Rigid body movements of optical elements due to opto-mechanical factors

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ABSTRACT

Techniques and formulas will be presented that demonstrate an effective means of characterizing the rigid body motions of optical elements from their nominal positions as caused by manufacturing tolerances and thermal effects. These techniques allow accurate prediction of the final position of a mechanically held lens element to be determined relative to mechanical datums. Even a single lens element with entirely nominal dimensions often needs to be positioned relative to a mechanical reference; the effects of any inherent inaccuracy of the mounting process can be over-looked and/or over-simplified. Tolerances on lens seats, element radii, bore diameters as well as thermal effects need to be accounted for in a design in order to accurately predict the final optical performance of a system in an "as built" condition. The differences in accounting for the mounting tolerances of edge mounted, cell mounted, and surface-centered elements are discussed. The work presented will aid in linking the tools available to the optical engineer in the form of optical design software, with the data available to the mechanical engineer in the form of manufacturing and fabrication tolerances.

Keywords: opto-mechanical, tolerance analysis, positioning, fabrication tolerances, optical mounting

1. INTRODUCTION

Modern lens design programs provide powerful tools for the design and optimization of multi-element lenses or optical systems, but a thorough understanding of how each element will interact with the mounting mechanics is required to fully account for all of the tolerances in the design. In the case of multi-element lenses, the process of turning the lens design into fabrication prints is often the most difficult part.

While there are many "rules of thumb" available and assumptions that designers use when tolerancing a lens, this paper provides both explanation of the various types of lens mounts used, as well as techniques and guidance to determine how the mechanical and optical fabrication tolerances can be fully accounted for within the optical design software.

Though of a slightly different nature, thermal changes within a lens system cause the overall form of the optical system to change. Like tolerancing errors, the mechanical structure that holds the lenses must be understood to accurately account for the shifts and shape changes, as temperature changes within the system. This paper will provide explanation of how to accurately account for these changes and demonstrate a case in which the assumptions built into most optical design programs will yield very different results than reality, unless special measures are taken.

2. BACKGROUND

For the purposes of this paper, element mounting methods have been classified as one of three types: Edge-mounted, Cell-mounted and Surface-centered. These three mounting methods represent a large majority of the opto-mechanical designs done to date and serve to cover most of the commonly implemented techniques.

Edge-mounted refers to a design where the lens rests on a lens seat and is centered by the contact of the edge of the element to the inside of a bore formed by the metals, as shown in Figure 1. This technique has also been called "drop-in assembly"[1], referring to the fact that no special care is taken to align the elements within the mount. This is a simple method to implement, and the rigid body movements that result from the tolerances can be easily seen. While the benefit of this technique is the simplicity, the detriment is often the tight tolerances necessary to achieve high performance.

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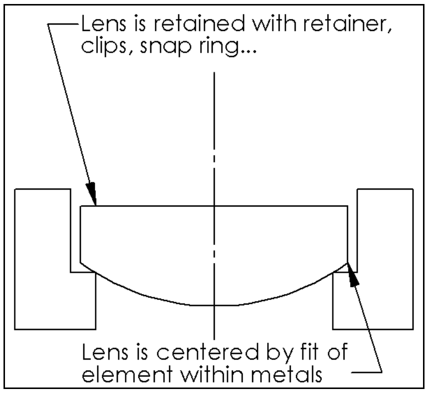


Figure 1: Edge-mounted

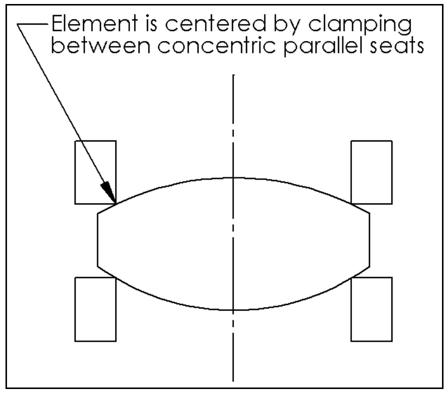


Figure 2: Surface-centered

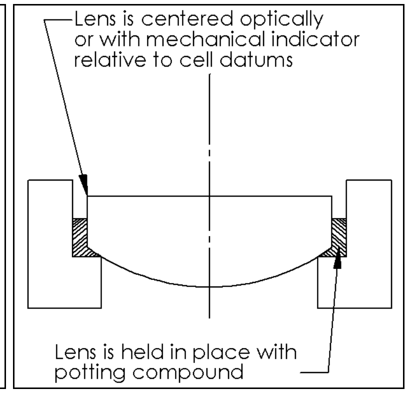


Figure 3: Cell-mounted

Surface-centered elements are sometimes referred to as "bell-chucked" or "bell-clamped". In this method an element is held within the optical system by the optical surfaces alone, as shown in Figure 2. While seemingly simple, this is a hard method to implement, and a hard technique to accurately predict the performance of within the optical system.

Cell mounted designs are often implemented to overcome the very tight tolerances that would be required of an edge-mounted design in a high performance system. Cell-mounted designs are a class of opto-mechanical designs where the element is aligned within a sub-cell, this has also been referred to as "poker-chip assembly"[1]. For the purposes of this paper we will assume that the element is bonded into the cell, and that the element and cell become a subassembly within the lens, as shown in Figure 3.

