Dries, L., Kvamme, T., and Barrett, S., "Compact and stable dual fiber optic refracting collimator," *Proceedings of SPIE* **5176**, 2003:192.
- 39. Stubbs, D.M. and Bell, R.M., "Fiber optic collimator apparatus and method," U.S. Patent No. 6,801,688, 2004.
- Krist, J.E., Beichman, C.A., Trauger, J.T., Rieke, M.J., Someretein, S., Green, J.J., Horner, S.D., Stansberry, J.A., Shi, F., Meyer, M.R., Stapelfeldt, K.R., and Roellig, T.L., "Hunting planets and observing disks with the JWST NIRCam coronagraph," *Proceedings of SPIE* 6693, 2007.
- 41. Jamieson, T.H., "Decade wide waveband optics," *Proceedings of SPIE* 3482, 1998:306.
- 42. Kvamme, E.T., Earthman, J.C., Leviton, D.B., and Frey, B.J., "Lithium fluoride material properties as applied on the NIRCam instrument," *Proceedings of SP IE*, **59040N**, 2005.
- 43. Johnson, W.G., and Gilman, J.J., "Dislocation velocities, dislocation densities, and plastic flow in lithium fluoride," J. Appl. Phys., 30, Feb. 1959.
- 44. Kvamme, E.T., Trevias, D., Simonson, R., and Sokolsky, L., "A low stress cryogenic mount for space-borne lithium fluoride optics," *Proceedings of SPIE* **58770T**, 2005.
- 45. Goddard Environmental Vibration Specification GEVS-SE Rev.A, 1996.
- 46. Kvamme, E.T., and Michael Jacoby, "A second generation low stress cryogenic mount for space-borne lithium fluoride optics," *Proceedings of SPIE* 669201, 2007.
- Huff, L.W., Ryder, L.A., and Kvamme, E.T., "Cryo-test results of NIRCam optical elements," *Proceedings of SPIE* 6692, 2007:6692G.
- Kvamme, E.T. and Jacoby, M., "Opto-mechanical testing results for the Near Infrared camera on the James Webb Space Telescope," (To be published in *Proceedings* of SPIE 7010, 2008).
- 49. Fortini, A.J., "Open-cell silicon foam for ultralight mirrors," *Proceedings of SPIE* 3786, 1999:440.
- 50. Jacoby, M.T., Montgomery, E.E., Fortini, A.J., and Goodman, W.A., "Design, fabrication, and testing of lightweight silicon mirrors," *Proceedings of SPIE* **3786**, 1999:460.
- Goodman, W.A. and Jacoby, M.T., "Dimensionally stable ultra-lightweight silicon optics for both cryogenic and high-energy laser applications," *Proceedings of SPIE* 4198, 2001:260.

- 52. Jacoby, M.T., Goodman, W.A., and Content, D.A., "Results for silicon lightweight mirrors (SLMS)," *Proceedings of SPIE* **4451**, 2001:67.
- 53. Goodman, W.A., Müller, C.E., Jacoby, M.T., and Wells, J.D. "Thermo-mechanical performance of precision C/SiC mounts," *Proceedings of SPIE* **4451**, 2001:468.
- 54. Goodman, W.A., Jacoby, M.T., Krödel, M., and Content, D.A., "Lightweight athermal optical system using silicon lightweight mirrors (SLMS) and carbon fiber reinforced silicon carbide (Cesic) mounts," *Proceedings of SPIE* **4822**, 2002:12.
- 55. Jacoby, M.T., Goodman, W.A., Stahl, H.P., Keys, A.S., Reily, J.C., Eng, R., Hadaway, J.B., Hogue, W.D., Kegley, J.R., Siler, R.D., Haight, H.J., Tucker, J., Wright, E.R., Carpenter, J.R., and McCracken, J.E., "Helium cryo testing of a SLMS[™] (silicon lightweight mirrors) athermal optical assembly," *Proceedings of* SPIE **5180**, 2003:199.
- 56. Eng, R., Carpenter, J.R., Foss, C.A., Jr., Hadaway, J.B., Haight, H.J., Hogue, W.D., Kane, D., Kegley, J.R., Stahl, H.P., and Wright, E.R., "Cryogenic performance of a lightweight silicon carbide mirror," *Proceedings of SPIE* **58680Q**, 2005.
- 57. Goodman, W.A. and Jacoby, M.T., "SLMS athermal technology for high-quality wavefront control," *Proceedings of SPIE* 6666, 2007:66660Q.
- Paquin, R. A., "Properties of Metals," Chapt. 35 in *Handbook of Optics*, 2nd ed., Vol. II, Optical Society of America, Washington, 1994.
- 59. Jacoby, M.T. and Goodman, W.A., "Material properties of silicon and silicon carbide foams, *Proceedings of SPIE* **58680J**, 2005.
- Boy, J. and Krödel, M., "Cesic lightweight SiC composite for optics and structures," *Proceedings of SPIE* 586807, 2005.
- 61. Krödel, M., "Cesic -- Engineering material for optics and structures," *Proceedings of SPIE* **58680A**, 2005.
- Peng, C.Y., Levine, M., Shido, L., Jacoby, M., and Goodman, W., "Measurement of vibrational damping at cryogenic temperatures for silicon carbide foam and silicon foam materials," *Proceedings of SPIE* 586801, 2005.

APPENDIX A Unit Conversion Factors

To facilitate conversion from U.S. Customary System (USC) units to metric or Systèm International (SI) units, we list here the standard factors for changing the units commonly used in measuring selected physical parameters mentioned in this text. This involves multiplying the value in USC units by the factor listed. Conversion in the reverse direction requires division by the same factor.

```
To change length in:
          inches (in.) to meters (m), multiply by 0.0254
          inches (in.) to millimeters (mm), multiply by 25.4
          inches (in.) to nanometers (nm), multiply by 2.54 \times 10^7
          feet (ft) to meters (m), multiply by 0.3048
To change weight in:
          pounds (lb) to kilograms (kg), multiply by 0.4536
          ounces (oz) to grams (g), multiply by 28.3495
To change force or preload in:
          pounds (lb) to newtons (N), multiply by 4.4482
          kilograms (kg) to newtons, multiply by 9.8066
To change linear force in:
          lb/in. to N/mm, multiply by 0.1751
          lb/in. to N/m, multiply by 175.1256
To change spring compliance in:
          in./lb to m/N, multiply by 5.7102 \times 10^{-3}
To change temperature dependence of preload in:
          lb/°F to N/°C, multiply by 8.0068
To change pressure, stress, or units for Young's modulus in:
          lb/in.<sup>2</sup> (psi) to N/m<sup>2</sup> (pascals), multiply by 6894.757
          lb/in.^2 (psi) to megapascals (MPa), multiply by 6.8948 \times 10^{-3}
          lb/in.<sup>2</sup> (psi) to N/mm<sup>2</sup>, multiply by 6.895 \times 10^{-3}
          atmospheres to MPa, multiply by 0.1103
          atmospheres to lb/in.<sup>2</sup>, multiply by 14.7
          torr to pascals, multiply by 133.3
To change torque or bending moment in:
          lb-in. to N-m, multiply by 0.11298
          oz-in. to N-m, multiply by 7.0615 \times 10^{-3}
          lb-ft to N-m, multiply by 1.35582
To change volume in:
          in.<sup>3</sup> to cm<sup>3</sup>, multiply by 16.3871
To change density in:
          lb/in.3 to g/cm3, multiply by 27.6804
To change acceleration in:
          gravitational units (g) to m/sec<sup>2</sup>, multiply by 9.80665
          ft/sec^2 to m/sec^2, multiply by 0.3048
```

To change temperature in:

degrees F to degrees C, subtract 32 and multiply by 5/9 degrees C to degrees F, multiply by 9/5 and add 32 degrees C to K, add 273.1

APPENDIX B Mechanical Properties of Materials

This appendix provides tables of material properties derived from various sources as noted. Included are:

- Table B1 Mechanical properties of 50 Schott optical glasses.
- Table B2 Mechanical properties of seven radiation resistant Schott glasses.
- Table B3 Selected optical and mechanical characteristics of some optical plastics.
- Table B4 Optomechanical properties of selected alkali halides and alkaline earth halides.
- Table B5 Optomechanical properties of selected IR-transmitting glass and other oxides.
- Table B6 Optomechanical properties of diamond and selected IR-transmitting semiconductor materials.
- Table B7 Optomechanical properties of selected IR-transmitting chalcogenide materials.
- Table B8a Mechanical properties of selected nonmetallic mirror substrate materials.
- Table B8b Mechanical properties of selected metallic and composite mirror substrate materials.
- Table B9 Comparison of material figures of merit especially pertinent to mirror design.
- Table B10a Characteristics of aluminum alloys used in mirrors.
- Table B10b Common temper conditions for aluminum alloys.
- Table B10c Characteristics of aluminum matrix composites.
- Table B10d Beryllium grades and some of their properties.
- Table B10e Characteristics of major silicon carbide types.
- Table B11 Comparison of metal matrix and polymer matrix composites.
- Table B12 Mechanical properties of selected metals used for mechanical parts in optical instruments.
- Table B13 Typical characteristics of a generic optical cement.
- Table B14 Typical characteristics of representative structural adhesives.
- Table B15 Typical characteristics of representative elastomeric sealants.
- Table B16 Minimum values for fracture strength S_F of infrared window materials.

