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Lens mounting techniques

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Abstract

This paper reviews a variety of techniques used for securing lenses in their mechanical surrounds and for the design of those surrounds so as to ensure proper function of the assembled parts in the intended environment. Mounting arrangements for individual lenses and for combinations of lenses are described briefly. The treatment progresses from simple, low-precision designs to more complex and higher precision versions. Component sizes considered here range from a few centimeters to tens of centimeters in diameter. Techniques for analyzing glass-to-metal interface designs to determine their adequacy when exposed to varying environmental conditions are summarized.

Introduction

In this paper we review a variety of ways to mount rotationally symmetric (i.e., centered) lenses in the size range of approximately 0.5 to 10 in (1 to 25 cm) diameter. Techniques for mounting individual refracting and reflecting elements are described first. Combinations of components that form optical subassemblies are then considered. In general, we progress from simple, low precision designs to more complex and higher precision versions.

Mounting individual lenses

Burnishing the lens permanently into its cell is illustrated by Figure 1 and is accomplished by deforming the cell lip after the lens is inserted. The cell material must be malleable rather than brittle. Brass and aluminum alloy are commonly employed for this reason. A radial clearance of 0.001 to 0.005 inch (0.025 to 0.125 mm) between lens and cell is common practice (Horne 1972). The technique is inexpensive, requires no extra parts, and holds the lens firmly in position if properly done. The operation is performed by chucking the cell in a lathe, inserting the lens, rotating the cell, and bringing a hardened tool against the projecting lip, forcing it over against the lens. Care must be exercised to not excessively strain the lens during the burnishing operation. The lens should be held firmly against its seat while the lateral force is exerted on the metal so as not to tilt the lens at the early stage of the process before it is secured.

A modification of the burnishing method has a coil spring inserted behind the lens in the cell and slightly compressed as the lip is spun over the lens. Jacobs (1943) suggested that this method is useful when the assembly is to be subjected to severe shock or when a low-cost means of avoiding stress in the lens is needed. A thin slotted brass tube may be used as the spring.

A snap ring that drops into a groove in the inside surface of the cell frequently is used to hold lenses in a cell. Figure 2 shows such a design with a ring of circular cross section.

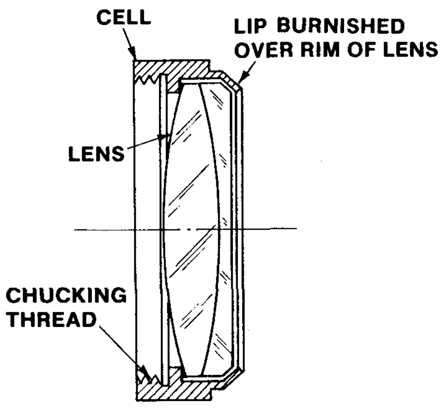


Figure 1. Lens burnished into a cell made of a malleable metal

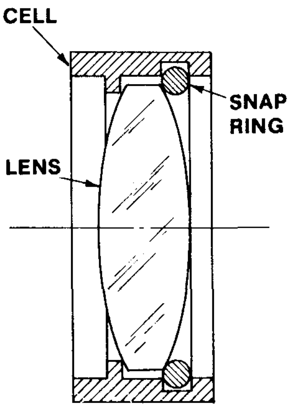


Figure 2. Technique for holding a lens in a cell with a snap ring in a groove

Rectangular section snap rings also can be used. In both cases the ring is cut through at some point on its periphery to allow it to spring into place. Variations in lens thickness as well as in groove location and depth and snap ring cross section dimensions affect the interface between the lens and ring. Use of a v-groove and a circular cross section snap ring tends to alleviate this problem.

A continuous ring can be used without a grooved cell if the parts are dimensioned and toleranced so an interference fit is achieved between the ring and the inner wall of the cell. If the assembly is to be exposed to large temperature changes, the ring and cell materials should have reasonably close thermal expansion coefficients to prevent loosening. This is a rather permanent assembly technique, since it is difficult to remove the ring without damaging the optic.

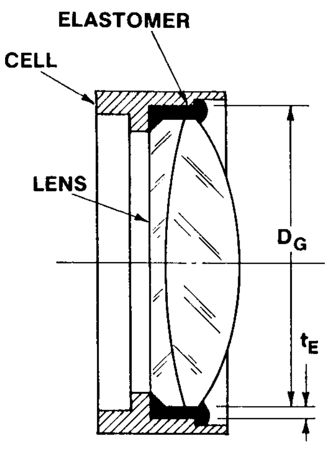
Another simple mounting technique is illustrated in Figure 3. Here a radial spacing is allowed between the lens and cell and the annular void filled with resilient material such as a room temperature vulcanizing (RTV) elastomer. Hopkins (1980) listed General Electric RTV-566 and Dow-Corning RTV Clear -732 as appropriate materials. Positioning of the lens in the center of the cell cavity can be accomplished by mechanical fixturing or by using three narrow plastic (Mylar or Teflon) shims of suitable thickness to center the lens temporarily. Application of the elastomer is commonly accomplished with a hypodermic syringe. The lens must be held axially against its seat while this material is injected and cured.

