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Design Considerations for Mirrors with Large Diameter to Thickness Ratios

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ABSTRACT

As optics increase in size, so increases the difficulty in achieving optical performance goals. Such challenges, however, can be accomplished by careful consideration of design, mounting, and material characteristics to lead to a selected concept of large ground or space based optics with high aspect ratios.

This paper outlines some of the large mirror design methodology and understanding required to meet specified performance criteria for both monolithic lightweight and thin solid meniscus design approaches. These criteria are many. The optic must be shown capable of meeting performance goals during a changing gravity environment, for ground based telescopes, often looking from horizon to zenith, or be tested for the zero-g environment of space. It must further be insensitive to, or provide accommodation for, thermal swings about the nominal fabrication temperature. At temperature, it must preclude excessive error causes by real time thermal gradients. It must resist vibratory and wind loadings as well, and be of adequate strength to withstand all phases of handling and transportation in various stages of completion.

The mirror must be generated, cut, ground, polished, tested, mounted, and coated. During these stages, the effects of residual stress, temperature, assembly tolerances, mount force errors, tool pressure, bimetallic distortion, surface flaws, and humidity must be duly considered to meet the stringent criteria of fractional visible wavelength performance and fractional arc second encircled energy requirements.

The body of this paper presents an overview of the detailed analyses required to meet such criteria. Discussed are methods of supporting, mounting, and controlling such optics. The analytical tools of finite element math modeling are presented which assist in determining effects of material inhomogeneities, thermal strain, initial stress conditions, lightweighting, and mirror curvature, which play an important role as the diameter to thickness ratio increases. Material phenomenon and comparative trades are developed to aid in the choice of ideal candidates. Design examples from experience are finally given to illustrate the sensitivity of proven optics to the design criteria which have met stringent performance goals. These include lightweight monolithic optics with diameter to thickness ratios in excess of 15:1 and thin meniscus designs with ratios in excess of 100:1. In the latter case, active control is demanded, and correctability to both fabrication and operational errors is discussed, including actuator count determination.

1.0 INTRODUCTION

The laws of physics which govern the design of mirrors with high aspect (diameter to thickness) ıatios are, or course, the same as for those of low values. The difference, however, lies in the magnitude of distortion induced in the high aspect ratio designs by the operational, fabricating, or test environments. Depending on the particular environmental conditions, performance errors can be orders of magnitude higher for these optics than those of the more common and lower ratios. What is "in the noise" or unmeasurable for that latter class suddenly becomes a design driver.

It is in this vain that the subject paper is presented. Design, material, and mounting considerations for large diameter to thickness ratio optics will be discussed from an analytical standpoint to point out difficulties which are often unseen in conventional optics. The presentation to follow however, is not meant to be an esoteric theoretical discussion involving advanced mathematics. Rather, a critical review, or overview, of some of the issues which must be addressed is made, as outlined in the abstract.

2.0 DEFINITION

A definition of large diameter and large diameter to thickness ratio is relative. Large diameter to some may be small to others. In turn, large diameter to thickness ratio is a subjective, relative quantity. In this latter regard, it is noted that aspect ratios of less than 6:1 are generally considered as conventional. For the purpose of this paper, aspect ratio in excess of 15:1 will be considered large. Indeed, designs will be discussed with ratios in excess of 200:1. It is noted further that environmental effects on optical distortion usually scale as the power of diameter in direct proportion. To this end, performance of mirrors with high aspect ratio will be discussed for mirrors with diameters in excess of one meter. With this arbitrary definition, issues can be address for a certain class of optics where not only analysis has been made but also facts have been revealed through interpretation verified by the data of actual test results and performance measurements.

3.0 BACKGROUND

For the class of optics being presented, two questions need to be addressed. The first is: how will the mirror meet performance goals in its operational environment? And the second is: how will you make it in the first place?

The first question concerns itself with the site, whether ground, aircraft, or space based. In any event common to those environments are gravity (or the lack of it) and temperature. It is those two environments to which questions must be largely addressed. In the former, analysis and design must be accomplished to maintain performance for varying angular orientations ground or in the air, from zenith or nadir to horizon. In space, the dilemma of performing a meaningful ground test must be assessed. In the latter, temperature soak changes even of a few degrees only, must be considered, as well as the potential for variations and gradients about the operating temperature as may be induced by sun, wind, or on station heat sources.

The second question concerns itself with whether one can get there in the first place. Can an optic be so large in aspect ratio that one cannot fabricate it to the required precision? Can an optic be so large in aspect ratio that you cannot test it? These questions address the concerns that, although critical, are less so in the more massive designs of low aspect ratio. Among those concerns are the effects of grinding, polishing, cutting, testing, mounting, coating, temperature, residual stress, handling, assembly, metrology, humidity, surface flaws, tolerances, mass uncertainties, force uncertainties, hysteresis, and peculiar material properties. These effects, along with those of the gravity and operational environments are discussed in the body of this report.

4.0 LIGHTWEIGHT CONSIDERATIONS

Along with the criteria for risk, schedule, cost, and performance is the requirement for lightweight, a feature of large diameter of thickness ratio optics. These can be classified as either thin solid optics (thin meniscus) or somewhat thicker but lightweighted, monolithic optics. This latter class is comprised of classical sandwich construction, i.e. a lightweighted core with top facesheet and bottom facesheet with or without holes. For very large diameter optics made of glass or glass ceramic, this type of construction often becomes impractical due to fusion or frit bonding requirements, particularly for the near zero thermal expansion candidates. (The Steward Observatory Mirror Laboratory has had much success in spin casting very large optics using borosilicate glass of relatively high thermal expansion.) To reduce risk, schedule, and cost to fabricate monolithic and lightweight optics, the use of "open back" construction is made, whereby the optic can be formed by machining away glass from an initially solid blank, where over 90 percent of the material can be removed. Such designs, in order to optimize stiffness, should consist of triangular cell cores. In the further interest of weight reduction, design optimization programs have been developed to minimize weight while maximizing stiffness, yet resist manufacturing pressures with no undue quilting, or "print-thru", during polish. While this technique is routinely used, advances using "fragile polish" removal techniques, in which nearly zero not pressure is applied while maintaining equal or greater removal rates as conventional methods allow, have obviated quilting concerns. This approach has been used with much success at Litton Itek Optical Systems.

An example of such a lightweight open back monolith is shown in Figure 1(a) which exhibits a diameter to depth ratio of 15:1. The equivalent solid thickness of this blank, i.e.. one giving the same stiffness, results in a effective aspect ratio of 30:1. Shown in Figure 1(b) is a thin meniscus solid, lightweight only by virtue of its small thickness. This optic has a diameter to thickness ratio in excess of 100:1. Performance comparisons of these optics are later given. It is noted that the mounting for each of the optics is different, the former performing on three points and the latter requiring multipoint. It is important to note that for the thin meniscus designs, and in may cases for the monolithic ones of high aspect ratio, some form of active control is generally demanded to meet the stringent tolerances specified for visible wavelength imagery. The need for multipoint mounting and active control is now presented.