3. ACCOUNTING FOR TOLERANCES IN THE LENS DESIGN

Optical design programs offer many methods of accounting for the tolerances that control the final position of the lens elements. These rigid body movements can be broken down into a combination of axial shifts and lateral shifts (decenter) of both elements as a whole, and as single surfaces. Axial shifts are straightforward to account for and can be determined through accurate accounting of spacer, element, and seat tolerances. The remaining perturbations, and the drivers that cause them, are shown in Table 1. Section 4 covers the simulation of the physical results in more detail.

	Driver	**Physical Result**
Edge-centered	Seat Tilt	Element Tilt
	Radial Gap (Element to Bore)	Element Tilt and Decenter
	Element Wedge	Tilt of Free Surface
	Seat Decenter	Element Decenter
Surface-centered	Seat Decenter	Element Tilt and Decenter
	Seat Tilt	Element Tilt and Decenter
	Cupping Angles	Element Wedge
Cell-mounted	Radial Gap (Cell to Bore)	Element Decenter
	Tilt of Cell Subassembly	Element Tilt
	Concentricity of Seat	Element Tilt and Decenter
	Parallelism of Seat	Element Tilt and Decenter
	Element Roll	Element Tilt and Decenter

Table 1: Element perturbations and drivers

4. MOTIONS DUE TO FABRICATION TOLERANCES

4.1 Edge-centered

For an edge-centered element the determination of decenter, tilt and wedge is relatively straightforward. Figure 4 shows an element on a lens seat that is allowed to "roll" until the edge touches the ID of the metals. Some optical design programs offer the ability to tolerance directly on "roll" by identifying the radial distance over which the element may roll. Given that this option is not always available it is useful to break the "roll" of a lens into a tilt and decenter of the surface that is not resting on the lens seat.

It is important to note that in many cases, particular rigid body movements of an entire optical element are often easiest to model as single surface movement instead. Given that the edges of the element are not of optical importance, they can be ignored when predicting performance of the system. In the case shown in Figure 4, the element has been allowed to "roll" on the surface against the lens seat. The "roll" can be modeled in two equivalent ways. The entire element could either be rotated about the center of the radius that is against the seat, or just the top surface could be rotated about the same center. Given that rotating the lower surface about it's center of curvature does not result in an optical change, only the motion of the top surface is of consequence. The optical designer may either choose to take the necessary steps to rotate the free surface about the center of the mounting surface, or use a realistic simplification that approximates this movement.

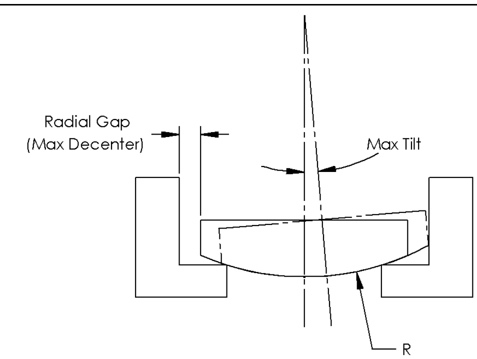
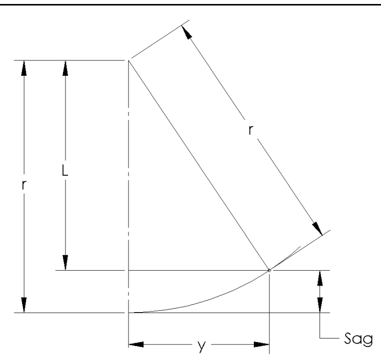


Figure 4: Roll of an edge centered element

Figures 5 and 6 show how to simulate the roll as a combination of a tilt and a decenter of the upper surface. Figure 5 shows how the sag of a surface at a given distance from the centerline is calculated. Figure 6 then shows how we can use the sag equation to determine the tilt and decenter of the top surface that results from rolling the element of Figure 4 on the lower surface until the edge contacts the inner diameter of the mount. The shift (decenter) of the top surface at the limit of the "roll" is given by equation 5 where CT equals center thickness of the lens element.

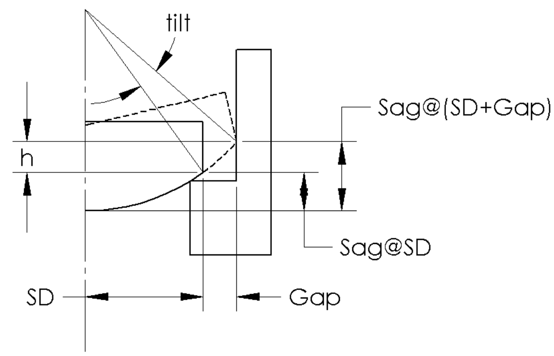


$$y^2 + L^2 = r^2 \qquad \text{Eq. 1}$$

$$L = \sqrt{r^2 - y^2} \qquad \text{Eq. 2}$$

$$Sag = r - \sqrt{r^2 - y^2} \qquad \text{Eq. 3}$$

Figure 5: Sag determination



$$tilt = \arctan\left(\frac{\sqrt{gap^2 - h^2}}{r}\right) \qquad \text{Eq. 4}$$

$$shift = (r - CT)\cdot \sin(tilt) \qquad \text{Eq. 5}$$

Figure 6: Accounting for "roll" of a edge mounted element

4.2 Surface-centered

Surface-centered refers to a class of mechanical mounts that utilize only the optical surfaces of the lens element to hold the elements orientation and position. Ideally, when an element is mounted in this method, it will self-align, such that the centers of curvature for the two surfaces lie on a line that is normal and concentric to the lens seats. This method can work well with lenses that have strong and opposed curvatures; ideally the lens would have almost a spherical shape. Lens elements that have surfaces that are nearly concentric do not fair well with this technique.