The appropriate radial thickness of the annular elastomer layer between the glass element and the metal cell can be computed using the equation shown in the figure. This calculation assumes thermal equilibrium throughout the assembly and compensates for differential expansion by deformation of the elastomer.

The values shown to the right of Figure 3 are typical values that we will use in a sample calculation. Substituting, we obtain t_E = 0.093 in. Here the cell is aluminum and the glass is BK 7. If a metal more closely matching BK 7 in thermal expansion coefficient were to be used in this cell, the elastomer layer could be thinner.

A common method for mounting an individual lens or mirror with rotational symmetry is to secure it against a shoulder in a cell with a threaded retaining ring as shown in Figure 4. This type design has many advantages: it gives a firm mounting that can be assembled and disassembled relatively easily, it compensates for axial thickness variations of the element, it lends itself easily to environmental sealing with an elastomer or O-ring, and it is compatible with mounting multiple elements in the same cell or housing.

The retaining ring loads the element axially against an annular seat which may be cut square to the axis (as shown in the figure), tangent to the radius of curvature of the element or cut and lapped to the same radius of curvature as the lens surface. The retainer may also be square, tangent, or ground spherically.



ELASTOMER LAYER THICKNESS TO ABSORB DIFFERENTIAL THERMAL EXPANSION IS

$$t_E = D_G(\alpha_M - \alpha_G)/(2)(\alpha_E - \alpha_M)$$

WHERE FOR TYPICAL CASE:

D_G = LENS DIAMETER = 2.0 INCHES
α_M = METAL CTE = 13.3 PPM/°F
α_G = GLASS CTE = 3.94 PPM/°F
α_E = ELASTOMER CTE = 114 PPM/°F

SUBSTITUTING, t_E = 0.093 INCH

Figure 3. Technique for holding a lens in a cell with a layer of cured-in-place elastomer

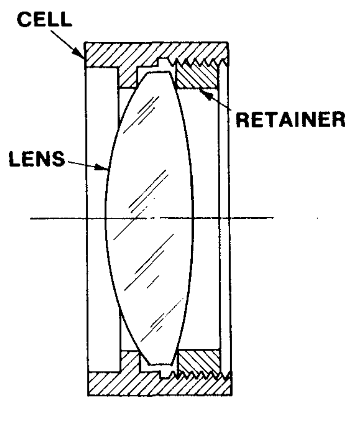


Figure 4. Common configuration for mounting a double-convex lens with a threaded retaining ring. Interfaces with glass are at "sharp" (90°) corners.

The square configuration is the most commonly used type and is the easiest to machine; the contacting edge is left sharp, i.e., just free of burrs. This usually results in a minute chamfer or radius.

The angle at which the surface of a lens seat, retaining ring, or spacer should be machined in order for it to be tangent to a given convex spherical lens surface (see Figure 5a) is given by the equation $\alpha = 90^{o} - \text{arc sin}\ (y_t/R)$. This is the half-angle of the corresponding right-circular cone. The annular radius of contact is y_t and R is the lens surface radius of curvature. It is common practice to define y_t so contact occurs midway between the lens clear aperture and its outside diameter.

The tolerance on α in a given design depends primarily upon the radial width of the conical annulus provided on the metal part and the allowable error in axial location of the lens. If the angle is machined too steeply (i.e., α is larger than nominal), the actual contact will occur nearer the axis and the tangent condition could degenerate into line contact at the inside edge of the metal part. An error in annular radius y_t could misposition the lens axially if calculations are relied upon and measurements of axial location are not made during assembly. A typical tolerance on α is $\pm 1^{o}$.

Figure 5b illustrates the spherical contact lens-cell interface. It is hard to make accurately and, hence, is expensive.

A preferred embodiment of the square seat and retainer design (Hopkins 1980) is shown in Figure 5c. The metal is beveled at 45^{o} so that the corner that contacts the glass is obtuse ($\sim 135^{o}$). This helps minimize nicks or burrs on the machined edge that would prevent line contact over a full circle.