5.0 MOUNT CONSIDERATIONS

The mounts which support the optic to its bezel or reaction structure frame must serve several purposes. They must be stiff in all directions to maximize rigid body frequencies, rigid enough to preclude buckling, and strong enough to resist operational and non operational stresses under acceleration loadings. They must also be flexible (in the appropriate degrees of freedom) to minimize mount induced assembly loading mirror distortion, to which the high aspect ratio optics are most susceptible. They must be flexible enough as well to minimize deflection caused by thermal expansion mismatch. To achieve such a design, the mounts must be near kinematic in nature. Truly kinematic mounts which rely on bearings, universal joints, pins, slots, rollers, or the like are to be avoided generally due to the unpredictable effects of stiction and friction. A near kinematic mount usually takes advantage of flexible parts in the appropriate degrees of freedom which are both predictable and accountable in the design.

An example of a three point near kinematic mount is shown in Figure 2(a) and 2(b). The former is a bipod design in which the flexibility requirements are met by necking down the members at each end. The stiffness requirements are met by the inherent axial member stiffness of the bipod legs themselves. The latter design is a three point leaf flexure mount with rotational pivot, the flexural providing the required stiffness and flexibility in all degrees of freedom except radial rotation which is provided by the pivot.

The ideal three point mount system often becomes impractical for mirrors of large diameter to thickness ratio, largely due one of the main enemies of those optics - gravity. Gravitational errors in conjunction with mount design are given in the section below.

6.0 GRAVITATIONAL CONSIDERATIONS

It is difficult to address mounting issues without addressing the gravity issue. This section will address some of the ways of mounting to resist gravity deflection in both optical axis vertical and horizontal configurations, particularly as the aspect ratio increases. A rather extensive discussion on mount configurations is given by Yoder[1], which provides an excellent reference. It is not the intent here to review detail but rather, provide an overview of the issues.

For mirrors with optical axis vertical, deflection under gravity on three points at the mirror periphery is given by

$$y = \frac{k_1 \rho D^4}{E t^2} \qquad (1)$$

for a thin meniscus solid, in which ρ is the material density, D the diameter, t the thickness, E the modulus, and k_1 a geometry/mount constant. (Nomenclature for this and all equations is given in Table I). When the optical axis is horizontal, deflection is given by

$$y = \frac{k_2 \rho D^5}{R_s E t^2}, \tag{2}$$

in which R_S is the mirror radius of curvature

Note the rapid increase as the diameter increases and thickness decreases. As the allowable performance is exceeded, the mount locations can be moved inboard to reduce distortions by up to a factor of four in both orientations. For the horizontal axis, however, care must be given to locate the mount intersection near the mirror center of gravity. As the mirror aspect ratio increases, the designer may be forced into a multipoint mount of six, nine, or more points. To maintain kinematics, a hindle, or whiffle tree, mount is then required for the optical axis vertical direction. One such mount is shown in Figure 3 for segment of the Keck telescope mirror, with aspect ratio of 25:1. Note here that the mount attachment is not truly kinematic, but the necessary compliance is achieved by flexured posts.

Other vertical axis mounts at many points include pneumatics, hydraulics, and counterweight systems, which serve to float the optic, leaving only gravitational residual error in between the mount points. Extreme care is needed in design to avoid problems of stiction and friction, however, and such designs will most often need some type of force sensing or active control.

For horizontal axis loading, a different scheme is required for multipoint loading. This can take the form of edge mounting with sinusoidally distributed forces, or mercury tubes. The sinusoidal forces can be achieved by lateral edge counterweights, as an example. Schewesinger[2] has provided an excellent theoretical treatise on the subject in several papers. This approach, too, has its limitations for high aspect ratio optics. It has been used successfully for mirrors up to four meters. But for the aspect ratios being considered here, distortion may still be too high for high acuity optical systems. In this case, supporting the optic edge tangentially can provide a good support. One such optic is shown in Figure 4, in which tangent rods provide the necessary lateral restraint yet maintain compliance in the axial direction. Angling the rods out of plane and other combinations with radial load distributions can further minimize distortion. As mirrors increase to the 8 meter class of aspect ratios in excess of 25:1, again some form of active control is still demanded.

The next level of support in terms of minimizing passive errors of gravity in the horizontal direction consists of multipoint restraints interior to the mirror edge. Again counterweights or hydraulics can be used, and combined to provide axial restraint as well. In fact, collocation of both axial support and lateral support systems can optimize the design in all orientations. The Hale telescope uses this concept. For thin meniscus designs, this approach minimizes distortion, but active control is usually demanded. Also, extreme care need again be taken to minimize mount induced rotational moments which may yield uncorrectable surface errors. A schematic example of a dual acting interior mount and rotationally compliant attachment scheme is shown in Figure 5, with provision for active control. Attachment to the optic back surface is achieved through epoxy bonding in the case of glass, to an attachment button matched in thermal expansion coefficient.

An example of a completed optic of aspect ratio greater than 100:1 is shown in Figure 6. Here, the optimized back internal multipoint support is used to resist gravity in all orientations. Each support also is under active control to correct errors from sources other than gravity. In this case no counterweights or hydraulics are required, since the weight per support is quite low. Lateral support is achieved through rotationally compliant bipods and tripods attached to the mirror back which point to the neutral surface of the mirror (i.e. the midpoint of its thickness), as seen in Figure 7.

7.0 MATERIAL CONSIDERATIONS

Properties of some of the potential common mirror materials are, in general readily available. When discussing large aspect ratio aspects, it is important to understand all the design criteria which might drive a design. Figures of merit in themselves are nice selling points for the manufacturer, but may have no bearing on the design. For example, common mirror choice materials include glass ceramics such as Zerodur, glasses such as borosilicate, fused silica, and ULE, metals such as beryllium and aluminum, and silicon carbide. A combined thermal mechanical figure of merit may look like

$$\frac{EKS}{\rho\alpha\Delta\alpha} \tag{3}$$

which E is the stiffness (Young's modulus) K the thermal conductivity, S the strength, ρ the density, α the coefficient of thermal expansion, and $\Delta\alpha$ the expansion homogeneity. Materials such as glasses have poor conductivity and hence do not resist thermal gradients, but the low expansion glasses make up for this shortfall. Beryllium has excellent conductivity but its benefit is negated relative to glass at room temperature due to relatively high expansion coefficient. At colder temperatures, however, conductivity to expansion ratios are better than the glasses. In the absence of heat flux, such as solar exposure, the figures of merit may have no bearing. Gravitational sag and weight may now drive the design (E/ρ). Strength may be insignificant if a design is stiffness driven. In the presence of soaks, expansion homogeneity is important, as will be discussed, particularly for large optics. Not addressed by usual figures of merit are the effects of coatings, residual stress, curvature, surface flaws, hysteresis, etc., all of which may play an active role in material selection for the high aspect ratio optics being considered. Some of the peculiar behavior of those optics are addressed in the remainder of this paper.

8.0 TEMPERATURE CONSIDERATIONS

In order to understand the behavior of the large optics detailed mathematical models are required to determine distortion during thermal soak and gradient conditions. This is important not only for the operational environment but for the fabrication environment as well, where active control is not as readily invoked. Figure 8 shows a typical method used to determine the effects of expansion coefficient inhomogeneity during thermal soak. In that case, a finite element model is prepared which maps, one for one, measured CTE data from a Corning ULE glass boule, where CTE values are expressed in parts per million. Although near zero in thermal expansion, it is the variability about the nominal which is often of concern in the approach to large optics design. Errors due to expansion inhomogeneity are found approximately proportional to the relation.

$$y = \frac{k_3 T \Delta\alpha D^2}{t}. \tag{4}$$

where T is the soak temperature.