	F	Table B1 C	B1 Optomechanical properties of 50 Schott optical glasses (Page 1 of 3).	inical prop	berties of 5	0 Schott o	optical gla	sses (Pag	e 1 of 3).		
		Int'l	Young's		Poisson's	Thermal expansion	xpansion				
Rank &	Glass	glass	modulus (E_G)	(0)	ratio (coefficent (α_G)	(α_G)	$K_G = (1 - v^2)/E_G$	$^{2})/E_{G}$	Density (p)	
(reference)	name	code	(Ib/in. ²)	(MPa)	v _G)	(1/°F)	(1/°C)	(in. ² /lb)	(1/Pa)	(lb/in. ³)	(g/cm ³)
1 (b)	N-FK5	487704	9.00E+6	6.20E+4	0.232	5.1E-6	9.2E-6	1.05E-7	1.53E-11	0.089L	2.45L
2 (a)	K10	501564	9.44E+6	6.50E+4	0.190L	3.6E-6	6.5E-6	1.02E-7	1.48E–11	0.091	2.52
3 (a)	N-ZK7	508612	1.0E+7	7.00E+4	0214	2.5E-6L	4.5E-6L	9.54E-8	1.34E-11	0.090	2.49
4 (a)	K7	511604	1.0E+7	6.90E+4	0.214	4.7E-6	8.4E-6	9.52E-8	1.38E-11	0.091	2.53
5 (a)	N-BK7	517642	1.2E+7	8.20E+4	0.206	3.9E-6	7.1E-6	8.05E-8	1.17E-11	0.091	2.51
6 obsolete	BK7	517642	1.17E+7	8.20E+4	0.208	3.9E-6	7.1E-6	8.18E-8	1.18E-11	0.091	2.51
7 (a)	N-K5	522595	1.03E+7	7.10E+4	0.224	4.6E-6	8.2E-6	9.24E-8	1.34E-11	0.094	2.59
8 (a,b)	N-LLF6	532489	1.05E+7	7.20E+4	0.211	4.3E-6	7.7E-6	9.15E-8	1.33E-11	0.091	2.51
9 (a)	N-BaK2	540597	1.03E+7	7.10E+4	0.233	4.4E-6	8.0E-6	9.18E-8	1.33E-11	0.103	2.86
10 (a)	LLFI	548459	8.71E+6	6.00E+4	0.208	4.5E-6	8.1E6	1.10E-7	1.59E-11	0106	2.94
11 (a)	N-PSK3	552635	1.22E+7	8.40E+4	0.226	3.4E-6	6.2E-6	7.85E–8	1.14E–11	0.105	2.91
12 (a)	N-SK11	564608	1.15E+7	7.90E+4	0.239	3.6E-6	6.5E-6	8.23E-8	1.19E-11	0.111	3.08
13 (a)	N-BAK1	573575	1.06E+7	7.30E+4	0.252	4.2E-6	7.6E–6	8.84E-8	1.28E-11	0.115	3.19
14 (a)	N-BaLF4	580538	1.12E+7	7.70E+4	0.245	3.6E6	6.5E-6	8.42E-8	1.22E-11	0.112	3.11
15 (a)	LFS	581409	8.56E+6	5.90E+4	0.223	5.1E-6	9.1E-6	1.11E-7	1.61E-11	0.116	3.22
16 (a)	N-BaF3	583466	1.19E+7	8.20E+4	0.226	4.0E-6	7.2E-6	8.00E-8	1.16E–11	0.101	2.79
17 (a)	FS	603380	8.42E+6	5.80E+4	0.220	4.4E-6	8.0E-6	1.13E-7	1.64E-11	0.125	3.47
18 (a)	N-BaF4	606437	1.23E+7	8.50E+4	0.231	4.0E-6	7.2E-6	7.68E–8	1.11E-11	0.104	2.89
19 (a)	F4	617366	8.13E+6	5.60E+4	0.222	4.6E-6	8.3E-6	1.17E-7	1.70E–11	0.129	3.58
20 (a)	N-SSK8	618498	1.22E+7	8.40E+4	0.251	4.0E-6	7.2E–6	7.69E–8	1.12E-11	0.118	3.27
21 (a)	F2	620364	8.27E+6	5.70E+4	0.220	4.6E-6	8.2E-6	1.15E-7	1.67E-11	0.130	3.61
22 (a)	N-F2	620364	1.19E+7	8.20E+4	0.228	4.3E-6	7.8E-6	7.97E-8	1.16E–11	0.096	2.65
23 (b)	N-SK16	620603	1.29E+7	8.90E+4	0.264	3.5E-6	6.3E-6	7.21E-8	1.04E-11	0.129	3.58
							ļ	1.19E-		0.00	
24 (a)	SF2	648339	7.98E+6L	5.5E+4L	0.227	4.7E6	8.4E6	7H	1.72E-11H	0.139	3.86
25 (a)	N-LaK22	621229	1.31E+7	9.00E+4	0.266	3.7E-6	6.6E-6	7.12E- 8L	1.03E-11L	0.135	3.73
26 (b)	N-BaF51	652450	1.32E+7	9.19E+4	0.262	4.7E-6	8.4E6	7.06E-8	1.03E-11	0.120	3.33

	Tal	Table B1 Op	otomechani	Optomechanical properties of 50 Schott optical glasses (Page 2 of 3).	es of 50 Sc	chott opti	cal glass	es (Page :	2 of 3).		
		Int'l	Vound's		Poisson's	Thermal					
Rank &	Glass	glass	modulus (E)		ratio	coefficent (α_G)	(α ^C)	$K_G = (1 - v^2)/E_G$	$^{2})/E_{G}$	Density (p)	_
(reference)	name	code	(lb/in. ²)	(MPa)	(v _G)	(1/°F)	(1/°C)	(in. ² /lb)	(1/Pa)	(lb/in. ³)	(g/cm ³)
27 (b)	N-SSK5	658509	1.28E+7	8.80E+4	0.278	3.8E-6	6.8E–6	7.23E8	1.05E- 11	0.134	3.71
28 (a)	N-BaSF2	664360	1.22E+7	8.40E+4	0.247	3.9E-6	7.1E-6	7.71E-8	1.12E- 11	0.114	3.15
29 (a)	SF5	673322	8.13E+6	5.60E+4	0.233	4.6E–6	8.2E6	1.16E-7	1.69E- 11	0.147	4.07
30 (a)	N-SF5	673322	1.26E+7	8.70E+4	0.237	4.4E6	7.9E6	748E-8	1.08E- 11	0.103	2.86
31 (a)	N-SF8	689313	1.28E+7	8.80E+4	0.245	4.8E–6	8.6E–6	7.23E–8	7.07E- 11	0.105	2.90
32 (a)	SF15	699301	8.71E+6	6.00E+4	0.235	4.4E6	7.9E-6	1.09E-7	1.57E- 11	0.147	4.06
33 (a)	N-SF15	699302	1.31E+7	9.00E+4	0.243	4.4E–6	8.0E-6	7.21E–8	1.04E- 11	0.105	2.92
34 (a)	SFI	717295	8.13E+6	5.60E+4	0.232	4.5E-6	8.1E-6	1.16E-7	1.69E- 11	0.161	4.46
35 (a)	N-SF1	717296	1.31E+7	9.00E+4	0.250	5.1E-6	9.1E6	7.18E-8	1.04E- 11	0.109	3.03
36 (b)	N-LaF3	717480	1.39E+7	9.60E+4	0.286	4.2E–6	7.6E-6	6.66E–8	9.66E- 12	0.150	4.14
37 (a)	SF10	728284	9.29E+6	6.40E+4	0.232	4.2E-6	7.5E–6	1.02E-7	1.48E– 12	0.155	4.28
38 (a)	N-SF10	728285	1.26E+7	8.70E+4	0.252	5.2E-6	9.4E-6	7.42E8	1.08E- 11	0.110	3.05
39 (b)	N-LaF2	74449	1.39E+7	9.60E+4	0.288H	4.5E-6	8.1E-6	6.73E-8	9.76E- 12	0.155	4.30
40 (b)	LaFN7	750350	1.16E+7	8.00E+4	0.280	2.9E–6	5.3E6	7.94E-8	1.15E- 11	0.158	4.38

	r-	Table B1 C	Table B1 Optomechanical properties of 50 Schott optical glasses (Page 3 of 3).	nical proper	ties of 50 S	schott op	tical glas	ses (Page	3 of 3).		
		ľnt'l	Voling's			Thermal				Dometer (o	
Rank &	Glass	glass	modulus (E)		Poisson's	coefficent (α_G)	t (a _G)	$K_G = (1 - v^2)/E_G$	$^{2})/E_{G}$	(lb/in. ³)	
(reference)	name	code	(lb/in. ²)	(MPa)	ratio (v _G)	(1/°F)	(1/°C)	(in. ² /lb)	(1/Pa)		
									9.65E-		
41 (b)	N-LaF7	749348	1.39E+7	9.60E+4	0.271	4.0E-6	7.3E-6	6.65E-8	12	0.135	3.73
									1.68E-		
42 (b)	SF4	755276	8.13E+6	5.60E+4	0.241	4.4E-6	8.0E-6	1.15E-7	11	0.173	4.79
						5.3E-	9.5E-		1.04E		
43 (b)	N-SF4	755274	1.31E+7	9.00E+4	0.256	H9	6H	7.16E-8	11	0.114	3.15
									1.45E-		
44 (a)	SF14	762265	9.44E+6	6.50E+4	0.231	3.7E-6	6.6E-6	1.00E-7	11	0.164	4.54
									1.43E-		
45 (a)	SF11	785258	9.58E+6	6.60E+4	0.235	3.4E-6	6.1E-6	9.87E-8	11	0.170	4.74
									1.65E-		
46 (a)	SF56A	785261	8.27E+6	5.70E+4	0.239	4.4e-6	7.9E-6	1.14E-7	11	0.178	4.92
						1			1.03E-		
47 (a)	N-SF56	785261	1.32E-7	9.10E+4	0.255	4.8E-6	8.7E-6	7.08E-8	11	0118	3.28
									1.71E-		
48 (a)	SF6	805254	7.98E+6L	5.50E+4L	0.244	4.5E-6	8.1E-6	1.18E-7	11	0.187H	5.18H
									$1.00E^{-}$		
49 (a)	N-SF6	805254	1.35E+7	9.30E+4	0.262	5.0E-6	9.0E6	6.90E-8	11	0.122	3.37
				·					8.35E-		
50 (a)	LaSFN9	850322	1.60E+7H	1.10E+5H	0.286	4.1E-6	7.4E-6	5.76E-8	12	0.160	4.44
Ratio: (high/low)	h/low)		1.75		1.51	2.12		2.07		2.11	
Glass selection	ion from (a) W	Walker, B.H.,	Glass selection from (a) Walker, B.H., <i>The Photonics Design and Applications Handbook</i> , Lauren Publishing, Pittsfield, 1993: H-356; and (b)	s Design and A	lpplications H	landbook, L	auren Publ	ishing, Pittsf	ield, 1993: F	I-356; and	(q)

ã Į0 Zhang, S. and Shannon, R.R., Opt. Eng. 34, 1995: 3536

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	Glace	Int'l olace			Young's	_	Poisson's ratio	Thermal expanding $T_{(\alpha_{c})}$	Thermal expansion coefficent (a.c)	Density (n)	
Rank		code	n_d	\mathbf{v}_d	(lb/in. ²)	(MPa)	(v_G)	(1/°F)	(1/°C)	(lb/in. ³)	(g/cm ³)
	BK7G18	520636	1.51975	63.58	1.19E7	8.2E4	0.204	3.9E6	7.0E–6	0.091	2.52
	LF5G19	597399	1.59655	39.89	8.12E6	5.6E4	0.242	5.9E-6	10.7E-6	0.108	3.30
	LF5G15	584408	1.58397	40.83	8.70E6	6.0E4	0223	5.1E-6	9.1E-6	0.105	3.23
	K5G20	523568	1.52344	56.76	9.86E6	6.8E4	0.222	5.0E-6	9.0E-6	0.084	2.59
	LAK9G15	691548	1.69064	54.78	1.58E7	10.9E4	0.284	3.5E-6	6.3E-6	0.112	3.43
	F2G12	621366	1.62072	36.56	8.41E6	5.8E4	0.220	4.5E-6	8.1E-6	0.117	3.60
	SF6G05	809253	1.80906	25.28	7.98E6	5.5E4	0.244	4.3E-6	7.8E6	0.169	5.20