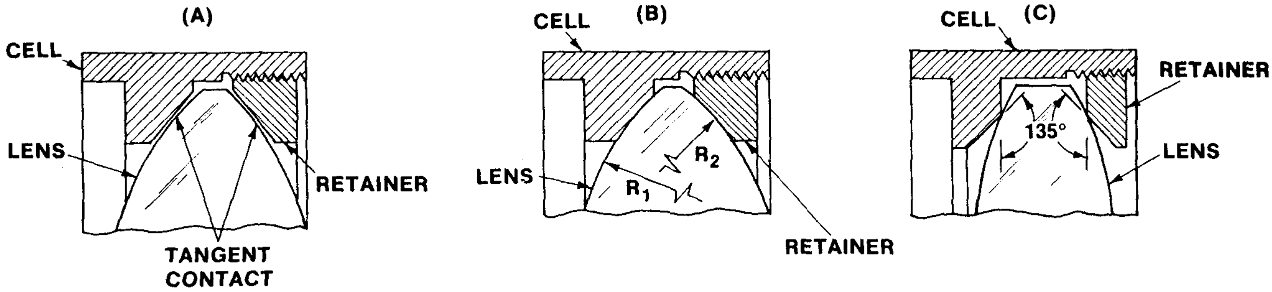


Figure 5. Three configurations for glass-to-metal interface in retaining ring type lens mounts (a) tangential, (b) spherical, (c) obtuse angle

Figure 6a shows a commonly used mounting for a double-concave lens. The seat and retaining ring both contact flat bevels on the lens. If the flat bevels are not both perpendicular to the optical axis through the centers of curvature, it will be impossible to center both surfaces simultaneously on a common mechanical axis by translating the lens. This problem can be reduced only by applying close tolerances to the annular flats. The obtuse angle line contact configuration of Figure 6b is less susceptible to these problems, but is more expensive to fabricate. A tangential interface with flat metal surfaces cannot be used with a concave lens surface. It can be approximated with curved metal surfaces of radius shorter than the lens surface.

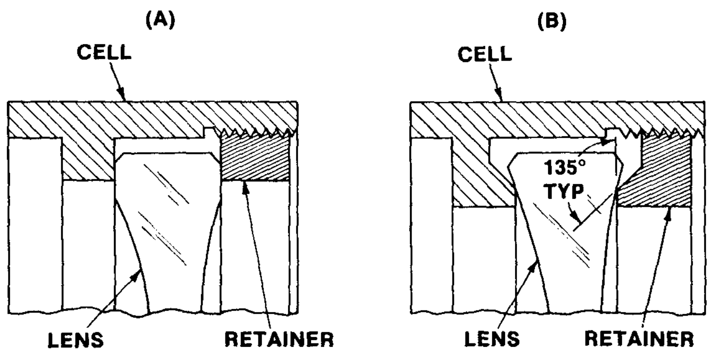


Figure 6. Configurations for mounting double-concave lenses. (a) conventional flat bevel interface, (b) obtuse angle interface.

Analysis of worst-case axial stresses in the glass-to-metal interface

To select the proper seat, spacer, and/or retaining ring configurations for a specific design, the worst-case axial compressive stress or preload, imparted to the glass by tightening the retainer and, additionally, by differential thermally induced contractions, should be computed. Delgado and Hallinan (1975) outlined a method (illustrated in Figure 7) for defining the stress concentration in the line contact of a square-corner spacer on a lens by considering the contact between the lens surface and the spacer to be analogous to the line contact of two cylinders of different diameters whose axes are parallel and uniformly loaded against each other. The larger cylinder represents an element of the lens surface and the smaller cylinder an element of the spacer. A "sharp" corner on the spacer or seat may be assumed to have a radius of 0.002 in. (0.05 mm).

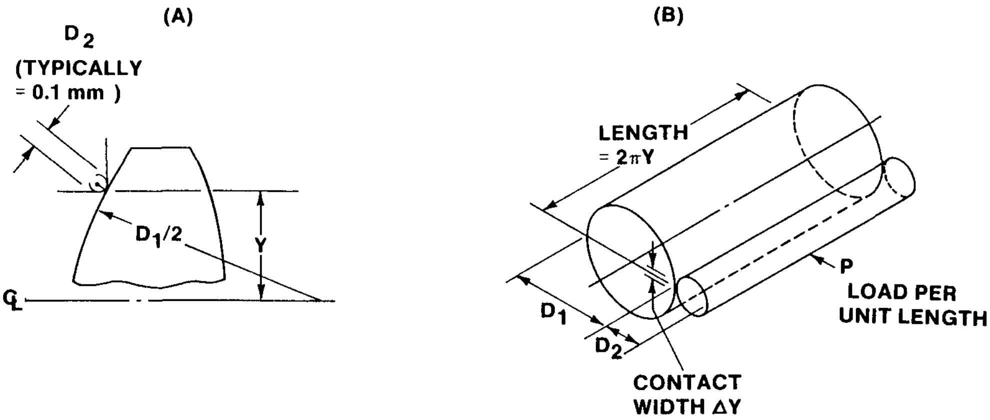


Figure 7. Elastic stress in a lens due to axial loading at "square" seat or retainer. (a) geometric configuration, (b) analytical equivalent.

Using an equation for elastic body stress from Roark (1954), we can compute the axial compressive stress S_A as shown in Figure 8.