As an example, an optic exhibiting a D/t ratio of 60:1 and presented inhomogeneity is shown in Table 2, using the finite element modeling technique for a proposed 7 meter diameter optic. Notice the effects even after power (focus) removal of the relatively small variaiion in expansion coefficient for only 1°C change.

Thermal gradients through the thickness can also be a driver in design of thin meniscus mirrors. Due to curvature (shell effects), errors so realized are not entirely focusable. The same example for the 60:1 optic shows the results before and after focus in the same table. The radius of curvature is 25 meters. For low aspect ratios, error is related to

$$y = \frac{k_4 \alpha \Delta T D^2}{t} \tag{5}$$

where ΔT is the axial thermal gradient. The error is nearly all focusable, for conventional optics.

For high aspect ratios, with finite radius of curvature, however, the mirror shall under the described gradient no longer remains in the stress free state afforded by the flat condition, so that the above equation no longer applies. In this case, the thermal bending of a circular segment of a thin spherical shell subjected to a constraint temperature gradient through its thickness is examined. The physical parameters and governing equations, which use complex Bessel (kelvin) functions, are described by Barnes.[3] A significant amount of spherical aberration is found as the aspect ratio increases.

To illustrate what happens to the residual error before and after focus correction, a detailed NASTRAN model of a 40 inch diameter mirror was made, with the mirror thickness varied accordingly. The model was initially run as a flat plate, and the results found to agree with the theoretical solution to within one-half of a percent. A radius of curvature of 175 inches was then included to represent a typical curved optic.

Utilizing a constant unit gradient through the mirror thickness, the resulting displacement from the math model were then input to a post-processor to determine residual error both before and after focus. The results are shown in Figure 9 along side the theoretical solution afforded in the reference above. The correlation is markedly good.

As evidenced by the curves, the departure from the flat plate error is most pronounced as the diameter to thickness ratio increases. For a D/t of 50:1, the error is about 2.5 times less than the flat plate without correction. After focus, however, about 16 percent residual still remains, whereas the flat plate error is entirely focusable. For a D/t of 10:1, the uncorrected error is very close to that of the flat plate, while the residual error remaining after focus is less than 1 percent.

We conclude that the shell effect reduces the residual error before a focus fit, under a thermal axial gradient. After focus, the residual error may be significant.

9.0 RESIDUAL STRESS

It is of extreme importance to understand the nature of residual stresses in the optic to be fabricated. For small diameter optics with relatively low D/t ratios (<10:1), residual stress effects are often negligible. For large diameter optics with increasing aspect ratios, the effect could be significant in understanding behavior during cutting, lightweighting, or edging operations. Stresses due to cooling processes for Zerodur, for example, while low, could cause excessive warping after release. Stress residual magnitude for ULE and other glassy materials are of similar magnitude, and could be significantly higher for metal materials. Finite element analyses have correlated markedly well with measurements made on several large optics programs.

Consider for example, a circular disc subjected to hoop compressive stresses at its outer periphery and tensile stresses toward its center. This is a likely scenario in the forming process of Zerodur glass ceramic, in which the outer periphery, due to heat transfer laws, is cooled more rapidly than the center. As the center continues to shrink (positive expansion coefficient) during cooling after ceramizing and anneal against the already cooled, more rigid outer edges, tensile stresses are set up near the center with hoop compression near the edge as shown in Figure 10. The locked-in stresses are a function of the expansion characteristic of the material, and since this value is small, so are the stress levels. Similar effects can be found in ULE, where low levels of residual stress are present due to small but finite differences in expansion coefficients throughout a boule due in part to variations in composition. Generally outer zones of the boules are in residual tension. For other materials, residual stress levels can be expected to be of greater magnitude. For the glasses, birefringence test can be made to correlate results with analysis. As with any polarized measurement, only the difference in principal stresses are readily made, with more complex techniques requiring oblique incidence viewing and shear integration required to isolate the stresses. Thus, an element in pure (bi-directional) tension gives no indication of stress if the usual technique is applied. Nonetheless, measurements near the edge, where radial stress is zero, give a good indication of the magnitude and direction of internal edge stress.

Again finite element modeling can be used to predict the magnitude of springing for various operations. Table 3, for example, is an example of warping after edge cutting and central hole cutting on a Keck telescope segment. These analytical results correlated very well with test data.[4] Stress level residuals are on the order of 30 psi only. Note that power is not the only anticipated change, but higher order symmetrical aberrations as well are realized, due to the shallow shell effects. The segment has a D/t ratio of about 25:1.

Springing effects are related to

$$y = \frac{k_5 \sigma_r D^2}{E t^2} \qquad (6)$$

in which σr is the residual stress level.

10.0 COATING CONSIDERATIONS

Coating of optics are subject to residual stresses during deposition and differential stresses during thermal excursions due to CTE mismatch. However, for normal reflective coatings measured in Angstroms of thickness, distortion to even large optics is of minimal concern. On the other hand, the use of much thicker metallic coatings for use on aluminum or beryllium for example, required for fine polish, can result in significant distortion. Over normal temperature ranges and accounting for backside coating as well to minimize this bimetallic effect, errors in excess of several hundred micro inches (tens of waves) are easily realized for an 8 meter optic with D/t of 40:1. While active control can alleviate this problem, correctability must be invoked continually and actuator density must be high. During fabrication, thermal changes could mask out what is trying to be interpreted.

Under thermal soak or residual coating stress, distortion is related to

$$y = \frac{k_6 \Delta\varepsilon h D^2}{t^2} \tag{7}$$

$\Delta\varepsilon$ is the strain residual in the coating ($T\Delta\alpha$ or σ_c/E) are h its thickness.

11.0 SURFACE FLAW CONSIDERATIONS

While surface flaws must be addressed due to slow crack growth phenomena, additionally during grinding large changes can be observed due to the Twymann effect, which is related to the residual stress discussion above and follows the same relationship. A material is removed, subsurface flaws and hence locked in stress will cause radius changes and higher order aberrations to a lesser degree. These will disappear after final polish, but must often be accounted for in the grind to understand what is happening. These effects could exceed several microns, for example, in grind of a glass optic with D/t of 40:1.

Another phenomenon related to surface flaws is humidity, which can cause chemical reactions and residual stress if moisture is allowed to propagate in flaws. Again, a few microns of power change can be realized with marked humidity changes for optics with aspect ratios in excess of 50:1 if surface flaws are left unchecked.

12.0 METROLOGY CONSIDERATIONS

Of prime importance to the large aspect ratio optics is the ability to fabricate and test it. A metrology mount is usually required to "float" the thin meniscus in order to perform a test of the mirror surface quality in the "absence" of gravity. Such mounts can take on several forms, including air bag, discrete piston, a combination of air and pistons, or perhaps the final mount itself. Mount spacing is chosen to minimize gravity sag in between supports, and often the use of active control correctability can be invoked, since what the mount puts in can be largely taken out by actuators. For these optics, however, a rather large sensitivity to force errors is possible. These errors can be caused by pressure variations stiction, mass uncertainties, force hysteresis, friction, and the like. Errors due to force uncertainty propagate as

$$y = \frac{k_7 F D^2}{E t^3} \tag{8}$$

in which F is the force.