4.2.1 Judging if an element will self-center

The following equation[2] can be used to judge if a lens element will self-center by determining if the following inequality is satisfied:

$$\left({}^{y_C}\!/_{2\cdot R}\right)_1 - \left({}^{y_C}\!/_{2\cdot R}\right)_2 > 0.07 \qquad \text{Eq. 6}$$

where y_c is the semi-diameter of the associated lens seat and R is the radius of the surface on that lens seat. While this may serve as a guideline, it assumes that if the condition of this formula is fulfilled, then the element will perfectly center. In reality, a lens element will center only as long as there is sufficient force to overcome the friction between the glass surfaces and the lens seat. Clearly geometry is the largest contributor to this, in that the shape of the surfaces will directly affect the leverage on the element, but items such as metal coatings, surface finish and materials all play a roll. In order to try to quantify the residual wedge present, when an item is bell clamped, we can look to lens fabricators. Fabricators use the following formula to determine what ETD (Edge Thickness Difference) is achievable given the "cupping angles" of a lens element.

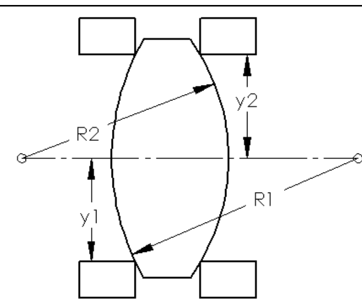


Figure 7: Cupping angles

$$\beta = \arcsin({}^{y_1}\!/_{R_1}) - \arcsin({}^{y_2}\!/_{R_2}) \qquad \text{Eq. 7}$$

β =Difference between cupping angles (see Figure 7)

$$\beta > 20^\circ \rightarrow \text{ETD} < 0.005\text{mm}$$

$$11^\circ < \beta < 20^\circ \rightarrow 0.020\text{mm} > \text{ETD} > 0.005\text{mm}$$

$$\beta < 11^\circ \rightarrow \text{Not Recommended}$$

The achievable ETD values are given as an example but are variable given the materials and geometries that will be used. Accurate values should be arrived at experimentally for a given set of conditions.

4.2.2 Effect of a decentered seat on a surface-centered element

If we assume that an element has suitable geometry to self-center we must then look at the tolerances of the lens seats that the element will be bell-clamped with. The tolerances on seat decenter and seat tilt are especially significant. If we regard one seat to be perfect, or a datum, and the other seat is decentered, then we see that the center of curvature of the datum seat remains unchanged, while the center of curvature of the surface touching the decentered seat is decentered by exactly the same amount. This results in a tilt and decenter of the element, but like the case of the edge centered element we need only consider one surface. In this case the tilted and decentered element can be simulated in the optical design software by simply applying the seat decenter to the element surface that is in contact with the displaced seat, as shown in Figure 8. The decenter of this single surface is nearly equivalent to the rigid body motion that would have resulted, as shown in Figure 9.

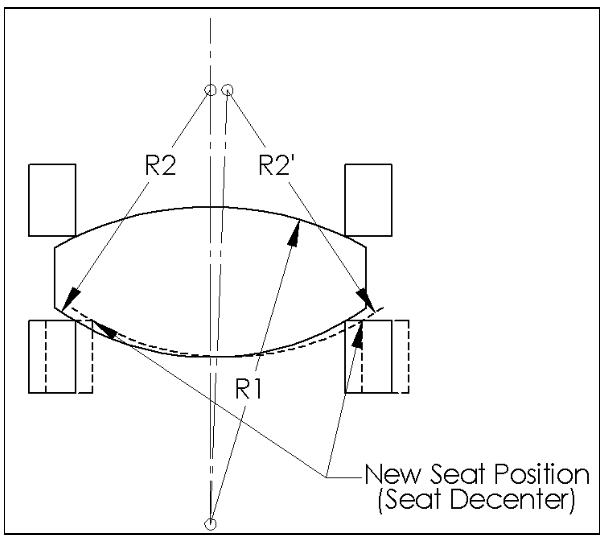


Figure 8:Effect of seat decenter

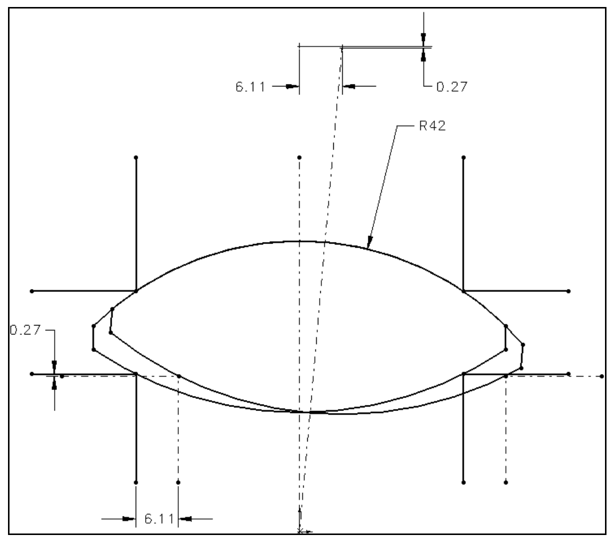


Figure 9: CAD model of surface-centered element

Figure 9 also shows that there is a secondary motion along the axis of the system that is likely insignificant and in this case equals about $1/20^{th}$ of the horizontal motion. Therefore, the actual motion of the surface as the element "rolls" to accommodate the shifted seat is a rotation about the center of curvature, of the upper surface, but can be approximated by a shift. If the radii are particularly short, or spacings are particularly critical, then this approximation should be re-visited.

