6 MOUNTING OPTICAL COMPONENTS

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6.1 GLOSSARY

a_G	acceleration factor
D_G	diameter of optic
E	Young's modulus
ID	internal diameter
K	constant factor
m	mass
OD	outer diameter
P	total preload
S_Y	yield stress
SPDT	single point diamond turning
t	thickness
Δ	deflection of spring or flange
ν	Poisson's ratio

6.2 INTRODUCTION AND SUMMARY

This chapter summarizes the techniques most commonly used to mount lenses, windows, small mirrors, and similar optical components as well as moderate-sized mirrors, and prisms within their mechanical surrounds to form optical instruments. Because of space limitations, mountings suitable for large (i.e., >85-cm diameter) optics are not discussed here. Two basic approaches for mounting optical components are considered: those in which the optic is held firmly against mechanical reference surfaces by applied forces (hard mounting) or those supported by benign means that do not inherently apply force (soft mounting). Descriptions of hard mountings include ones using threaded retaining rings, flanges, or springs while descriptions of soft mountings include ones using flexures, elastomeric encapsulation, or bonding to mechanical pads. With either type of mounting, the required location and orientation of the optic relative to other portions of the instrument, that is, its alignment, is established during assembly in order to maximize performance. An important aspect of mounting design considered here is how the adverse influences of shock, vibration, temperature change, and moisture on alignment and system performance can be minimized. References cited here provide equations for designing and analyzing a large variety of mountings. Although we speak here of optics as if they are always made of glass and to mechanical parts (housings, cells, spacers, retainers, etc.) as if they are always made of metal, it should be understood that many of these mounting considerations also apply to other materials such as crystals, plastics, and composites.

6.3 MOUNTING INDIVIDUAL ROTATIONALLY SYMMETRIC OPTICS

Hard Mounting Techniques

In order to constrain an optic and preserve its alignment relative to other critical components of an optical instrument, hard mountings apply compressive forces to the glass at discrete locations or along line contacts. These forces, called preloads, are established during assembly and generally are of sufficient magnitude to hold the optic against appropriately located mechanical reference surfaces in the mount under all environmental conditions, including shock, vibration, and temperature changes. The magnitude of the preload P in N applied along any axis should be at least $9.81ma_G$, where m is the mass of the optic and any related components to be held by a single constraining means and a_G is the worst case acceleration expected to be encountered by the subassembly. This term a_G is understood to be a multiple of ambient gravity. If the optic is rotationally symmetric, glass-to-metal contact can be provided at the optic's cylindrical rim, at its ground bevels, or at its polished surfaces. Five degrees of freedom (three translations and two tilts) must be controlled. The sixth degree of freedom (rotation about the optic axis) is also adjusted and controlled in some cases to improve performance in the presence of residual optical wedge or if nonsymmetrical aspheric surfaces are involved. All six degrees of freedom must be constrained for noncircular optics, such as prisms.

The forces applied at the interfaces as well as those from gravity or imposed accelerations may distort the optical surfaces (thereby affecting performance) and introduce stress into the glass. Stress is known to cause birefringence, or, in extreme cases, damage to the optic—especially at low temperatures where shrinkage of the metal exerts maximum force on the glass. To minimize these adverse effects, forces must be kept within acceptable limits. Very few closed-form equations are available for predicting refracting or reflecting surface deformations due to applied forces. Finite element analyses are most frequently used for this purpose.[1,2] Explanation of how this is done is beyond the scope of this presentation.

Relatively simple analytical means for estimating compressive and tensile stresses introduced by mounting forces are detailed in the literature.[3,4] Most of these techniques are based on adaptations of standard formulations by Roark[5] and Timoshenko and Goodier.[6] The magnitude of the stress generated by a force depends not only upon the magnitude of that force, but also on the shapes of the surfaces in contact and Young's modulus and Poisson's ratio values for the glass and metal involved.

Statistical analyses backed by experimentation indicate that an optical component made by conventional high-quality grinding and polishing methods can usually withstand tensile stress as large as ~6.9 MPa without failure. This value is generally accepted as a "rule-of-thumb" tolerance for survival of the optic under stress.[7] Optics made by "controlled grinding" techniques,[8] polished and assembled with great care, and not scratched or otherwise damaged during use, might well survive long-term stress about 1.7 times greater.[9]

Under the more benign conditions of the operating environment (wherein the instrument must perform to specifications), survival is not a concern, but distortions of optical surfaces

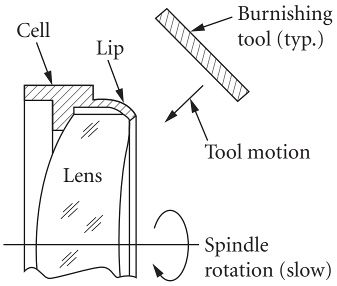

FIGURE 1 A small lens burnished into a cell made of malleable metal.

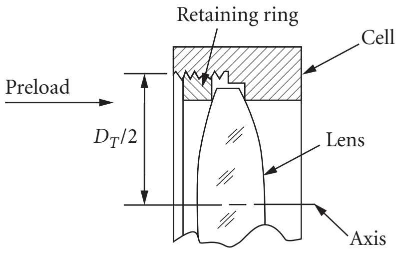

FIGURE 2 A lens preloaded in its cell with a threaded retaining ring.

due to mounting forces may degrade performance. High-performance optical systems and those using polarized light may also be especially sensitive to stress-induced birefringence. A commonly applied tolerance for stress in the glass in such cases is ~3.4 MPa. Analytical methods outlined in this chapter allow mounting stresses to be estimated to predict the potential success of a given design.

Burnished Mountings Figure 1 shows a very simple way to mount a lens element in a tubular cell. This cell has an internal shoulder against which the lens is to be held. The cell is mounted on a spindle, the lens is inserted and held in place gently, and the assembly is slowly rotated. The cell has a lip that extends beyond the rim of the lens. That lip is burnished with one or more hardened rod-shaped tool(s) over the edge of the lens as indicated in the right-hand view. The cell material must be malleable so it can be bent easily. Brass or annealed aluminum are common choices. The magnitude of the force, if any, introduced by the mounting cannot be quantified in this case because the bent metal tends to spring slightly away from the glass once tool pressure is removed. This type mounting is most suitable for use with small elements used in some endoscopes, simple microscope objectives, or low-cost cameras.

Mounts Using Threaded Retaining Rings This mounting, shown schematically in Fig. 2, is the type most frequently used to secure a lens element in its mount. Torque Q in N-mm applied to the ring with a wrench creates axial preload P to hold the lens against the shoulder very approximately as $5Q/D_T$, where D_T is the pitch diameter of the thread in millimeters.

The fit of the mating threads in the cell and retainer should be loose enough for the retainer to align itself to the centered lens surface; otherwise, lens alignment may be altered when the retainer is tightened. Such a fit may be specified as Class-1 or -2 per ANSI/ASME B1.1-2003.* During assembly, the lens should first be aligned in the cell and then held in place as the retainer is tightened to the required torque.

Mounts Using Annular Flanges Figure 3 shows a lens element preloaded against a shoulder by an annular flange that is deflected axially by a distance Δ from its nominal flat shape. Adapting an equation from Roark,[5] the deflection Δ required to produce a given preload P in N equals $(K_A - K_B)P/t^3$ where t is the flange thickness in millimeters and the constants K_A and K_B are determined by the material properties and the dimensions a and b indicated in the figure.

For a given design, the required deflection may be obtained by customizing the thickness of the spacer located under the flange and should be at least 10 times larger than the resolution capability of the device to be used to measure the flange deflection at the time of assembly. As the flange is bent, stress is developed within that component. To prevent damage to the flange, its thickness should equal $K_C P f_S / S_Y$ where the constant K_C depends upon the dimensions a and b and the flange material

*Unified Inch Screw Threads (UN and UNR Thread Form).

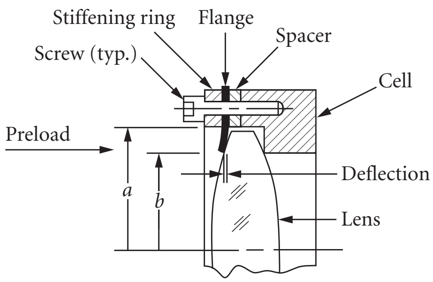

FIGURE 3 A lens preloaded in its cell with a deflected continuous ring flange.

properties. The quantity S_Y is the yield stress of the flange material and f_S is the desired safety factor. The stiffening ring shown next to the flange maintains uniform flange deflection between the attaching screws.

A significant advantage of the annular flange as compared to the threaded retaining ring is that the flange can be calibrated before installation by measuring the actual preload developed as a function of deflection. Then, one can be quite confident that the preload on the lens is as stated by the above relationship when that flange is deflected by the specific distance Δ. This level of confidence cannot be achieved with a threaded ring.