Figure 11 gives an example of the sensitivity to force error for a four meter optic with D/t ratio in excess of 200:1. Note that the mirror is gram sensitive with up to a micron of error realized for randomly distributed forces of ±1 gram maximum. This indicates the care which must be taken in understanding the subtle effects of metrology for such optics. For the subject case, a fluid, single zone, multi piston mount arrangement was chosen which utilized nearly hysteresis free diaphragm assemblies. Accounting for force uncertainties and active control correctability which would be collocated with some (but only a portion) of the pistons, high performance goals were analytically achieved and validated by test of a smaller optic with D/t of 100:1.

13.0 TEMPORAL INSTABILITY

What is considered stable with time for optics of small D/t ratio and small deflections may not be so for those with high D/t ratio and the resulting larger deflections which can be accommodated due to the flexible nature of the piece. The allowable deflection an optic can undergo within its stress limits is proportioned to

$$y = \frac{k_8 \sigma_a D^2}{t} \tag{9}$$

where σ_a is the allowable stress.

Thus, as D/t increases, capabilities exist in such areas as active control, as is planned for the Very Large Telescope program (VLT), for example, in which distortions of up to 25μm may be required for multiple use of the system.[5] The Keck Telescope program utilizes a stress polish technique[6], in which deflection of up to 100μm are required during fabrication. In other cases, large operational deflection under gravity or thermal loading, for example, may occur and be compensated for using active control.

One such phenomena that can occur and be noticed with large deflections is that of delayed elasticity, defined as that portion of strain under stress beyond the instantaneous value under load, and not immediately recovered upon unloading.

Much has been written about structural relaxation, viscous flow, delayed elasticity, hysteresis, and other dimensional instability phenomena of glass and ceramics at elevated temperatures. Less has been documented about similar effects at room temperature. The time-dependent phenomenon of delayed elastic exhibited by Zerodur, a low-expansion glass ceramic manufactured by Schott, has been studied at room temperature.[7] The effect is related to the alkali oxide content of the glass ceramic and to rearrangement of the ion groups within the structure during stress. The effect, apparent under externally applied load, is elastic and repeatable, that is, no hysteresis of permanent set, as measured at elevated temperature, is evident within measurement capabilities. Nonetheless, it must be accounted for in determining the magnitude of distortion under load (delayed elastic creep) and on load removal (delayed elastic recovery). This is particularly important for large and lightweight optics, which might undergo large strain during fabrication and environmental loading, such as experienced in gravity release or in dynamic control of active components.

Itek Optical systems first encountered the delayed elastic effect in Zerodur while fabricating the primary-mirror segments for the Keck 10 meter astronomical telescope. The effect was measured using a high performance mechanical profilometer. To confirm the Keck data for delayed elasticity further, we conducted additional experiments on both glass manufactured by Corning. The samples used were 12.7 cm-diameter stress relieved plates polished flat on one side. The Zerodur was a piece taken from an actual Keck blank. Measurements were made using interferometry to eliminated metrology errors. The load was applied for two weeks at room temperature. The setup was modeled with the NASTRAN finite-element computer program, and the predicted deflections were characterized as Zernike polynomials. The instantaneous delayed elasticity expected was 0.7% of the maximum deflection, whereas the approximate theory would indicate that the figure of the plate would return to its original condition linearly with log time over a period equal to the loading time. The actual data are plotted for Z5 (C 2, 0), Z7 (C3, -3) and Z13 (C 4,0) in Figure 12. When the measured data are compared with the predictions, the correlation is excellent.

The effect for glasses containing alkali oxides is up to 1% or more, in excess of ten times that of fused silica. For usual and most applications, the results are readily accommodated, since the effect is reversible, without hysteresis. Unless strains are large, it will never be noticed. In cases where strains are large, however, it must be understood, to prevent misinterpretations of the source of time changing data during metrology.

14.0 ACTIVE CONTROL CONSIDERATIONS

Most of the class of optics being discussed here will demand some form of active control, since environmental, fabrication, or test deflections may be in excess of allotted requirements, particularly for visible wavelength systems. It is noted that a thin active meniscus design can result in less expensive facesheet cost; is very lightweight, even when actuator and reaction structure weight is included; results in low thermal mass; and exhibits superior performance to the more massive and passive designs. In general, for a given and equally spaced actuator count, correctability for active mirrors is independent of both diameter and thickness, and hence D/t ratio, for a given shape and magnitude of error. The force requirements of course, increase for smaller diameters and higher thickness according to the relations earlier described. Actuator count can be determined by the need for minimizing inter-actuator gravity sag (which is a function of thickness) and the magnitude and shape of aberrations to be corrected. In the former regard, actuators are often made coincident with the gravitational operational mount. In the latter case, correctability algorithms must be used to determine count, and detail trades made to size the facesheet within total weight limitations. Residual error after correction is approximately proportional to actuator count. For example, as few as 50 discrete force actuators can correct low order astigmatic error to a residual of 1% of the original aberration; 100 actuators reduces that residual in half. For higher order, residual error increases for a given count of actuators; 100 actuators may reduce residual error to 10% of the input, for example, in the case of spherical aberration. Force requirements along with weight may drive the meniscus thickness trade. A 4 meter optic with D/t of 15:1 may operate passively in a gravity environment, but hydraulics or counterweights may be needed; increasing D/t to 40:1 requires actuator force sensing and local loop closure; increasing D/t to 100:1 will require active control and wavefront sensing, with the benefit of eliminating counterweights or hydraulics, directly attaching the facesheet to its supportive structure. The design of the latter thin meniscus must take into account all of the design principles included herein. Facesheet thickness versus actuator count plots can then be made for the multiple criteria, and an optimum combination of thickness and count found in which all requirements are achieved. These requirements, in addition to the gravity concern, may include thermal gradient, aberrated incoming wavefronts, weight, stress during acceleration or in flexure, fundamental frequency for control stability, manufacturability, metrology error force limitations, power, stroke, resistance to uncorrectable assembly loads, and manufacturing errors, among others.

15. PERFORMANCE COMPARISONS

A the outset of this paper, an arbitrary definition of size and aspect ratio was given for purpose of the above discussions, i.e. diameter in excess of 1 meter and D/t in excess of 15. In this section, a comparison is given in terms of sensitivity to mirror displacements based on the scaling relationships developed. This can give the reader a sense of why design considerations for large high aspect ratio optics are important, since displacements unseen in the smaller optics suddenly increase to significant proportions.

As an example, the summary of Table 4 is presented which compares a 1 meter optic with D/t ratio of 6:1, a low aspect ratio design, to one of 2 meters in diameter with D/t ratio of 15:1, another of 4 meters in diameter with aspect ratio 25:1, still another of 8 meter diameter with D/t of 40:1, and a final of 8 meter diameter of aspect ratio 150:1. The chart is normalized to the 1 meter design of D/t ratio 6:1 for purpose of the comparison. All of the designs, incidentally, are in the domain of optics already manufactured or under consideration, and in fact are all less sensitive than some large optics currently being fabricated.

