United States Naval Observatory Optical Interferometry Project

Preliminary Design Study of the Beam Collapser and Certain Optomechanical Aspects of the Siderostat

SUMMARY

Four topics were studied during this phase of the design of the USNO astrometric interferometer: 1) the optical configuration of the beam collapser; 2) the mirror mount for the primary mirror of the beam collapser; 3) the beam collapser structure, and 4) bearings for use in the siderostat. A Gregorian beam collapser configuration, with a 750-mm diameter f/2.7 primary mirror, a 116-mm diameter secondary mirror, with a radius of curvature of 623 mm, and a vertex-to-vertex spacing 2337 mm is recommended. Use of a Gregorian beam collapser reduces optical fabrication costs, with a small impact in overall system length and weight. A solid primary mirror, 100-mm thick and oversized, with a 850-mm diameter, is the simplest and most economical mirror to fabricate and mount. A roller chain would provide radial support for the primary mirror; a six-point whiffle tree located outside the optical clear aperture would provide axial support. A modified Serrurier truss connects the secondary mirror to the primary mirror; the upper forward section of the truss is removed to allow better sky coverage for the siderostat. Use of a modified Serrurier truss permits testing of the system in the optical axis horizontal position, and without loss of alignment in the 10° down position. In addition, this type of truss is center supported, permitting the use of a tapered pier to minimize concrete volume on site. Spacing from the primary mirror to the secondary mirror, a critical parameter for system performance, is controlled by a pair of metering rods made from the same type of material as the optics. Rolling element bearings, oil pad bearings, and air bearings were considered for use in the siderostat. Although an extensive literature survey was performed, and a number of vendors contacted, it is not yet established which bearings are optimum for the siderostat. Use of a relatively conservative Gregorian beam collapser, with a solid primary mirror, and modified Serrurier truss with metering rods, will provide the desired performance at relatively low risk and cost.

OPTICAL CONFIGURATION

Three options were considered for the beam collapser optical configuration: 1) classical Cassegrain; 2) classical Gregorian; and 3) off-axis combinations of the first two configurations. Although both are strictly afocal Mersenne systems, the terms "Cassegrain" and "Gregorian" are used to distinguish the corresponding convex-secondary and concave-secondary versions. The beam collapser specification is:

Primary clear aperture:	0.75 m
Primary central hole:	0.125 m > h_p < 0.1154 m
Primary optical quality:	0.05 wave peak-to-valley (1 wave = 633 nm) over any 0.3 m diameter of the clear aperture, 0.1 wave peak-to-valley over entire clear aperture
Primary cosmetic quality:	40-20
Secondary clear aperture:	Less than 0.125 m
Secondary optical quality:	0.05 wave peak-to-valley over entire clear aperture
Secondary cosmetic quality:	20-10
Beam compression:	6.5:1
Beam diameter variation:	Less then 1% collapser to collapser
Minimum focal ratio:	f/5000 (redundant)

An important consideration not listed in the above specification is cost. By traditional astronomical standards, the optical quality required of both primary and secondary mirrors of the beam collapser is unusually high. This high optical quality implies higher than normal fabrication costs for these mirrors. Because weight and physical size are relatively unrestricted, the possibility of trading these parameters against ease and cost of optical fabrication was considered.

Some consideration was given to the use of an off-axis configuration. A Brachyt type of system, using an off-axis Cassegrain or an off-axis Gregorian configurations, eliminates the central obscuration and provides good control of stray light. A Brachyt design might also allow the height of the pier to be lowered, reducing the required volume of concrete required on site.

Cost and ease of fabrication are major concerns with an off-axis system. The Optical Sciences Center has extensive experience in the design, fabrication, testing, and alignment of these types of systems, up to about 1.5-m aperture. This experience was used to evaluate the cost and risk of making off-axis systems of the type required for the beam collapser.

Two methods are used to make off-axis mirrors: 1) diamond turning of metal substrates; 2) and cutting the mirror (glass or metal) from a larger parent substrate. The current fabrication limit for diamond-turned metal surfaces is about 1 wave (1 wave = 633 nm), which is substantially worse than the system specification. A diamond-turned metal mirror would present scattering problems caused by grooves on the mirror surface. Long-term dimensional stability of a large metal mirror is unlikely to be better than about 1 wave. Scatter is reduced by plating the metallic mirror surface with nickel and performing post-diamond-turning polishing. Bi-metallic bending effects would cause a significant change in the mirror surface figure with temperature if electroless nickel-plated aluminum mirrors were employed.

Cutting the mirror from a larger parent mirror has some advantages. Two primary mirrors could be cut from a 1.7-m diameter parent, and four could be cut from a 2-m parent. This would allow some economy of fabrication. Fabrication cost is dependent on the mirror surface area and on mirror speed. A 1.7-m mirror has about four times the surface area of a single 0.85-m mirror, while a 2-m mirror has about six times the area of a single 0.85-m mirror. A 1.7-m mirror requires optical working of about twice the area of a 0.85-m mirror to produce one off-axis mirror; a 2-m mirror requires about 1.5 times the area. A 1.7-m parent would have a focal ratio of about 1.2. Fabrication of a 1.7-m f/1.2 mirror to a 0.1 wave peak-to-valley specification is marginal with current fabrication methods. A 2-m parent would have a focal ratio of 1. A f/1 2-m mirror, good overall to a peak-to-valley specification of 0.1 wave, is literally state-of-the art for today's fabrication. Even if this figure quality could be achieved, the mirrors must be cut from the parent. Experience with similar mirror sizes (1.1-m mirrors cut from a 1.5-m parent) indicates that some change in figure because of release of stress must be expected after cutting the mirror from the parent.[1] Post-cutting local figuring would be required, which again raises the cost of fabrication. It appears that an off-axis system fabricated from a large parent would be significantly more expensive than a conventional system.

Because the off-axis systems are too expensive for this application, the remaining choices are the selection of a conventional Cassegrain or Gregorian system, and the choice of the f/number of the system. Both Gregorian and Cassegrain systems use paraboloid mirrors, separated by the sum of their focal lengths, where the ratio of the focal lengths is equal in magnitude to the beam compression ratio.

Cassegrain systems are used in most astronomical applications.[2] The Cassegrain has a slightly smaller secondary obscuration than the Gregorian system, with a resulting reduction in diffraction and increase in transmission. For a given overall focal length and primary focal ratio, the Gregorian system will be longer than a Cassegrain system. This increase in length is critical in most astronomical applications. An increase in overall system length requires a longer tube, with a corresponding increase in weight and moment of inertia. The longer tube in turn requires a larger mounting, and a larger building or dome. Since dome costs scale as the cube of the dome diameter, a small increase in tube length results in a large increase in cost.

The situation in the astrometric interferometer is significantly different from the conventional astronomical telescope. The beam collapser is fixed, and weight is not critical. Overall length is still an issue, since an increase in length could encroach on the siderostat, or require a longer, more expensive housing. In addition, the optics of the beam collapser must be very much better than traditional astronomical optics.

Fabrication of the convex secondary mirror of a Cassegrain telescope requires either the use of an aspheric test plate or a large Hindle sphere. Of the two test methods, the Hindle sphere is the most common, and requires test optics of quality equal to or better than the secondary specification. The Hindle sphere must be at least as large as the primary mirror, and a high-quality calibrated collimator large enough to fill the secondary mirror is needed as well.

Use of auxiliary optics during testing substantially increases cost and risk during fabrication. Removing the errors in the test reference optic from the test is a difficult and time-consuming process. Interferometric testing of large optics to a precision (repeatability) of better than 0.05 wave is near the current state of the art. Fabrication and testing of the beam-collapser mirrors will require removal of test errors of this size.

Discussion with two experienced opticians at the Optical Sciences Center confirmed the above analysis. Auxiliary optics are undesirable and can be avoided by the use of a Gregorian system. In the case of a Gregorian configuration, both surfaces are concave paraboloids, and can be tested directly without the use of reference optics. Fabrication of the surfaces of a Gregorian system is therefore less costly and less risky.

Overall length of the beam collapser is constrained to about 2.25 m by the specification that the system be as short as possible with an f/3 primary. Use of a Gregorian system may not be cost effective if the ease if fabrication is offset by the need to reduce the f-ratio of the primary, to the point where primary mirror fabrication becomes expensive. If m is the ratio of the focal length of the secondary mirror to the focal length of the primary mirror, then for the same f/number, the Gregorian system will be $(1 + m)/(1- m)$ times as long as the Cassegrain. This factor also determines the ratio of the f/numbers for systems with the same length.

Experience with at least eight 1.8-m f/2.7 mirrors at the Optical Sciences Center indicates that a mirror of this speed is not significantly more difficult to fabricate than an f/3 mirror. Increasing the primary speed of the beam collapser from f/3 to f/2.7 permits the use of a Gregorian configuration without a penalty in overall length.

If a Gregorian system is used, the specifications for the two mirrors are:

1.	Both mirrors will be concave paraboloids			
2.	Primary focal length:	2025;	+5/-0	mm
3.	Primary clear aperture:	750		mm
4.	Secondary focal length:	311.5;	+0.5/-0	mm
5.	Secondary clear aperture:	116		mm
6.	Spacing tolerance, vertex-to-vertex, secondary to primary:	±2		μm
7.	Centering tolerance, primary to secondary:	±40		μm

Focal-length tolerances were derived from the requirement that the compression ratio for all four systems differ by less than 1%. Spacing and centering tolerances were derived from the 0.05-wave peak-to-valley error specification in the compressed wavefronts.

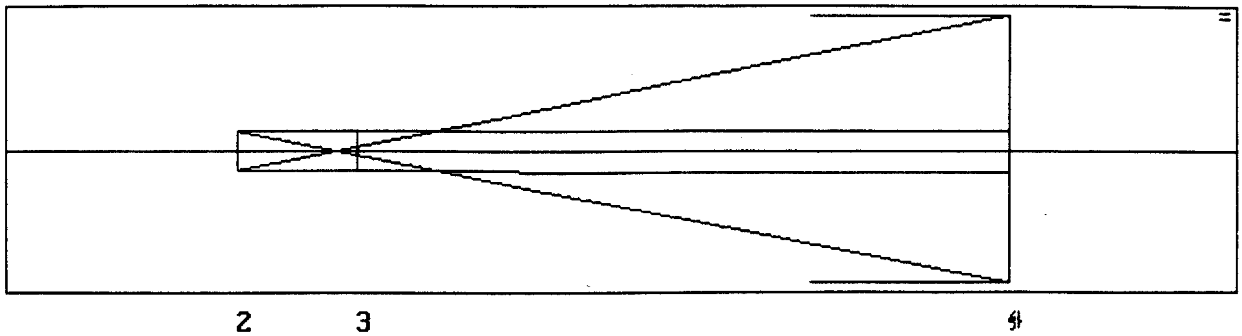
Control of stray light is another virtue of the Gregorian configuration. An intermediate, real focus is formed; a stop placed at this point effectively excludes light from outside the field of view. For the above system, this stop should be 1977 mm from the primary mirror, and 115.4 mm in diameter. For improved control of stray light, an opaque baffle tube, surrounding the secondary mirror and extending back towards the primary mirror to the location of the stop, is suggested.

One concern with the existing specification for the beam collapser is the cosmetic surface quality of 40-20. This cosmetic quality is likely to prove very expensive to obtain on a 0.75-m diameter f/2.7 mirror. It is recommended that this surface quality specification be re-examined prior to procurement of the mirror.

The system prescription and layout are shown in Figure 1.