Soft Mounting Techniques

Elastomeric Mountings A convenient technique for mounting a lens in a cell is to inject a continuous annular ring of an elastomeric material such as room temperature vulcanizing (RTV) sealing compound between the lens rim and the inside surface of the cell (see Fig. 4). This is sometimes called an "elastomeric ring mounting." The thickness t_e equals $K_D D_G$, where D_G is the lens diameter and the constant K_D is determined from the material properties using a relationship attributed by Herbert[10] to R. Vanbezooijen. The lens is then virtually free of radial stress at all temperatures. This is because the elastomer expands or contracts with temperature changes just enough to always fill the radial gap between the glass and the metal.

Some designs using the elastomeric ring approach also benefit from the fact that a continuous ring of this material effectively seals the lens to its mount so, if this subassembly forms part of the exterior skin of an optical instrument, leakage of gases and moisture through that interface is prevented.

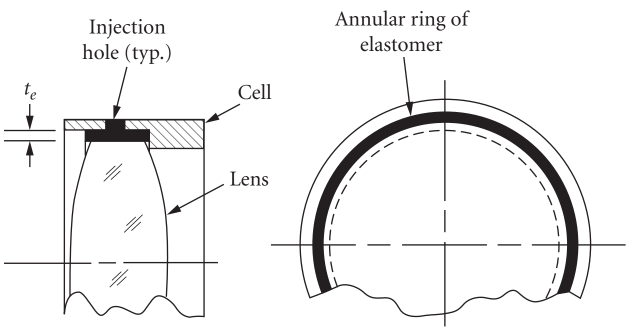

FIGURE 4 A lens supported in its cell by a continuous annular ring of elastomer.

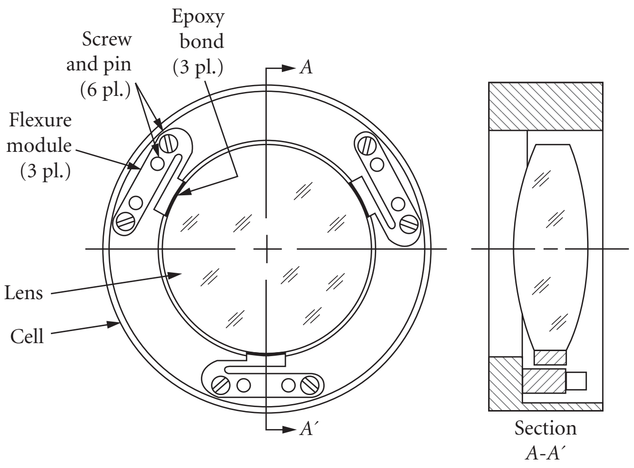

FIGURE 5 A lens bonded to three flexure modules attached to a cell.

Another type of elastomeric mounting for lenses uses discrete pads of elastomer located between the lens rim and the cell's inside surface. At least three such pads are needed to fully constrain the optic. They should be symmetrically distributed around the lens. The radial thicknesses of these pads can be sized as described above for the continuous ring.

Flexure Mountings High-performance optical systems require the optical axes of their lenses to be precisely centered mechanically with respect to some mechanical reference and to remain in that condition when the temperature changes. Because metal and glass components expand or contract with temperature at different rates, the optics may become decentered, tilted, or stressed when temperature changes occur in the above-described hard mountings. A properly designed support configuration using three or more symmetrically located identical flexures between the mount and lens rim will ensure that the lens stays as originally aligned and free of stress in spite of such changes.

Figure 5 shows a concept for a simple flexure mount design suggested by Ahmad and Huse.[11] Three identical flexure modules are made with narrow slots cut into them (by an electric discharge machining method) to form cantilevered flexure blades. Each blade has, at its free end, a curved pad shaped to interface with the lens rim. These modules are attached to the lens cell with screws passing through slightly oversized holes. In an alignment fixture, the optical axis of the lens is centered with respect to the axis of the cell. The modules are then adjusted to provide specific gaps between the pads and the lens rim and pinned in place. Epoxy is injected into those gaps and cured. Because the flexures are separate from the cell, they can be made from a material (such as titanium) with a higher yield stress than the cell. The cell is typically made of less expensive yet dimensionally stable material (such as stainless steel). More complex flexure designs and ones featuring a larger number of radial flexures have also been described.[12–16] Because of their inherent flexibility, flexure mountings should be analyzed to determine their responses to externally imposed shock and vibration.

6.4 MULTICOMPONENT LENS ASSEMBLIES

Groups of lenses used in optomechanical assemblies typically are individually mounted and constrained in seats machined into a housing or are separated axially by spacer rings within a common cylindrical bore in the housing. It is important for those lenses to have a common optical axis and the correct axial airspaces within allowable tolerances. We here consider several ways in which such assemblies can be designed.

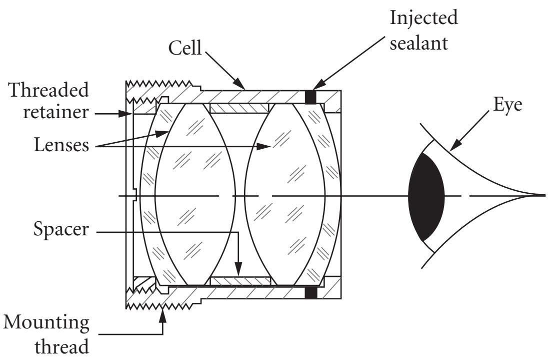

FIGURE 6 A telescope eyepiece with two lenses and a spacer assembled by the "drop-in" method.

Drop-In Assembly

If alignment requirements are not too demanding, lenses, housings, and spacers can be machined to reasonable tolerances and simply assembled without further machining or adjustment other than tightening a retainer to provide prescribed preload. Figure 6 shows an eyepiece for a low-power telescope assembled in this way. Radial clearances are typically ~0.075 mm, so individual lenses can easily be inserted into their seats. The eyepiece is configured to thread into a cylindrical opening in the telescope housing and to be focused by rotating the entire eyepiece. In some cases, the lenses are sealed to the cell and the threaded joint with the telescope also is sealed.

Many all-plastic lens assemblies used in consumer products are designed for swift drop-in assembly. The example shown in Fig. 7 is the objective for a rear-projection television system. Flexible tabs molded into the inside walls of both halves of the plastic housing project inward to form pockets for insertion of the three injection-molded plastic lens elements. Grooves (not shown) molded into the inside walls of the housings reduce stray light that otherwise could reduce contrast of the image. The housings are fastened together by self-tapping screws passing through flanges along each side, as indicated in the end view. Optical alignment relies on accuracy of the molding processes and is adequate for the application.

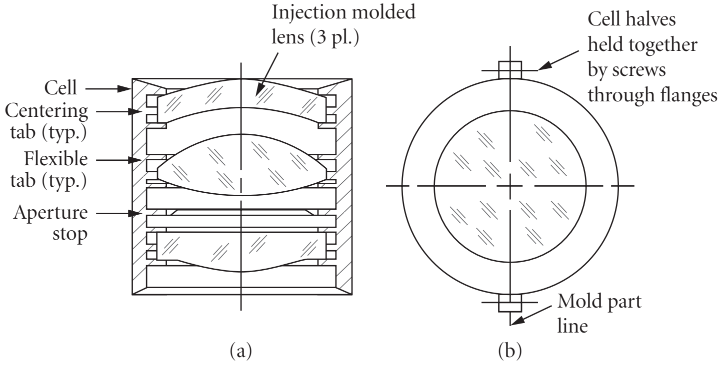

FIGURE 7 An all-plastic projection lens assembled by the "drop-in" method.

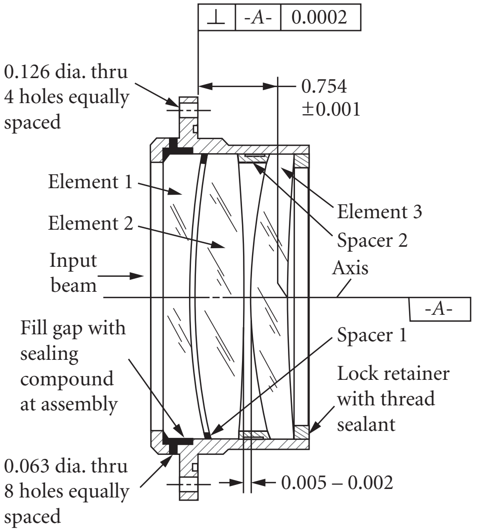

FIGURE 8 A telescope objective assembly with alignment resulting from tightly toleranced dimensions. Dimensions are in inches. (*From Yoder.*[17])

Tightly Toleranced Assemblies

When higher performance is required, the dimensional tolerances are tightened and better optical alignment is achieved. Figure 8 shows the objective lens assembly for a military telescope.[17] The three lenses are edged to fit the cell inside diameter with nominal radial clearances of 0.012 mm. All metal parts are made of stainless steel. The first spacer is made of sheet metal stock 0.025 ± 0.005 mm thick. It conforms to the spherical shapes of the adjacent lens surfaces under preload. The axial thicknesses of the lenses are toleranced to ± 0.005 mm. Residual optical wedge tolerances for the lenses are 12 arcsec. The beam deviation from these wedges is minimized at assembly by rotating (i.e., clocking) two lenses about their axes relative to the third lens to obtain maximum symmetry of the image of an on-axis artificial star. This image is observed with a microscope during alignment.