As evidenced by the table, sensitivities are increased by several orders of magnitude in some cases, highlights the care which must be given. This presentation is ended in this light, by a similar performance comparison of mirrors from actual or planned programs, involving the high aspect ratios being discussed here. This is given in Table 5, in which all values are ranked by sensitivity and normalized to the successful large optic of the European Southern Observatory (ESO), a 3.5 meter by 230mm meniscus with aspect ratio of approximately 15:1. Shaded programs are those completed or under completion at Itek Optical Systems.

16.0 CONCLUSION

Large optics with high aspect ratio are quite sensitive to the environment in which they are made and in which they must operate. In order to ensure their success, considerable consideration must be given to peculiar mirror properties, response to force errors, manufacturing approach, test methods, and the need for active control. Fabrication is not without challenge but can be thus approached with a degree of certainty in finishing to desired optical tolerances.

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TABLE 1
Nomenclature

F	=	random force error
E	=	material modulus of elasticity
α	=	coefficient of thermal expansion (CTE)
K	=	thermal conductivity
ΔT	=	axial thermal gradient
T	=	thermal soak
$\Delta\alpha$	=	material CTE inhomogeniety
ρ	=	density
σ_r	=	residual stresses
σ_c	=	coating stress
σ_a	=	material allowable stress
S	=	strength
y	=	performance error
D	=	facesheet diameter
t	=	facesheet solid thickness
k_n	=	geometrical/mount constant
R_s	=	mirror radius of curvature
$\Delta\varepsilon$	=	residual strain
Z	=	Zernnike polynomial

TABLE 2
Response to Soak and Gradient for a 300 x 5 Inch Thick ULE Curved Optic

	1°C Soak Change		1°C Axial Gradient	
	Error From Perfect CTE Value	Error From Radial CTE Variation	Error From Perfect CTE Value	Error From Radial CTE Variation
Uncorrected	0.0024	0.032	0.131	0.073
Focus	0.0000	0.019	0.037	0.046

- values in waves (surface) at 633nm

TABLE 3
Theoretical Warping (Normalized to 1.0 micron power)

Zernike Term	Edge Cut	Center Cut	Total
C(2,0)	+0.70	+0.30	+1.00
C(4,0)	-0.06	-0.13	-0.19
C(6,0)	0.00	+0.05	+0.05
C(8,0)	-0.01	-0.08	-0.09

- values correspond to 35psi stress in outer zone
- positive power is concave

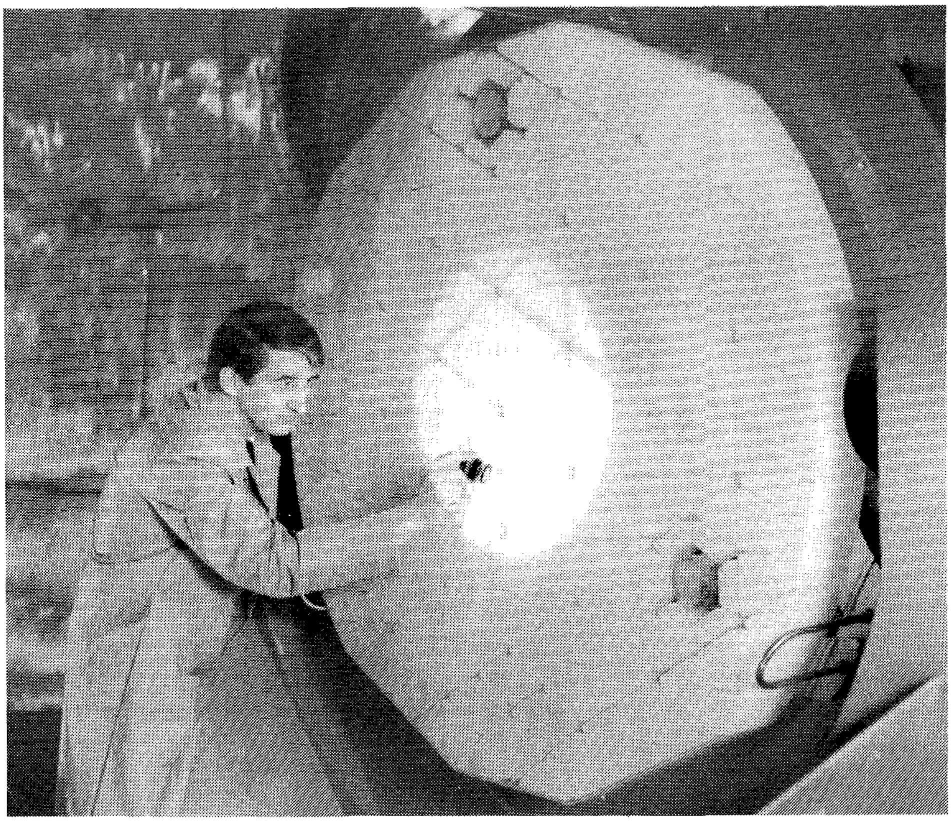
TABLE 4
Sensitivity of High Aspect Ratio Optics Relative to Conventional

Size (meters)	Aspect Ratio	Grind/Cut/Coat	Metrology	Thermal Soak	Gravity
8.0	150	580	1760	190	37300
8.0	40	40	33	51	2620
4.0	25	16	16	16	256
2.0	15	6	7	5	24
1.0	6	1	1	1	1

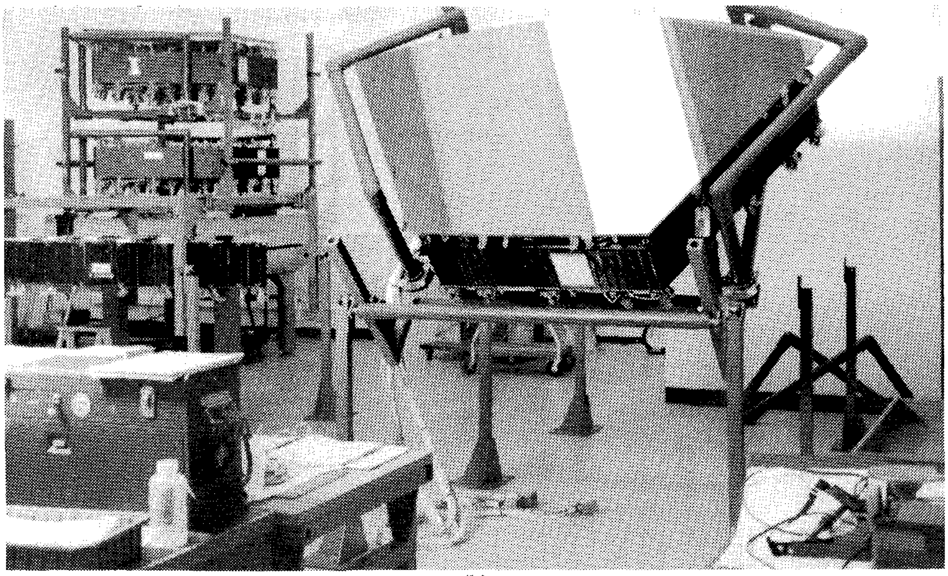
TABLE 5
Senstivity Ratings for Large Optics