PRIMARY MIRROR AND MOUNT

Selection of a primary mirror and mirror mount involves consideration of the thermal and stiffness characteristics of a variety of technologies. Because weight is not restricted, the use of lightweight mirror configurations is suggested only if the thermal response or cost benefits are significant. Many common mirror-mount designs used in conventional astronomical telescopes must be reconsidered. The beam collapser is fixed with respect to the gravity vector, while most telescope primary-mirror mounts must be designed to maintain mirror figure with respect to a gravity vector that changes direction.



GREG MERSENNE F/2.7 (MERSNC) 1/4/91

NO.	Y	YBAR	EFL	DIST
0	375.000	-0.107	0.000	599.697
1	375.000	0.000	2025.000	%-2336.580
2	-57.700	0.418	311.580	359.522
3	-57.700	0.000	0.000	1977.058
4	-57.700	-2.300	0.000	0.000

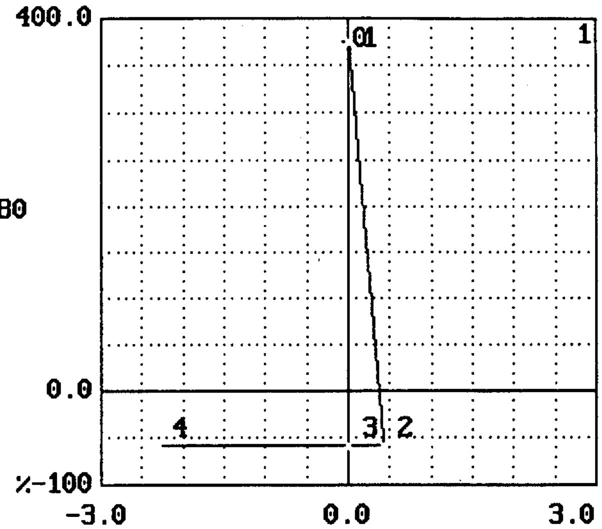


Figure 1. Beam collapser prescription and layout.

Two issues in the design of the primary mirror are the primary mirror material and the mirror configuration. Choice of the primary mirror material is based primarily on thermal response and cost. Choice of the mirror configuration is based on ease of fabrication, stiffness, and thermal response.

A mirror's response to temperature changes is determined by the material properties of thermal coefficient of expansion, the thermal distortion parameter, the spatial variation of thermal coefficient of expansion, and the thermal diffusivity. The thermal coefficient of expansion determines the overall change in physical size of the mirror when exposed to a uniform change in temperature. Changes in the physical shape of the mirror when exposed to a thermal gradient are determined by the thermal distortion parameter of the material. The length of time required for the mirror to reach thermal equilibrium is determined by the thermal diffusivity. Often overlooked, but very important, is the spatial variation of the thermal coefficient of expansion of the material. This property controls the change of shape of the mirror when a uniform change in temperature occurs.

If both primary and secondary mirrors are made of the same material, the value of thermal coefficient of expansion is not very critical. The only effect the thermal coefficient of expansion has on the beam collapser is its effect on the means of maintaining focus with temperature. Selection of a material with a low thermal coefficient of expansion requires either the use of a material with a low thermal coefficient of expansion in the beam collapser structure, or active focus control. A high value for the thermal coefficient of expansion in the optics could be readily offset by the use of materials with high thermal coefficients of expansion in the support structure.

Because the temperature at the observatory site changes during the night, the ability of the mirror to maintain its shape when exposed to a temperature gradient is important, and is controlled by the thermal distortion parameter. The thermal distortion parameter is the ratio of the material's thermal coefficient of expansion to its thermal conductivity. This parameter is given in units of m/W. As an example of the use of this parameter, the change in radius of curvature of a mirror exposed to a steady-state linear temperature gradient is given by

$$\Delta R \simeq \frac{\alpha}{K} q R^2 \quad ,$$

where: ΔR is the change in radius of curvature of the mirror;

α is the thermal coefficient of expansion of the mirror material;

K is the thermal conductivity of the mirror material;

q is the heat flux per unit area through the mirror; and

R is the radius of curvature of the mirror.

From the above equation, the smaller the value of the ratio of the thermal coefficient of expansion to thermal conductivity, or the smaller the value of the thermal distortion index, the smaller the thermal distortion when the mirror is exposed to a temperature gradient. Thermal distortion parameters for materials of interest for the primary mirror are: borosilicate glass (Pyrex), 2.9×10^{-6} m/W; fused silica, 410×10^{-9} m/W; Corning ULE, 23×10^{-9} m/W; Schott Zerodur, 31×10^{-9} m/W; and 6061-T6 aluminum, 136×10^{-9} m/W.

As indicated by the above values, aluminum has a better response when exposed to a thermal gradient than common mirror materials; only the "zero coefficient of expansion materials" such as ULE and Zerodur are better.

When exposed to a change in temperature, the mirror requires time to reach thermal equilibrium. The internal temperature of the mirror at some time after a sudden change in temperature is given by:

$$T' = T - \Delta T \exp(-t/\tau) \quad ,$$

where: T is the initial temperature of the mirror;

T′ is the temperature of the interior of the mirror;

ΔT is the change in temperature;

t is the time since the temperature change; and

τ is the thermal time constant of the mirror.

The thermal time constant is given by:

$$\tau = \frac{h^2}{\pi^2 D} = \frac{h^2}{\pi^2 \left[\frac{K}{\rho c_p} \right]} \quad ,$$

where: τ is the thermal time constant of the mirror;

D is the thermal diffusivity of the mirror material (units of m^2/sec);

K is the thermal conductivity of the mirror material;

ρ is the density of the mirror material; and

c_ρ is the specific heat of the mirror material.

According to the above equations, the greater the thermal diffusivity of the mirror material, the more rapidly the mirror will reach thermal equilibrium. Most glasses have about the same value of thermal diffusivity: borosilicate glass, 604×10^{-9} m^2/sec; fused silica, 840×10^{-9} m^2/sec; ULE, 777 $\times 10^{-9}$ m^2/sec; and Zerodur, 800×10^{-9} m^2/sec. Aluminum has a very favorable thermal diffusivity 66 $\times 10^{-6}$ m^2/sec.

Spatial variation of the thermal coefficient of expansion of the mirror material will cause a variation in the response of the mirror to changes in temperature, depending on location in the mirror. The effect of spatial variation in the thermal coefficient of expansion with temperature is complex, and is determined by the use of finite element analysis.[3] A simple illustration of the magnitude of this effect is the mirror surface deflection caused by a change in thermal coefficient of expansion from front to back in the mirror:

$$\delta = \frac{r^2}{2h} \Delta\alpha \, \Delta T \quad ,$$

where: δ is the surface deflection;

r is the mirror radius;

h is the mirror thickness;

$\Delta\alpha$ is the variation in thermal coefficient in expansion, from front to back in the mirror; and

ΔT is the change in temperature.

For most materials, the spatial variation in the thermal coefficient of expansion is no more than about 5%.[4] Lesser variations in thermal coefficient of expansion are therefore found for materials with low thermal coefficients of expansion. Spatial variations in the thermal coefficient of expansion for typical materials are: borosilicate glass (Ohara E-6), 50×10^{-9} m/m-K; fused silica (Corning 7940) 2×10^{-9} m/m-K; Corning ULE, 10×10^{-9} m/m-K; Zerodur, 20×10^{-9} m/m-K; and aluminum, 120×10^{-9} m/m-K.

Using the above thermal response equations and material data, an estimate is made of the thermal performance of beam collapser primary mirrors made from a variety of materials. Typical diurnal temperature changes for mountaintop observatories are about 10 K, with the fastest change occurring after sunset, at about 1.5 K/hr.[5] Table 1 was developed using this temperature-change data, and assuming a solid primary mirror, 0.12 m thick. The data in Table 1 should be considered pessimistic, or as a worst case. Mirror distortion caused by the thermal equilibrium effect after one hour was calculated on the basis of an instantaneous change of 1.5 K. In reality, the change in temperature is not instantaneous, and this effect is likely to be significantly smaller. Distortion arising from a lack of spatial uniformity of the thermal coefficient of expansion was calculated assuming a front-to-back or axial change equivalent to the entire spatial variation. This assumption is also pessimistic. Better estimates of performance can be provided using more sophisticated methods, but the relative rankings of the materials will not change.

Table 1 might seem to indicate that the best thermal performance, when exposed to sudden change in temperature, is provided by aluminum, however, as shown later, there are other considerations. Comparable in performance, and within a factor of two of the optical quality specified for the primary mirror, are ULE and Zerodur. Fused silica is a factor of ten worse than either ULE or Zerodur, while borosilicate glass is a factor of 100 worse than fused silica. Aluminum is the worst performer with respect to spatial variation, with an optical surface error that is a factor of ten worse than the specification. Fused silica and ULE are roughly comparable. Zerodur and borosilicate glass are similar, with a performance that is about ten times worse than fused silica or ULE.

Table 1. Thermal performance of primary mirror materials.

Material	α (m/m - K)	D (m^2/sec)	ΔR/m (waves)	$\Delta\alpha$ (m/mK)	δ/m (waves)
Borosilicate Glass (OHARA E-6)	3.3×10^{-6}	604×10^{-9}	149×10^{-6} (235)	50×10^{-9}	300×10^{-9} (0.48)
Fused Silica (Corning 7940)	56×10^{-9}	840×10^{-9}	1.41×10^{-6} (2.2)	2×10^{-9}	12×10^{-9} (0.02)
ULE (Corning 7971)	3×10^{-9}	777×10^{-9}	88×10^{-9} (0.14)	10×10^{-9}*	60×10^{-9} (0.10)
Zerodur (Schott)	5×10^{-9}	800×10^{-9}	139×10^{-9} (0.22)	20×10^{-9}**	120×10^{-9} (0.19)
Aluminum (6061-T6)	23×10^{-6}	66×10^{-6}	(0) (0)	120×10^{-9}	722×10^{-9} (1.14)

Assumptions:

1. Mirror thickness = 0.12 m
2. Mirror radius of curvature = 4 m
3. Mirror diameter = 0.75 m
4. ΔR computed on the basis of 10 K instantaneous change, ΔR is figure error after 1 hr.
5. δ computed on the basis of 10 K change, with $\Delta\alpha$ varying through the thickness of the mirror.

*Spangenberg-Jolley, J., and Hobbs, T., "Mirror substrate fabrication techniques of low expansion glasses," Proc. SPIE 966, pp. 284 (1988).

**Mueller, R. W., Hoeness, H. W., and Marx, T. A., "Spin-cast Zerodur mirror substrates of the 8-m class and lightweighted substrates for secondary mirror," Proc. SPIE 1236, pp. 723 (1990).

Based on the data from Table 1, ULE appears to exhibit the best performance. Zerodur is the second choice, if the spatial variation of thermal coefficient of expansion can be controlled to a level comparable to that of ULE. Fused silica would be a distant third choice. Neither borosilicate glass or aluminum appear suitable for this application.

The rejection of borosilicate glass deserves further comment, considering the current interest in large, lightweight borosilicate optics. Because borosilicate glasses have poor thermal performance, interest in these materials has centered around the possibility of offsetting poor material properties by careful mirror design. Lightweight mirror configurations with thin cross sections promise relatively short thermal time constants, with reduced thermal distortion effects.[6] The best thermal control of a lightweight borosilicate mirror achieved in a laboratory environment is a gradient of 0.4° C/1.8 m.[7] Scaling this gradient to a 0.12-m-thick borosilicate mirror produces a thermal deformation of about 12 μm, or about 19 waves in the visible (1 wave = 633 nm). Dramatic improvement in the thermal control of lightweight borosilicate mirrors must be achieved before this type of mirror can be considered for use in the primary mirror of the beam collapser.