Lathe Assembly

A technique that is frequently used to obtain lens centration by minimizing radial clearances between lens ODs and cell IDs is called “lathe assembly” because it is done on a machinist’s lathe. The diameters and thicknesses of a selected set of lenses are measured and recorded. The required central air spaces and their tolerances are obtained from the optical system design. Actual lens surface radii are obtained from interferometric measurements made during lens manufacture. This data accompanies the lenses to the machine shop where a partially machined cell or housing is customized to provide conical or toroidal interfaces with the polished surfaces of that particular set of lenses and to provide all other required dimensions for the optomechanical assembly within the required tolerances. Radial clearances of ~0.005 mm can be achieved by this method. This clearance is adequate for careful assembly of the lenses into the cell.

Figure 9[17] shows an air-spaced doublet lens subassembly created by this process. The individual lens seats are finish-machined at the time of assembly to fit those lenses. The length of the spacer (dimension *E*) and the location of the mounting flange (Datum B) relative to the front lens vertex

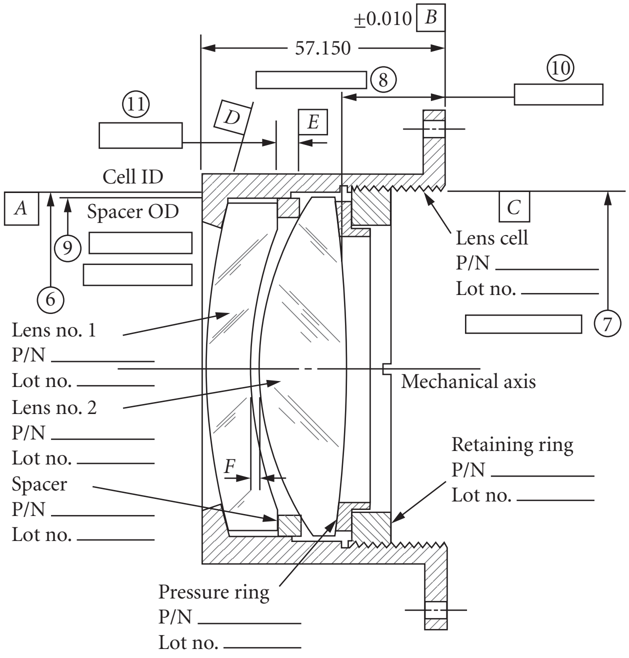

FIGURE 9 An air-spaced doublet assembled by the "lathe assembly" process. (*From Yoder.*[17])

are also machined to produce the proper air space and overall length. Both lenses are secured by the retainer. The actual values of the numbered dimensions are recorded in the boxes. If required, a complete pedigree of that particular assembly can be established for future reference from the measured data and inspection reports. This type of construction is especially suited for applications involving high accelerations. Bayar described an aerial camera lens assembled by this method.[18]

"Poker Chip" Assembly

Figure 10 is a partial section view through a lens assembly that features seven lenses: four doublets and three singlets. Each lens, except the largest, was centered interferometrically to the mechanical axis of its cell OD and held in place in that cell with annular rings of epoxy nominally 0.381 mm thick. After the adhesive was cured, the axial thicknesses of the cells were final machined so all axial air spaces would be within design tolerances. The cell subassemblies Numbers 6 through 2, which had been machined to the same ODs within tight tolerances, were then inserted into the stainless steel housing and secured by Cell No. 1 that was threaded into the housing to act as a retainer. The largest lens (No. 12) was held directly in the housing by its own retainer. Accuracy of internal alignment was built into the assembly by the fabrication process.[19] This type of construction is frequently referred to as "poker chip" assembly because the individual lens/cell subassemblies are stacked on top of each other inside the housing.

Lenses Adjusted at Assembly

Many complex lens assemblies to be used in very high-performance applications such as microlithographic projection systems need positional adjustment of a few carefully selected elements at the final stage of assembly. This is because application of the best possible optical and mechanical

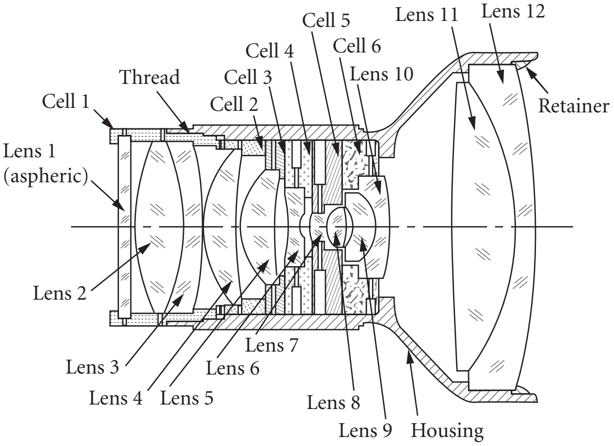

FIGURE 10 A projection lens comprising a stack of "poker chip" lens/cell subassemblies inserted into the bore of the mount. (*From Fischer.*[19])

manufacturing processes and extremely tight dimensional tolerances cannot make the lenses and mechanical parts accurately enough to obtain the full level of performance required by the application. An example is shown schematically in Fig. 11.[20]

This optomechanical system comprises twelve air-spaced "poker chip" subassemblies, stacked on top of each other with custom-made spacers placed between lapped coplanar pads on the cell faces

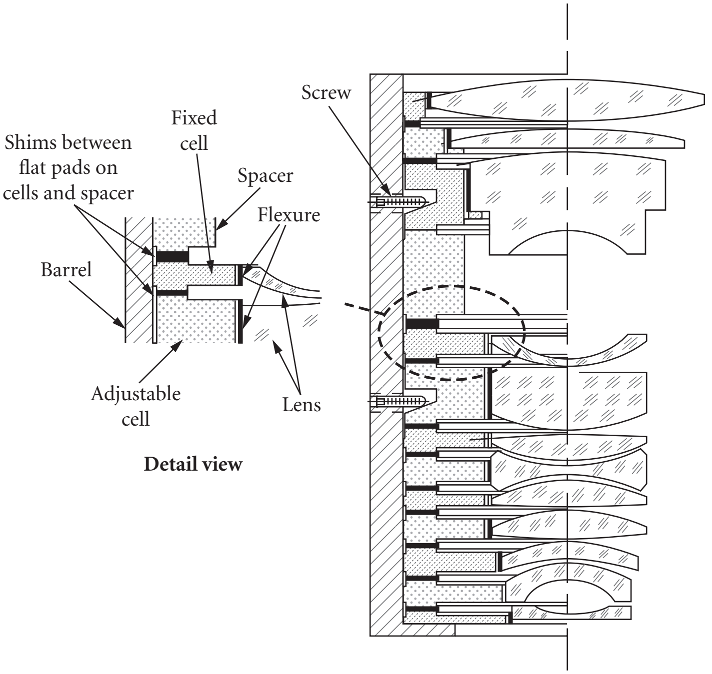

FIGURE 11 Partial section view of a "poker chip" lens assembly with two lenses adjusted after assembly to optimize performance. (*From Yoder.*[4])

to control axial air spaces. The lenses are mounted on flexures machined directly into the interior surfaces the cells. Ten of the subassemblies fit closely inside the stainless steel barrel. Two subassemblies are adjustable laterally in orthogonal directions from the outside. Optical performance of the system is measured interferometrically in near real time, while the adjustable lenses are moved very slightly until optimal performance is achieved. The adjustment mechanisms are then locked and the lens is installed into the microlithography system.