The locally deformed glass/metal interface over which this load is distributed is a rectangle of width Δy and length = $2\pi y$, where y is the radial distance from the lens axis to the contact midpoint. The equation for Δy is given in the figure.

As an example of the use of these equations, let us assume the values given at right in the figure for the various parameters. Substituting, we obtain S_A = 21,636 lb/in^2 and Δy = 0.00003 in. For y = 1.0 in. the contact length is 6.283 in. and the total contact area is 0.0002 in^2.

ASSUME ELASTIC BODIES. EQUATIONS PER ROARK (1954) APPLY.

$$\text{AXIAL STRESS} = S_A = 0.798 \left[\frac{(P)(D_1 + D_2)/(D_1 D_2)}{((1 - \nu_G^2)/E_G) + ((1 - \nu_M^2)/E_M)} \right]^{1/2}$$

$$\text{CONTACT AREA} = 2\pi Y \Delta Y$$

$$\text{WHERE } \Delta Y = 1.6 \left[\left(P D_1 D_2 / (D_1 + D_2) \right) \left(((1 - \nu_G^2)/E_G) + ((1 - \nu_M^2)/E_M) \right) \right]^{1/2}$$

PARAMETERS & TYPICAL VALUES ARE:

P = LOAD PER UNIT LENGTH OF LINE CONTACT	= 0.50 LB/IN
D_1 = LARGE CYLINDER DIAMETER = TWICE LENS RADIUS	= 60.000 IN
D_2 = SMALL CYLINDER DIAMETER = TWICE CORNER RAD.	= 0.004 IN
ν_G = POISSON'S RATIO FOR GLASS	= 0.208 (BK 7)
ν_M = POISSON'S RATIO FOR METAL	= 0.332 (ALUMINUM)
E_G = MODULUS OF ELASTICITY FOR GLASS	= 11.8 X 10^6 LB/IN^2
E_M = MODULUS OF ELASTICITY FOR METAL	= 10.0 X 10^6 LB/IN^2
Y = HEIGHT OF LINE CONTACT ON LENS	= 1.0 IN

COMPUTED VALUES: S_A = 21,636 LB/IN^2 (SAFE LOAD ≅ 50,000 LB/IN^2)
ΔY = 0.00003 IN
CONTACT AREA = 0.0002 IN^2

Figure 8. Typical axial stress calculation for "sharp" corner interface

This preload exists at the ambient temperature of assembly and might by itself be considered a safe compressive load on the glass, since that material generally can withstand over 50,000 lb/in^2 ($\sim$ 345 N/mm^2) distributed load. However, we have not considered the added stress that would occur with temperature changes.

Bayar (1981) gave an equation that evaluates the axial compressive stress introduced into a lens by a metal cell that shrinks more than the glass as the temperature drops. This is given in Figure 9. The value of ΔT shown here is the change from 68^{o}F (20^{o}C) to -80^{o}F (-62^{o}C), characteristic of military equipment survival requirements.

If, as would normally be the case, the lens is preloaded at assembly when the temperature is nominally 68^{o}F, the preload stress would add to that from the temperature change. In the example under consideration, the values to be added are 21,636 lb/in^2 and 7500 lb/in^2, giving 29,136 lb/in^2. This is the worst-case stress.

ASSUME: TEMPERATURE DROPS BY $\Delta T = 68°F - (-80°F) = 148°F$
METAL CONTRACTS MORE THAN GLASS.

APPLY EQUATION FROM BAYAR (1981):

$$\text{THERMAL STRESS} = S_A' = \frac{(\alpha_M - \alpha_G)(E_M E_G \Delta T)}{(E_M + E_G)}$$

PARAMETERS & TYPICAL VALUES ARE:

α_M = **COEFFICIENT OF THERMAL EXPANSION FOR METAL = 13.3 PPM/°F**
α_G = **COEFFICIENT OF THERMAL EXPANSION FOR GLASS = 3.94 PPM/°F**
E_G & E_M **AS PREVIOUSLY DEFINED (11.8 X 10^6 & 10.0 X 10^6 LB/IN^2)**

COMPUTED VALUE: $S_A' = 7500$ **LB/IN^2**
ADD PRELOAD $S_A = 21{,}636$ **LB/IN^2**

WORST CASE AXIAL STRESS = 29,136 LB/IN^2

SAFE LOAD = 50,000 LB/IN^2 SAFETY FACTOR $<$ 2

Figure 9. Additional axial stress due to differential thermal effects

The stress imparted to the glass surface could be reduced drastically without reducing the loading force of the retainer that might be needed to keep the lens positioned correctly during shock and vibration by changing the design to tangent contact between glass and metal parts. (See Figure 10.)

The axial stress S_A'' and the contact area width $\Delta y''$ in this situation can be computed from equations from Roark (1954). They pertain to a cylinder loaded against a flat surface. Here D is twice the lens radius of curvature and all other parameters are as defined earlier.