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Table B2 Optome
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		mondo nanceiro o	echanical charac	able by belected optomechanical characteristics of optical plastics.	ai piasucs.	
				Max. service		H ₂ O
				temperature	Thermal conductivity	absorption
Name	n_d	$CTE (\times 10^{-5})^{\circ}C)$	Density (g/cm ³)	(°C)	[cal/(sec·cm·°C×10 ⁻⁵)]	(% in 24 h)
Polymethyl methacrylate	1.4918	6.0	1.18	85	4-6	0.3
Polystyrene	1.5905	6.4-67	1.05	80	2.4-3.3	0.03
Methyl methacrylate						
styrene copolymer	1			č		
(NAS)	1.564	5.6	1.13	85	4.5	0.15
Styrene acrylonitrile						
(SAN)	1.5674	6.4	1.07	75	28	0.28
Polycarbonate	1.5855	6.7	1.25	120	4.7	0.2-0.3
Polymethyl pentene		11.7	0.835	115	4.0	0.01
Polyamide (Nylon)		8.2	1.185	80	5.1-5.8	1.5-3.0
Polyarylate		6.3	1.21		7.1	0.26
Polysulfone		2.5	1.24	160	2.8	0.1-0.6
Polyallyl diglycol						
carbonate (CR39)	1.504		1.32	100	4.9	
Polyethersulfone		5.5	1.37	200	3.2-4.4	
Polychloro-						
trifluoroethelyne		4.7	2.2	200	6.2	
From: Lytle, J.D., "Polymeric Optics", Chapter 34 in OSA Handbook of Optics II, 2 nd ed., McGraw-Hill, New York, 1995.	c Optics", C	Chapter 34 in OSA Hai	ndbook of Optics II, 2	nd ed., McGraw-Hill,	New York, 1995.	

anical characteristics of optical plastics. ĉ Table B3 Selected

Table	B4 Optomecha	Table B4 Optomechanical properties of selected alkali halides and alkaline earth halides (Page 1 of 2).	selected	alkali halides	and alkalin	e earth ha	lides (Pag	e 1 of 2).
Material name and (symbol)	Refractive index $n (\underline{a}) \lambda$ in μm	(<i>dn/dT</i>) _{REL} @ λ in μm & 20 to 40°C (×10 ^{-6/o} C)	CTE a (×10 ⁻ ¹⁰ /°C)	Young's modulus E (×10 ⁴ MPa)	Poisson's ratio (v _G)	Density (p) (g/cm ³)	Knoop hardness	$K_G = (1 - v_G^2)/E_G(x)$ $(1 - v_G^2)(1/MPa)$
Barium	1.463 @ 0.63 1.458 @ 3.8		(82	
fluoride (BaF ₂)	$1.449 ext{ (a) 5.3}$ $1.396 ext{ (a) 10.6}$	-16.0 @ 0.63	6.7 @ 75K	5.32	0.343	4.89	g ooc) (bad)	1.659
	1.4679 @ 0.248							
	1.4317 @							
	0.706							
	1.4285 @							
Calcium	1.060							
fluoride	1.411 @ 3.8	-10.4						
$(CaF_2)^*$	1.395 @ 5.3	@ 1.060	18.4	7.6	0.260	3.18	158	1.227
-	1.555 @ 0.6			•				
Potassium	1.537 @ 2.7						7	
bromide	1.529 @ 8.7	-41.9 @ 1.15	25.0 @				(200 g	
(KBr)	1.515 @ 14	-41.1 @10.6	75K	2.69	0.203	2.75	load)	3.564
	1.474 @ 2.7							
Potassium	1.472 @ 3.8						7.2	
chloride	1.469 @ 5.3	-36.2 @ 1.15	-				(200 g	
(KCI)	1.454 @ 10.6	-34.8 @ 10.6	36.5	2.97	0.216	1.98	load)	3.210
Lithium	1.394 @ 05	-16.0 @ 0.46	5.5 @				102-113	
fluoride	$1.367 extbf{@} 3.0$	-16.9 @ 1.15	77K				(600 g	
(LiF)	1.327 @ 5.0	-14.5 @ 3.39	7@20C	6.48	0.225	2.63	load)	1.465

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		$(dn/dT)_{\text{REL}} \otimes \lambda$						$K_{G} = (1 -$
Material	Refractive	in µm	CTE a	Young's		Density		$v_{G}^{2})/E_{G}$
name and	index $n \otimes \lambda$	& 20 to 40°C	(×10 ⁻	modulus E_G	Poisson's	(d)	Knoop	(×10 ⁻⁵) (1/
(symbol)	in µm	(×10 ⁻⁶ /°C)	¹⁰ /°C)	(×10 ⁴ MPa)	ratio (v_G)	(g/cm ³)	hardness	MPa)
	1.384 @							<u>.</u>
	0.40**							
	1.356 @							
Magnesium	3.80**		14.0					
fluoride	1.333 @							
(MgF_2)	5.30**	+0.88 @ 1.15	8.9 (1)	16.9	0.269	3.18	415	0.549
Sodium	1.525 @ 2.7						15.2	
chloride	1.522 @ 3.8						(200 g	
(NaCl)	1.517 @ 5.3	$-36.3 ext{ (a) } 3.39$	39.6	4.01	0.28	2.16	load)	2.298
Thallium	2.602 @ 0.6	-254 @ 0.6						
bromo-	2.446 @ 1.0	-240 @ 1.1					40.2	
iodide	2.369 @ 10.6	-233 @ 10.6					(200 g	
(KRS5)	2.289 @ 30	-152 @ 40	58	1.58	0.369	7.37	load)	5.467
* Values are fo	r Schott Lithotec-C	· Values are for Schott Lithotec-CaF ₂ 434952.318 material used in microlithographic optics.	rrial used in	microlithographic o	ptics.			
** Birefringent	** Birefringent material, O means ordinary axis.	ordinary axis.						

** Biretringent material, O means ordinary axis. Sources: P.R. Yoder, Jr., Opto-Mechanical Sy stems Design, 3rd ed., CRC Press, Boca Raton, 2005; W.J. Tropf, M.E. Thomas, and T.J. Harris, "Properties of crystals and glasses," Chapt. 33 in OSA Han dbook of Optic s, 2nd ed., Vol. II, McGraw-Hill, New York, 1995.

								$K_G = (1 -$
Material	Refractive		CTE (a)	Young's				$v_G^2)/E_G$
name and	index <i>n</i> @ \ in	$(dn/dT)_{\text{REL}}$	(×10 ⁻	modulus (E)	Poisson's	Density	Knoop	$(\times 10^{-5})$
(symbol)	mm	(X10 7°C)	(<u>)</u>	(XIU MPa)	rauo (v _G)	(m)(g)(d)	naruness	(I/ INITA)
	1.801 @ 0.5			,				
Aluminum	1.779 @ 1.0						1850	
oxynitride	$1.761 \overset{\circ}{@} 2.0$		5.65				(200g	
(ALON)	1.653 @ 5.0		30-200°C	32.3	0.24	3.69	load)	0.292
							1370	
Sapphire*	1.684 @ 3.8		5.6 ()				(1000g	
$(A1_2O_3)$	1.586 @ 5.8	13.7	5.0 (1)	40.0	0.27	3.97	load)	0.232
			-0.6 @					
	1.561 @ 0.193	10-11.2 @	73K					
Fused silica	1.460 @ 0.55	73K	0.58 @		·			
(Corning	1.433 @ 2.3	0.58 @ 273-	273-				500	
7940)	1.412 @ 3.3	473K)	473K)	7.3	0.17	2.202	(200g load)	1.333
* Birefringent material	naterial	-	-					

Table B5 Optomechanical properties of selected IR-transmitting glasses and other oxides.

Sources: P.R. Yoder, Jr., Opto-Mechanical Sy stems Design, 3rd ed., CRC Press, Boca Raton, 2005; W.J. Tropf, M.E. Thomas, and T.J. Harris, "Properties of Crystals and Glasses," Chapt. 33 in OSA Handbook of Optics, 2nd ed., Vol. II, McGraw-Hill, New York, 1995.

Table B6 (Table B6 Optomechanical properties of diamond and selected IR-transmitting semiconductor materials.	roperties of	diamond a	and selected II	R-transmitting	g semicor	iductor mate	erials.
			CTE					$K_{\tilde{G}} = (1 - 1)$
Material	Refractive		(a)	Young's		Density		$v_{\rm G}^2)/E_{\rm G}$
name and	index <i>n</i> @ λ in	$(dn/dT)_{\text{REL}}$	(×10 ⁻ 1 ⁰ / ₀ C)	modulus (E) (x10 ⁴ MPa)	Poisson's ratio (v.c.)	(p) (a/cm ³)	Knoop hardness	(×10 ⁻⁵) (1/ MPa)
(man fe)	Inn	(2 / 21)	010	(n 11 01)	(1) (1)	(B' WII)		(m m m)
								-
			0.8 @					
	2.382 @ 2.5		293K					
Diamond	2.381 @ 50		5.8 @		0.069			
(C)	2.381 @ 10.6		1600K	114.3	(CVD)	3.51	0006	0.094
Indium								
-								
antimonide								
(InSb)	$3.99 \otimes 8.0$	4.7	4.9	4.3		5.78	225	
Gallium arsenide	-							
(GaAs)	3.1 @ 10.6	1.5	5.7	8.29	0.31	5.32	721	1.090
			2300					
			100K					
	1055 @ 77							
			0.0 (k)					
	4.026 @ 3.8	474	200K					
	4.015 @ 5.3	<u>a</u> 250-	6.0					
Germanium(Ge)	4.00 @ 10.6	350K	300K	10.37	0.278	5.323	800	0.890
	3.436 @ 2.7							
	3 477 @ 3 8							
	3.422 (a) 5.3							
Silicon (Si)	3.148 @ 10.6	130	2.7-3.1	13.1	0.279	2.329	1150	0.704
Sources: P.R. Yoder	Sources: P.R. Yoder, Jr., Opto-Mechanical Systems Design, 3rd ed., CRC Press, Boca Raton, 2005; P.M. Amirtharaj and D.G. Seiler, "Optical Properties	Systems Design,	3 rd ed., CRC	Press, Boca Raton	, 2005; P.M. Amii	rtharaj and D	.G. Seiler, "Opt	ical Properties
2 Compared and the second second second second second and the second sec	11 100 . 10 10		1.75 1. 1. 0		T1 005			

of Semiconductors," Chapt. 36 in OSA Handbook of Optics, 2nd ed., Vol. II, McGraw-Hill, New York, 1995.