4.2.3 Effect of a tilted seat on a surface-centered element

A tilted seat can be handled in much the same manner. Figure 10 shows an element located between an ideal seat and a seat that has been tilted by 5º. Because the center of curvature of the lower surface is following an arc, as the lens "rolls" about the top surface, there will again be a slight, and usually negligible, amount of axial shift.

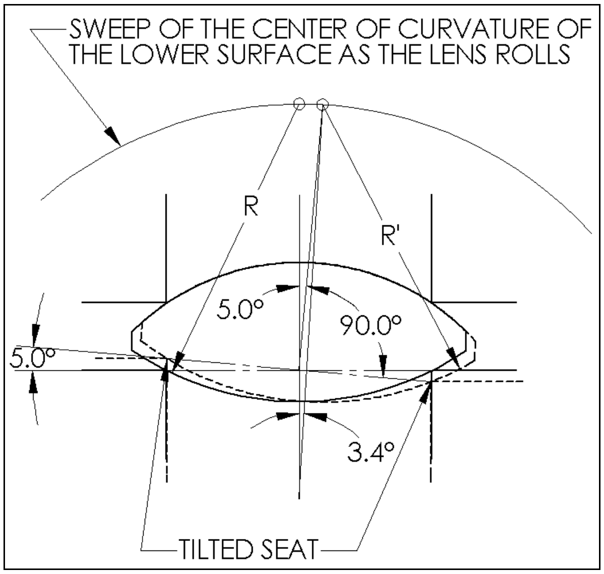


Figure 10: Shift of a surface centered lens

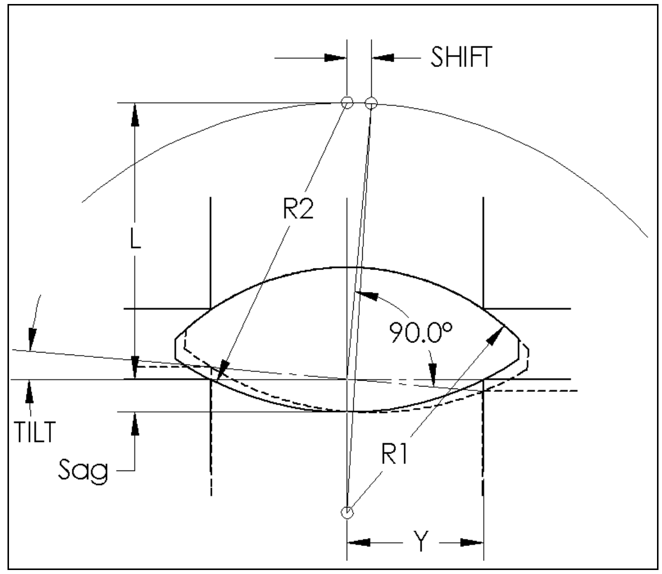


Figure 11: Calculation of shift

$$L = \sqrt{R^2 - Y^2} \quad \text{(Eq. 2)}$$

$$Shift = L \cdot \sin(TILT) \quad \text{Eq. 8}$$

While a tilted seat will again cause a tilt and decenter of the element as a whole, it can more simply be modeled as a single surface decenter applied to the surface in contact with the seat that is tilted. However, determining an appropriate amount of surface decenter is not as straightforward, because the motion of the seat is no longer equal to that of the element surface. Figure 11 and equations 2 and 8 show how to calculate the appropriate lateral shift (decenter) to apply to the lower element surface, to simulate the element tilt and decenter caused by the seat tilt.

4.3 Cell-mounted

Cell-mounted designs are most often used when high positioning tolerances (centrations and tilts) are required. Cell mounted designs offer the advantage of an intermediate alignment step to relieve some of the fabrication tolerances of the element. In general, an element is placed within a precision metal cell. Then the exposed surface of the optical element is aligned relative to the OD and one side of the cell. There are various techniques used to align the top surface and they range in sophistication. These techniques can be as simple as rotating the cell against a “V”, with a mechanical indicator on the top surface, to cells aligned on air-bearing spindles equipped with laser based centration measurement. The benefit of this technique is that the edge diameter and ETD, or element wedge, do not need to be accurately held. The lens seat controls the centration of the lower surface and the element is allowed to “roll” on that lower surface until the center of curvature of the top surface lies coincident with the axis of the cell (see Figure 13). The “potting tolerance” is the tolerance that controls how well the top surface will be located within the cell. It is often measured as a run-out of the top surface, measured normal to the surface and at the clear aperture diameter.