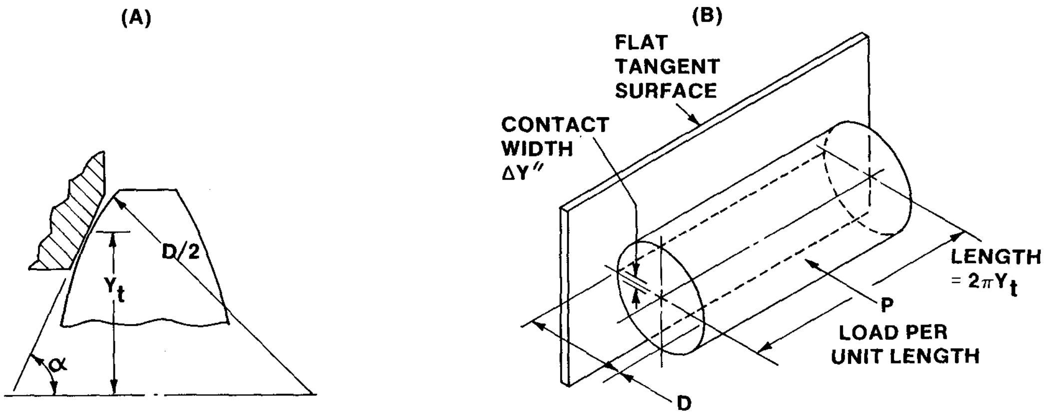


Figure 10. Elastic stress in a lens due to axial loading at tangential seat or retainer. (a) geometric configuration, (b) analytical equivalent.

Substituting the values from our last example into the equations given in Figure 11, we find that the preload stress with tangent contact is reduced to 177 lb/in^2. Even in the worst-case thermal exposure ($\Delta T = -148^{o}F$) the total stress is only 177 + 7500 = 7677 lb/in^2, which is down by a factor of 6 from the danger value. We could then consider the tangential contact design as acceptable from a low temperature survival viewpoint.

CHANGE DESIGN TO TANGENTIAL CONTACT AT HEIGHT Y_T
APPLY EQUATIONS PER ROARK (1954)

$$\text{AXIAL STRESS} = S_A'' = 0.798 \left[\frac{(P/D)}{((1-\upsilon^2_G)/E_G) + ((1-\upsilon^2_M)/E_M)} \right]^{1/2}$$

$$\text{CONTACT AREA} = 2\pi Y_T \Delta Y''$$

$$\text{WHERE } \Delta Y'' = 1.6 \quad \left[(PD)\,(((1-\upsilon^2_G)/E_G) + ((1-\upsilon^2_M)/E_M))\right]^{1/2}$$

PARAMETERS & TYPICAL VALUES ARE:

P	= LOAD PER UNIT LENGTH OF LINE CONTACT	= 0.50 LB/IN
D	= CYLINDER DIAMETER = TWICE LENS RADIUS	= 60.000 IN
υ_G	= POISSON'S RATIO FOR GLASS	= 0.208 (BK 7)
υ_M	= POISSION'S RATIO FOR METAL	= 0.332 (ALUMINUM)
E_G	= MODULUS OF ELASTICITY FOR GLASS	= 11.8 X 10^6 LB/IN^2
E_M	= MODULUS OF ELASTICITY FOR METAL	= 10.0 X 10^6 LB/IN^2
Y_T	= HEIGHT OF CONTACT ON LENS	= 1.0 IN

COMPUTED VALUES: $S_A'' = 177$ LB/IN^2
CONTACT AREA = 0.023 IN^2
WORST CASE @ - 80°F TOTAL $S_A'' = 177 + 7500 = 7677$ LB/IN^2

Figure 11. Typical axial stress calculation for tangential interface

Analysis of worst-case radial stresses in the glass-to-metal interface

So far we have considered lens mounting techniques in which the lens was edged to a diameter considerably smaller than the inside diameter of the mating cell. Clearances were adjusted so at worst-case of manufacturing tolerances the lens would not be "unduly" compressed radially by the cell at the lowest expected temperature.

Some lens assemblies are designed with very little radial clearance between glass and metal. A case in point is one intended for use in a severe shock and/or vibration environment. Nominal clearances as small as a few ten thousandths of an inch might be provided and tolerances specified so the maximum clearance does not exceed ~ 0.0005 in. In such designs, it is important to match the coefficient of expansion of the cell material closely to that of the glass, since extreme temperature decreases could otherwise cause excessive radial compression of the cell onto the glass and perhaps break the lens.

This stress is called "hoop stress" and can be evaluated with the help of the equation from Bayar (1981) shown in Figure 12. Substituting the given values (that pertain to the same BK 7/aluminum lens design example considered earlier) into this equation we find S_R = 1566 lb/in^2.