Aluminum is rejected as a primary mirror material for reasons in addition to the poor uniformity of its thermal coefficient of expansion. Bare aluminum has poor scattering characteristics, and must be plated, typically with nickel, to improve surface scatter. Unfortunately, the thermal coefficient of expansion of nickel, 13×10^{-6} m/m-K, is different from the thermal coefficient of expansion of aluminum. A change in temperature will induce a bi-metallic bending effect in a plated-aluminum mirror.[8] Plating both sides of the mirror is not a complete solution, since plating thickness changes during fabrication. Even if bending of the mirror surface is avoided, stresses in the aluminum/nickel interface may become high enough to induce permanent deformation of the mirror.[9] Long-term dimensional stability of aluminum is typically limited to about 1 wave, which is a factor of ten worse than the optical surface specification for the mirror.[10]

Possible mirror shapes for the primary mirror of the beam collapser include the traditional solid, a contoured back shape, a thin meniscus, an open-back ribbed shape, and a sandwich structure. Shapes that differ from the traditional solid are normally selected on the basis of weight reduction and thermal control. Since weight reduction is not required for the beam collapser primary, and since good thermal performance is obtained with proper materials selection, justification for departure from a solid mirror is difficult.

Contoured back shapes include the single arch, double arch, and double concave shapes.[11] Both double arch and single arch provide optimum stiffness-to-weight ratios in the optical axis vertical position, and are not suited for a horizontal axis application. The double concave shape provides significant reduction in deflection when the optical axis is horizontal. Use of the double concave shape might be indicated if a more detailed analysis found excessive self-weight-induced surface deflection. Since the double concave shape has a concave back, it is significantly more difficult to fabricate than the traditional flat-back mirror.

There has been considerable interest recently in thin meniscus mirrors for large astronomical optics. Thin meniscus mirror blanks are often lower in cost than traditional solid blanks.[12] The uniform thickness of the meniscus improves thermal control, and provides for uniform areal density. The latter is an important consideration when the mirror is in the axis vertical position. Unfortunately, the low stiffness of a thin meniscus mirror makes fabrication very difficult. Lack of a symmetric section induces substantial self-weight deflection when the mirror is in the horizontal position, requiring relatively complex support systems.[13]

Open-back ribbed shapes are a traditional solution to thermal equilibrium problems.[14] Provided a material with a very low thermal coefficient of expansion is selected for the primary mirror, thermal control through the use of thin mirror sections will not be required. Open-back ribbed mirror shapes are relatively expensive to produce. Such mirrors are cast in borosilicate glass, or machined from a solid. Borosilicate glass has undesirable thermal properties for this application, and machining from a solid involves significant risk of mirror breakage. It is common to break one out of three mirrors during machining an open-back configuration from a solid. During polishing, pressure from the lap can deform the thin face plate of the mirror between the reinforcing ribs. A permanent periodic surface deformation called "quilting," corresponding to the rib position, is produced. This deformation can significantly lower mirror performance. Reducing quilting requires reduced polishing pressure, which in turn increases polishing time. Increased polishing time increases polishing cost with respect to a conventional solid mirror. Open-back mirrors are more difficult to support than conventional solid mirrors. Contrary to popular belief, for equal weight or equal thickness, the open-back ribbed mirror has about the same stiffness as a solid mirror.[15]

The sandwich mirror has essentially the same advantages and disadvantages of the open-back ribbed mirror. Unlike the open-back ribbed mirror, the sandwich has better stiffness than solid mirrors of comparable weight or height. This stiffness-to-weight advantage is not of importance in the beam collapser primary mirror application. Sandwich mirrors are even more expensive to produce than open-back ribbed mirrors. Support and ventilation (for thermal control) of sandwich mirrors are difficult design problems, far more so than for solid mirrors.

As seen from the above discussion, the use of a traditional solid shape for the primary mirror of the beam collapser will result in the lowest cost, simplest mirror mount, and minimum fabrication risk. Improved thermal control is the only virtue of the more exotic mirror shapes. Unless a material such as fused silica or borosilicate glass is selected, such thermal control is not required.

A wide variety of support schemes for large astronomical mirrors have been developed in the past. Since the beam collapser primary mirror is fixed in a near-horizontal position (down pointing at 10° with respect to horizontal), most traditional support schemes are not applicable. The primary contribution to mirror-surface deformation is astigmatism caused by a change in the mirror radius in the vertical plane under self-weight.[16]

Significant reduction in surface deformation of an optical axis horizontal mirror is possible through the use of a roller chain support. A roller chain support is a variation on the classical metal band support. The chain used in a roller chain support consists of a conventional metal conveyor chain with oversize rollers. The oversize rollers contact the edge of the mirror, and reduce friction between the mirror and chain, while providing many individual support points. Because there are a large number of support points, the influence of any single support point on the mirror surface is small. Properly designed, a roller chain support can reduce the optical surface deflection of a mirror to a value lower than that produced with classic multi-point support systems.[17]

Because the mirror is pointed away from horizontal, a roller chain cannot provide complete support of the mirror. Some type of axial support is required as well. The simplest type of support would bear on the edge of the mirror, and the optimum edge support would consist of a continuous ring. A uniform distribution of support force with a continuous edge ring is difficult to achieve in practice. More practical, and virtually equivalent in support efficiency to a continuous ring, is a ring of six equal-spaced discrete point supports.[18]

A rough order-of-magnitude calculation of mirror surface deflection was performed for the primary mirror of the beam collapser using closed form expressions. These calculations are given in Appendix 1. Caution is required, since these expressions do not include correction for shear effects. These calculations indicate that a simple roller chain radial support combined with a six-point axial support will reduce self-weight-induced mirror surface error to below the performance specification.

A one-inch pitch ANSI standard roller chain with oversize rollers would be used to provide support for the lower 180° of the mirror. Two such chains, equi-spaced on either side of the plane of the center of gravity, would be used. The chains would be extended parallel to each other from the mirror horizontal. Each chain would be connected to a chain hanger assembly, which provides for adjustment of the chain position on the mirror. A universal joint connects the chain hanger to the mirror cell. This design is based on the very successful chain support used on the Multi-Mirror Telescope (MMT).

Axial support would be provided with six discrete support points at the edge of the mirror. Two options exist for the radial support points: if the mirror is made oversize, the support can bear on the forward surface of the mirror; if the mirror is not made oversize, the support points must be attached to the back of the mirror. The oversize option is preferred. Making the mirror oversize simplifies the design of the support system, since gravity presses the mirror into contact with the support points. Attachment of the support points to the mirror back requires the use of adhesives, or machining a complex socket into the mirror back. Adhesives may deteriorate with time, and ultimately fail, leading to catastrophe. Complex mechanical sockets avoid the use of adhesives, but are expensive to produce and may influence mirror surface figure. There is also a risk of mirror breakage during the machining operation.

The preferred axial support scheme uses a mirror of 0.85-m diameter, with a support ledge 0.05-m wide located on the front surface of the mirror, just below the optical surface. This support ledge serves as a bearing surface for the six support points. Each discrete support point consists of a 25-mm diameter stainless steel swivel pad in contact with the support ledge. According to kinematic theory, six points of support are redundant, and will overly constrain the mirror. To avoid this condition, each pair of adjacent points is connected together by a link; the link in turn is connected by means of an axially stiff universal joint to the mirror mount. This three-link/six-point system is called a whiffle tree, and provides good kinematic location with uniform support.

Because a roller chain is dynamically unstable, additional constraint must be provided in the radial direction. Three tangential straps or flexures attached to the mirror edge provide radial constraint. Tangential straps or flexures are individually radially compliant, allowing expansion or contraction of the mirror cell with respect to the mirror. At the same time, the vector sum of the tangential stiffness of the radial straps provides good stiffness against radial disturbance.[19]

The final recommended design of the beam collapser primary mirror uses a solid ULE or Zerodur blank, 0.85 m in diameter and 0.12 m thick. A support ledge 0.05 m wide is located just below the optical surface on the front face of the mirror. The mirror back is flat to simplify fabrication. A roller chain provides support in the radial direction. Axial support is provided by a six-point whiffle tree bearing against the support ledge.

BEAM COLLAPSER STRUCTURE

Three types of structure are possible for the beam collapser: 1) a closed tube configuration; 2) a "bed frame" type support; and 3) a modified Serrurier truss. In addition to selecting a structure, the material used in the structure must also be considered. The support structure must maintain optical alignment and stability of focus with respect to time and temperature. Like the primary mirror and mount design, conventional astronomical structures are not suited for this application. Astronomical structures are intended for use when the direction of the gravity vector is constantly changing. Since the beam collapser is fixed with respect to the direction of the gravity vector, this aspect of conventional astronomical structural design is absent.

A key design parameter for the support structure is the deflection tolerance from the primary to secondary mirrors, which is 40 μm with respect to centering, and 2 μm with respect to vertex-to-vertex spacing. Maintaining the centering tolerance requires adequate stiffness in the support structure. Maintaining the spacing tolerance requires careful thermal design of the support structure.

The classic support for a telescope assembly is a cylindrical tube. A conventional cylindrical tube provides very good stiffness to weight, and may be fabricated from a wide variety of materials. For a 0.75-m aperture system, a cylindrical tube may provide better structural efficiency than a truss design.[20] To provide better sky coverage, the portion of the cylindrical tube near the secondary mirror must be cut away. This cut should extend from the height of the optical centerline back to the primary mirror. Removing material in this location lowers the stiffness of the tube significantly. A good analogy of the effect of removing this material is removing the upper flange of an "I" beam.

Loss of stiffness can be offset by providing supports for the tube at the points of attachment of the optics. Because the unsupported center span of the tube induces moments into the end supports, some rotation of the optics with respect to each other must be expected. It is unrealistic to assume the supports will be sufficiently stiff to prevent rotation. The amount of rotation would depend on the tube orientation. Alignment of the optics in the 10° down position would be required to offset this rotation. This is inconvenient, although the siderostat mirror could perhaps be used as an auto-collimation flat.

Thermal performance of the tube is another concern. Neglecting material considerations, a closed or semi-closed tube is not well ventilated. Current research into local seeing effects indicates that a well-ventilated support for the optics is desirable.[21] Ideally, this support should have a very short thermal time constant to avoid inducing any air currents into the optical beam path.[22] Attempts to ventilate closed tubes with fans have not been very successful in previous telescope projects.[23]

The structural inefficiency of the cylindrical tube design, combined with the poor thermal performance of this type of structure, led to the consideration of a "bed frame" type of structure for the beam collapser. The "bed frame" structure consists of a pair of large-diameter parallel tubes; the secondary and primary mirrors are mounted atop these tubes and at either end of the structure. Two vanes, each at 45° with respect to vertical, connect the secondary mirror to the pair of tubes. Shorter tubes tie the two large tubes together, and the entire structure is supported at three points. Two of the points are side by side near one end of the structure. Each of these points is attached directly to one of the large tubes. The third support point is located near the other end of the structure, and on the centerline. This support point is connected to one of the cross members of the "bed frame." The similarity between an ordinary bed frame with headboard and the beam collapser structure (the primary mirror cell looking like the headboard of a bed frame) led to the selection of the unusual name.