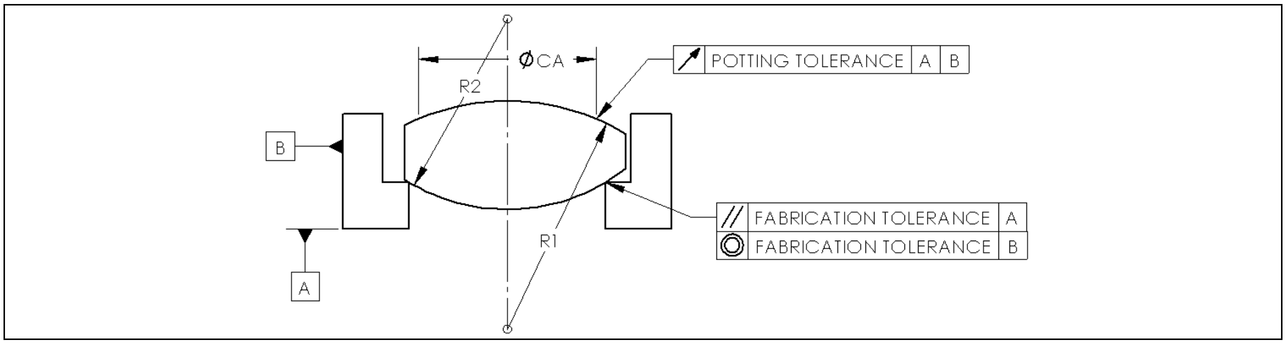


Figure 13: Element with extreme wedge, centered in cell

The effect of a tilted seat or a decentered seat is the same as it would be in a surface-centered element. If you consider that the top surface will be centered or aligned to the cell datums such that the center of curvature of the top surface lies on the cell axis, then any tilt or decenter of the lens seat will cause a combination tilt and decenter of the lens element. The equations presented for determining the motion of a surface-centered element can be used for determining the rigid body movements of a cell-mounted element as well.

In addition to these effects, one must consider how well any one cell within a multi-element lens is located relative to the other cells. Variables that need to be accounted for include cell parallelism, cell run-out, radial clearance within lens barrel, spacer parallelism, and others. In the case of the fit of a cell within a lens barrel, the tolerance would be modeled as element decenter. Tolerances on spacer parallelism and adjacent cell parallelism could cause tilts of the entire lens cell; these tolerances would be modeled as element (or group) tilts.

5. TOLERANCING METHODS

Tolerance models of complex systems are often only partially developed. Instead, many designers use intuition and rules of thumb. This often works because the tolerances are overly tight, causing the assemblies to be more expensive and more difficult to assemble than necessary. On the other hand, it is possible for the necessary precision of an assembly to be unrealized. Therefore the individual tolerances, of the mechanical parts, may not be consistent with the accuracy required.

Worst case tolerancing is often used for simpler systems. It has the advantage of being easy to use and understand, while guaranteeing that if the parts are built to specification, the assembly will work. The disadvantage of this method is that it tends to yield overly tight tolerances, especially as the assemblies become more complex. If one is to look at lens centration in a cell mounted design, the lens centration is a function of the nominal radial gap between cell and barrel, the tolerance of the OD of the cell, and the tolerance of the ID of the barrel. The centration of one lens relative to another is then dependent on five different inputs, assuming a common barrel ID. If worst case is assumed, the tolerances will be much tighter than need be, because the tolerances apply for a case that is statistically improbable.

To avoid overly tight tolerances, statistical tolerancing methods are often used. Statistical tolerancing methods assume a normal distribution of dimensions within a tolerance band. While this means that is it possible to put together multiple parts that are all within specification and still end up with a non-performing assembly, though this is improbable if there is a sufficient number of variables. If y represents the output tolerance for an assembly and x represents the individual contributions of the parts within an assembly, the statistical tolerance can be calculated by the following equation:

$$\Delta_y = \sqrt{\Delta_{x_1}^2 + \Delta_{x_2}^2 + \ldots + \Delta_{x_n}^2} \qquad \text{Eq. 9}$$

In Equation 9, Δ_x would equal $6\sigma_x$ by the standards of the six sigma approach[3]. This approach has been used successfully in the tolerancing of multi-element lens systems and is an appropriate alternative to worst-case tolerancing methods.

6. ACCOUNTING FOR MOVEMENTS DUE TO TEMPERATURE CHANGE

When thermal requirements are included with a lens specification, the analysis concentrates on absolute and deterministic movements of lens elements to a new nominal value at a given temperature. The new values can be used to predict the optical performance of the system at the new operating temperature. [NB a complete thermal analysis should also include the variation of index of refraction with temperature and pressure. The index changes can be more significant drivers of optical performance than the mechanical changes. However, since the index variation does not cause any physical movements, such discussion is outside the scope of this paper]

The heart of the mechanical analysis rests with the approximation (accurate for most common materials at terrestrial temperature ranges) for three of the four lens parameters that have units of length assigned: radius of curvature, center thickness, and diameter. These parameters scale linearly with the Coefficient of Thermal Expansion (CTE) of the optical material. However, we will see that lens sag distances and lens-to-lens air space separations do not vary linearly when CTE's of dissimilar materials are not equal. When taking opto-mechanical design factors into consideration, lens sag distances and lens-to-lens air space separations are usually a function of two or more CTEs (glass and metal). In addition, different lens-to-lens air space separations are obtained depending on the mechanical design approach for how the lens elements are mounted and constrained. The formulas governing these lens movements are unique to the design approach and should be considered for accurate thermal modeling of optical performance.