It should be noted that ΔT should be the change from the temperature at which the metal just touches the glass. If ΔT is measured from the temperature at which a nominal radial clearance δy exists in a particular design, then the value of S_R must be multiplied by a factor K_R, as defined by the equation shown at the bottom of Figure 12.

FROM BAYAR (1981) "HOOP" STRESS IS:

$$S_R = (\alpha_M - \alpha_G)\,\Delta T / ((1/E_G) + (Y/E_M T))$$

PARAMETERS & TYPICAL VALUES FOR BK 7 & ALUMINUM ARE:

α_M	= METAL COEFFICIENT OF THERMAL EXPANSION	= 13.3 PPM/°F
α_G	= GLASS COEFFICIENT OF THERMAL EXPANSION	= 3.94 PPM/°F
ΔT	= TEMPERATURE CHANGE	= - 148°F
E_G	= GLASS MODULUS OF ELASTICITY	= 11.8 X 10^6LB/IN^2
E_M	= METAL MODULUS OF ELASTICITY	= 10.0 X 10^6LB/IN^2
Y	= ½ OUTSIDE DIAMETER OF LENS	= 1.0 IN
T	= CELL WALL THICKNESS	= 0.125 IN

COMPUTED VALUE: $S_R = 1566$ LB/IN^2
IF RADIAL CLEARANCE EXISTS, MULTIPLY BY:
$K_R = 1 - (\delta Y/(Y\Delta T(\alpha_M - \alpha_G))) = 0.64$, FOR $\delta Y = 0.0005$ IN
THEN $S_R = 1000$ LB/IN^2 (SAFE LOAD IS ~50,000 LB/IN^2)

Figure 12. Calculation of "hoop stress" due to temperature change

Assuming δy to be 0.0005 in in our example, we compute K_R as 0.64 and correct the above-computed value of S_R to 1000 lb/in^2. The safe value for BK 7 glass is about 50,000 lb/in^2, so this design has a large safety factor.

Analysis of stresses under operating conditions

The operating temperature range for an optical instrument is undoubtedly more benign than its survival range. The changes in mounting stresses due to these lesser temperature changes would therefore not be catastrophic but rather ones that affect performance of the optics. Structural contractions and expansions that change element locations and alignments lie beyond the scope of this paper. Here we limit our considerations to effects within the specific lens mounting. The two effects of interest are birefringence and surface deformation.

A generally accepted criteria for compressive stress that does not introduce noticeable birefringence due to exercise of the material's stress-optic characteristics is $\sim$ 500 lb/inch (3.45 N/mm^2).

The axial stress equation given earlier can be applied to operational conditions in the same manner as previously illustrated for worst-case survival by letting ΔT represent the change in temperature associated with operational requirements. For example, if the aluminum/BK 7 lens with tangent glass-to-metal contacts of our previous example were to be used in a controlled thermal environment so ΔT is only $\pm$ 5°F from ambient at assembly, the stress due to thermally induced cell compression would be only 253 lb/in^2. This is about one-half the total stress allowable from a birefringence viewpoint. The balance could be allocated to preload stress. Fortunately, birefringence does not exert a strong influence on performance of most optical systems, since few can be used in such tightly controlled temperature conditions. In the case of laser interferometers, polarimeters, and other devices that function by virtue of transmission of polarized light, special care should be exerted in design to prevent undue stress of the optics.

The tightening torque applied to a retaining ring should be minimized, while keeping it compatible with the applicable shock and vibration requirements. In some cases, the mounted lens can be examined in a polariscope to detect excessive stresses. Interferometric evaluation of mounting stresses is commonly applied in high-precision applications (see Bayer 1981).

Analytical methods for predicting the deformations of surfaces under mounting stresses include solution of closed-form equations and finite element analysis - both these methods are too complex for consideration here.

Multi-element mounting configurations

We will now consider examples of various ways in which two or more lenses may be mounted together to form opto-mechanical subassemblies. The treatment is by no means all inclusive; the examples were chosen to illustrate use of the design features described earlier as techniques for mounting individual elements and components.

Stack-mounted assemblies are those in which the lenses are inserted in sequence into the cell or lens barrel with spacer rings to separate them by the proper airspaces. A single retainer usually holds all these parts in place.

Figure 13 shows a fixed focus eyepiece subassembly for a military telescope. Here, two identical achromatic doublets back-to-back form a symmetrical eyepiece of the Ploessl type (Rosin 1965). Both lenses and the spacer fit into an internal bore in an aluminum cell with typically 0.003 inch (0.075 mm) diametrical clearance. The first inserted lens registers against the squared seat shown at the right in the figure. The spacer is of the square configuration as is the threaded retainer that holds both lenses in place. The doublets are generally edged after cementing, so both elements have the same outside diameter.