Determination of which lens elements to move in a given optical system to correct residual aberrations is a job for the lens designer working with mechanical engineers and metrology experts who help decide how to incorporate the needed mechanisms and to conduct the necessary tests. The sensitivities of spherical, coma, astigmatism, and distortion aberration contributions from each lens to lateral and axial displacements are determined by raytracing. The ideal candidates for correcting each aberration are lens shifts that modify that aberration significantly, but do not excessively affect the other aberrations. The results are reviewed to determine which lens movements are best to minimize each aberration. Williamson[20] outlined a procedure in which the aberration contributions of the optical system shown in Fig. 12*a* for specific axial and lateral displacements of each element are plotted as shown in Fig. 12*b* and *c*, respectively. Using phase-measuring ultraviolet interferometry

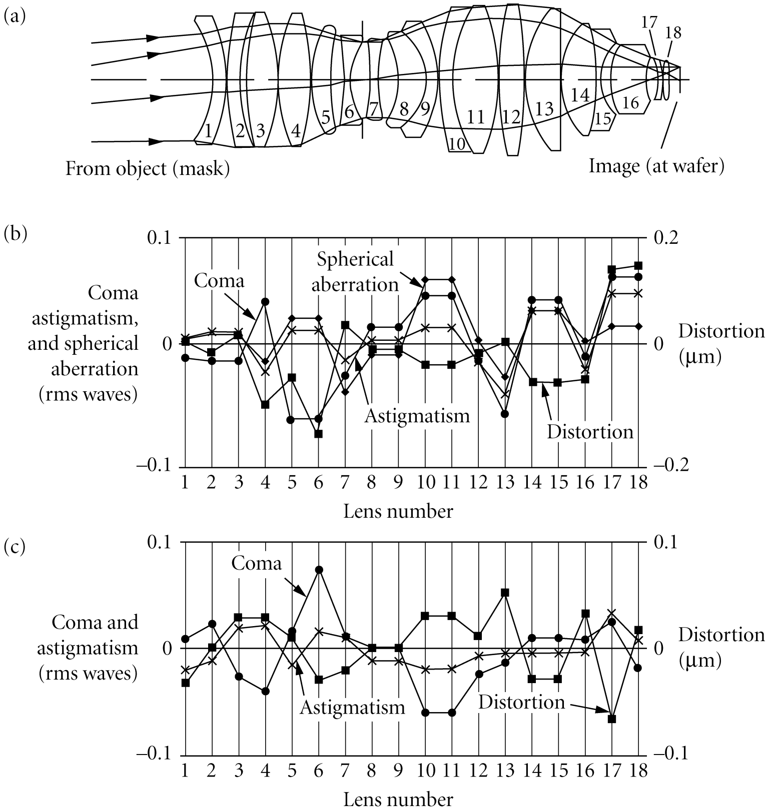

FIGURE 12 (a) Optical schematic of an 18-element lithographic projection lens. (b) The effects on aberrations of individually displacing each element axially by 25 µm. (c) Similar effects of displacing each element laterally by 5 µm. The best lenses to adjust for optimum system performance can be determined. (*From Williamson.*[20])

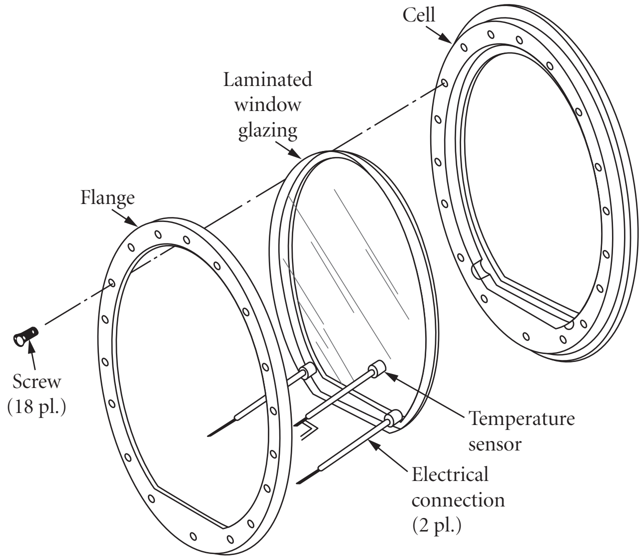

FIGURE 13 Exploded view of a heated window assembly used in a military application. (*Courtesy of Goodrich Corporation, Danbury, CT.*)

as the image quality monitor, all these aberrations can be significantly reduced by iteration of lens movements and the final system performance of production lens assemblies greatly enhanced.

6.5 MOUNTING WINDOWS AND DOMES

Small circular windows usually are secured in a mount with a threaded retainer or by an elastomeric ring. Noncircular ones are best held in place with elastomer. The continuous flange-mounting method can be used to advantage with larger windows. The one shown in Fig. 13 has an elliptical aperture of 20.32×30.5 cm.[4] The electrical connections shown provide current to a conductive coating on a buried surface, which keeps the window free of fog in high-humidity situations. The flange preloads the window into an aluminum cell. An elastomeric sealant is injected into a groove around the window's rim to seal it to the cell. The cell is sealed to the instrument housing with a gasket or an O-ring.

Figure 14 shows typical mountings for deeply curved spherical windows, called *shells* or *domes*. That in Fig. 14*a* is sealed and secured with a Neoprene gasket clamped in place by a flange[21] while that in Fig. 14*b* is secured and sealed with a continuous ring of elastomer.[4] In some more elaborate designs, an elliptically shaped sapphire dome is brazed with special metallic alloys to a titanium mount.[22]

6.6 MOUNTING SMALL MIRRORS AND PRISMS

General Considerations

The appropriateness of designs for mechanical mountings for small mirrors and prisms depends upon a variety of factors including: tolerable rigid body movement of the optic and distortion of its reflecting and/or refracting surface(s); the magnitudes, application locations, and directions

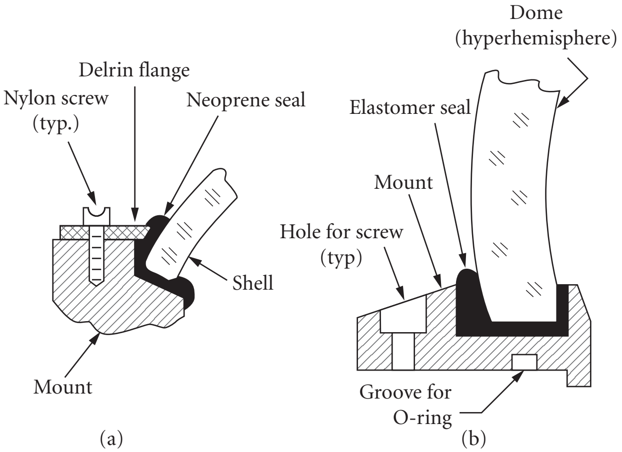

FIGURE 14 Typical mountings for (*a*) a thin shell and (*b*) a hyperhemispherical dome. [(*a*) *From Vukobratovich.*[21] (*b*) *from Yoder.*[4]]

of forces tending to move the optic with respect to its mount; steady-state and transient thermal effects (including gradients); the sizes and kinematic compatibility of interfacing optomechanical surfaces; and the rigidity and long-term stability of the structure supporting the optic. In addition, the designs must be compatible with assembly, maintenance, package size, weight, and configuration constraints, as well as being cost effective. The representative mounting designs described in the following sections illustrate proven mounting techniques.

Mechanically Clamped Mountings

Figure 15 shows a simple means for attaching a first surface flat mirror to a mechanical bracket.[23] Three cantilevered springs press the reflecting surface against three pads that have been lapped coplanar. The contacts between the springs and the mirror's back face are directly opposite the pads to minimize bending moments. This design constrains one translation and two tilts. Translations in the plane of the reflecting surface can most easily be constrained by dimensioning the spacers supporting the springs so as to just clear the rim of the mirror at minimum temperature. Rotation in that plane usually does not need to be constrained. Given the number of springs N, the spring material's Young's modulus E_M, its yield stress S_Y, and its Poisson's ratio ν_M, the spring lengths L and widths b, and an appropriate safety factor f_S, the spring thickness t that will provide a total preload P to the mirror is determined as $[K_{S1}PLf_S/(bS_YN)]^{1/2}$. The length of the spacer located under each spring is chosen to cause that spring to be deflected from its flat condition by Δ equal to $(K_{S2}L^3)(1-\nu_M{}^2)/(E_Mbt^3N)$. In these relationships, K_{S1} is 4 and K_{S2} is 0.75.

Elastomeric Mountings for Mirrors

Small mirrors can often be mounted in the manner illustrated by Fig. 4 for a lens. In applications where the optic does not need to be sealed in place with a continuous ring of elastomer, three or more discrete pads located between the lens rim and the cell ID can support it. Vanbezooijen's equation is again used to determine the pad thicknesses.[10] The lateral dimensions of the pads have, in some designs, been determined by finite element analysis that predicts the dynamic response of the subassembly to vibration inputs from the environment.[24]

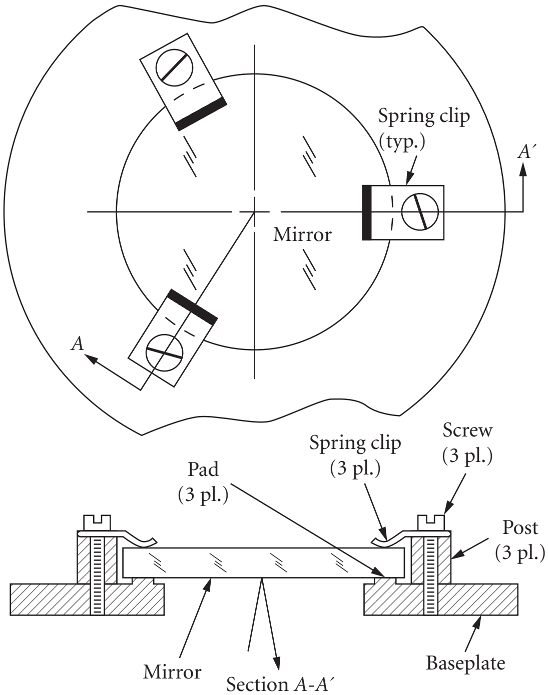

FIGURE 15 A simple mounting for a flat mirror preloaded with deflected cantilevered springs. (*From Yoder.*[23])