The reader is cautioned that the thermal analysis features incorporated into commercial lens design software are typically a subset of the formulas given for the expanded axial air distances.[4,5] Accurate optical performance modeling may be accomplished by manually implementing these formulas for the specific mechanical design cases through the use of macros or other computational methods when using lens design software.

The scaling equation governing dimensional change with temperature can be written as:

$$L' = L(1 + \alpha\Delta T), \qquad \text{Eq. 10}$$

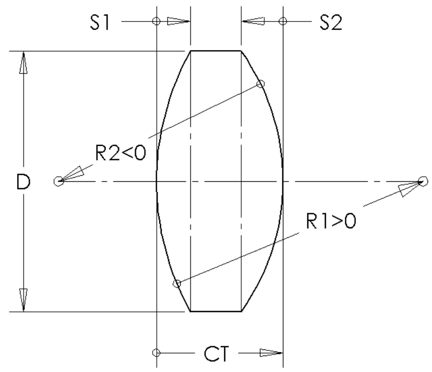
where *L'* is the new value of a parameter with unit of length after the temperature has changed and the material has reached a steady-state condition; *L* is the nominal value of the parameter; α is the CTE of the material; and ΔT is the difference in temperature from nominal to the new operating temperature.

Shown below is a sampling of representative CTE's for selected materials commonly found in opto-mechanical lens systems, valid for temperatures from 0 °C to 80 °C:

optical materials	CTE α_g /°C	mechanical materials	CTE α_m /°C
Zerodur	0.05 *10E-6	Invar	0.5 - 1.5 *10E-6
Fused Silica	0.5 *10E-6	Stainless Steel 300 series	10 *10E-6
BK7	7.1 *10E-6	Stainless Steel 400 series	16 - 18 *10E-6
all optical glass types	3.7 - 14.6 *10E-6	Brass	20 - 21 *10E-6
PMMA, Polycarbonate	60 - 70 *10E-6	Aluminum	23 - 25 *10E-6
NOA61	225 *10E-6		

Table 2: CTE's of common materials

The discussion will apply to all four possible lens shapes: convex/convex (cx/cx), convex/concave (cx/cc), concave/convex (cc/cx), concave/concave (cc/cc). For clarity, Figures 1-3 depict cx/cx lens singlets. For the other three possible lens shapes, the +/- sign before the square root in the sag equation must be chosen appropriately. We start with the singlet made of material with CTE α_g and floating in air:



Sign convention used in optical design for radius of curvature:

$R > 0$ if center of curvature lies to the right of the surface (as for R_1)

$R < 0$ if center of curvature lies to the left of the surface (as for R_2)

The sign of a dimension is indicated by the direction of the arrow in all figures:
pointing right = positive
pointing left = negative

Figure 14: Singlet floating in air with dimensional sign convention

The sag equation yields

$$S_1 = R_1 - \sqrt{R_1^{\,2} - \left(\frac{D}{2}\right)^2} \quad > 0 \ \text{ for } R_1 > 0 \qquad \text{Eq. 11}$$

$$S_2 = R_2 + \sqrt{R_2^{\,2} - \left(\frac{D}{2}\right)^2} \quad < 0 \ \text{ for } R_2 < 0. \qquad \text{Eq. 12}$$

After a thermal soak:

$$CT' = CT\left(1 + \alpha_g \Delta T\right) \qquad \text{Eq. 13}$$

$$D' = D\left(1 + \alpha_g \Delta T\right) \qquad \text{Eq. 14}$$

$$R_1' = R_1\left(1 + \alpha_g \Delta T\right) > 0 \qquad \text{Eq. 15}$$

$$R_2' = R_2\left(1 + \alpha_g \Delta T\right) < 0 \qquad \text{Eq. 16}$$

$$S_1' = R'_1 - \sqrt{R'^{\,2}_1 - \left(\frac{D'}{2}\right)^2} = S_1\left(1 + \alpha_g \Delta T\right) \qquad \text{Eq. 17}$$

$$S_2' = R'_2 + \sqrt{R'^{\,2}_2 - \left(\frac{D'}{2}\right)^2} = S_2\left(1 + \alpha_g \Delta T\right) \qquad \text{Eq. 18}$$

These formulas are simple and straightforward to understand. They describe the changes to the mechanical dimensions for a singlet floating completely in air. In reality, a singlet must be held in space by some mechanical means. In this case, the equality in Equations 17 and 18 break down if the CTE of the optical and the mounting materials are dissimilar.