Although the structural efficiency of this type of structure is low, this inefficiency is offset by favorable deflection characteristics. Careful selection of the three support points will produce beam deflections at the point of the optics that are of zero slope. At the same time, variation of the cross section of the large tubes will produce equal deflections at both the primary and secondary mirrors. With the primary and secondary mirrors deflecting parallel to each other, and with the same amount of deflection, optical alignment is preserved. This design is similar in principle to the Airy support points for a uniformly loaded beam.[24] Deflection and optical alignment of the structure are independent of the orientation of the gravity vector. Alignment in the convenient axis horizontal position is possible, with little or no change in alignment when the structure is placed in the 10° down position.

The open nature of the bed frame design provides essentially "open air" conditions for the optics, to provide for optimum local seeing conditions.[25] Access to the optics is excellent, and there is no obstruction to the field of view of the siderostat. Construction cost is very low.

Unfortunately, late in the development of the bed frame structural concept, concern was voiced by the USNO about the size of the support pier. Apparently the cost of supplying concrete to the site is very high. A structural design that minimized the size of the pier is therefore preferred. The minimum volume possible for a stiff pier is that of a pyramid. An analogy is a cantilever beam of uniform stress, which tapers from point of applied load to its support. Use of such an optimum pier shape requires a support structure that is tied to the pier at a single point.

One such structure is a modified Serrurier truss. The Serrurier truss is a two-bay, center-supported truss designed to produce equal and parallel end-ring deflections.[26] As a very open design, the Serrurier truss provides nearly the same kind of open-air environment for the optics as the bed frame design. The center support of the Serrurier truss is located at the center of gravity of the truss. A support box or ring is located at this center support. A truncated pyramidal pier could then be connected to this central ring or box structure.

A disadvantage of the standard Serrurier truss is the end ring located at the secondary end of the structure, and the upper member of the truss. Both parts of the truss block the field of view of the siderostat. The upper part of the truss is used as a metering structure, to insure that the end rings remain parallel to each other. Total deflection of the end ring is determined by the triangular side members of the truss. It is possible to remove the upper part of the truss and replace it with a pair of links lower on the side of the truss. These lower side mounted links work with the remaining lower portion of the truss to guarantee the parallel deflection of the end rings. If the upper part of the truss is removed, there is no longer a need for a continuous end ring, and a semi-circular end ring can be used. Removing the upper parts of the Serrurier truss and cutting the end ring in half dramatically improves the field of view of the siderostat. It is recommended that the upper portion of the truss be removed only at the secondary end of the structure.

A preliminary analysis of the design of a modified Serrurier truss was performed, and is given in Appendix 2. A conventional steel tube structure was assumed. The final structure is about 2.4 m long, 1.2 m high, and 1.2 m wide. Estimated weight of structure and optics is about 1000 kg.

Maintenance of mirror spacing to 2 μm during the specified temperature change of -20° to +27° requires a structure made from a material with a thermal coefficient of expansion of less than 18×10^{-9} m/m-K. This thermal coefficient of expansion is much lower than normal structural materials, and about twice that of Invar.[27] If the spacing between primary and secondary were adjusted when the temperature range exceeded 10°C, Invar could be used for the structure. Since diurnal variation in temperature at the average observatory site is about this great, use of Invar implies a spacing adjustment at least once a night. Superinvar has a much lower thermal coefficient of expansion than Invar, and would expand the operating temperature between adjustment. Unfortunately, superinvar has poor dimensional stability, and may drift up to 20.5×10^{-6} m/m-day.[28] Although superinvar would reduce the need for spacing adjustment because of temperature changes, daily spacing adjustment would still be necessary due to lack of dimensional stability of the material.

Invar is an expensive material, and is not easily machined. Cold work, or heating due to welding operations, can change the effective thermal coefficient of expansion of Invar. Since welding and extensive cold working is necessary to fabricate an Invar beam collapser structure, a change in the thermal coefficient of expansion would be inevitable. This change, the poor thermal expansion coefficient of invar, and the lack of dimensional stability of superinvar, led to rejection of this class of materials for the structure.

A similar problem was solved during development of the Hubble Space Telescope through the use of a composite truss.[29] Composite materials can be tailored to produce virtually any desired thermal coefficient of expansion, have excellent stiffness-to-weight, and are available in tubular form suitable for use in a truss. Unfortunately, composite materials expand or contract with moisture absorption. Typical values of moisture-induced dimensional instability for composites with low thermal coefficients of expansion are in the range of $80\text{-}155 \times 10^{-6}$ m/m-%M, where %M is the moisture absorption of the composite.[30] Although the frequency of spacing adjustment necessitated by a change in temperature is reduced through the use of a composite structure, adjustment with humidity would be necessary and frequent.

A support structure made of the same material as the optics, ULE or Zerodur, would solve the spacing adjustment problem. Such a structure is prohibitively expensive and very fragile. The principle of same-material athermalization is valid, and can still be applied by using fused-silica metering rods to control the spacing between the secondary and primary mirrors. A conventional steel truss would be used to support the optics while spacing was maintained by low expansion rods. This type of structure has been used in previous astronomical telescopes when spacing was critical.[31]

Two fused-silica metering rods extend from the support ledge on the front surface of the primary mirror to the secondary mirror. These rods are parallel to the optical axis and outside the clear aperture of the system. Each of the two support vanes for the secondary mirror are attached to the end ring of the modified Serrurier truss by means of parallel spring guides. The parallel spring guides are soft in the direction of the optical axis and stiff in all other directions. Each metering rod is fastened to one of the support vanes. A change in temperature causes the metering rods to move the secondary mirror relative to the surrounding structure, and maintains spacing. Flexures are ideal for the secondary mechanism, being free of friction and hysteresis. In addition, flexures are zero-maintenance devices, and do not require periodic lubrication and adjustment.

Stress in the 2.4-m-long fused-silica metering rods is reduced by supporting each rod at its Airy points. An Invar sleeve is bonded using a semi-flexible adhesive to the Airy points of the metering rod. A diaphragm flexure connects the sleeve to an outer steel housing or tube. This tube provides protection and a rigid support for the metering rods; only the end is exposed. A similar approach was used to maintain spacing in the successful OAO-C satellite.[32]

The recommended structural configuration for the beam collapser is a modified Serrurier truss, 2.4-m long and 1.2-m wide. A pyramidal pier is used, connected to the truss at the center of gravity of the system, about 0.8 m from the primary. Steel tubing is used throughout the truss. The upper part of the secondary end of the truss, and the upper part of the secondary end-ring are removed to improve the field of view of siderostat. Equal and parallel end ring deflection in the truss is insured by adding two side links to the secondary end of the truss. Spacing between the primary and secondary mirrors is maintained by a pair of fused-silica metering rods in contact with the primary mirror, and moving the secondary mirror through a flexure system.

BEARING STUDIES

A literature survey is underway to determine the practical performance limits for the bearings in the siderostat. Three types of bearings are under study: 1) air bearings; 2) oil bearings; and 3) rolling-element bearings. All three bearings have been used with success in high-precision applications. Since the baseline configuration uses air bearings, considerable effort has been placed on determining the limitations of this type of bearing. Of relevance to the siderostat is the performance of the air bearings used in the Kuiper Airborne Observatory[33] and the planned successor to this successful instrument, the Stratospheric Observatory for Infrared Astronomy.[34] If an air or oil bearing is used, a Yates-type bearing will be required to provide stiffness in all directions.[35]

Contacts with bearing manufacturers were limited by the holidays, and are now underway.

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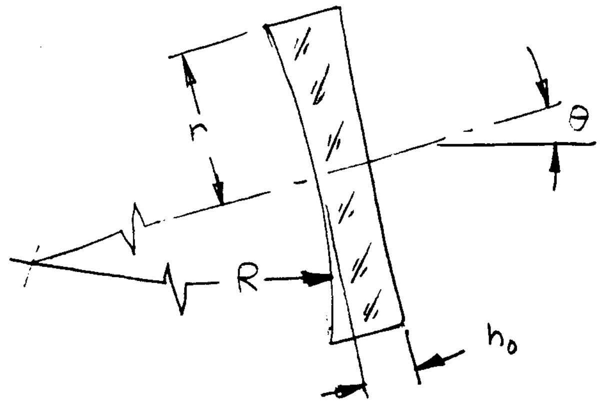
APPENDIX 1



PRIMARY MIRROR RMS SURFACE ERROR

$$\delta_{rms} \simeq \left[(\delta_R \cos\theta)^2 + (\delta_A \sin\theta)^2 \right]^{1/2} \quad (1)$$
(REF. 1)

WHERE:
- δ_{rms} = RMS SURFACE DEFLECTION
- δ_R = RADIAL DEFLECTION
- δ_A = AXIAL DEFLECTION
- θ = ANGLE, MIRROR AXIS TO HORIZONTAL

AXIAL RMS SURFACE ERROR

ASSUME MULTI-POINT SUPPORT:

$$\delta_A \simeq C \frac{q A^2}{D} \quad (2)$$
(REF. 2)

WHERE:
- C = DEFLECTION COEFFICIENT
- q = AREA DENSITY = ρh
- A = AREA = πr^2
- D = FLEXURAL RIGIDITY

$$= \frac{E h^3}{12(1-\nu^2)} \quad (3)$$

- ρ = DENSITY
- h = THICKNESS
- E = ELASTIC MODULUS
- ν = POISSON'S RATIO

RADIAL RMS SURFACE ERROR

ASSUME CHAIN (SLING) SUPPORT

$$\delta_R \simeq (a_0 + a_1\gamma + a_2\gamma^2)\,\frac{2\rho r^2}{E} \qquad (4) \text{ (REF. 3)}$$

WHERE: a_0, a_1, a_2 = DEFLECTION COEFFICIENTS

$$\gamma = \frac{r^2}{2h_0 R} \qquad (5)$$

R = OPTICAL SURFACE RADIUS

MIRROR PROPERTIES

ρ: 0.0796 LBS/IN3 } FUSED SILICA
E: 10.6×10^6 PSI
ν: 0.17
r: 15.75"
R: 159.45"
h_0: 4 - SAG = 4 - 0.685 = 3.315"
θ: 10°

DEFLECTION

AXIAL:

$$\delta_A = C\,\frac{qA^2}{D} = C\,\frac{\rho h(\pi r^2)}{\frac{Eh^3}{12(1-\nu^2)}}$$

$$= C\,12\pi\,\frac{\rho r^2}{Eh^2(1-\nu^2)}$$

$$= (19\times10^{-4})(12)\,\pi\,\frac{(.0796)(15.75)^2}{(10.6\times10^6)(3.315)^2\left[1-(.17)^2\right]}$$

$$= 12.50 \times 10^{-9}\ ''$$

RADIAL: $\gamma = \dfrac{(15.75)^2}{2(3.315)(159.45)}$

$= 234.65 \times 10^{-3}$

$\delta_R = \left[0.0073785 + 0.106685(235 \times 10^{-3}) + 0.03075(235 \times 10^{-3})^2\right] \dfrac{2(0.0796)(15.75)^2}{10.6 \times 10^6}$

$= 127.0637 \times 10^{-9}$"

TOTAL DEFLECTION

$\delta_{RMS} = \left\{\left[(127.1 \times 10^{-9})(\cos 10)\right]^2 + \left[(12.5 \times 10^{-9})(\sin 10)\right]^2\right\}^{\frac{1}{2}}$

$= 125.2 \times 10^{-9}$"