		lechanical pro	perues or self	sup n-u aus	SITILUTING CITAL	codernae ma	llerials.	
Material	Refractive			Young's modulus				$K_G = (1 - v_G^2)/E_G$
name and (symbol)	index <i>n</i> @ λ in μm	$(dn/dT)_{\rm REL}$ (×10 ⁻⁶ /°C)	CTE (α) (×10 ⁻¹⁰ /°C)	(E) (×10 ⁴ MPa)	Poisson's ratio (v _G)	Density(p) (g/cm ³)	Knoop hardness	(×10 ⁻⁵) (1/MPa)
Arsenic trisulfide (AsS ₃)	$\begin{array}{c} 2.521 @ 0.8 \\ 2.412 @ 3.8 \\ 2.407 @ 5.0 \end{array}$	85 @ 0.6 17 @ 1.0	26.1	1.58	0.295	3.43	180	5.778
Ge ₃₃ As ₁₂ Se ₅₅ (AMTIR-1)	2.605 @ 1.0 2.503 @ 10.0	$\begin{array}{c} 101 @ 1.0 \\ 72 @ 10.0 \end{array}$	12.0	2.2	0.266	4.4	170	4.224
Zinc sulfide (ZnS)	2.36 @ 0.6 2.257 @ 3.0 2.246 @ 5.0 2.192 @ 10.6	63.5 @ 0.63 49.8 @ 1.15 46.3 @ 10.6	4.6	7.45	0.29	4.08	230	1.229
Zinc selenide (ZnSe)	$\begin{array}{c} 2.61 @ 0.6\\ 2.438 @ 3.0\\ 2.429 @ 5.0\\ 2.403 @ 10.6\end{array}$	91.1 @ 0.63 59.7 @ 1.15 52.0 @ 10.6	5.6 @ 163K 7.1 @ 273K 8.3 @ 473K	7.03	0.28	5.27	105	1.311
Sources: P.R. Yo "Properties of Cr.	12.	inical Systems Design, 3 rd ed., CRC Press, Boca Raton, 2005; W.J. Tropf, M.E. Thomas Chapt. 33 in OSA Handbook of Optics, 2nd ed., Vol. II, McGraw-Hill, New York, 1995.	gn, 3 rd ed., CRC Pr andbook of Optics,	ess, Boca Raton, 2 2nd ed., Vol. II,	2005; W.J. Trop McGraw-Hill, N	f, M.E. Thomas, ew York, 1995.	and T.J. Harris,	

Table B7 Optomechanical properties of selected IR-transmitting chalcogenide materials.

			Young's				Young's		
	-		modulus						
		CTE	(E)						
		$(\alpha) \times 10^{-6}$	×10 ⁴		Density	Specific	Thermal		Best
Material		/°C	MPa		(d)	heat (C_P)	conductivity	Knoop	surface
name and		(×10 ⁻⁶	(×10 ⁶	Poisson's	g/cm ³	J/kg·K	(<i>k</i>)	hardness	smoothness
(symbol)	Manufacturer	/°F)	lb/in. ²)	ratio (v)	(lb/in. ³)	(Btu/lb·°F)	(Btu/hr·ft·°F)	(kg/mm²)	(Å rms)
		3.2	6.17		2.23	835	1.02		
Duran 50	Schott	(1.8)	(8.9)	0.20	(0.081)	(0.20)	(0.59)		
		3.3	6.30		2.23	1050	1.13		
Pyrex 7740	Corning	(1.86)	(6.1)	0.2	(0.081)	(0.25)	(0.65)		~5
Borosilicate		2.8	5.86		2.18				
crown E6	Ohara	(1.5)	(8.5)	0.195	(0.079)				~5
	Corning or	0.58	7.3		2.205	741	1.37		
Fused silica	Heraeus	(0.32)	(10.6)	0.17	(0.080)	(0.177)	(0.8)	500	~5
		0.015	6.76		2.205	766	1.31		
ULE 7971	Corning	(0.008)	(9.8)	0.17	(0.080)	(183)	(0.76)	460	~5
		0 ± 0.05	9.06		2.53	821	1.64		
Zerodur	Schott	(0 ± 0.03)	(13.6)	0.24	(0.091)	(0.196)	(0.95)	630	~5
		0 ± 0.05	8.9		2.57	810	1.6		
Zerodur M	Schott	(0 ± 0.03)	(12.9)	0.25	(0.093)	(0.194)	(0.92)	540	~5
Sources: Manufac Academic Press, N Bellingham, 2001.	turer's data shee Jew York; R.A.]	sts; W.P. Barnes Paquin, "Materia	s, Jr., "Optical als Properties	Materials - Re and Fabricatio	eflective," Cf n for Stable (ıapt. 4 in <i>Applie</i> Optical Systems	ets; W.P. Barnes, Jr., "Optical Materials - Reflective," Chapt. 4 in <i>Applied Optics and Optical Engineering</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Paquin, "Materials Properties and Fabrication for Stable Optical Systems," <i>SPIE Short Course Notes SC219</i> , Page 2018,	al Engineering rse Notes SC21	, VII, 9,

Table B8a Mechanical properties of selected nonmetallic mirror substrate materials.

l able B		di properies	הו מפופרופר					
Material name and		Voiino's						Best
(symmet)	CTE (a)	modulus (E)		Density (p)	Specific heat (C_p)	Thermal		smooth-
	×10 ⁻⁶ /°C	$\times 10^4 M Pa$	Poisson's	g/cm ³	J/kg·K	conductivity (k)	Hardness	ness
	$(\times 10^{-6} / {}^{0}F)$	$(\times 10^6 \text{ lb/in.}^2)$	ratio (v)	(lb/in. ³)	(Btu/lb·°F)	(Btu/hr·ft·°F)	(kg/mm ²)	(Å rms)
Aluminum 6061-T6	23.6	6.82 (9.9)	0.332	2.68	960 (0.23)	167 (96)	30-95 Brinell	~200
Borryllium I 70.A	11.3	28.9		0.08	1820	194		60-80*
Bervilium O-30H	11.46 (6.37)	30.3 (44)	0.08	(0.067) (0.067)	1820 (0.436)	215-365* (125-211)	80 Rockwell B	15-25
Copper OFHC**	16.7 (9.3)	11.7 (17)	0.35	8.94 (0.323)	385 (0.092)	392 (226)	40 Rockwell F	40
Molybdenum TZM	5.0 (2.8)	31.8 (46)	0.32	10.2 (0.368)	272 (0.065)	146 (84.5)	200 Vickers	10
Silicon carbide RB-30% Si	2.64 (1.47)	31.0 (45)		2.92 (0.106)	660 (0.16)			
Silicon carbide RB- 12% Si	2.68 (1.49)	37.3 (54.1)		3.11 (0.112)	680 (0.16)	147 (85)		
Silicon carbide CVD	2.4 (1.3)	46.6 (67.6)	0.21	3.21 (0.116)	700 (0.17)	146 (84)	2540 Knoop (500g load)	
SXA metal matrix of 30% SiC in 2124 Al***	12.4 (6.9)	11.7 (17)		2.90 (0.105)	770 (0.18)	130 (75)		
Graphite epoxy GY-70/x30	0.02 (0.01)	9.3 (13.5)		1.78 (0.064)		35 (20)		
* Sputtered, ** Oxygen free high conductivity, *** With SiC particles of mean size 3.5 µm (0.0014 in.) per Composite Materials Corp., San Diego. Sources: Manufacturer's data; W.P. Barnes, Jr., "Optical materials -Reflective," Chapt. 4 in <i>Applied Optics and Optical Engineering</i> , VII, Academic	n free high cond 's data; W.P. Ba	luctivity, *** With arnes, Jr., "Optica	h SiC particles I materials -Re	s of mean size 3 eflective," Chap	th SiC particles of mean size 3.5 μ m (0.0014 in.) per Composite Materials Corp., San Diego. al materials -Reflective," Chapt. 4 in <i>Applied Optics and Optical Engineering</i> , VII, Academi	Composite Material and Optical Enginee	ls Corp., San Die ering, VII, Acad	emic

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Press, New York; R.A. Paquin, "Materials Properties and Fabrication for Stable Optical Systems," SPIE Short Course Notes SC219, Bellingham, 2001.

_		Table B9 Comparison of material figures of merit for mirror design	erial figures of n	lerit for mirror de	sign.	
	Weight and self-weig	and self-weight deflection proportionality factors	rtionality factors		I hermal distortion factors	ion factors
	$(E/\rho)^{1/2}$ Resonant	p/E Mass or	$\rho^{3/E}$	$(\rho^{3}/E)^{1/2}$		
	frequency for same	deflection for	Deflection for	Mass for same	a/conductivity	a/diffusivity
Material	geometry	same geometry	same mass	deflection	Steady state	Transient
Preferred value	large	small	small	small	small	small
Pyrex	5.3	3.53	1.76	0.420	2.92	5.08
Ohara E6	5.2	3.72	1.71	0.420		
Fused silica	5.7	3.04	1.46	0.382	0.36	0.59
ULE	5.5	3.30	1.59	0.401	0.02	0.04
Zerodur	6.0	2.78	1.78	0.422	0.03	0.07
Zerodur M	5.9	2.89	1.91	0.437	0.03	
Al 6061	5.0	3.97	2.90	0.538	0.13	0.33
Al Metal matrix*	6.3	2.49	2.11	0.459	0.10	0.22
Be I-70H/I-220H	12.5	0.64	0.22	0.149	0.05	0.20
Cu, OFHC	3.6	7.64	61.1	2.471	0.53	0.14
Glidcop TM	3.8	6.80	53.1	2.305	0.05	0.17
Invar 36	4.2	5.71	37.0	1.924	0.10	0.38
Super Invar	4.3	5.49	36.3	1.906	0.03	0.12
Molybdenum	5.6	3.15	32.8	1.812	0.04	0.09
Silicon	7.5	1.78	0.97	0.311	0.02	0.03
SiC: HP alpha	11.9	0.70	0.72	0.268	0.02	0.03
SiC: CVD beta	12.0	0.69	0.71	0.267	0.02	0.03
SiC: RB-30% Si	10.7	0.88	0.73	0.270	0.01	0.03
CRES 304	4.9	4.15	26.5	1.629	0.91	3.68
CRES 416	5.2	3.63	22.1	1.486	0.34	1.23
Titanium 6Al4V	5.1	3.89	7.63	0.873	1.21	3.03
* 30% SiC. Source properties and fabr	* 30% SiC. Sources: P.R. Yoder, Jr., <i>Opto-Mechanical Systems Design</i> , 3 rd ed., CRC Press, Boca Raton, 2005; R.A. Paquin, "Materials properties and fabrication for stable optical systems," <i>SPIE Short Course Notes SC219</i> , Bellingham, 2001.	-Mechanical Systems systems," SPIE Sho	s Design, 3 rd ed., CF rt Course Notes SC	t Press, Boca Rato 219 , Bellingham, 20	n, 2005; R.A. Paqu 001.	iin, "Materials