Spring-Loaded Mountings for Prisms

A spring-loaded mounting for a prism is illustrated in Fig. 16.[25] Here, a penta prism is preloaded against three coplanar pads on the baseplate by three cantilevered springs supported by posts with spacers machined to produce the necessary spring deflection and resultant preload, as described earlier for a mirror mounting. Constraint in the plane parallel to the pad surfaces is provided by a single spring (called a straddling spring) that is supported at each end and presses against the end of the prism. The dimensions and deflection of this spring are chosen to preload the prism against three locating pins that are pressed or threaded into strategically located holes in the baseplate. The relationships for t and Δ given in Sec. "Mechanically Clamped Mountings" apply also to the straddling spring, but K_{S1} equals 0.75, $N = 1$, and $K_{S2} = 0.0625$. How the direction of the force exerted by the straddling spring can be optimized to nearly equalize the stresses created in the prism at the interfaces with the pins has been explained in the literature.[26]

Bonded Mountings for Small Mirrors and Prisms

A widely used and successful technique for mounting small mirrors and prisms is to bond them directly to a plate or bracket with an adhesive such as epoxy. Any alignment adjustments that are needed should be built into the mount rather than into the glass-to-metal joint. Figure 17 shows a typical mirror mount of this type.[23] It has proven satisfactory for cases where the diameter-to-thickness ratio for the mirror substrate is at least 6:1. The mirror should then be stiff enough not to be excessively

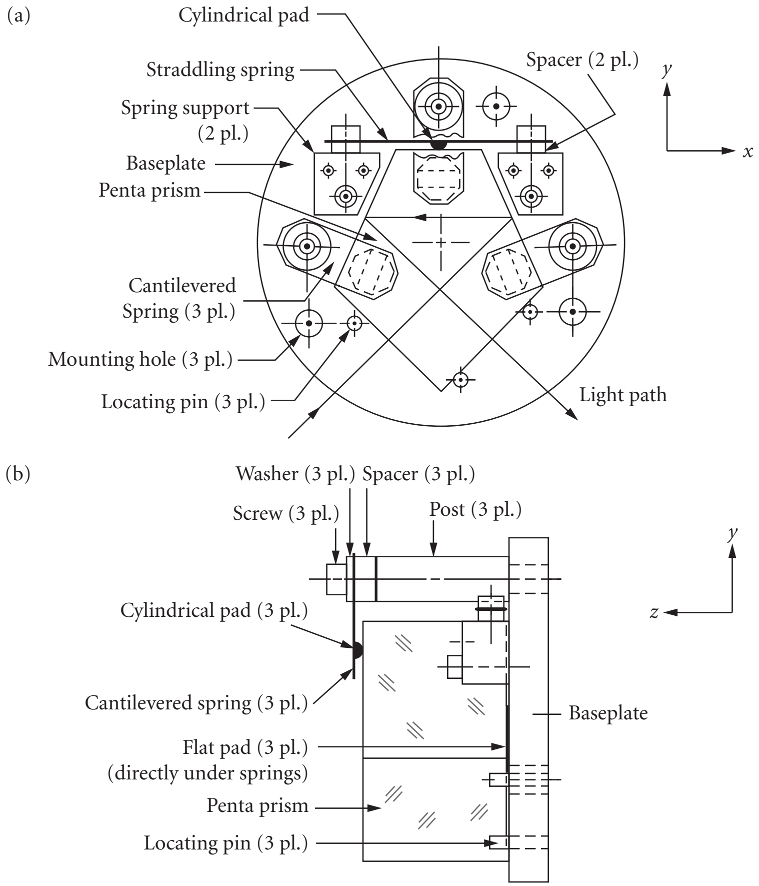

FIGURE 16 A penta prism preloaded against lapped pads on a baseplate with cantilevered and straddling springs. (*From Yoder.*[25])

distorted by shrinkage of the adhesive as it cures. Most prisms are thick enough that they are not distorted by this shrinkage.

All adhesive bonds need to have sufficient area for the joint to be strong enough to support the optic under all anticipated levels of acceleration. The minimum bond area Q_{MIN} is $9.81ma_G f_S/J$, where J is the strength of the cured adhesive joint and all other terms are as previously defined.[27] Bonding should be done on a fine ground surface of the optic for maximum joint strength. A typical value for J for a two-part epoxy such as 3M 2216B/A with bond thickness of 0.100 ± 0.025 mm is ~17.2 MPa. For conservative design, the factor f_S should be ~4. Successful bonding requires careful cleaning of the surfaces to be bonded and adequate curing time. The adhesive manufacturer's recommendations should be followed unless tests indicate otherwise for a specific application.

The 29-mm aperture roof penta prism in Fig. 18 is bonded in cantilevered fashion to a bracket nominally oriented vertically. The circular bond area is adequate to withstand a severe military shock and vibration environment. Some designs work better if the prism is supported from both sides. Figure 19 shows one way to do this. It was adapted from Beckmann.[28] The mount is designed with two arms, one of which has a hole bored through it. The prism is supported by a fixture in the proper location and orientation relative to the mount and epoxy bonded to the flat pad on the left arm. A plug made of the same

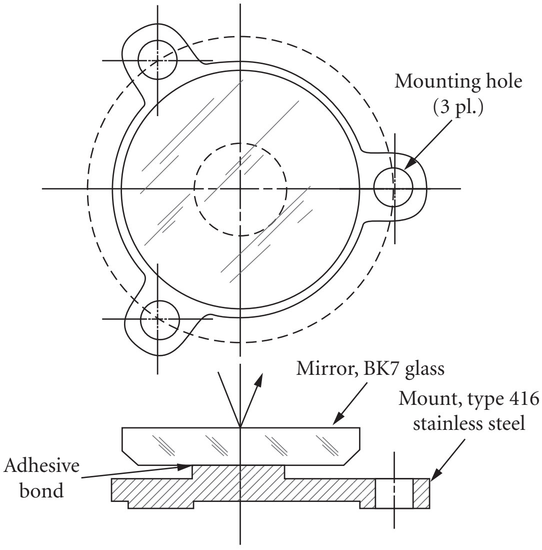

FIGURE 17 A flat mirror bonded on its back to a pad on its mount. (*From Yoder.*[23])

metal as the mount is then centered in the hole in the right arm and bonded to the right surface of the prism. When those bonds have cured, the plug is bonded into the right arm.

Flexure Mountings for Small Mirrors and Prisms

Circular mirrors as large as ~15-cm diameter have been successfully mounted on flexures in the general manner shown in Fig. 5 for a lens. Usually, these are image-forming mirrors, perhaps aspheric, that need to have constant centration relative to a system axis.

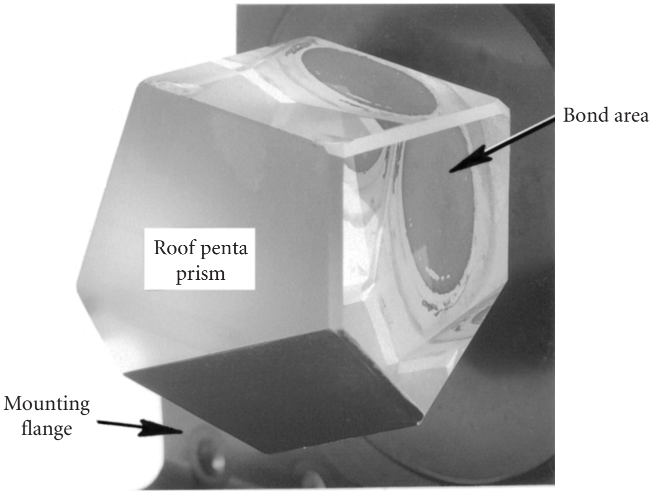

FIGURE 18 A roof penta prism bonded to a pad on a mounting flange.