The right-hand lens is sealed against the environment by injecting sealing compound into one or more holes through the cell wall. These holes penetrate into a rectangular groove around the inside of the cell adjacent to the rim of the lens. Injection continues until sealant emerges from all holes and a continuous seal can be observed through the ground lens rim. After curing, this seal is quite effective throughout the common military environment, including exposure to -80° and + 160°F temperatures and pressure differentials of at least $\pm$ 5 lb/inch^2.

The two rectangular grooves on the outside of the cell are worth mentioning. This subassembly is designed to be inserted into a cylindrical bore in the telescope housing. Two set screws through the housing project into the right-hand groove and clamp the eyepiece after it has been focused. Sealing compound is then injected into the left-hand groove through holes in the housing to seal the eyepiece subassembly to that housing.

The opto-mechanical configuration of a multiple-element objective for a relatively high-performance military telescope intended to withstand a severe shock and vibration environment is illustrated by Figure 14. The three singlets have the same outside diameter and fit into a stainless steel cell with nominally 6 μm diametrical clearance. All lenses are inserted from the right side. The first is Schott SF 4 glass and has a plano entrance face that registers against a square shoulder on the cell. The first spacer is 66 ± 0.005 mm thick and is made of stainless steel sheet. The second lens is Schott SK 16 glass, while the third is Schott SSK 4 glass. Spacer No. 2 is made of the same steel as the cell and is shaped for tangential contact on the adjacent convex surfaces. The retainer also is made of the same steel as the cell and is machined square to interface with an annular flat on the third lens. All metal parts are passivated and dyed black.

The wedge tolerances on the lenses and spacers are 10 arc seconds and the maximum edge thickness variation from the annular flat to the first surface on the third element is toleranced at 10 μm. At assembly, the lens elements are phased by rotation about the axis for maximum symmetry of the on-axis image.

The radial compression of the cell is slightly greater than that of the glass when the temperature of the assembly is reduced. With the help of the equations given earlier, we can calculate the worst-case "hoop stress" on the glass. Using appropriate values for the design parameters, we compute this stress to be 2925 lb/in^2 after correcting for the 3 μm radial clearance. This is far below the limit, so a large safety factor exists.

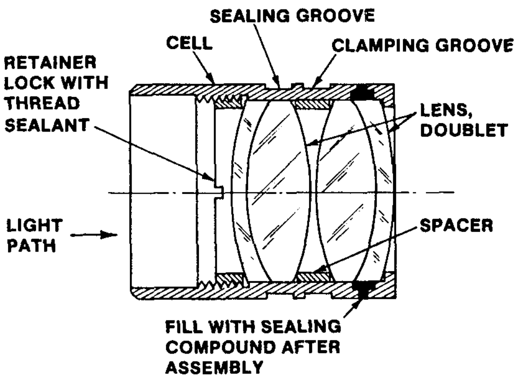


Figure 13. Example of a fixed focus eyepiece subassembly for a military telescope.

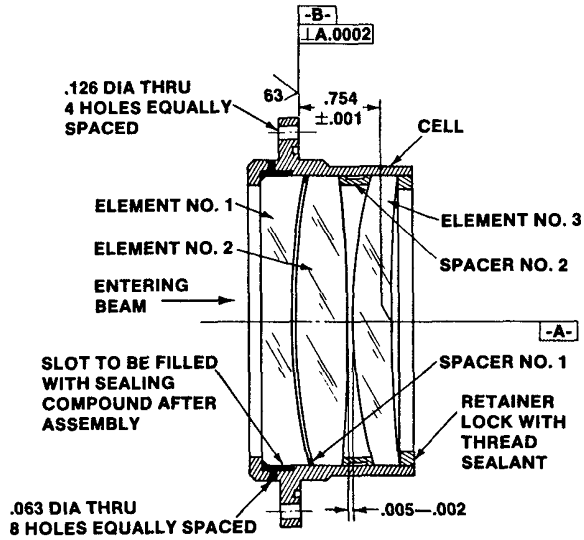


Figure 14. Example of a high performance telescope objective subassembly. Dimensions are inches.

Lathe assembly techniques

In this type of mounting, elements are radially positioned by the inside diameter (ID) of the mating cell or housing. The outside diameter (OD) of each element must be precision ground to a high degree of roundness and measured. The ID of the mating metal part is then machined to fit that specific element. In multi-element designs, the axial positions of the various elements are established by properly locating the machined seats while cutting the IDs. Since this machining process is traditionally done on a lathe or similar machine tool spindle, it has come to be known as a "lathe assembly".

In a high-performance lens assembled in this manner, nominal diametrical clearance between the OD of the element and the ID of the metal part may be as small as 0.0002 inch (5 μm).

The adequacy of a design in regard to radial loading of the lens due to shrinkage of the cell at low temperature may be evaluated by calculating the "hoop stress" using the equation given in Figure 12.