$\cong 0.005\ \lambda$

APPENDIX 2

EST WT PRIMARY & CELL

PRIMARY WT: 263 LBS
16' - 4" x 4" x .25" STL TUBING: 192 LBS
MISC HARDWARE: 150 LBS

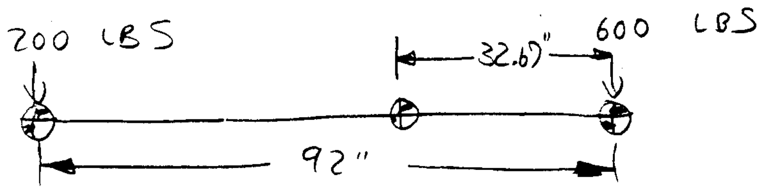
TOTAL: 605 LBS EST 600 LBS

SECONDARY ASSY:

PRIMARY & CELL WT: 15 LBS
VANES: 5.6' x 6 X .25 STL 80 LBS
END RING: 8' - 4" x 2" X .25" STL TUBING: 70 LBS
MISC HARDWARE: 35 LBS

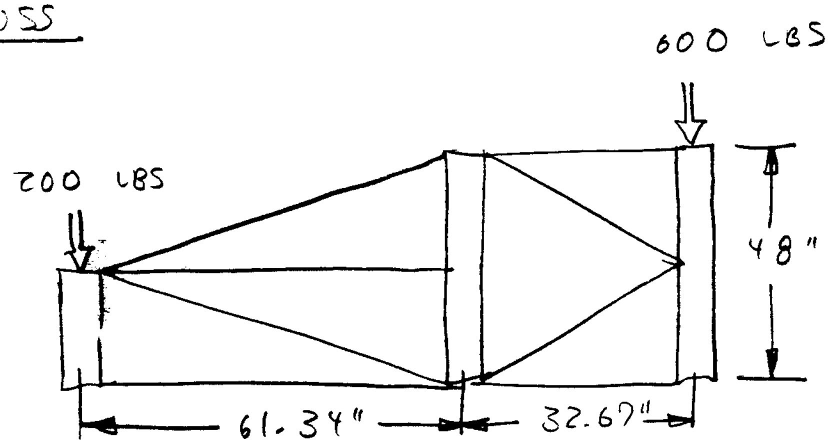
TOTAL: 200 LBS

CG



$$200(92-x) = 600x$$
$$(200)(92) - 200x = 600x$$
$$x = \frac{(200)(92)}{600} = 32.67''$$

TRUSS



$E = 29 \times 10^6$ PSI (STL)

$$\frac{W_1}{\cancel{E_1} A_1}\left(\frac{4\ell_1^2}{b^2}+1\right)^{3/2} = \frac{W_2}{\cancel{E_2} A_2}\left(\frac{4\ell_2^2}{b^2}+1\right)^{3/2}$$

$$\frac{A_2}{A_1} = \frac{W_2}{W_1}\,\frac{\left(\frac{4\ell_2^2}{b^2}+1\right)^{3/2}}{\left(\frac{4\ell_1^2}{b^2}+1\right)^{3/2}}$$

$$= \frac{600}{200}\,\frac{\left[\frac{4(32.67)^2}{(48)^2}+1\right]^{3/2}}{\left[\frac{4(61.34)^2}{48^2}+1\right]^{3/2}}$$

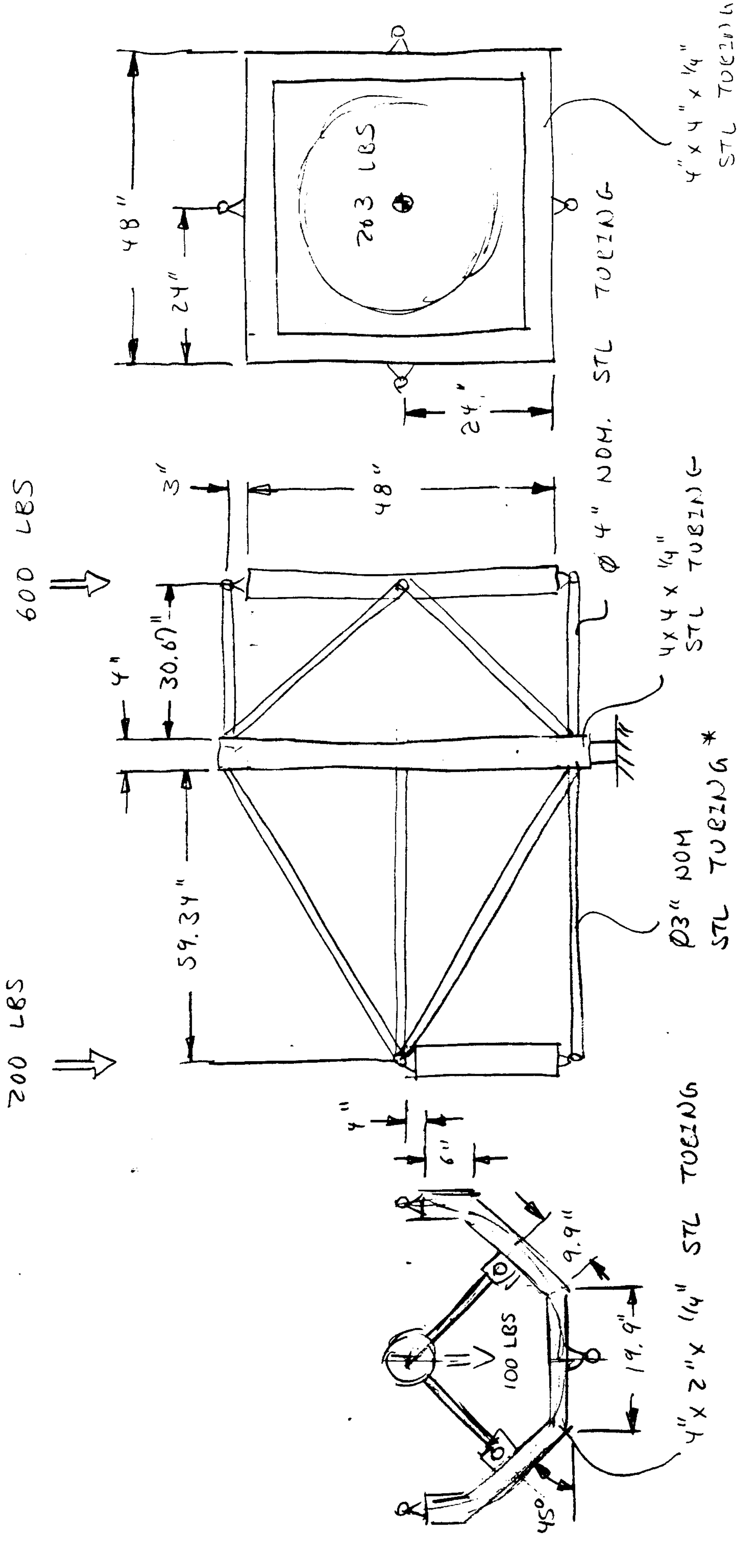
$$= .699 \simeq .7$$

$A_2 = .7\,A_1$

COMBO #1: $A_1 = \varnothing$ 4.50 O.D.
$\varnothing$ 4.026 I.D.
3.17 IN^2
$A_2 = \varnothing$ 3.50 O.D
$\varnothing$ 3.068 I.D
2.23 IN^2
$\epsilon = 0.993$

COMBO #2: $A_1 = \varnothing$ 2.875 O.D.
$\varnothing$ 2.469 I.D.
1.70 IN^2
$A_2 = \varnothing$ 2.375 O.D.
$\varnothing$ 2.067 I.D.
1.07 IN^2
$\epsilon = 0.900$

USNOI TRUSS 1 FEB 91 DAN VUKOBRATOVICH 3/3



* NOTE: LOWER HORIZONTAL TRIANGLE ONLY ON SECONDARY TRUSS SIDE

A-G DAVIS

Gagemakers

JANUARY 8, 1992

UNIVERSITY OF ARIZONA
P.O. BOX 40370
TUCSON, AZ 85717
602-621-1747

ATTN: MR. DANIEL VUKOBRATOVICH,
ASSOC. RESEARCH SCIENTIST

SUBJECT: CONCEPTUAL DESIGN STUDY FOR THE SIDEROSTAT OF THE OSNO STELLAR INTERFEROMETER
P.O. NO. 190462 DATED 11-08-91

DEAR MR. VUKOBRATOVICH:

IN REFERENCE TO SUBJECT DESIGN STUDY AND RECENT MEETINGS, A.G. DAVIS IS PLEASED TO OFFER THE FOLLOWING INFORMATION FOR YOUR EVALUATION.

THE ENCLOSED OUTLINE DRAWINGS ILLUSTRATES THE FEASIBILITY OF UTILIZING GAS BEARINGS FOR THE AZIMUTH AND ELEVATION BEARING, PLUS A SPECIAL GAS BEARING FLOAT SYSTEM FOR ALIGNMENT OF MIRROR CENTER IN RESPECT TO THE AZIMUTH BEARINGS.

TRUNNION BEARINGS (ELEVATION)

THE TRUNNION JOURNAL GAS BEARING WILL SUPPORT THE MIRROR CRADLE CASTING. THE LEFT SIDE JOURNAL BEARING WILL INCLUDE A BI-DIRECTIONAL GAS THRUST AND RADIAL GAS BEARING. THE LEFT TRUNNION SHAFT WILL INCLUDE MOUNTING HARDWARE FOR THE ELEVATION AXIS ENCODER. THE RIGHT SIDE JOURNAL GAS BEARING WILL HOUSE AN INTEGRAL SERVO DRIVE MOTOR.

UNIVERSITY OF ARIZONA
MR. DANIEL VUKOBRATOVICH
JANUARY 8, 1992

SPECIFICATIONS

AIR REQUIREMENTS:
AIR PRESSURE.................80 P.S.I.
AIR FLOW....................3.6 CU.FT/MIN.
AIR FILM THICKNESS:
THRUST (AXIAL)...............0.000400 INCH
RADIAL (JOURNAL).............0.000800 INCH
LOAD CAPACITIES:
VERTICAL (WEIGHT)............5200 LBS.
AXIAL (TRUNNION).............500 LBS.
STIFFNESS:
RADIAL.......................14.0 LBS/MICROINCH
AXIAL........................5.8 LBS/MICROINCH
MOMENT.......................2000 LB.FT/ARC SECOND
BEARING:*
RADIAL.......................4 MICROINCHES
AXIAL........................2 MICROINCHES

ELEVATION (TRUNNION) BEARING CENTERING HARDWARE

TO ACHIEVE THE ELEVATION PRELOADED GAS BEARING ASSEMBLY ON THE ELEVATION SUPPORT SYSTEMS THERE ARE TWO CONSIDERATIONS. THE FIRST ELEVATION BEARING ALIGNMENT AND SECOND MIRROR CENTERING ALIGNMENT. THE ELEVATION BEARING ALIGNMENT WILL BE ACCOMPLISHED BY MANUFACTURING PROCESS. A GAS FLOAT TYPE BEARING WILL BE INCLUDED BETWEEN THE TRUNNION BEARINGS AND THE BEARING SUPPORT STRUCTURE FOR ASSISTANCE IN X-Y MOVEMENT OF THE CRADLE STRUCTURE WITH MECHANICAL ADJUSTMENTS TO OBTAIN THE REQUIRED CENTER ALIGNMENT IN REFERENCE TO THE AZIMUTH CENTER.

AZIMUTH BEARING

THE AZIMUTH PRELOADED GAS BEARING WILL MEET THE BELOW LISTED SPECIFICATIONS.