Now consider the singlet with CTE α_g mounted in a metal cell with CTE α_m as depicted in Figures 1, 2, or 3. The element is loaded against the seat by a retainer, an adhesive, or some other force. The seat is the datum to which the thermal expansion will be relative. After a thermal soak, the metal cell will have expanded in diameter and thickness a different amount from the glass lens element. The constraint is that the seat remains in contact with the lens surface, but is free to slide along the surface. In this case the sag will be a function of both α_g and α_m, and its expansion no longer scales linearly:

$$S_1'\left(\alpha_g, \alpha_m\right) = R'_1\left(\alpha_g\right) - \sqrt{R'^{\,2}_1\left(\alpha_g\right) - \left(\frac{D'\left(\alpha_m\right)}{2}\right)^2} \neq S_1\left(1 + \alpha_g \Delta T\right) \qquad \text{Eq. 19}$$

$$S_2'\left(\alpha_g, \alpha_m\right) = R'_2\left(\alpha_g\right) + \sqrt{R'^{\,2}_2\left(\alpha_g\right) - \left(\frac{D'\left(\alpha_m\right)}{2}\right)^2} \neq S_2\left(1 + \alpha_g \Delta T\right) \qquad \text{Eq. 20}$$

To calculate the axial shift of the element vertices, we depict the four possible cases of two lens elements (lens A with CTE α_A and lens B with CTE α_B), separated by a vertex-to-vertex distance, *t*, mounted on two independent seats (see Figure 15).

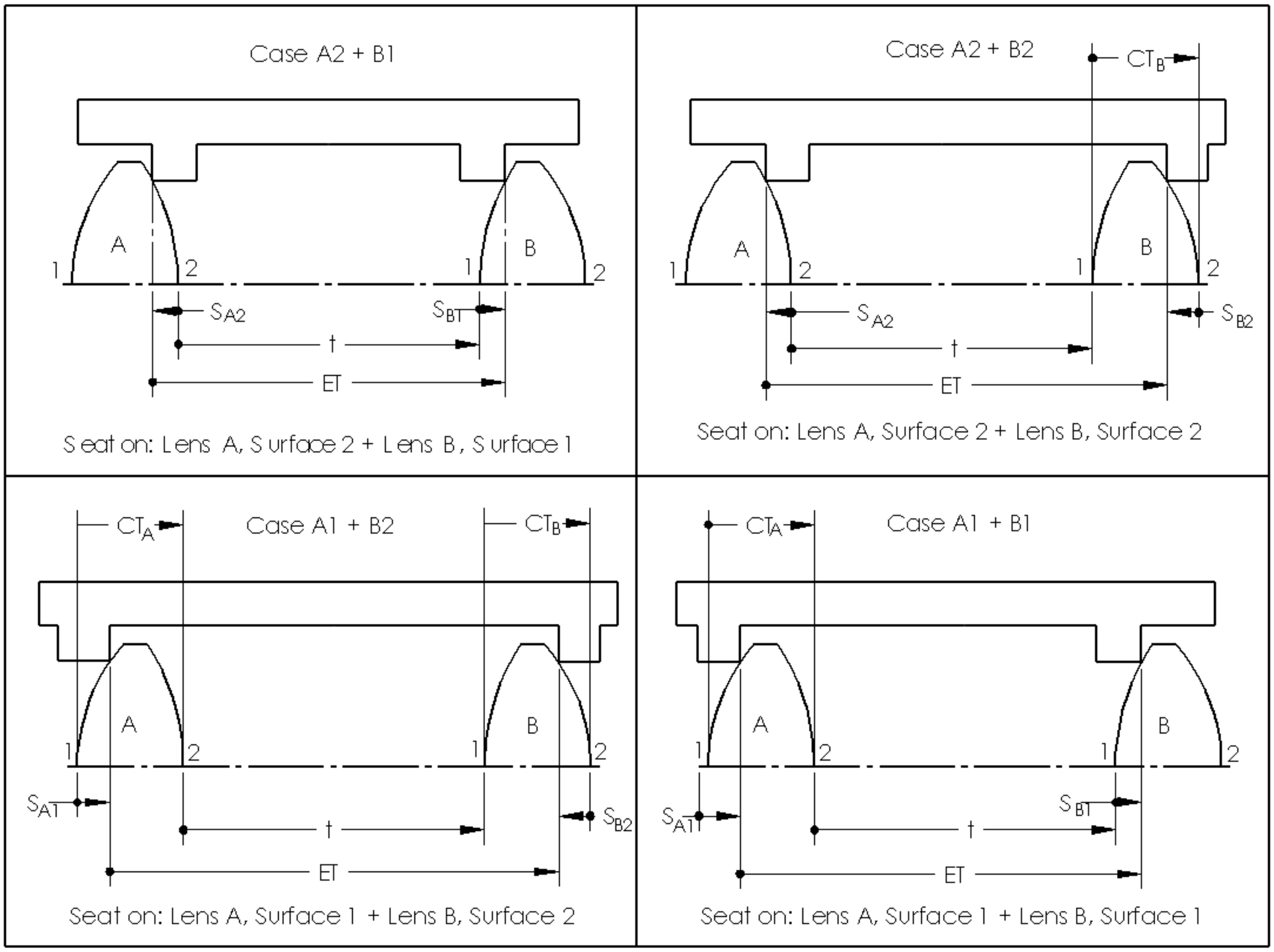


Figure 15: Possible seat positions and dimensions

Which side of lens A and B the two seats are chosen to be on, is usually left to the discretion of the mechanical designer, who takes into consideration a multitude of performance factors and general design criteria. The thermal performance may vary widely for each case: each one requires its own computation for *t'*. Case A2 + B1 (alternately referred to as the "stacked lenses," "stacked spacers," or "rubber tube" model) is the default mounting assumption built into the computation of *t'* in the major lens design software packages. Seldom does this model apply to all elements of a multiple element lens system.