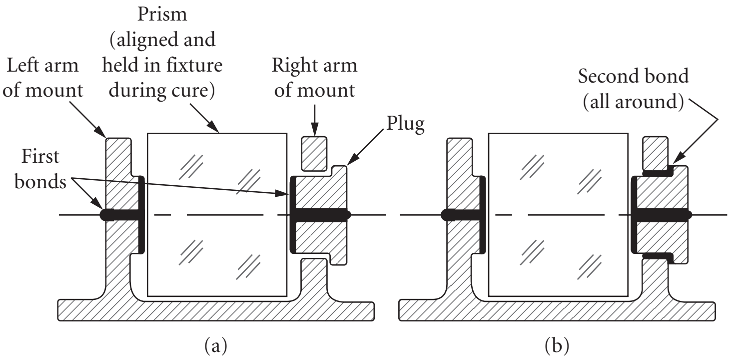

FIGURE 19 Concept for supporting a prism from both sides. (*a*) Prism bonded to left arm of mount and plug bonded to prism. (*b*) Plug bonded into right arm of mount. (*Adapted from Beckmann.*[28])

Prisms intended for use in relatively benign environments can be mounted on flexures. One way to do this is by attaching the prism to instrument structure through three posts with integral flexures at each end. Figure 20 shows a large multiple component Zerodur prism mounted in this manner. It has two wing prisms optically contacted to a third (base) prism, to which the flexures are bonded. The wing prism surfaces are perpendicular to each other and form a 15.2-cm-wide roof mirror that is inclined at 45° to the vertical. The reflected image is inverted horizontally as the optic turns the incident beam axis 90°. The orientations of three of the flexure joints are

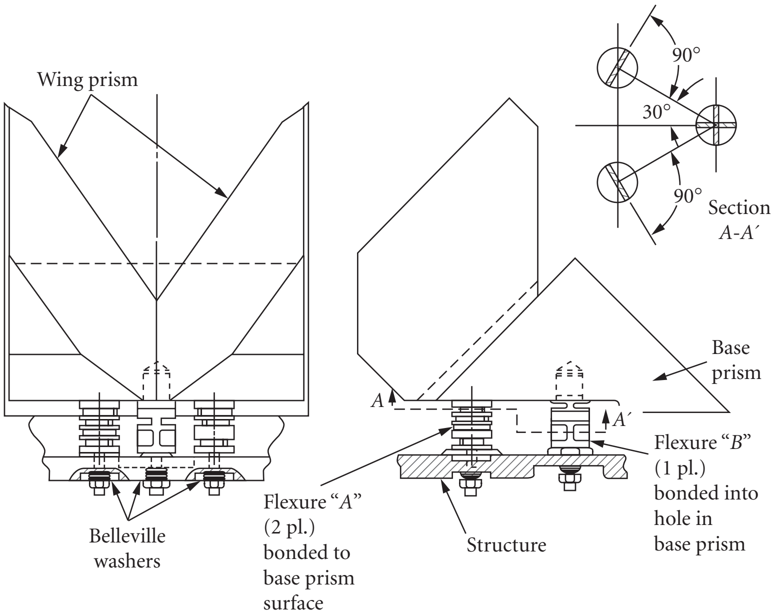

FIGURE 20 Optomechanical configuration of a large prism assembly with three flexure mounting posts to isolate the optic from dimensional changes under temperature changes. (*Courtesy of ASML Lithography, Wilton, CT.*)

as indicated in the section view A-A′. If attached to a structure that expands or contracts more than the prism as the temperature changes, the flexures simply bend very slightly and prevent the introduction of mounting forces that could distort the reflecting surfaces and interfere with performance of the optical system.

6.7 MOUNTING MODERATE-SIZED MIRRORS

General Considerations

The simple mirror mountings described earlier are not satisfactory for mirrors larger than about 15-cm diameter because they are too flexible to be treated as rigid bodies. The important criteria for selecting a suitable mounting are orientation with respect to gravity, performance level required, substrate material stiffness, and weight limitations. The mounting for a mirror to be used in a fixed horizontal- or vertical-axis orientation can be figured during polishing to compensate for gravity effects. Variable orientation applications require mounts that change their force distribution with inclination to keep surface deflections within tolerance. Both axial and radial supports are required. Mirrors to be used in space have the added requirement of release of gravitational force after being fabricated, tested, and installed into the instrument in a normal gravity environment. Choice of mounting depends strongly on the substrate configuration. Weight constraints generally lead to solid substrates with shaped back surfaces or ones built-up from multiple parts that are attached together. We here describe a few typical ways to support mirrors of various shapes as large as ~85 cm. Designs appropriate to both nonmetallic and metallic mirrors are considered.

Substrate Configurations

Figures 21*b* through *e* shows half-section views of four first-surface mirror solid substrates of the same diameter and material with concave surfaces of the same radius. Their back surface shapes differ and reduce the mirror weight as compared to a flat-back baseline design (Fig. 21*a*). All these mirrors, except one, can be supported within the telescope housing on a hub passing through the mirror's central perforation. For example, see Fig. 22. Here, the hub has a toroidal-shaped raised land that supports the 41-cm-diameter meniscus-shaped mirror radially and a shoulder that locates it axially. The radial support lies in the mirror's neutral plane where fore and aft bending moments are balanced. A threaded retaining ring provides axial preload. To focus, the locating ring is moved on the hub and secured with the clamping ring. The substrate configuration from Fig. 21 that cannot be hub mounted is the double arch configuration (Fig. 21*e*). It is best supported on flexures at three or more points spaced equally around the zone of greatest thickness.

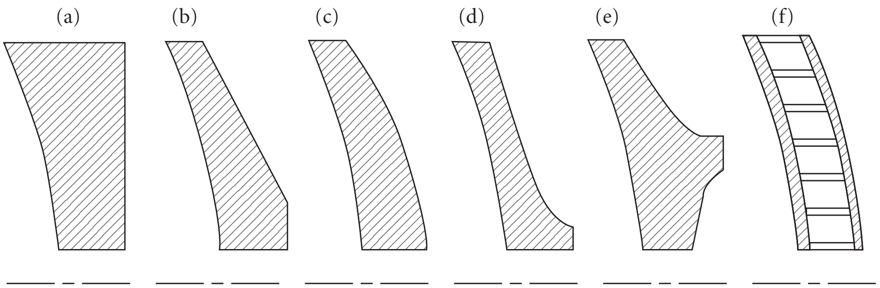

FIGURE 21 Sectional views of baseline concave-plane (*a*) and lightweighted mirror substrates (*b*) through (*e*) with contoured backs. Figure (*f*) shows a built-up substrate configuration.

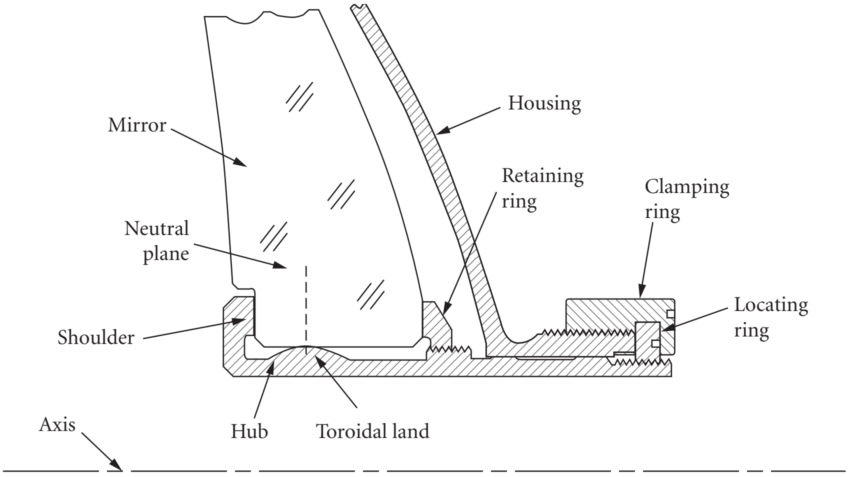

FIGURE 22 Hub mounting for a meniscus-shaped telescope mirror. Focus adjustment means is illustrated.

Lighter-weight mirror constructions typically employ built-up substrates such as that shown in Fig. 21*f*. A very successful type is the monolithic meniscus construction illustrated by Fig. 23. Such mirrors are usually made of Corning ULE. Strips of the material form the webs of a core to which front and back facesheets are fused. All joints in the core also are fused together. The spacing of the webs is large except at locations where axial and radial supports attach to the substrate. There, the spacing is considerably smaller to increase strength.

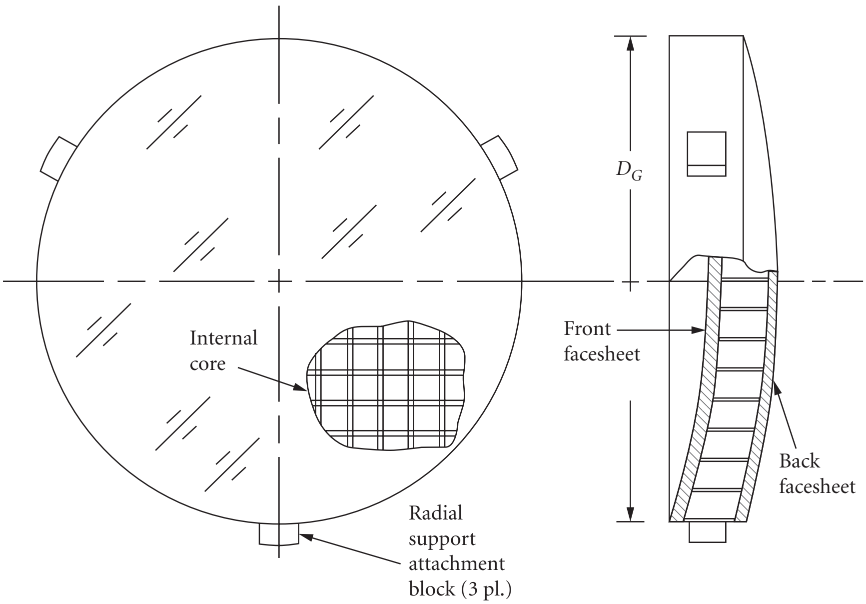

FIGURE 23 A completely fused (monolithic) built-up lightweight mirror substrate.