AIR REQUIREMENTS:
AIR PRESSURE.................80 P.S.I.
AIR FLOW....................5 CU.FT/MIN.
AIR FILM THICKNESS:
THRUST (AXIAL)...............0.000500 INCH
RADIAL (JOURNAL).............0.000560 INCH
LOAD CAPACITIES:
THRUST (WEIGHT)..............11000 LBS.
RADIAL.......................2500 LBS.
STIFFNESS:
RADIAL.......................10.0 LBS./MICROINCH
AXIAL........................72.0 LBS./MICROINCH
MOMENT.......................4550 LB.FT/ARC SECOND

BEARING MOTION:*
RADIAL........................8 MICROINCHES
AXIAL.........................4 MICROINCHES
TILT..........................0.1 ARC SECOND

* NOTE-AIR BEARING CONTRIBUTION ONLY. DOES NOT INCLUDE STRUCTURE FLEXURE.

THE AZIMUTH BEARING WILL BE DESIGNED AND WILL HOUSE BOTH THE INTEGRAL SERVO DRIVE MOTOR AND THE AZIMUTH ENCODER, WITH ATTACHMENT MEANS TO THE SIDEROSTAT PIER. A FLATNESS OF .0004 T.I.R. WILL BE REQUIRED FOR THIS SURFACE. IF THIS FLATNESS IS NOT FEASIBLE A THREE POINT MEEHANITE CAST IRON BASE CAN BE MANUFACTURED TO INTERFACE BETWEEN THE PIER AND THE AZIMUTH BEARING ASSEMBLY. THIS CASTING WOULD BE APPROXIMATELY 18 INCHES THICK. PROPERLY RIBBED AND GUSSETTED TO MAINTAIN THE REQUIRED STABILITY AND FLATNESS.

IN SUMMARY, THE ABOVE CONCEPT IS A FEASIBLE DESIGN TO FULFILL THE SIDEROSTAT BEARING SPECIFICATIONS.

HOWEVER, THE BELOW LISTED THREE (3) ITEMS ARE POSSIBLE PROBLEM AREAS AND ARE OF CONCERN.

1. **ELEVATION AXIS GRAVITATIONAL SYMMETRY:**

THE RUNOUT OF THE MIRROR CENTER WITH RESPECT TO THE ELEVATION AXIS IS CRITICAL. THIS RUNOUT COMES FROM MANY SOURCES OTHER THAN THE AIR BEARINGS. THE MAJOR RUNOUT ERROR CONTRIBUTION WILL COME FROM THE TRUNNION CRADLE CASTING. THE CRADLE CASTING ROTATES WITH RESPECT TO GRAVITY (MIRROR HORIZONTAL TO MIRROR VERTICAL). THE CRADLE CASTING WILL OBVIOUSLY SAG MORE WHEN THE MIRROR IS HORIZONTAL THEN WHEN IT IS VERTICAL THIS IS DUE TO THE CASTING STRENGTH DIFFERENCE BETWEEN THE HORIZONTAL AND VERTICAL PLANES. WE RECOMMEND THAT THE CRADLE DESIGN BE COUNTERBALANCED IN SUCH A WAY THAT IF THE CRADLE WAS CUT IN HALF PERPENDICULAR TO THE ELEVATION AXIS THE TWO HALVES WOULD BALANCE AT THE CENTER OF THEIR RESPECTIVE JOURNAL BEARINGS. THESE COUNTERWEIGHTS MUST ALSO BALANCE THE TRUNNION FOR TORQUE AROUND ITS AXIS.

2. **MIRROR ALIGNMENT VS ELEVATION BEARING ALIGNMENT:**

THE REQUIREMENT FOR FIELD MIRROR ALIGNMENT TO THE ELEVATION AXIS CONFLICTS WITH THE CRITICAL MANUFACTURING ALIGNMENT REQUIRED FOR THE JOURNAL BEARINGS. THESE CRITICAL ALIGNMENTS MUST BE INDEPENDENT. WE THEREFORE RECOMMEND THAT THE FIELD MIRROR ADJUSTMENT CAPABILITY BE DESIGNED BETWEEN THE MIRROR AND THE TRUNNION CRADLE CASTING. THE JOINTS BETWEEN THE TRUNNION CRADLE CASTING AND THE JOURNAL BEARINGS MUST BE USED ONLY TO MAINTAIN THE ALIGNMENT REQUIRED BY THE BEARINGS.

3. GAS SUPPLY FAILURE:

GAS SUPPLY FAILURE DURING MOTION CAN BE A SOURCE OF TROUBLE FOR CONVENTIONAL GAS BEARINGS. WE RECOMMEND THAT A RESERVOIR BE PROVIDED NEAR THE GAS BEARING WITH APPROPRIATE ELECTRICAL SWITCHES AND INTERLOCKS TO STOP MOTION IMMEDIATELY. THE RESERVOIR WILL MAINTAIN BEARING LEVITATION FOR A SHORT PERIOD OF TIME AFTER THE FAILURE. SINCE FEW SYSTEMS ARE EVER FOOLPROOF WE ALSO RECOMMEND THAT THE MATERIALS USED IN THE GAS BEARINGS COMPLETELY PREVENT THE POSSIBILITY OF GALLING WHEN THE GAS BEARINGS LOSE GAS PRESSURE DURING MOTION.

WE HOPE THE ABOVE INFORMATION IS SATISFACTORY. HOWEVER, SHOULD YOU REQUIRE ADDITIONAL INFORMATION, PLEASE CONTACT THE WRITER.

SINCERELY,

LEE CHENOWITH
TABLES DIVISION

DS

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Design Specifications

I) Optics

A) 1.0 m siderostat flat

B) 0.75 m 6.5:1 afocal Gregorian beam collapser

C) Metrology system

II) Enironment

A) Flagstaff, AZ climate

B) "Standard" earthquake zone

C) Worst case assumptions

Design Specification Continued

III) Siderostat

A) 0.1 wave P-V (1 wave = 633 nm) mirror figure

B) Elevation: +10° to +85° Azimuth: ± 60°

C) Bearing accuracy: 2.5 microns

D) Axis temperature stability
   1) 250 microns/20°C
   2) ±5 arc-sec/20°C

E) Axis stable to 20 microns in 9 m/s wind

F) Slewing speed: 2 deg/sec

G) Encoder accuracy: 20 bits = 1.24 arc-sec

IV) Beam Collapser

A) 0.1 wave P-V (1 wave = 633 nm) primary mirror figure

B) Operates at -10° elevation

C) Optical alignment stability over 15°C

D) Focus temperature stability: 4 microns/8°C
E) Alignment stable in 9 m/s wind

Project Status

I) Optical design
→ Complete

II) Environmental study
→ Complete

III) Siderostat conceptual design

A) Siderostat flat mirror
→ Complete

B) Siderostat flat cell
→ Complete

C) Bearings
→ Complete

D) Motors
→ Complete

E) Encoders
→ Selected

F) Structure
→ Two configurations studied

Project Status Continued

IV) Beam collapser conceptual design

A) Primary mirror
→ Complete

B) Primary mirror cell
→ Complete

C) Secondary Assembly
→ Complete

D) Truss
→ Complete

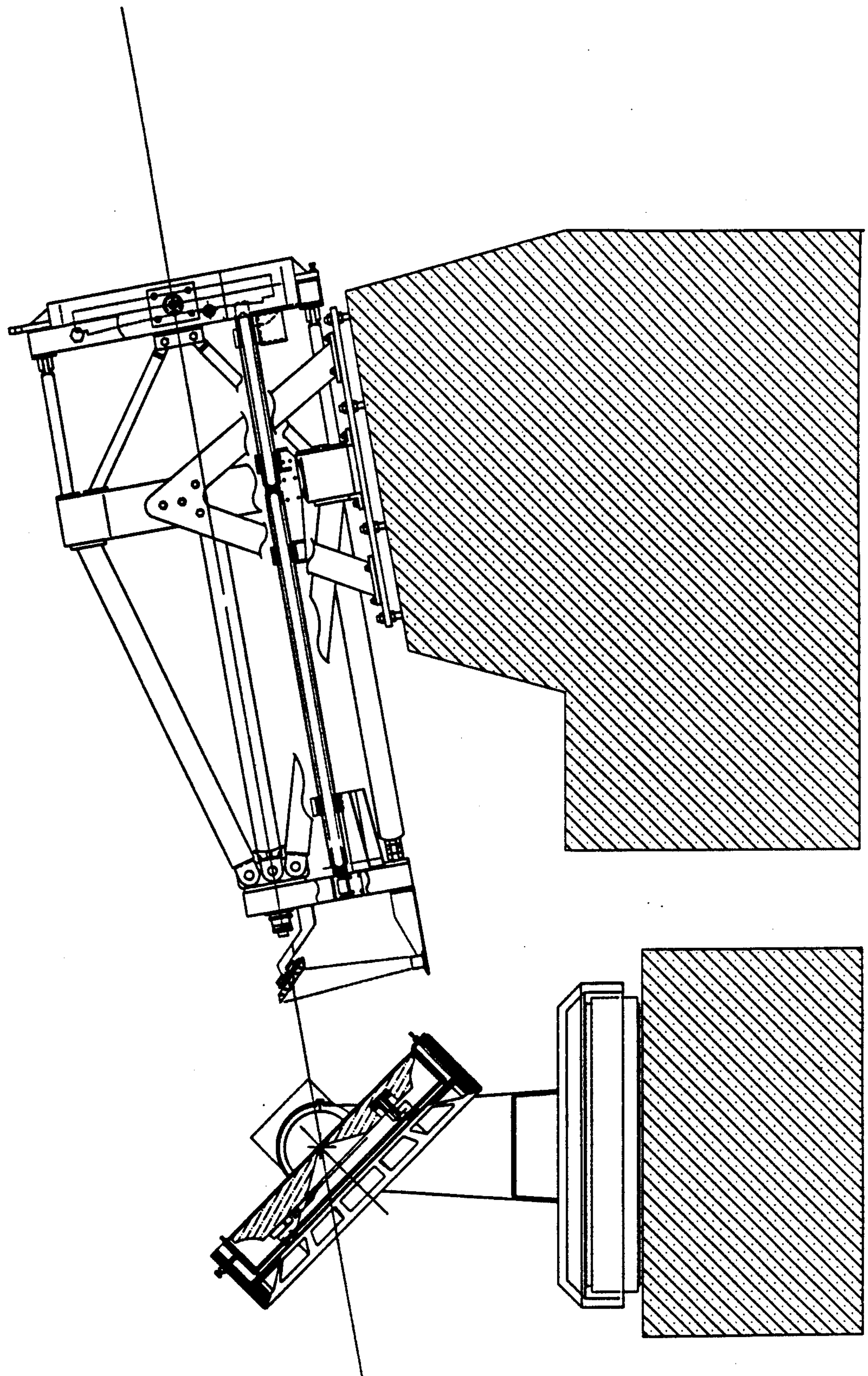
F) Pier attachment
→ Studied

V) Thermal response
→ Studied

IV) Remaining work (following USNO response)

A) Siderostat
→ Detailed drawings

B) Beam collapser
→ detailed drawings



Environmental Specification

I) Wind

A) Operate at wind speeds up to 9 m/s (20 miles/hr)
B) Survive without damage wind speeds up to 31 m/s (70 miles/hr)
(OSC recommended maximum wind speed is 100 miles/hr)