Size (Meters)		Aspect Ratio		Grind/Cut Coat		Metrology		Thermal Soak		Gravity	
VLT	8.2	LOS	235	LOS	240	LOS	3250	LOS	18	LOS	78
JNLT	8.1	ALOT	153	ALOT	100	ALOT	1350	VLT	7	ALOT	55
LOS	4.0	LAMP	118	LAMP	60	LAMP	810	ALOT	7	VLT	52
AEOS	3.6	VLT	47	VLT	9.5	VLT	12	JNLT	6	JNLT	38
ESO	3.5	JNLT	41	JNLT	7.0	JNLT	8	LAMP	4.5	LAMP	20
ALOT	2.6	KECK	25	KECK	2.8	KECK	8.6	AEOS	1.6	AEOS	2.6
LAMP	2.0	AEOS	24	AEOS	2.5	AEOS	3.8	ESO	1	ESO	1
KECK	1.9	ESO	15	ESO	1	ESO	1	KECK	.9	KECK	.8

• Listed by Degree of Difficulty Relative to ESO

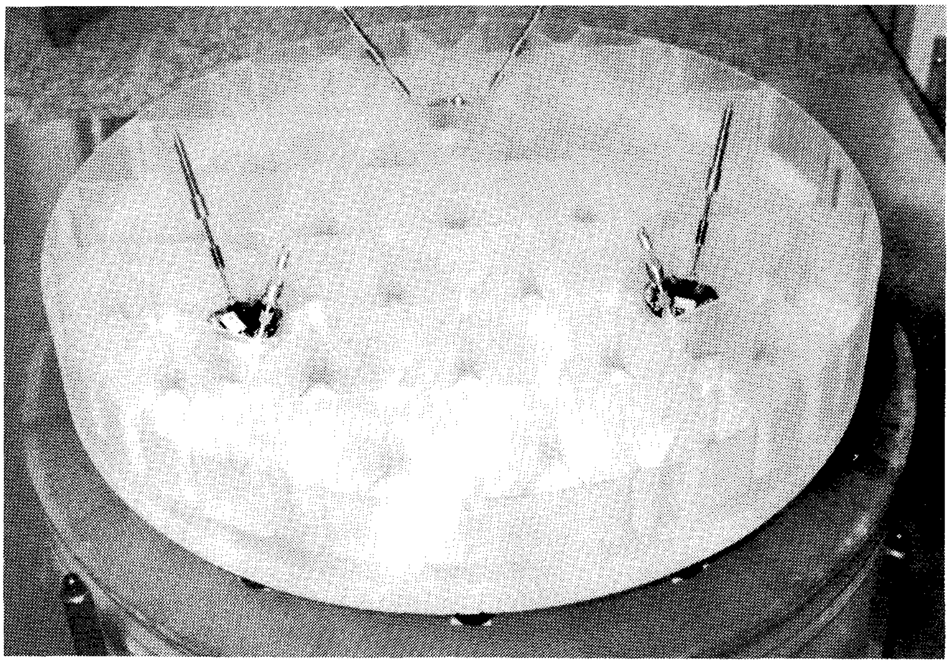


(a)

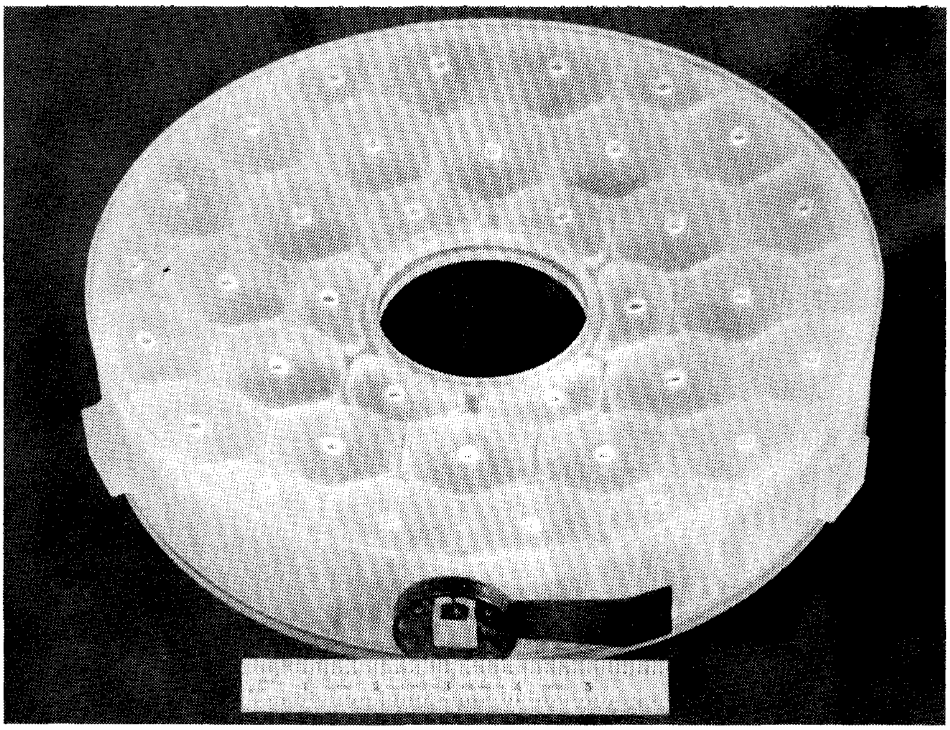


(b)

Fig. 1. Large-aspect-ratio optics.



(a)



(b)

Fig. 2. Three-point near-kinematic mounts

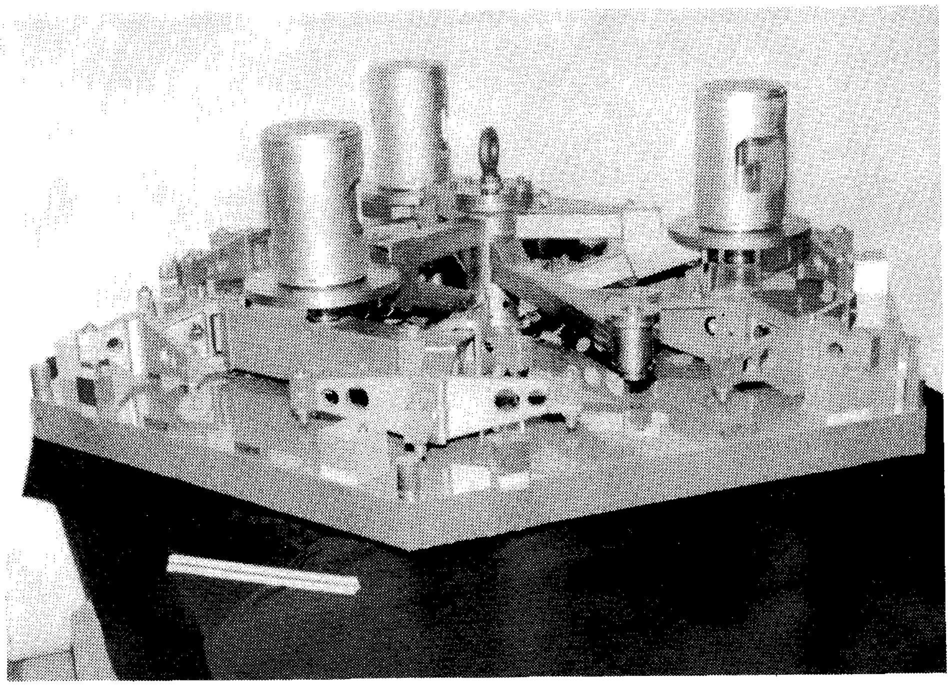


Fig. 3. Multi-point whiffle tree mount.