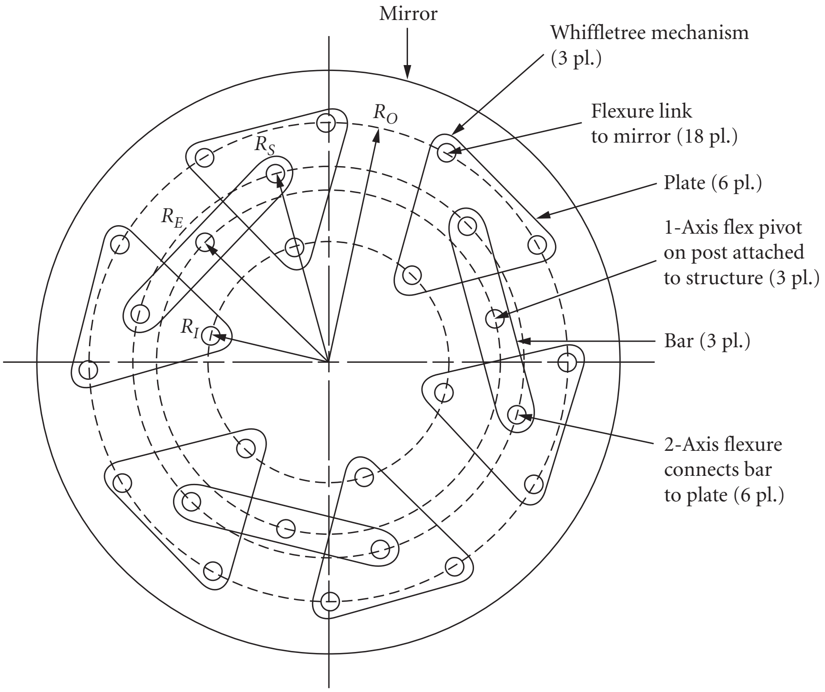

FIGURE 24 An 18-support Hindle-type mirror mount supporting the optic at multiple points on rings of radii R_O and R_I from three posts attached to structure at radius R_E. The whiffletree plates are centered on the ring of radius R_S.

Lever Mechanism Mountings

Because of the flexibility of lightweighted mirrors, axial support is frequently provided at many points on the back of the substrate. Hindle mounts[29] using multiple lever mechanisms (called "whiffletrees") are commonly used. Figure 24 shows such a mount with 18 supports for the mirror. The number of supports needed is the minimum number that keeps the gravitational sag of the reflecting surface between support points smaller than the deflection tolerance when the mirror axis is vertical.[4] To avoid friction effects, flexures (sometimes called "Flex-Pivots") are typically used as single-axis bearings in these mounts. Dual-axis bearings are usually necked-down posts that serve as flexures.

A mirror on a Hindle mount also needs radial support if it is to be used in any orientation other than axis vertical. This might be in the form of three or more mechanical links with universal-joint flexures at each end that are oriented tangent to the rim of the mirror and connect the mirror rim to the surrounding structure. Provision for such a support is shown in the mirror of Fig. 23. Multiple-point whiffletree radial supports have also been used for this purpose.[21]

Mountings for Metallic Mirrors

Metallic mirrors are generally easier to support than nonmetallic ones because attachments can be made directly to the substrate through, for example, threaded holes for screws. The metallic substrate may also be stiffer than the glass counterpart. An example is the aluminum mirror shown by section and back views in Fig. 25*a*.[30] Here, a single-point diamond-turning (SPDT) method is employed to machine the optical surface and the axial and radial mounting interface surfaces on the mirror's back. In this method, many extremely fine cuts are made with a precision diamond tool as the substrate rotates about a common mechanical and optical axis. The tool moves on a prescribed path under interferometric control. This results in very accurate surface shapes and surface interrelationships, as well as smooth surfaces and very low residual stresses in the parts. The mating surfaces on the mount are also created by diamond turning. The mirror is shown installed in its mount in Fig. 25*b*. Optical surface distortions due to mounting forces are minimal because the contacting surfaces on the optic and its mount are parallel when drawn together.[31] When the mirror and its mount are made of the same material, the effects of temperature changes are minimized.

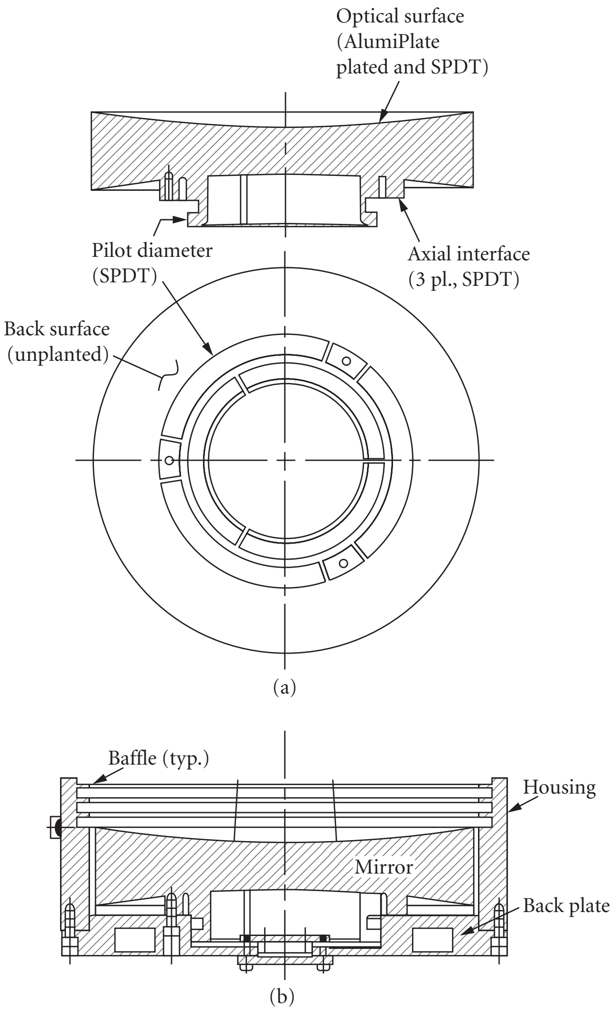

FIGURE 25 Optomechanical configuration (*b*) of an aluminum mirror (*a*) with optical and mounting surfaces machined by SPDT methods. The radial and axial interfaces are shown. (*From Vukobratovich et al.*[30])

6.8 CONTACT STRESSES IN OPTICS

The shape of the mechanical surface touching a lens, mirror, or prism surface is typically spherical, cylindrical, conical, or flat. Point contacts occur at spherical pads attached to the ends of springs while short line contacts occur if cylindrical pads are used on springs or if pins are used to locate the optic. Lenses, windows, and mirrors preloaded against mechanical constraints by a retaining ring or flange typically have circular line contacts with the metal around the edges of their apertures. The metal surface typically is conical for a convex glass surface and toroidal for a concave glass surface. The preloads applied through all of these interfaces cause elastic deformations of the glass and metal parts. Associated with these deformations are compressive and tensile stresses in those materials. Up-to-date analytical methods for estimation of these stresses have been presented in detail elsewhere.[4] Space constraints preclude discussion of those methods here. Once the tensile stress to be expected in a given optomechanical design has been quantified, it can be compared to the aforementioned rule-of-thumb tolerance to predict success or failure of the optomechanical design. Should the stress appear to be too large, certain design changes that can be made to reduce it are suggested in the referenced publication.

6.9 TEMPERATURE EFFECTS ON MOUNTED OPTICS

General Considerations

Because the temperature environment of any optical instrument is seldom constant, we should anticipate changes in dimensions of all parts, in refractive indices, and in material parameters [such as coefficient of thermal expansion (CTE) and Young's modulus] to occur throughout the lifetime of the device. These changes may defocus the system, change aberration balance, or degrade alignment. Athermalized designs are created in a manner to reduce the magnitudes of these effects to tolerable levels.

Prevention of Axial Gaps

Differential expansions and contractions of all types of materials with temperature changes may change the axial and/or radial relationships, that is, alignment between optics and their mechanical reference surfaces. Optomechanical assemblies that are adequately preloaded at assembly will tend to maintain optic-to-mount contact, but this preload will change as the temperature changes. It may disappear completely at elevated temperatures. Then the optics may be free to move if externally disturbed, as by vibration or shock. These component shifts may become permanent if the optic is decentered or tilted when the temperature drops and the mount reapplies forces to the optics.