			I ADIE DIVA UTIAI ACTELISUUS OLI AIUTIITIUTII AITOYS USEU ITI ITIITTOIS.
Alloy type	Form	Hardenable?	Remarks
1100	Wrought	No	Relatively pure, low strength, can be diamond turned
2014/2024	Wrought	Yes	High strength and ductility, multiphase, must be plated
5086/5486	Wrought	No	Moderate strength when annealed, weldable, available in large plates
			Low alloy, all-purpose, reasonably high strength, weldable, can be diamond turned
6061	Wrought	Yes	and/or plated, all forms readily available
7075	Wrought	Yes	Highest strength, usually plated, strength more temperature sensitive than other alloys
B201	Cast	Yes	Sand or permanent mold cast, high strength, can be diamond turned
-			Sand or permanent mold cast, moderate strength, most common form, extensive
A356/357	Cast	Yes	processing for dimensional stability
713/Tenzalloy	Cast	Yes	Sand or permanent mold cast, moderate strength,
771/Precedent			Sand cast, moderate strength, very stable, expensive casting procedures required, easiest
71A	Cast	Yes	to machine
Source R A Paquin "Materia	n "Material	s for Precision In	als for Precision Instruments " SPIF Short Course Notes SC016 2002

Table B10a Characteristics of aluminum allovs used in mirrors.

Г

Source: R.A. Paquin, "Materials for Precision Instruments," SPIE Short Course Notes SC016, 2002.

Condition:	Description:
	As fabricated. Applies to products shaped by cold working, hot working, or
	casting processes in which no special control over thermal conditions or
F	strain hardening is employed.
	Annealed. Applies to wrought products that are annealed to obtain the lowest
	strength temper and to cast products that are annealed to improve ductility
0	and dimensional stability.
	Strain hardened (wrought products only). Applies to products that have been
	strengthened by strain hardening, with or without supplementary heat
H	treatment.
	Solution heat treated. An unstable temper applicable only to alloys that
	naturally age spontaneously at room temperature after solution heat
W	treatment.
	Heat treated to produce stable tempers other than F, O, or H. Applies to
	products that are thermally treated, with or without supplementary heat
Т	treatment. The tempers are followed by one or more digits.

Table B10b Common temper conditions for aluminum alloys.

Source: Adapted from Boyer, H.E. and Gall, T.L., Eds., *Metals Handbook-Desk Edition*, Am. Soc. for Metals, Metals Park, OH, 1985.

Duonoutry	Incharant cash a	On the stand of the	Structural grade
Property	Instrument grade	Optical grade	
Matrix alloy	6061-T6	2124-T6	2021-T6
Volume percentage SiC	40	30	20
SiC form	Particulate	Particulate	Whisker
CTE $(\times 10^{-6}/K)$	10.7	12.4	14.8
Thermal conductivity (W/m·K)	127	123	not available
Young's modulus (MPa)	145	117	127
Density (g/cm ³)	2.91	2.91	2.86

Table B10c Characteristics of aluminum matrix composites.

Source: W.R. Mohn and D. Vukobratovich, "Recent applications of metal matrix composites in precision instruments and optical systems," Opt. Eng. 27, 1988: 90.

Property	O-50	I-70-H	I-220-H	I-250	S-200-Н	О-30-Н
Maximum beryllium						
oxide content (%)	0.5	0.7	2.2	2.5	1.5	
Grain size (µm)	15	10	8	2.5	1.5	7.7
2% offset yield strength						
(MPa)	172	207	345	544	296	295-300
Microyield strength (MPa)	10	21	41	97	34	24-25
Elongation (%)	3.0	3.0	2.0	3.0	3.0	3.5-3.6

Table B10d Beryllium grades and some of their properties.

Sources: R.A. Paquin, "Metal mirrors," Chapt. 4 in Handbook of Optomechanical Engineering, CRC Press, Boca Raton, 1997; Brush Wellman, Inc., Elmore, OH; and T. Parsonage, private communication.

			s of major since	in carbiac typ	
	Structure/	Density	Fabrication		
SiC type	composition	(%)	process	Properties*	Remarks
			Powder	High <i>E</i> , ρ,	Simple
	>98% alpha		pressed in	k_{LC} , MOR;	shapes only;
Hot pressed	plus others	>98	heated dies	lower k	sizes limited
					Complex
			Hot gas		shapes
	>98%		pressure on	High E , ρ ,	possible, size
Hot isostatic	alpha/beta		encapsulated	k_{LC} , MOR;	limited by
pressed	plus others	>99	preform	lower k	facility
					Thin shell or
Chemically				High <i>E</i> , ρ, <i>k</i> ;	plate forms;
vapor			Deposition on	lower k_{LC} ,	built-up
deposited	100% beta	100	hot mandrel	MOR	shapes
					Complex
					shapes readily
					formed;
			Cast, prefired,		large sizes;
			porous		properties are
			preform fired	Lower E , ρ ,	silicon -
Reaction	50-92% alpha		with silicon	k, MOR	content
bonded	plus silicon	100	infiltration	lowest k_{LC}	dependent

Table B10e Characteristics of major silicon carbide types.

* MOR is modulus of rupture, k_{LC} is plane strain fracture. Source: R.A. Paquin, "Materials properties and fabrication for stable optical systems," SPIE Short Course Notes, SC219, 2001.

			Typical
Material	Advantages	Disadvantages	Applications
	Metal N		
SiC/Al (Discontinuous SiC	IsotropicLarge database	• Most not weldable	 Truss fittings Brackets
particles)	• 1.5 times the modulus and strength of	 Machinable, but high tool wear Lower ductility 	 Mirrors and optical benches
	aluminum alloys at the same mass	than conventional aluminum alloys	
B/Al (Continuous boron fiber)	 High strength vs. weight Low CTE	 Anisotropic Limited flight heritage Expensive 	• Truss members
	Polymer	Matrix	
Aramid/Epoxy (e.g., Kevlar or / Spectra fibers with epoxy	 Impact resistant Lower density than graphite/epoxy 	 Absorbs water Outgasses Low compressive 	 Solar array structures Solar array structures
matrix)	 High strength vs. weight Very high 	 strength Negative CTE Outgasses 	Radomes Truss members
(high strength fiber)	strength vs. weight • High modulus vs. weight • Low CTE • Flight heritage	 (matrix dependent) Absorbs water (matrix dependent) 	 Fruss members Face sheets for sandwich panels Optical benches
Graphite/Epoxy (high modulus fibers)	 Very high modulus vs. weight High strength vs. weight Low CTE High thermal conductivity 	 Low compressive strength Ruptures at low strain Absorbs water Outgasses (matrix dependent) 	 Truss members Face sheets for sandwich panels Optical benches Monocoque cylinders
Glass/Epoxy (Continuous glass fiber)	 Low electrical conductivity Well-established manufacturing processes 	 Higher density than graphite/epoxy Lower strength and modulus than graphite/epoxy 	 Printed circuit boards Radomes

Table B11 Comparison of metal matrix and polymer matrix composites.

From Sarafin, T.P., Heymans, R.J., Wendt, R.G., Jr., and Sabin, R.V., "Conceptual Design of Structures," Chapter 15 in *Spacecraft Structures and Mechanisms*, Sarafin, T.P., ed., Microcosm Inc., Torrance and Kluwer Academic Publishers, Boston, 1995: 507.

Table B12 Mechanical properties of selected metals used for mechanical parts in optical instruments. (Page 1 of 2)	nanical propert	ies of selected	I metals used	for mecha	nical par	ts in optical in	struments. (Page 1 of 2).
		Young's			Density	Thermal conductivity		$K_M =$
	CTE	modulus	Yield stress	Poisson's	(d)	, (k)		$(1-v_M^2)/E_M$
Material	$(\alpha) \times 10^{-6} / ^{\circ}C$ (x10 ⁻⁶ / ^{\circ}F)	$(E) \times 10^{4} \text{ MPa}$ (×10 ⁶ lh/in. ²)	$(S_Y) \times 10 \text{ MPa}$ $(\times 10^3 \text{ lb/in.}^2)$	ratio	g/cm ³ (lh/in ³)	W/m·K (Btu/hr-ft·ºF)	Hardness	$\times 10^{-11} \text{ m}^2/\text{N}$ ($\times 10^{-8} \text{ in }^2/\text{h}$)
	23.6	6.89	3.4-15.2	(htt.)	2.71	218-221	23-24	
Aluminum 1100	(13.1)	(10.0)	(5-22)		(0.098)	(126-128)	Brinell	
	22.9	7.31	7.6-39.3		2.77	119-190	47-130	1.22
Aluminum 2024	(12.7)	(10.6)	(11-57)	0.33	(0.100)	(69-110)	Brinell	(8.41)
	23.6	6.82	5.5-27.6		2.68	167	30-95	1.30
Aluminum 6061	(13.1)	(6.9)	(8-38)	0.332	(0.097)	(96.5)	Brinell	(8.99)
	23.4	7.17	10.3-50.3		2.79	142-176	60-150	
Aluminum 7075	(13.0)	(10.4)	(15-73)		(0.101)	(82-102)	Brinell	
	21.4	7.17	17.2-20.7		2.68	150-168	02-09	-
Aluminum 356	(11.9)	(10.4)	(25-30)		(0.097)	(87-97)	Brinell	
	11.5	27.6-30.3	20.7		1.85	220	80-90	
Beryllium S-200	(6.4)	(40-44)	(30)		(0.067)	(127)	Rockwell B	
	11.5	27.6-30.3	34.5		1.85	220	100	3.28
Beryllium I-400	(6.4)	(40-44)	(50)	0.08	(0.067)	(127)	Rockwell B	(2.37)
	11.3	28.9			1.85	194		3.28
Beryllium I-70A	(6.3)	(42)		0.08	(0.067)	(112)		(2.37)
	11.46	30.3			1.85	215	. 80	3.28
Beryllium O-30H	(6.37)	(44)		0.08	(0.067)	(125)	Rockwell B	(2.27)
Copper								
C10100	16.9	11.7	6.9-36.5		8.94	391	10-60	7.54
(OFHC)	(9.4)	(17)	(10-53)	0.343	(0.323)	(226)	Rockwell B	(5.16)
Copper	1							
C17200	17.8	12.7	107-134		8.25	107-130	27-42	7.23
(BeCu)	(9.9)	(18.5)	(155-195)	0.285	(0.295)	(62-75)	Rockwell C	(4.97)
Copper 360	20.5	9.65	12.4-35.9		8.50	116	62-80	9.30
(brass)	(11.4)	(14.0)	(18-52)	0.32	(0.307)	(67)	Rockwell B	(6.41)