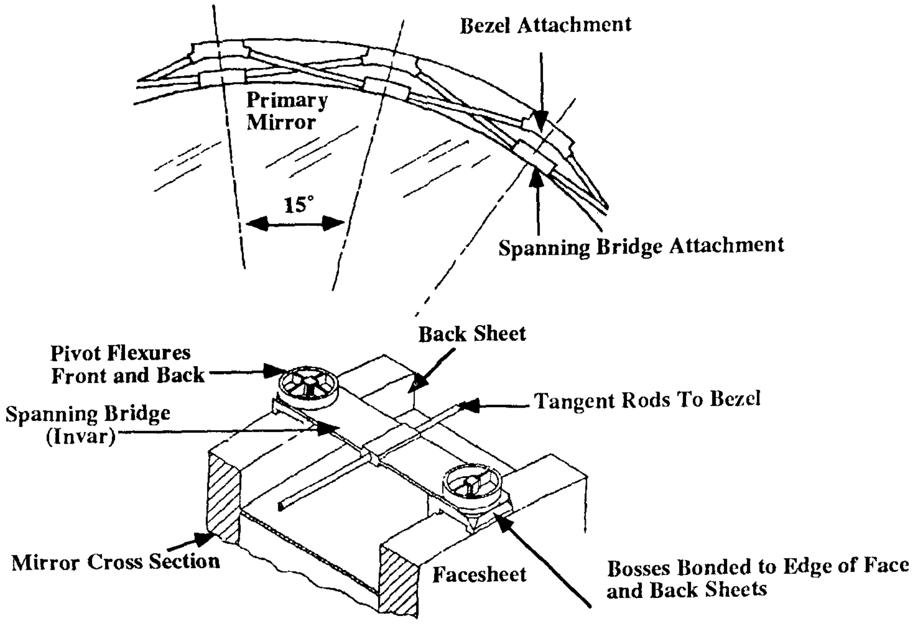


Figure 4
Tangent Rod Design

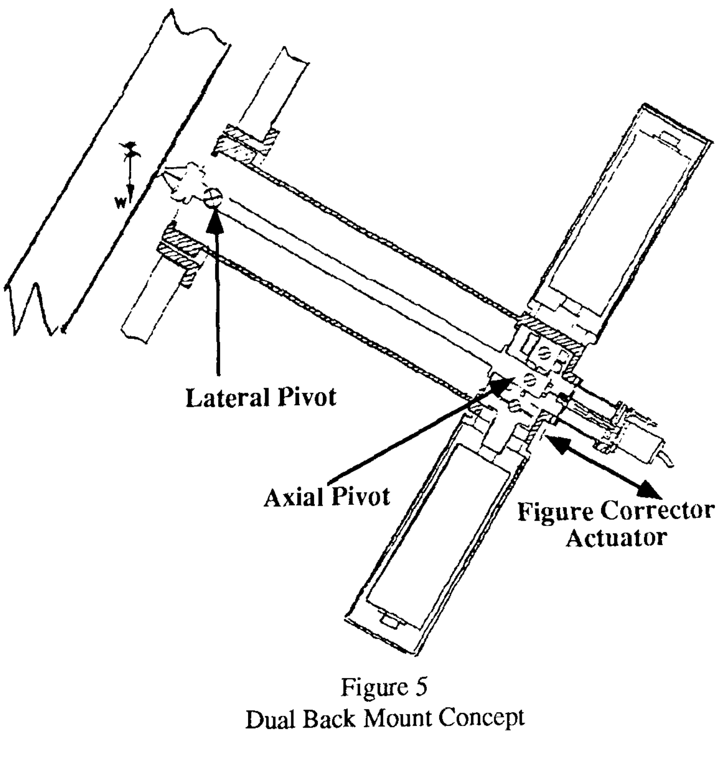


Figure 5
Dual Back Mount Concept

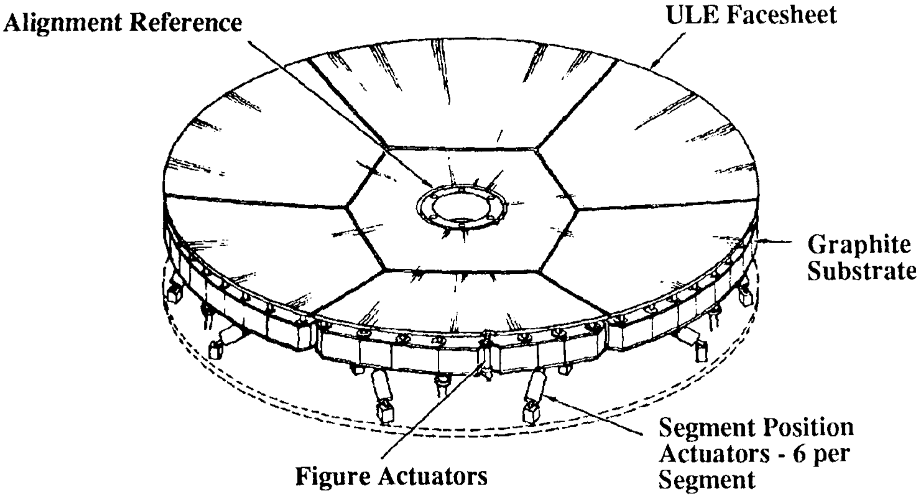


Figure 6
Lightweight Active Optic

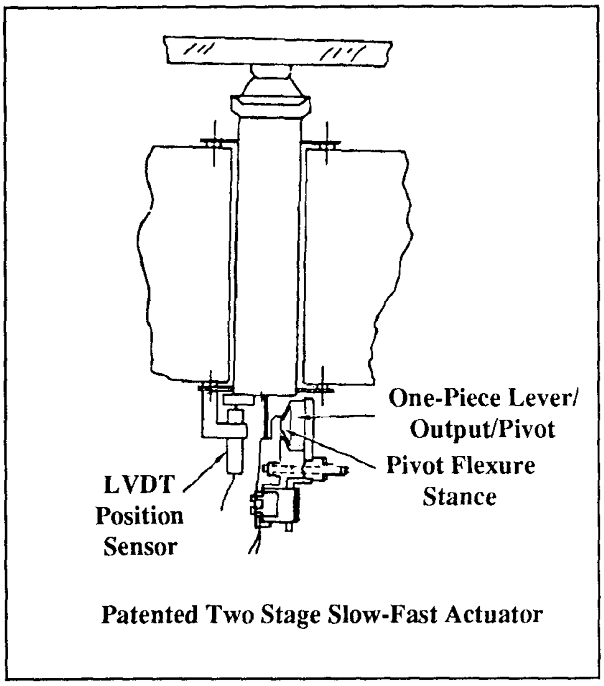


Figure 7
Surface Control Actuator Schematic

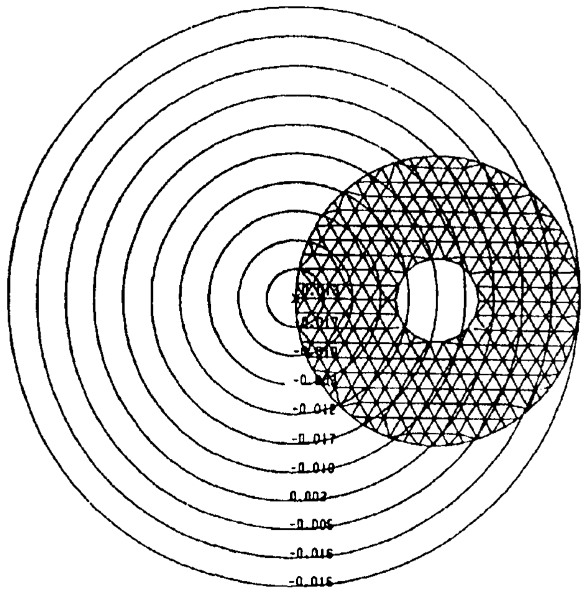


Figure 8
Prescribed ULE Variation

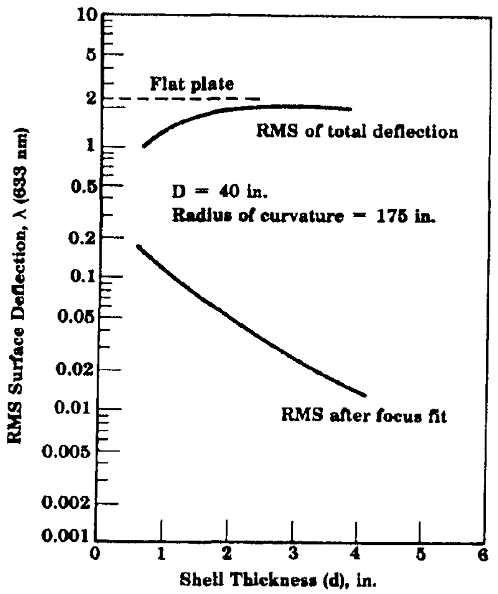


Figure 9
The Effect of Inhomogeneities on Performance - Solid Mirror,
ULE-Prescribed Variation in Plan

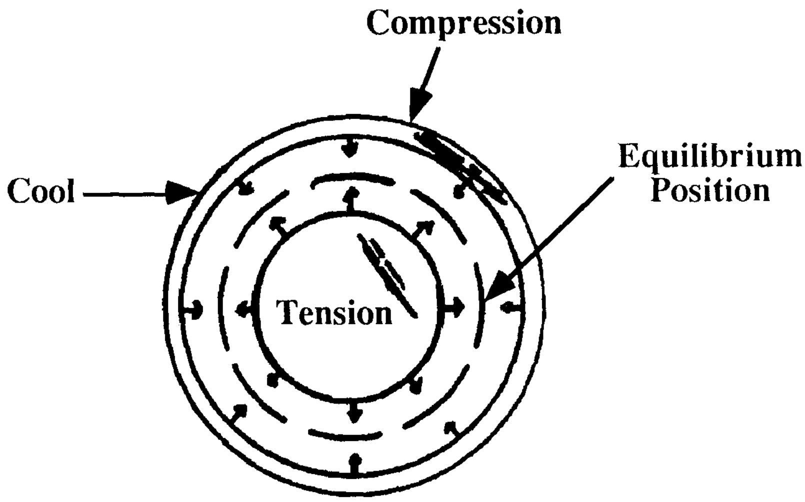


Figure 10