To compute the correct formula for *t'*, the following algorithm is applied:

1. compute seat-to-seat distance, *ET*, as function of *t* and sags
2. express vertex-to-vertex distance, *t*, as function of *ET*
3. apply thermal expansion individually to each term in expression for *t*

Case A2 + B1:

$$ET = -S_{A2} + t + S_{B1} \quad \text{Eq. 21}$$

$$t = ET + S_{A2} - S_{B1}$$

$t' = ET' + S'_{A2} - S'_{B1}$, where $ET' = ET(1 + \alpha_m \Delta T)$ and S'_{A2} from equation 19 and S'_{B1} from equation 20

Case A2 + B2:

$$ET = -S_{A2} + t + CT_B + S_{B2} \quad \text{Eq. 22}$$

Case A1 + B2:

$$ET = -S_{A1} + CT_A + t + CT_B + S_{B2} \quad \text{Eq. 23}$$

Case A1 + B1:

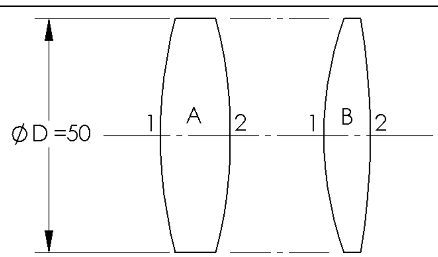
$$ET = -S_{A1} + CT_A + t + S_{B1} \quad \text{Eq. 24}$$

The equations for *ET* hold for all four lens types as long as the sign convention is followed.

Case	Seat on	*t'*	*ET'*
1	A2 + B1	$t'_{21} = ET'_{21} + (S'_{A2}) - (S'_{B1})$	$ET'_{21} = [(-S_{A2}) + t + (S_{B1})] \cdot [1 + \alpha_m \Delta T]$
2	A2 + B2	$t'_{22} = ET'_{22} + (S'_{A2}) - (CT'_B + S'_{B2})$	$ET'_{22} = [(-S_{A2}) + t + (CT_B + S_{B2})] \cdot [1 + \alpha_m \Delta T]$
3	A1 + B2	$t'_{12} = ET'_{12} + (S'_{A1} - CT'_A) - (CT'_B + S'_{B2})$	$ET'_{12} = [(-S_{A1} + CT_A) + t + (CT_B + S_{B2})] \cdot [1 + \alpha_m \Delta T]$
4	A1 + B1	$t'_{11} = ET'_{11} + (S'_{A1} - CT'_A) - (S'_{B1})$	$ET'_{11} = [(-S_{A1} + CT_A) + t + (S_{B1})] \cdot [1 + \alpha_m \Delta T]$

Table 3: Summary matrix

A two element lens has been defined in Figure 16 that will serve as an example of how the results differ between the four cases that have been presented. Table 4 lists the values of t' for the four cases, assuming a 40 degree rise in temperature.



Lens	Surface (mm)	Radius (mm)	Thickness (mm)	CTE (ppm/C)
A	1	100	15	7.4
	2	-100		
Airspace			20	23.6
B	1	75	10	7.4
	2	-150		

Figure 16: Example multi-element lens

Case	t (mm)	t' (mm)	t' - t (um)
A2 + B1	20	20.014	13.803
A2 + B2	20	20.025	24.613
A1 + B2	20	20.039	38.586
A1 + B1	20	20.028	27.776

Table 4: Results of a rise in temperature of 40º

Table 4 shows us that the expected growth of the air space, t'-t, is between 13.8 and 38.6 um. Case A2 + B1 is the default case that would be simulated by lens design programs and it predicts the smallest growth. In this example, Case A1 + B2 predicts ≈2.8 times the value of Case A2 + B1. While the difference between the two is only 24.8 um, it is reasonable to assume that if the designer had reason to be concerned about the effects of the temperature change, then the differences between the cases may very well be significant enough to account for, as accurately as possible. It is important to note that the separation between the cases grows with increasing CT's of the elements as well as an increase in the mismatch of CTE's.

7. SUMMARY

Many lens designs require precise positioning of optical lens elements within the optical system in order to achieve the desired performance. Lens design programs offer the means to determine the nominal position for each element and also provide the tools necessary to determine the performance when the fabrication tolerances are taken into account. This thorough tolerancing of a lens design can only happen after a mechanical concept or layout has been established. Once this mechanical concept is available it becomes possible to determine all of the possible rigid body movements of the lens elements within the system. This paper has described these movements in detail and provided the equations necessary to calculate the resultant movements of lens elements that are located by imperfect mechanical features. This paper has also outlined the three major categories of lens mounts and how they differ from each other. Finally, the effects of a temperature change on a multi-element lens were discussed and the four cases of two elements separated by a homogeneous mount were presented.

REFERENCES

1. Yoder, Paul, *Mounting Lenses in Optical Instruments*, SPIE Tutorial Text Vol. TT21, Chapter 4, SPIE Press Bellingham, 1995
2. Yoder, Paul, "Principles for Mounting Optical Components", *SPIE Short Course*, San Diego, CA, July 2001.
3. Taylor, Wayne, "Process Tolerancing: A Solution to the Dilemma of Worst-Case Versus Statistical Tolerancing", Oct. 1995.
4. CodeV Reference Manual V9.5, Vol. III, Optical Research Associates, Aug 2004.
5. Zemax Reference Manual, p. 506, Zemax Development Corporation, June 15, 2005.