Q 2 C

Table B12 N	Mechanical pro	operties of sele	ected metals u	used for m	echanical p	oarts in optical	Table B12 Mechanical properties of selected metals used for mechanical parts in optical instruments. (Page 2 of 2).	age 2 of 2).
		Young's			Doneity	Thermal		<i>K</i> =
	CTE	modulus	Yield stress	Poisson's	(p)	(k)		$\frac{1}{(1-v_M^2)/E_M}$
	$(\alpha) \times 10^{-6} / ^{\circ}C$	$(E)\times 10^4$ MPa	$(S_Y) \times 10 \text{ MPa}$	ratio	g/cm ³	W/m·K		$\times 10^{-11} \text{ m}^2/\text{N}$
Material	$(\times 10^{-6} / {}^{\circ}F)$	$(\times 10^6 \text{ lb/in.}^2)$	$(\times 10^3 \text{ lb/in.}^2)$	(N_M)	(lb/in. ³)	(Btu/hr·ft·°F)	Hardness	(×10 ⁻⁸ in. ² /lb)
	20.0	11.0	7.6-44.8		8.52	121	55-93	
Copper 260	(11.1)	(16)	(11-65)		(0.308)	(020)	Rockwell B	
	1.26	14.1	27.6-41.4		8.05	10.4	160	0.662
Invar 36	(0.7)	(2.4)	(40-60)	0.259	(0.291)	(0.0)	Brinell	(4.57)
	0.31	14.8	30.3		8.13	10.5	160	0.629
Super Invar	(0.17)	(21.5)	(44)	0.29	(0.294)	(6.1)	Brinell	(4.34)
Magnesium	25.2	4.48	14.5-25.5		1.77	97	73	1.95
AZ-31B-H241	(14)	(6.5)	(21-37)	0.35	(0.064)	(56)	Brinell	(13.5)
Magnesium	25.2	4.48	12.4-17.9		1.77	138	42-54	
MIA	(14)	(6.5)	(19-26)		(0.064)	(2.6)	Brinell	
	14.7	19.3	51.7-103		8.0	16.2	83 Rockwell B	0.48
CRES 304	(8.2)	(28)	(75-150)	0.27	(0.29)	(9.4)	42 Rockwell C	(3.31)
	6.6	20.0	82.7-106		7.8	24.9	83 Rockwell B	0.46
CRES 416	(5.5)	(29)	(120-154)	0.283	(0.28)	(14.4)	42 Rockwell C	(3.17)
Titanium	8.8	11.4			4.43	7.3	36-39	0.79
6Al4V	(4.9)	(16.5)		0.34	(0.16)	(4.2)	Rockwell C	(5.47)
	2.6 @ 330K							
	(1.4 @ 68°F	23.5			2.65	~135		
CESIC®	<0.5 @ <90K	(34.1)			(0.095)	(~77)		
Sources: R.A. P	Sources: R.A. Paquin, "Materials p	properties and fabrication for stable optical systems," SPIE	cation for stable o	ptical systems	s," SPIE Short	Course SC219, 20	Sources: R.A. Paquin, "Materials properties and fabrication for stable optical systems," SPIE Short Course SC219, 2001; Muller, C., Papenburg, U.,	nburg, U.,

Goodman, W.A., and Jacoby, M., Proceedings of SPIE 4198, 2001:249; T. Parsonage, private communication, 2004.

Refractive index (n) after curing	1.48 to 1.55 @ 25°C
CTE (α):	
(a) 27 to 100°C	$\sim 63 \times 10^{-6} / ^{\circ} C (35 \times 10^{-6} / ^{\circ} F)$
@ 100 to 200°C	$\sim 56 \times 10^{-6} / ^{\circ}C (\sim 31 \times 10^{-6} / ^{\circ}F)$
Shear modulus	\sim 386 MPa (\sim 5.6 × 10 ⁴ lb/in. ²)
Young's modulus	$\sim 1.1 \times 10^3$ MPa ($\sim 1.6 \times 10^5$ lb/in. ²)
Poisson's ratio	~0.43
Shrinkage during curing	~4%
Viscosity (before curing)	275 to 320 cP
Density	$\sim 1.22 \text{ g/cm}^3$ ($\sim 0.044 \text{ lb/in.}^3$)
Hardness (after curing)	~85 (Shore D)
Total mass loss in vacuum (outgassing)	< 3%

Table B13 Typical characteristics of a generic optical cement.

	Table B14 Typi	ical characte	Table B14 Typical characteristics of representative structural adhesives (Page 1 of 2).	entative struc	tural adhes	ives (Page 1	of 2).	
					CTE		Young's Modulus	
Matorial	Dacemended	Uncured Visconity.	Shear strength	Temperature	(a)×10 ⁻	Joint	(E)	
(Mfr. Code)*	Curing time @ °C	v iscosity (cP)	MFa (10/111.) @ °C	•C (°F)	/(×10 ⁻⁶ /°F)	I nickness mm (in.)	MFa (lb/in. ²)	ratio (v)
			One-part epoxies					
			20.7 (3000)					
			@ -55					
			31.0 (4500)					
			(a) 24					
			31.0(4500)				-	
		Thixotropic	<i>a</i> 82					
2214 Boorles Com		paste	10.3 (1500)	101 -7 63			0213	
(3M)	60 min @ 121	(aluminium filled)	(W 121 2.7 (400) @ 177	-33 to 121 (-63 to 250)	(1,1) $(2,1)$ $(0,1)$ $(0,1)$		$(\sim 7.5 \times 10^{5})$	
/+		(Two-nart anavias					
							592	
			(1007) / / / 1		((006,c8)	
DuodiiM			(a - b)		62(a) - 54		(a) ->0°C	
I:I wt. mix			14.5 (2100)	-54 to 70	to 20		158	
ratio	$\frac{3 \text{ hr } @ 71}{2 \text{ l} 0.02}$			(-65 to 158)	(72 @ 20	0.381 ± 0.025	(23,000)	
()()	(day (<i>a</i>) 22		2.5 (10/0) @ /0		to /U)	(100.0 ± 10.01)	$(a) 20^{\circ}C)$	
			13.8 (2000)					
			a^{-50}		102 (57)			
2216			17.2 (2500)		@ 0 to			
B/A Gray	30 min @ 93				40°C			
2:3 vol. mix	2 hr @ 66		2.3 (400) @ 82	-55 to 150	134 (74) @	0.102 ± 0.025	$\sim 6.9 \times 10^{4}$	
ratio (3M)	7 day @ 24	~80,000	1.3 (200) @ 121	(-67 to 302)	40 to 80 °C	(0.004 0.001)	$(\sim 1.0 \times 10^{5})$	~0.43
			20.7 (3000)					
			<u>a</u> -55		81 (45) @			
2216 B/A			13.8 (2000)		50 to °C			
Translucent	60 min @ 93		@ 24		207 (115)			
1:1 vol. mix	4 hr @ 66		1.4 (200) @ 82	-55 to 150	@ 60 to	0.102 ±0.025	$\sim 6.9 \times 10^{4}$	5
ratio (3M)	30 day (a) 24	~10,000	0./ (100)@121	(-0/ to 302)	150 °C	(0.004 ± 0.001)	(~1.0×10 ⁻)	~0.43

-	avic e i t i pica						./	
					CTE		Young's Modulus	
Material	Recommended Curing time @	Uncured Viscosity	Shear strength MPa (lb/in. ²) @	Temperature Range of use	(α)×10 ⁻ ^{6/0} C	Joint Thickness	(E) MPa	Poisson's
(Mfr. Code)*	°C Č	(cP)	°C	°C (°F)	(×10 ^{-6/°} F)	mm (in.)	(lb/in. ²)	ratio (v)
			Urethanes					
3532 B/A Brown 1:1 vol. mix ratio		000 05	13.8 (2000)@ -40 13.8 (2000)@24 21 (200) @ 82		~0.127			
(INTC)	+7 m m +7	000,00	70 m (000) 1:2		(20010)	0.076 to		
U-05FL off-white 2:1 vol. mix ratio (L)	24 hr @25 & 50% RH		5.2 (7.5) (a) 25			0.229 (0.003 to 0.009	- 8 % bess	
			UV Curing					
	UV cure @ 100 mW/m ²							
	Fix: <8 sec @							
340 Cinala	~ 0 gap Enll: 36 car @			-54 to 130		< 0.35		
component (L)	0.25 gap	~9500	11.0 (1600)	-65 to 266	80 (44)	(<0.014)		
OP-30	UV cure @200						1	
Single-component low stress (DY)	mW/cm ² 10 to 30 sec	400	5.2 (750)	<150 (<302)	111 (62) @ 125°C		17.2 (2500)	
OP-60-LS					27(15)			
>ungre component <0.1%	UV cure @<300				<u>a</u> < 50°C			
shrinkage	mW/cm ² 5 to 30 sec	80.000	31 7 (4600)	-45 to 180	66 (37) @ > 50°C		(1.0×10 ⁶)	
(17)	222.22	222,222	Cyanoacrylates					
	Fix: 1 min @ 22 Full: 24 hr @ 22							
460 (L)	@ 50% RH	45	11.7 (1700)		80 (44)	Very small		
* Mfr code: (3M) is 31	* Mfr code: (3M) is 3M; (SO) is Summers Optical; (DY) is Dymax ; (L) is Loctite.	ical; (DY) is Dy	max ; (L) is Loctite.					

Table B14 Typical characteristics of representative structural adhesives (Page 2 of 2).