II) Temperature

A) Operate over -20°C to +27°C
(-40°F to +81°F)
(OSC recommended temperature range is -23°F to 100°F)
B) Survive temperature range of -35°C to +50°C (-31°F to +122°F)

III) Humidity

(OSC assumes 0 - 100% RH)

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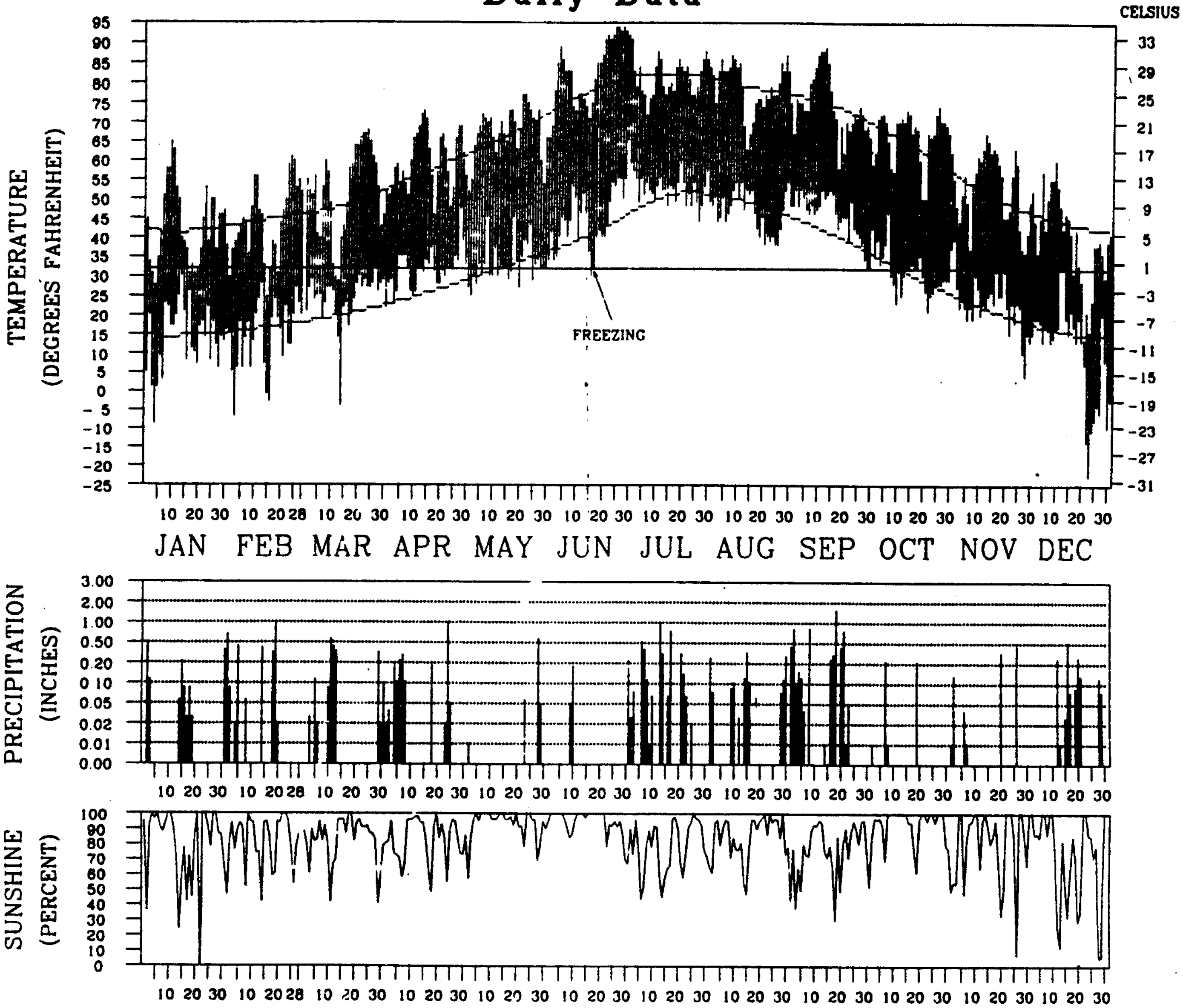
1990 LOCAL CLIMATOLOGICAL DATA
ANNUAL SUMMARY WITH COMPARATIVE DATA
FLAGSTAFF, ARIZONA

Daily Data



TEMPERATURE DEPICTS NORMAL MAXIMUM, NORMAL MINIMUM AND ACTUAL DAILY HIGH AND LOW VALUES (FAHRENHEIT)
PRECIPITATION IS MEASURED IN INCHES. SCALE IS NON-LINEAR
SUNSHINE IS PERCENT OF THE POSSIBLE SUNSHINE

I CERTIFY THAT THIS IS AN OFFICIAL PUBLICATION OF THE NATIONAL OCEANIC AND ATMOSPHERIC ADMINISTRATION, AND IS COMPILED FROM RECORDS ON FILE AT THE NATIONAL CLIMATIC DATA CENTER, ASHEVILLE, NORTH CAROLINA, 28801

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NATIONAL OCEANIC AND ATMOSPHERIC ADMINISTRATION

NATIONAL ENVIRONMENTAL SATELLITE, DATA AND INFORMATION SERVICE

NATIONAL CLIMATIC DATA CENTER ASHEVILLE NORTH CAROLINA

Kenneth D Hadeen
DIRECTOR
NATIONAL CLIMATIC DATA CENTER

METEOROLOGICAL DATA FOR 1990

FLAGSTAFF, ARIZONA

LATITUDE: 35 °08' N LONGITUDE: 111 °40' W ELEVATION: FT. GRND 7006 BARO 6997 TIME ZONE: MOUNTAIN WBAN: 03103

	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEP	OCT	NOV	DEC	YEAR
TEMPERATURE °F:													
Averages													
-Daily Maximum	43.4	44.5	52.6	60.4	67.6	83.9	81.6	78.9	74.4	66.5	53.0	40.1	62.2
-Daily Minimum	13.7	14.0	23.9	31.7	32.9	44.9	52.1	46.3	45.2	31.2	22.2	10.1	30.7
-Monthly	28.6	29.3	38.3	46.1	50.3	64.4	66.9	62.6	59.8	48.9	37.6	25.1	46.5
-Monthly Dewpt.	13.4	16.1	21.0	26.9	21.4	26.2	43.6	38.6	42.0	26.6	19.0	8.0	25.2
Extremes													
-Highest	65	61	68	73	77	94	91	87	89	74	67	60	94
-Date	10	24	24	14	22	28	1	29	13	26	13	10	JUN 28
-Lowest	-9	-7	-4	23	25	30	44	38	32	21	4	-23	-23
-Date	4	3	14	3	2	16	27	26	30	22	28	23	DEC 23
DEGREE DAYS BASE 65 °F:													
Heating	1124	996	822	558	449	84	10	93	167	494	816	1231	6844
Cooling	0	0	0	0	0	76	76	28	18	0	0	0	198
% OF POSSIBLE SUNSHINE	79	82	83	83	93	95	78	85	76	92	79	72	84
AVG. SKY COVER (tenths)													
Sunrise - Sunset	4.5	6.0	6.5	7.0	3.6	2.4	5.7	4.5	5.8	2.0	3.5	4.3	4.7
Midnight - Midnight													
NUMBER OF DAYS:													
Sunrise to Sunset													
-Clear	13	9	9	5	17	21	6	13	8	24	16	16	157
-Partly Cloudy	9	5	3	7	9	6	16	13	9	3	8	5	93
-Cloudy	9	14	19	18	5	3	9	5	13	4	6	10	115
Precipitation .01 inches or more	9	9	10	11	4	2	16	11	15	4	6	10	107
Snow, Ice pellets 1.0 inches or more	6	7	5	2	1	0	0	0	0	0	4	6	31
Thunderstorms	0	0	1	5	2	1	15	12	13	2	1	0	52
Heavy Fog, visibility 1/4 mile or less	1	0	0	0	0	0	0	0	3	0	2	1	7
Temperature °F													
-Maximum													
90° and above	0	0	0	0	0	10	1	0	0	0	0	0	11
32° and below	5	4	2	0	0	0	0	0	0	0	0	7	18
-Minimum													
32° and below	31	28	31	19	18	2	0	0	2	21	29	31	212
0° and below	1	3	1	0	0	0	0	0	0	0	0	8	13
AVG. STATION PRESS. (mb)													
RELATIVE HUMIDITY (%)													
Hour 05 (Local Time)	71		74		54	40	61		73	66	66	64	
Hour 11 (Local Time)	47	49	42	40	25	19	36	34	44	33	42	46	38
Hour 17 (Local Time)	45	45	40	37	23	15	40	37	43	30	38	45	37
Hour 23 (Local Time)	63		66	59		29			69	59	60	63	
PRECIPITATION (inches):													
Water Equivalent													
-Total	1.54	3.20	2.17	2.32	0.73	0.24	4.32	1.71	6.18	0.49	1.09	1.68	25.67
-Greatest (24 hrs)	0.61	1.25	0.81	1.03	0.66	0.24	1.00	0.37	1.93	0.25	0.53	0.62	1.93
-Date	2- 3	18-19	12-13	23-24	28-29	9-10	13	15-16	17-18	7- 8	26	16-17	SEP 17-18
Snow, Ice pellets													
-Total	24.2	45.5	25.0	4.2	1.4	0.0	T	T	T	T	9.6	22.3	132.2
-Greatest (24 hrs)	8.1	15.2	13.1	2.8	1.4	0.0	T	T	T	T	6.4	8.4	15.2
-Date	2- 3	1- 2	12-13	24	28-29		22	10	18	19	26	16-17	FEB 1- 2
WIND:													
Resultant													
-Direction (!!!)	181	208	231	223	215	221	231	205	157	194	129	229	214
-Speed (mph)	0.6	2.5	2.2	2.9	4.7	4.1	1.8	1.2	0.5	0.8	1.0	1.1	1.8
Average Speed (mph)	5.0	5.6	4.9	5.5	6.7	5.7	4.0	4.6	3.4	4.5	5.3	5.2	5.0
Fastest Obs. 1 Min.													
-Direction (!!!)													
-Speed (mph)													
-Date													
Peak Gust													
-Direction (!!!)													
-Speed (mph)													
-Date													