Internal Stress Due to Cooling

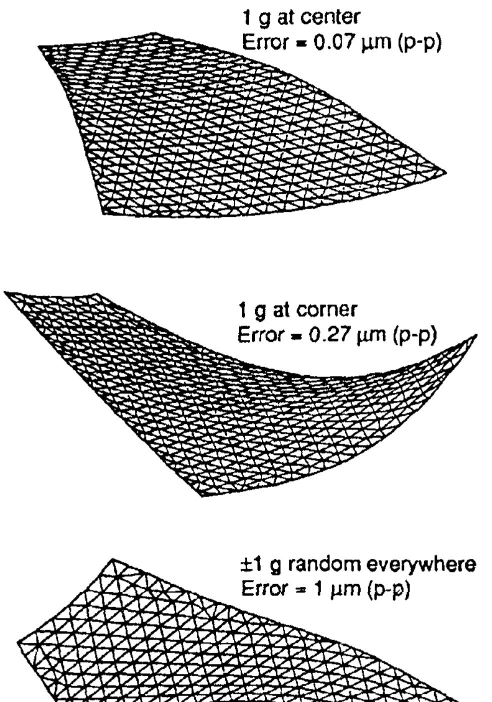
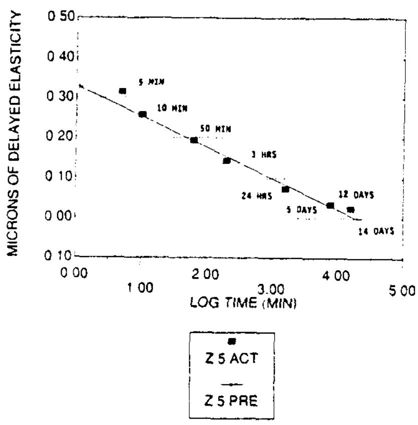
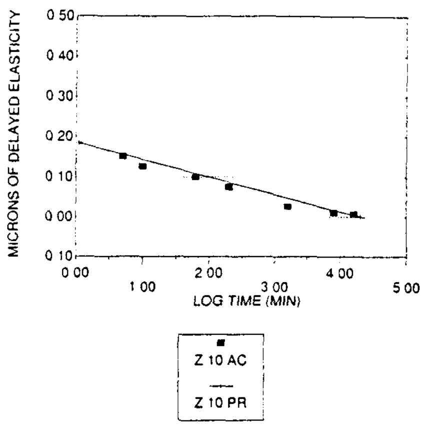


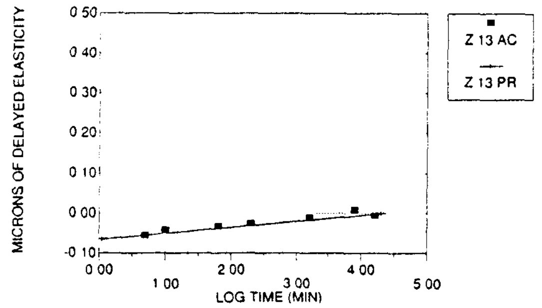
Figure 11
Fabrication/Metrology Mount
Sensitivity to Force Errors



(a) Delayed elasticity after 2 weeks loading
(30 kg) in Zerodur



(b) Delayed elasticity after 2 weeks loading
(30 kg) in Zerodur.



(c) Delayed elasticity after 2 weeks loading
(30 kg) in Zerodur.

Figure 12
Delayed Elasticity over Time - Actuals (squares) vs. Predicted (line) in Zerodur for Z(5), Z(10), and Z(13)