To reduce this effect, each optical assembly might be designed to compensate for axial dimensional changes so axial preload changes are reduced to insignificance.[32] For example, the air-spaced triplet assembly of Fig. 26*a* is constructed of three optical glasses, an aluminum cell, and two aluminum spacers. The scale of the figure is as indicated. At maximum temperature, the physical separation of the interfacing points *A* and *B* in this particular assembly changes by 0.015 mm if computed for a path through the lenses and spacers, but changes by 0.030 mm if computed for a path through the cell. One or more axial air gaps totaling 0.015 mm would then exist somewhere within the assembly and the lenses might move or tilt within that space. If the design were to be modified by changing the metals in the cell and in one spacer, lengthening that spacer, and providing space for the larger spacer by adding a step bevel to the second lens—as indicated in Fig. 26*b*—the chosen materials and component dimensions would make the A-to-B separation remain equal for both paths for all temperature changes. Preload would then remain unchanged and misalignment would not occur.

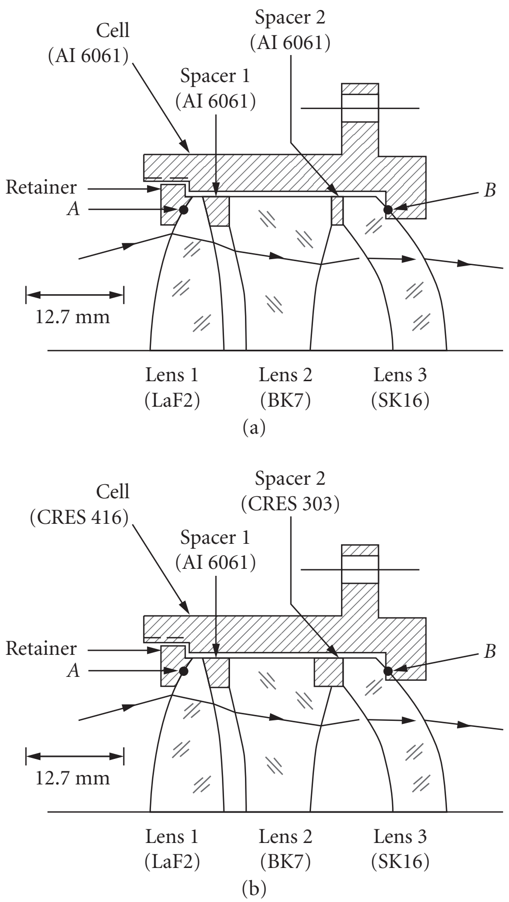

FIGURE 26 An air-spaced triplet lens assembly (*a*) in which an axial gap between glass and metal parts exists at high temperature, possibly allowing the lens to become misaligned. Modified design (*b*) is athermalized to maintain registry of the optics in the mount. (*Adapted from Yoder and Hatheway.*[32])

Focus Athermalization Techniques

Single Material Designs Figure 27 shows a reflecting telescope made of a single material, in this case, aluminum.[33] All dimensions change, but the assembly remains in alignment and the optical performance is unchanged (other than a small change in image scale) as the temperature changes. This telescope is an example of the use of single-point diamond-machining methods as all optical and mounting surfaces are precisely made in the proper geometric relationships so alignment accuracy is built-in.

Passive Athermalization Figure 28*a* illustrates the use of materials with dissimilar CTEs and carefully chosen axial dimensions so the axial distance between optical components (in this case, the two mirrors) remains constant when the temperature changes.[34] This keeps the optical performance within required limits. Control of the mirror separation of this telescope is modeled schematically in Fig. 28*b*. Positive signs associated with lengths of low and high CTE materials indicate how the

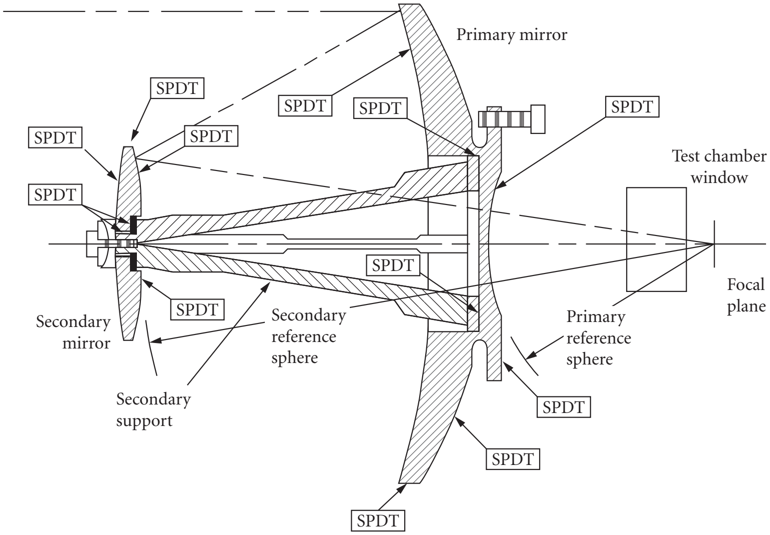

FIGURE 27 Schematic of an all-aluminum (athermalized) telescope objective with optical and mechanical interface surfaces finished by SPDT for ease of assembly without alignment. (*From Erickson et al.*[33])

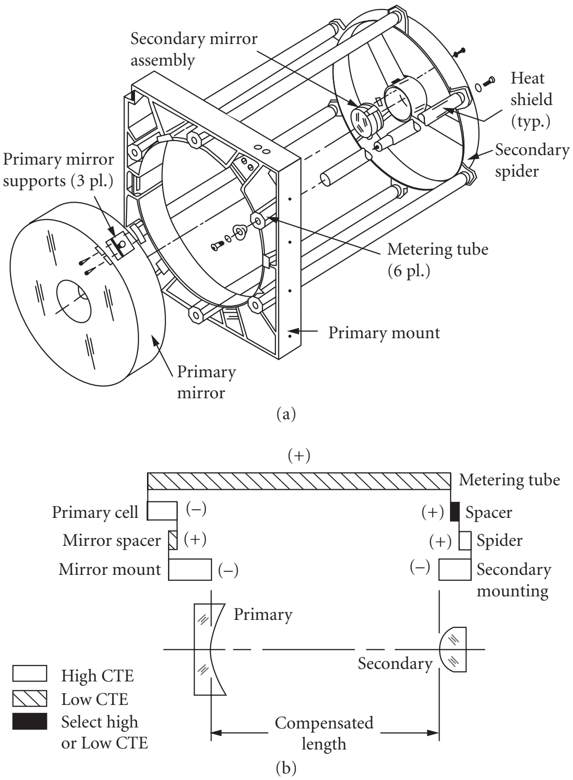

FIGURE 28 A passively athermalized telescope structure using Invar metering tubes to connect the primary and secondary mirror mounts. (*a*) Exploded view of the telescope. (*b*) Model of the compensation system. (*From Zurmehly and Hookman*[34])

mirror separation changes as the temperature rises. Proper choices of materials for their coefficients of thermal expansion and dimensions make the mirror separation remain constant as the temperature changes.

Active Athermalization When a source of power is available, components in an optical system can be physically moved to compensate for the effects of temperature changes. For example, Fig. 29*a* shows a concept for a zoom lens system in which locations of the moveable lenses are varied by motors as commanded by an internal microprocessor that monitors the temperature of the system.[35] As indicated in Fig. 29*b*, desired magnification inputs from the operator are automatically converted into the lens shifts required to focus properly on the object at the measured temperature.

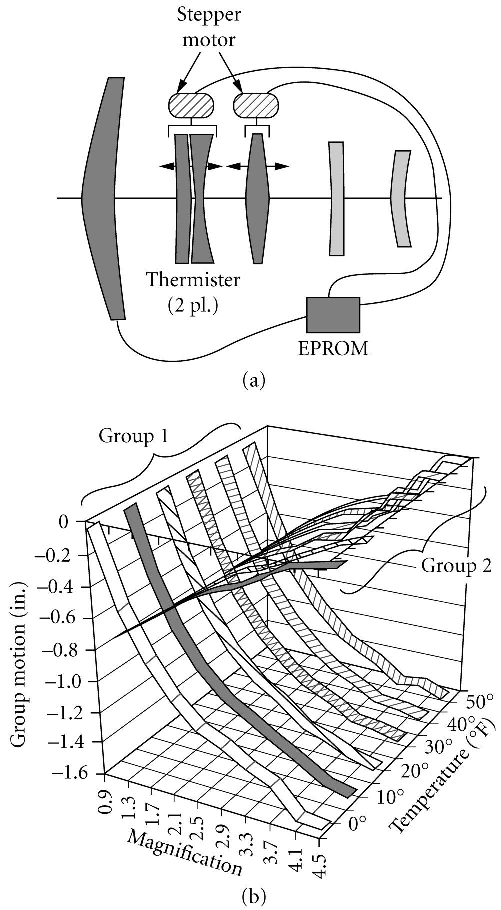

FIGURE 29 An actively athermalized zoom lens system that drives two lens groups to maintain focus at selectable magnification settings in spite of temperature changes. (*From Fischer and Kampe.*[35])

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