	Table B15 Ty	oical charact	eristics of	representative	elastomeri	5 Typical characteristics of representative elastomeric sealants (Page 1 of 2).	ge 1 of 2).	
			Cured		Shrinkage	Effluent or		Tensile
Material	Recommended	Uncured	hardness	Temperature	(%) after	mass loss	CTE (a)	strength
(Mfr. Code)*	Curing time (a) °C	Viscosity (P)**	(Shore A)	Range of use °C (°F)	3 days @ 25°C	(%) after hrs @ °C	×10 ⁻⁰ /°C (×10 ⁻⁶ /°F)	MPa (lb/in. ²)
				One-part silicones				
		320 g/min		-60 to 177				
		0.125 in.		(-76 to 350)				
	24 hr @ 25 &	orifice a		continuous				
	50% RH (0.125	90 lb/in. ²		<204 (400)				2.2
732 (DC)	bead)	air pressure	25	intermittent		acetic acid		(325)
				<204 (400)				
RTV112				continuous				
(GE)	24 hr @ 25 for 3			<260 (500)				
	mm thickness	200	25	intermittent	1.0	acetic acid	270 (150)	2.2 (325)
			Two	Two-part silicones				
93-500						-		
10:1 wt.						0.16 @ 24 hr		
mix ratio	7 day @ 77 %			-65 to 200		@ 125 &	300	
(DC)	50% RH		40	(-85 to 392)	nil	<10 ⁻⁶ Torr	(167)	
RTV88				-54 to 260				
200:1				(-65 to 500)				
wt.mix	24 hr @ 25 &			continuous				
ratio***	50% RH			<316 (600)				
(GE)		8800	58	intermittent	0.6	methanol	210 (111)	5.7 (830)

	Table B15 T	pical charac	teristics of r	ypical characteristics of representative elastomeric sealants (Page 2 of 2).	elastomeric s	ealants (Page :	2 of 2).	
Material (Mfr. Code)*	Recommended Curing time @ °C	Uncured Viscosity (P)**	Cured hardness (Shore A)	Temperature Range of use °C (°F)	Shrinkage (%) after 3 days @ 25°C	Effluent or mass loss (%) after hrs @ °C	CTE (α) ×10 ^{-6/ο} C (×10 ^{-6/ο} F)	Tensile strength MPa (lb/in. ²)
RTV560 200:1 wt. mix ratio (GE)	24 hr @ 25 & 50% RH	300	55	-115 to 260 -175 to 500)	1.0		200 (110)	4.8 (690)
RV8111 ~33:1 wt. mix ratio (GE)	<72 hr @ 25 & 50% RH	66	45	54 to 204 (65 to 400)	1.0	methanol	250 (140)	24 (350)
			0	Other products				
EC801B/A polysulfide (3M)	EC801B/A tack free: polysulfide <72 hr @ 25 (3M) full cure: 1 wk @ 25	viscous liquid	>35 to 60 (40 Rex)	-54 to 82 (-65 to 180)				
* Mfr. code:	* Mfr. code: (DC) is Dow Corning, (GE) is General Electric, (3M) is 3M Corp. ** Poise. *** Vacuum de-aerate before use.	ing, (GE) is Ge	meral Electric,	(3M) is 3M Corp	** Poise. ***	Vacuum de-aera	te before use.	

S _F (MPa)	$S_{\rm F}$ (lb/in. ²)
142	20,500
67	97,100
300	43,500
100	14,500
100	14,500
600**	87,020**
120	17,400
100	14,500
90	13,000
60	8700
50	7250
	142 67 300 100 100 600** 120 100 90 60

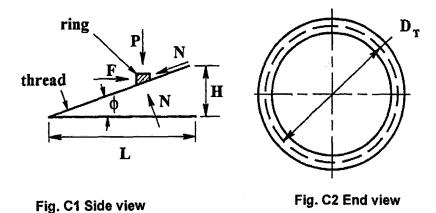
Table B16 Fracture strength S_F of infrared materials.*

* Values depend upon surface finish, fabrication method, material purity, type of test, and size of the sample tested. From: D. Harris, *Materials for Windows and Domes, Properties and Performance*, SPIE Press, Bellingham, WA, 1999.

** Updated through personal communication from D. Harris, U.S. Naval Warfare Center, 2008.

APPENDIX C Torque-Preload Relationship for A Threaded Retaining Ring

The threaded retainer acts as a body moving on an inclined plane. Figure C1 shows the geometry and the forces acting on the body. We follow the general guidelines of Chapter 10 in Boothroyd and Poli¹ to derive the appropriate equations.



For horizontal equilibrium, $F = \mu N \cos \varphi - N \sin \varphi = 0$, where μ is the coefficient of

For horizontal equilibrium, $F = \mu N \cos \varphi - N \sin \varphi = 0$, where μ is the coefficient of friction.

Solve for N:

$$N = F/(\mu \cos \varphi + \sin \varphi). \tag{C1}$$

For vertical equilibrium, $P + \mu N \sin \varphi - N \cos \varphi = 0$.

Solve for N:

$$N = P/(-\mu \sin \varphi + \cos \varphi).$$
(C2)

Equate equations (C1) and (C2) for N to get

$$F = \frac{(P)(\sin \varphi + \mu \cos \varphi)}{\cos \varphi - \mu \sin \varphi}$$

Divide by $\cos \phi$ to get

$$F = \frac{(P)(\tan \varphi + \mu)}{(1 - \mu \tan \varphi)}.$$

Since $\tan \varphi = H/L = H/(\pi D_T)$ where H = thread pitch, L = thread circumference, and $D_T =$ thread pitch diameter, we have:

$$F = \frac{\left[P\frac{H}{(\pi D_T)}\right]}{1 - \left[\frac{\mu H}{\pi D_T}\right]}.$$

The torque applied to the retainer is

$$Q = F \frac{D_T}{2} = \frac{\left(P \frac{D_T}{2}\right) \left(H + \pi \mu D_T\right)}{\pi D_T - \mu H}$$

We assume that the threads are triangular with a half-angle γ , so they wedge together and increase the terms involving friction by $1/\cos \gamma = \sec \gamma$. This factor is 1.155 for a 60-deg thread. Hence:

$$Q = \left(P\frac{D_T}{2}\right) \frac{(H + \pi\mu D_T)(1.155)}{[\pi D_T - (\mu H)(1.155)]}$$

Since $H \leq D_T$, we can safely neglect it.

So

$$Q = \left[P\frac{D_T}{2}\right] \frac{(\pi\mu D_T)(1.155)}{(\pi D_T)} = \frac{1.155PD_T\mu}{2} = 0.577PD_T\mu.$$

But there is another term to consider. This accounts for friction between the end of the retaining ring and the lens and is $Q_L = P \mu_G y_C$. We add this to the first term, approximate y_C as $D_T/2$, and get

$$Q = 0.577 P D_T \mu_M + \left(P \frac{\mu_G D_T}{2} \right) = (P D_T) (0.577 \mu_M + 0.5 \mu_G).$$

Hence,

$$P = \frac{Q}{\left[(D_T) (0.577\mu_M + 0.5\mu_M) \right]}.$$
 (C3)

Measurements of the angle of inclination for a slowly sliding dry-anodized aluminum plate on a dry-anodized aluminum inclined plane yield a value for μ_M of about 0.19. Similar measurements for BK7 glass on anodized aluminum yield a μ_G of about 0.15. Substituting these values into Eq. (C1) gives

$$P = 5.42 \frac{Q}{D_{\tau}} \tag{C4}$$

Vukobratovich,² Kowalski,³ and Yoder,⁴ state that this equation is usually written as:

$$P = 5\frac{Q}{D_{\tau}}.$$
 (C5)

The correlation between Eqs. (C4) and (C5) is within about 8%. It is doubtful if the coefficients of friction are known this accurately in any real situation.

References

- 1. Boothroyd, G., and Poli, C., *Applied Engi neering Mechani cs*, Marcel Dekker, New York, 1980.
- 2. Vukobratovich, D., "Introduction to optomechanical design," in SPIE Short Course Notes SC-014, 1993.
- 3. Kowalskie, B.J., "A user's guide to designing and mounting lenses and mirrors," Digest o f Pa pers, OSA Workshop on Optical Fabricat ion and Testi ng, N orth Falmouth, MA, Optical Society of America, Washington, 1980: 98.
- 4. Yoder, P.R., Jr., *Opto-Mechanical Systems Design* 3rd ed., CRC Press, Boca Raton, 2005.

APPENDIX D Summary of Methods for Testing Optical Components and Optical Instruments under Adverse Environmental Conditions^{*}

1. Cold, Heat, Humidity Testing

The following methods of conditioning in a test chamber are specified:

- *Met* hod 10, Cold: Condition for 16 hr to 1 of 10 degrees of severity with temperature ranging from 0 to -65°C.
 - Method 11, D ry heat: Condition for 16 hr to 1 of 4 degrees of severity with temperature ranging from 10 to 63°C and < 40% RH. Two additional 6 hr conditionings may apply with temperatures of 70 or 85°C and <40% RH.
 - Method 12, Damp heat: Condition at 40°C and 92% RH to 1 of 5 degrees of severity ranging from 16 hr to 56 days. Two additional 6 or 16 hr conditionings at 55°C and 92% RH may apply.
 - Method 13, Condensed water: Condition at 40°C and approximately 100% RH for 1 of 6 degrees of severity ranging from 6 hr to 16 days.
 - Method 14, Cycling exposure conditions, slow temperature change: Condition for 5 cycles to 1 of 9 degrees of severity ranging from 40°C high and -65°C low to 85°C high to -65°C low, with a rate of change between 0.2 and 2°C/min.
 - Method 15, Cycling exposure conditions, rapid temperature change (thermal shock): Condition for 5 cycles to 1 of 5 degrees of severity ranging from 20°C high and -10°C low to 70°C high to -65°C low within 20 sec for equipment to 10 kg and within 10 min for larger equipment. Dwell at extreme temperatures until stabilized.
 - Method 16, Cycling exposure conditions, damp heat: Condition for 5 to 20 cycles at specified rates of change to 1 of 3 degrees of severity ranging from a low of 23°C with 82 % RH and a high of 40°C with 92 % RH to a low of 23°C and a high of 70°C with unspecified RH.

2. Mechanical Stress Testing

The following methods of conditioning at ambient atmospheric conditions on a shock machine, acceleration facility, or electrodynamic shaker are specified:

^{*} Based on a preliminary version of ISO 9022.

- Method 30, Shock: Condition with 3 shocks in each direction along each axis to 1 of 8 degrees of severity ranging from 10 to 500-g acceleration with half-sine wave pulse durations of 0.5 to 18 msec.
- Method 31, Bump: Condition with 1000 to 4000 shocks in each direction along each axis to 1 of 8 degrees of severity ranging from 10 to 40-g acceleration with half-sine wave pulse durations of 6 to 16 msec.
- Method 32, Drop and topple: Condition with 1 of 3 degrees of severity involving 25 to 100-mm drops on each corner plus topple about each edge.
- Method 33, Freefall: Condition in transport container or unprotected (if so designed) with 2 to 50 falls ranging in severity from 25 to 1000 mm, depending on mass of specimen.
- Method 34, Bounce: Condition to 1 of 3 degrees of severity ranging from 15 to 180 min with double amplitude 25.5 mm and 4.75 Hz frequency on an approved bounce table.
- Method 35, St eady-state acc eleration: Condition to 1 of 3 degrees of severity ranging from 5 to 20-g acceleration for 1 to 2 min in each direction along each axis.
- Method 36, Vibration, sinusoidal sweep frequencies: Condition at ambient conditions to 1 of 10 degrees of severity involving displacements ranging from 0.035 to 1.0 mm and accelerations of 0.5 to 5 g sweeping at 1 octave/min within frequency bands ranging from (lowest) 10 to 55 Hz for equipment used on ships, near heavy machinery, or in general industrial applications to (highest) 10 to 2000 Hz for equipment used in aircraft and missiles. This test may be followed by conditioning for 1 of 3 degrees of severity ranging from 10 to 90 min vibration along each axis at characteristic frequencies indentified under sweep frequency tests or identified in the applicable specification.
- Method 37, Vibration, random: Condition to 1 of 26 degrees of severity ranging from a power spectral density of 0.001 to 0.2 g^2/Hz with random frequencies of 20 to 2000 Hz and conditioning times of 9 to 90 min.