To illustrate the actual lathe assembly process, let us consider the design of Figure 15. Here two lens elements are to be hard-mounted into a cell with a small radial clearance between metal and glass parts. The seat for the convex surface of Lens No. 1 is to be cut tangent to the surface and to the proper depth to position the vertex of that optical surface at 57.150 ± 0.010 mm from flange mounting surface B. The airspace F is to be controlled to within ± 15 μm of the nominal value from the lens design. Both lenses are to be constrained by a single threaded retaining ring acting through a pressure ring.

The first step is to measure the actual lenses to be assembled. Lens No. 2 is nominally ∿0.5mm larger than Lens No. 1. Figure 16 indicates the five data items to be recorded. After these measurements are made, the lens surfaces might well be protected temporarily with peelable lacquer except for areas to be mounted or measured subsequently.

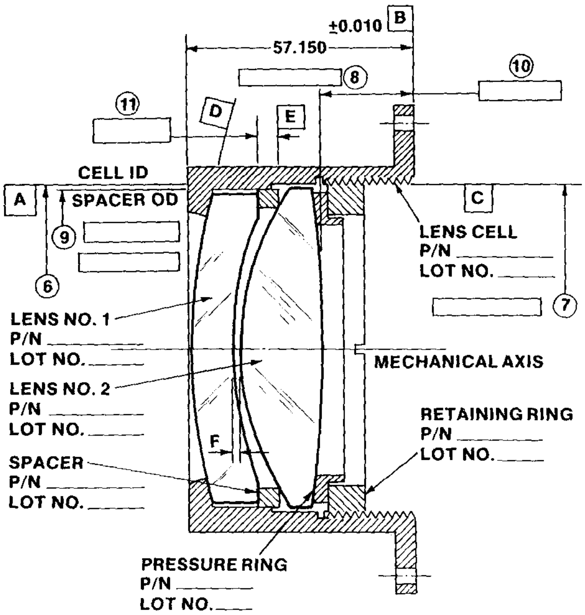


Figure 15. Illustration of opto-mechanical assembly by lathe assembly process.

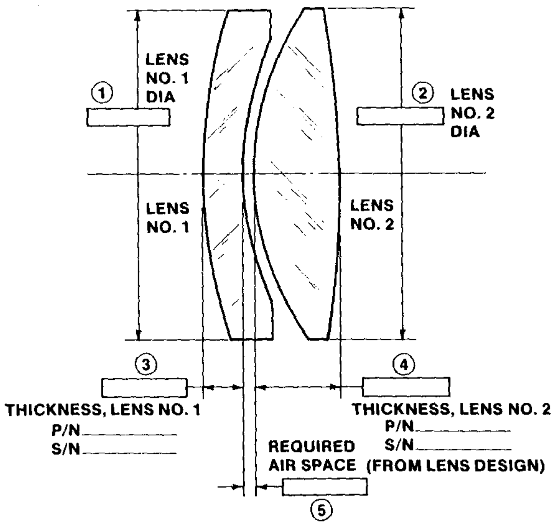


Figure 16. Data to be recorded for lathe assembly.

The lens cell flange is mounted via an adapter plate (not shown in the figure) to a precision lathe spindle. The cell ID surface A is machined perpendicular within 15 arc-minutes to surface B and to obtain 7 to 12 μm diametrical clearance over the actual OD of Lens No. 1. The cell ID is measured and recorded as dimension 6. Cell ID surface C is then machined concentric within 8 μm to surface A, and to obtain 7 to 12 μm diametrical clearance over the actual OD of Lens No. 2. This ID is measured and recorded as dimension 7.

The seat surface D is then machined at an angle of 75.0 ± 0.5 degrees from the mechanical axis. The axial location of surface D is determined by iterating the facing operation with trial installations of the lens and measurement of the convex vertex location relative to the required axial location from the flange. When within tolerance, the actual value of dimension 8 is recorded.

The spacer is separately machined so its interface with Lens No. 2 is "square", its OD is perpendicular within 15 arc-minutes to its surface E, and to obtain 7 to 12 μm diametrical clearance with respect to the ID of the cell (dimension 6). This OD is measured and recorded as dimension 9. The spacer is inserted into the assembly so as to contact Lens No. 1 and Lens No. 2 inserted. The axial location of the exposed vertex of Lens No. 2 relative to surface B is measured and recorded as dimension 10. The required thickness reduction of the spacer is computed as (dim 8 - dim 3 - dim 4 - dim 5 - dim 10). A little less than this amount of material is removed from the spacer and the calculation repeated using a new measured value for dimension 10. When the error in thickness falls within the ± 15 μm tolerance (from the lens design) the spacer dimension is recorded as dimension 11.

The lenses and metal parts are cleaned and the pressure ring and retaining ring installed to complete the assembly.

Acknowledgements

Many of the designs shown here were previously described by Jacobs (1943), Horne (1972), Richey (1974), Hopkins (1980), Kowalskie (1980), and Bayar (1981). The author acknowledges their contributions to our understanding of opto-mechanical design.

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