3. Salt Mist Testing

Representative samples of components or materials to be used in optical instruments that will experience exposure to salt atmosphere are to be tested. Complete instruments are tested only in exceptional cases. The tests are not considered reliable representations of actual exposure, but serve only as indications of suitability or unsuitability. Method 40 specifies the following:

The test chamber shall have a volume of at least 400 L and be heated to 30°C during the test. Precautions are taken to prevent direct impingement of spray onto specimens or condensate from dripping onto them. The salt mist is injected pneumatically through plastic nozzles at a rate that delivers a prescribed volume of 5% aqueous solution of sodium chloride/hr. The purity of ingredients must be high and the pH of the solution

APPENDIX D

must be controlled. Conditioning is to 1 of 7 degrees of severity ranging from 2 hr to 8 day duration.

4. Cold, Low Air Pressure Testing

Method 50 specifies that the hardware shall be conditioned in a chamber to low pressure with and without exposure to condensation and freezing of moisture to simulate exposure in unheated aircraft or missiles or operation/transport in high mountainous regions. Conditioning is for 4 hr to 1 of 8 degrees of severity ranging from -25° C and 60-kPa pressure (3500 m altitude) to -65° C and 1-kPa pressure (31,000-m altitude).

5. Dust Testing

This test, *Method 52*, evaluates the resistance of the specimen to blowing dust that may impair function of moving parts or cause unacceptable wear of surfaces. Unless otherwise specified, optical surfaces are covered during exposure. The dust consists of sharp-edged particles, not less than 97% silicon dioxide. Particle size ranges from 0.045 to 0.1 mm, with the majority (90%) smaller than 0.071 mm.

Conditioning is to 1 of 3 degrees of severity involving 6 to 34 hr exposure to 8 to 10 m/sec velocity air containing 5 to 15 g/m³ sand. Temperature is held at 18 to 28°C and RH controlled at <25%.

6. Drip, Rain Testing

The following methods of conditioning in a test chamber are specified:

- Method 72, Drip testing: Means shall be provided for decalcified or desalted water to drip through a perforated plate (0.35-mm holes) onto the specimen from a distance >1 m. The specimen shall be rotated in the chamber. Condition to 1 of 9 degrees of severity ranging from 1 to 30 min exposure to 1.5 to 5.5 mm/min.
- Method 73, Steady rain: Shower heads shall be arranged within the test chamber so as to distribute simulated rainfall to the rotating specimen at 5 or 20 mm/min rate for 30 min.
- Method 74, Driving rain: Wind driven water shall be directed onto the specimen at velocities of 18 or 33 m/sec for 1 of 6 degrees of severity corresponding to exposure times of 10 to 30 min. Rainfall shall be at 2 or 10 mm/min.

7. High-pressure, Low-pressure, Immersion testing

The following methods of conditioning are specified:

Method 80, Internal high pressure: Conditioning for 10 min to 1 of 13 degrees of severity involving either 100 or 400 Pa pressure difference with

associated allowable drops in the internal pressure ranging from 75% (least severe) to 2% (most severe).

- Method 81, Internal low pressure: Identical to the above test except with higher pressure outside the specimen.
- Method 82, Immersion: Conditioning by submersion of the specimen 1 to 400 m under water for 2 hr.

8. Solar Radiation

Under *Method 20*, a specimen is tested in a heated test chamber having a source capable of irradiating the specimen to a specified level (in W/m^2) within each of six spectral bands representative of solar energy. Removal of ozone, if generated, is required. Two degrees of conditioning severity expose the specimen to about 1 kWh/m² for 1 to 5 24 hr cycles with chamber temperature varying between 25 and 55°C and <25 % RH. Two additional degrees of conditioning apply to representative samples tested for longer periods of time (to 240 hr) to evaluate photochemical influences and achieve artificial aging.

9. Combined Sinusoidal Vibration, Dry Heat, or Cold Testing

The following methods of conditioning are specified:

- Method 61, Combined sinusoidal vibration, dry heat: Condition to 1 of 13 degrees of severity involving three elevated test chamber temperatures (40 to 63°C) with RH < 40% and displacements ranging from 0.035 to 1.0 mm and accelerations of 0.5 to 5 g, sweeping at 1 octave/min within frequency bands ranging from (lowest) 10 to 55 Hz to (highest) 10 to 2000 Hz. This test may be followed by conditioning for 1 of 3 degrees of severity ranging from 10 to 30-min vibration along each axis at characteristic frequencies identified under sweep frequency tests or identified in the applicable specification.
- Method 62, Combined sinusoidal vibration, cold: Condition to 1 of 17 degrees of severity, involving six reduced test chamber temperatures (-10 to -65° C) with RH < 40% and displacements, accelerations, and frequencies per Method 61. This test may be followed by conditioning for either 10 or 30-min vibration time along each axis at characteristic frequencies identified under sweep frequency tests or identified in the applicable specification. Guidance for choice of test severity is given in terms of application of the instrument to astronomical, industrial, ground vehicle, naval vessel, or aircraft/missile/special applications.

10. Mold Growth Testing

Method 85 specifies that representative samples, such as mounted optics, materials samples, or surface coatings, be conditioned for 28 or 84 days in a closed test chamber, with a temperature of approximately 29°C and high humidity. Complete instruments are tested only if required by the specification. Tests require innoculation of the test samples with mixed viable spores of 10 specified types of fungi. Control strips of sterilized filter paper are inoculated and placed in the test chamber along with the test samples. Mold growth on the control strips must be visible 7 days into the test period for the test to be considered valid. At the conclusion of the test, all samples are examined for mold growth and physical damage (such as coating damage, surface etching, or corrosion). If the specification requires evaluation of possible effects on optical performance, control samples are exposed for the same time period at the same temperature and humidity conditions, but without mold spores. These are compared with the innoculated samples at the conclusion of the test.

Note that the sequence of environmental testing can have an effect on results. Fungus testing should not follow salt mist or sand/dust exposure because salt tends to suppress mold growth and sand/dust may provide nutrients for mold growth.

11. Corrosion Testing

Condition representative samples such as mounted optics, material samples, or surface coatings for specified periods in contact with felt pads saturated with specified substances at ambient atmospheric conditions. Complete instruments are tested only if required by the specification. Post-test evaluation classifies specimens to five levels of damage ranging from no visible degradation to heavy degradation and structural damage. Basic test methods follow.

- Method 86, Ba sic cosmetic sub stances and artificial ha nd swea t: Condition in contact with paraffin oil, glycerine, Vaseline, lanolin, cold cream, and artificial hand sweat for 1 to 30 days and inspect.
- Method 87, Laboratory agents: Condition in contact with various agents including sulfuric, nitric, hydrochloric, and acetic acids and potassium hydroxide in various dilutions with water for 10 to 120 min as well as agents like ethanol, acetone, and xylene for 5 to 60 min and inspect.
- Method 8 8, P roduction pl ant resources: Condition in contact with hydraulic oil, synthetic oil, cooling lubricant, and general-purpose detergent for 2 to 16 hr and inspect.
- Method 89, F uels and res ources f or aircraft, naval ve ssels, a nd l and ve hicles: Condition in contact with specified materials including gasoline, fuel oil, lubricating oil, hydraulic oil, brake fluid, deicing fluid, antifreeze agent, fire extinguishing agent, detergent, alkaline, and acid battery electrolyte, etc. and inspect.

12. Combined Shock, Bump, or Free Fall; Dry Heat, or Cold Testing

The following methods of conditioning at elevated or reduced temperatures on a shock machine, acceleration facility, or electrodynamic shaker are specified:

- Method 64, Shock, dry he at: Condition with three shocks in each direction along each axis to 1 of 15 degrees of severity ranging from 15 to 500 g acceleration with half-sine wave pulse durations of 1 to 11 msec. Four temperatures ranging from 40 to 85°C and < 40% RH apply.
- Method 65, Bump, dry heat: Condition with 1000 to 4000 shocks in each direction along each axis to 1 of 8 degrees of severity ranging from 10 to 25 g acceleration with half-sine wave pulse durations of 6 msec. Three temperatures ranging from 40 to 63°C and < 40% RH apply.
- Method 66, Shock, cold: Condition with three shocks in each direction along each axis to 1 of 25 degrees of severity ranging from 15 to 500 g acceleration with half-sine wave pulse durations of 1 to 11 msec. Six temperatures ranging from -10 to -65°C apply.
- Method 67, Bump, cold: Condition with 1000 to 4000 shocks in each direction along each axis to 1 of 14 degrees of severity ranging from 10 to 25 g acceleration with half-sine wave pulse durations of 6 msec. Six temperatures ranging from -10 to -65°C apply.
- Method 68, Freefall, dry heat: Condition in transport container or unprotected (if so designed) with 2 to 50 falls ranging in severity from 100 to 1000 mm, depending on mass of specimen. Three temperatures ranging from 40 to 85°C and <40% RH apply.
- Method 69, Freefall, cold: Condition in transport container or unprotected (if so designed) with 2 to 50 falls ranging in severity from 100 to 1000 mm, depending on mass of specimen. Five temperatures ranging from -25 to -65°C apply.

13. Dew, Hoarfrost, Ice Testing

Exposure to dew (*Method 75*), hoarfrost (*Method 76*), or ice (*Method 77*) results from rapid change in environmental conditions in a chamber or from transfer of the specimen from a cold chamber to a conditioned room. Instrument parts normally protected from frost or ice should be protected during the test. Each test is conducted in three steps:

- 1. Stabilization at temperatures ranging from 10 to -25°C in accordance with 5 degrees of severity,
- 2. exposure to 30°C and 85% RH (dew formation) or −5 to −25°C with water spray (ice formation) until temperature is stabilized or ice has reached thickness up to 75 mm, as applicable, and
- 3. exposure to 30°C and 85% RH to stabilize.

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