(!!!) See Reference Notes on Page 68

NORMALS, MEANS, AND EXTREMES

FLAGSTAFF, ARIZONA

LATITUDE: 35°08'N LONGITUDE: 111°40'W ELEVATION: FT. GRND 7006 BARO 6997 TIME ZONE: MOUNTAIN WBAN: 03103

	(a)	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEP	OCT	NOV	DEC	YEAR
TEMPERATURE °F:														
Normals														
-Daily Maximum		41.7	44.5	48.6	57.1	66.7	77.6	81.9	78.9	74.1	63.7	51.0	43.6	60.8
-Daily Minimum		14.7	16.9	20.4	25.9	32.9	40.9	50.3	48.7	40.9	30.6	21.5	15.9	30.0
-Monthly		28.2	30.7	34.5	41.6	49.9	59.2	66.1	63.8	57.5	47.2	36.3	29.8	45.4
Extremes														
-Record Highest	41	66	71	73	80	87	96	97	92	90	85	74	68	97
-Year		1971	1986	1988	1989	1974	1970	1973	1978	1950	1980	1977	1950	JUL 1973
-Record Lowest	41	-22	-23	-16	-2	14	22	32	24	23	-2	-13	-23	-23
-Year		1971	1985	1966	1975	1975	1955	1955	1968	1971	1971	1958	1990	DEC 1990
NORMAL DEGREE DAYS:														
Heating (base 65°F)		1141	960	946	702	468	194	34	76	229	552	861	1091	7254
Cooling (base 65°F)		0	0	0	0	0	20	68	39	0	0	0	0	127
% OF POSSIBLE SUNSHINE	11	77	73	77	82	89	85	74	77	81	79	76	73	79
MEAN SKY COVER (tenths)														
Sunrise - Sunset	39	5.3	5.2	5.2	4.7	4.0	3.0	5.4	5.0	3.7	3.6	4.1	4.7	4.5
MEAN NUMBER OF DAYS:														
Sunrise to Sunset														
-Clear	39	12.4	11.1	11.7	12.6	15.5	18.6	8.9	10.2	15.7	17.1	15.5	14.0	163.1
-Partly Cloudy	39	6.4	6.1	7.9	8.5	9.2	7.7	13.1	12.9	9.6	7.1	6.5	6.5	101.4
-Cloudy	39	12.3	11.1	11.4	8.9	6.3	3.7	9.1	7.9	4.8	6.8	8.1	10.5	100.7
Precipitation .01 inches or more	41	7.4	6.7	8.3	5.8	4.2	2.8	11.8	11.3	6.6	4.8	5.3	6.3	81.3
Snow, Ice pellets 1.0 inches or more	40	4.2	3.9	4.9	2.4	0.6	0.0	0.0	0.0	0.*	0.5	2.0	3.5	22.0
Thunderstorms	30	0.*	0.3	0.6	1.3	2.6	3.7	16.6	15.7	6.7	2.2	0.7	0.2	50.5
Heavy Fog Visibility 1/4 mile or less	30	1.8	1.8	1.6	1.2	0.2	0.*	0.1	0.3	0.5	0.9	1.1	1.8	11.4
Temperature °F														
-Maximum														
90° and above	41	0.0	0.0	0.0	0.0	0.0	1.3	1.5	0.3	0.*	0.0	0.0	0.0	3.2
32° and below	41	4.6	2.7	1.7	0.2	0.0	0.0	0.0	0.0	0.0	0.1	1.1	4.5	15.0
-Minimum														
32° and below	41	30.4	27.6	30.0	25.0	14.1	2.9	0.1	0.1	2.9	18.5	27.9	30.3	209.8
0° and below	41	3.4	1.5	0.7	0.*	0.0	0.0	0.0	0.0	0.0	0.*	0.5	2.2	8.3
AVG. STATION PRESS. (mb)	5	786.7	786.9	783.0	784.8	786.4	789.2	791.7	791.6	790.6	789.7	788.2	787.6	788.0
RELATIVE HUMIDITY (%)														
Hour 05	33	73	73	71	66	63	54	68	70	73	71	70	71	69
Hour 11 (Local Time)	35	53	50	44	35	29	23	34	40	38	38	44	50	40
Hour 17	27	50	45	40	32	26	21	39	43	37	36	42	51	39
Hour 23	22	67	64	61	54	47	40	60	68	64	63	63	71	60
PRECIPITATION (inches):														
Water Equivalent														
-Normal		2.10	1.95	2.13	1.35	0.75	0.57	2.47	2.62	1.47	1.54	1.65	2.26	20.86
-Maximum Monthly	41	6.52	7.81	6.75	5.62	2.16	2.92	6.62	8.06	6.75	9.86	6.64	7.30	9.86
-Year		1980	1980	1970	1965	1979	1955	1986	1986	1983	1972	1985	1967	OCT 1972
-Minimum Monthly	41	0.00	T	T	0.01	T	0.00	0.32	0.26	T	T	T	T	0.00
-Year		1972	1967	1972	1989	1974	1971	1963	1962	1973	1952	1989	1958	JAN 1972
-Maximum in 24 hrs	41	2.10	2.53	2.96	1.79	1.11	2.79	2.55	3.04	3.43	2.73	3.69	3.11	3.69
-Year		1979	1980	1970	1985	1965	1956	1964	1936	1965	1972	1978	1951	NOV 1978
Snow, Ice pellets														
-Maximum Monthly	40	63.4	45.5	77.4	58.3	8.2	T	T	T	2.0	24.7	40.7	86.0	86.0
-Year		1980	1990	1973	1965	1975	1955	1990	1990	1965	1971	1985	1967	DEC 1967
-Maximum in 24 hrs	40	23.1	23.1	26.3	17.2	6.6	T	T	T	2.0	13.5	18.4	27.3	27.3
-Year		1980	1987	1970	1977	1965	1955	1990	1990	1965	1974	1985	1967	DEC 1967
WIND:														
Mean Speed (mph)	23	7.0	6.9	7.5	7.9	7.5	7.1	5.7	5.3	6.0	6.1	7.1	7.0	6.8
Prevailing Direction through 1963		NE	S	SSW	SSW	SSW	SSW	SSW	S	S	N	NNE	NE	SSW
Fastest Mile														
-Direction (!!!)	11	SW	SW	SW	SW	SW	SW	NW	SW	W	NW	SW	NE	SW
-Speed (MPH)	11	38	34	37	40	46	35	39	30	33	34	39	38	46
-Year		1975	1980	1974	1974	1975	1984	1976	1978	1974	1978	1978	1982	MAY 1975
Peak Gust														
-Direction (!!!)														
-Speed (mph)														
-Date														

(!!!) See Reference Notes on Page 6B

Key Siderostat Specifications

I) Range of Motion

A) Elevation: $+10^{\circ}$ to $+ 85^{\circ}$

B) Azimuth: ± 60

C) No mechanical interference with metrology beams

II) Bearings

A) Runout: $\approx$ 2.5 microns

B) Axis alignment: 10 microns

C) Total axis error: 100 microns

D) Perpindicularity of axes: 05 arc-sec

III) Environment

A) Operational temperature range: $-35^{\circ}C$ to $+50^{\circ}C$

B) Operational wind speed: 9 m/s

C) Operational humidity: 100% RH (OSC)

Key Siderostat Specifications Continued

IV) Structural

A) Vertical change due to temperature change: 250 microns

B) Static deflection: 20 microns

C) Fundamental frequency: 10 Hz

SIDEROSTAT MIRROR -- DESIGN SPECIFICATION

- **MIRROR MATERIAL:** CORNING ULE
- **MIRROR DIAMETER:** 1.1 M MECHANICAL, 1.0 M CLEAR APERTURE
- **MIRROR THICKNESS:** AS REQUIRED BY DESIGN
- **MIRROR WEIGHT:** LESS THAN 170 KG
- **ORIENTATION:** +10 TO +85 DEGREES
- **CENTRAL HOLE:** 50 MM DIAMETER,
  CLEAR 120 DEGREE ACCESS FROM BACK
  WITH CLEARANCE FOR PASSAGE OF
  3.0 MM DIAMETER BEAM
- **CAT'S EYE:** ATTACHMENT WITH NON DESTRUCTIVE ADHESIVE
- **GROUND ANNULUS:** REAR SURFACE AROUND CENTRAL HOLE, FINE
  GROUND FLAT. ANNULUS PARALLEL TO FRONT
  SURFACE WITHIN 1 ARCMINUTE
- **SURFACE CURVATURE:** FLAT
- **SURFACE ACCURACY:** .05 λ P-V OVER ANY 30 CM DIA.
  .10 λ P-V OVER CLEAR APERUTRE
  (BOTH EXCLUDE 0.06 M RADIUS FROM CENTER
- **COSMETIC:** 40 - 20 SCRATCH DIG OR BETTER

1.1 M FLAT MIRROR
USNO INTERFEROMETER - SUMMARY OF OPTICAL DEFLECTIONS

Mirror Material = ULE : E =9.8 E 6 psi, ν = 0.17, ρ = 0.08 $\frac{lb}{in^3}$

Mirror supported on back side with whiffle tree and socket supports

Optical deflections as calculated by PCFRINGE and expressed in waves, λ = 0.633 micron

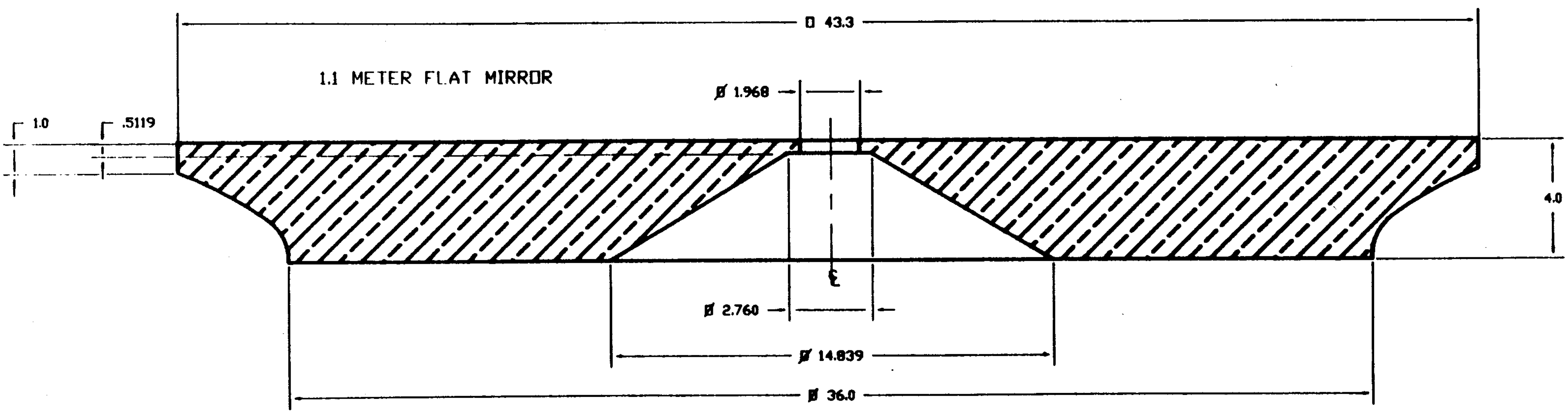
MODEL	THICK. (in.)	SUPPORT PTS.	ORIENTATION (degrees)	OPTICAL DEFLECTION (P-V)	(RMS)
1	4.0	ring at r =.650	0	.437	.166
1	4.0	ring at r =.635	0	.272	.088
2	5.2	ring at r =.635	0	.170	.056
2	5.2	ring at r =.600	0	.376	.135
3	6.0	ring at r =.635	0	.126	.041
1	4.0	r =.407 (6pt), r =.770 (6pt)	0	.266	.055
2	5.2	r =.499 (6pt), r =.770 (6pt)	0	.329	.067
3	6.0	r =.566 (6pt), r =.770 (6pt)	0	.279	.060
1	4.0	r =.407 (12pt), r =.770 (12pt)	0	.065	.023
1	4.0	r =.407 (12pt), r =.770 (12pt) 12 sockets at r =.589	90	.046	.007
1	4.0	r =.407 (12pt), r =.770 (12pt) 6 sockets at r =.589	90	.053	.007
1	4.0	r =.407 (12pt), r =.770 (12pt) 3 sockets at r =.589	90	.072	.010
1	4.0	r =.407 (12pt), r =.770 (12pt) 3 sockets at r =.589 2 lb cat's eye weight added	0	.100	.020
1	4.0	r =.407 (12pt), r =.770 (12pt) 3 sockets at r =.589 2 lb cat's eye weight added	90	.073	.010

Tolerances: Weight : 374 lb, Optical Deflection : .1λ P-V

Mirror Weights : 4 inch thick = 382 lb, 5 inch thick = 481 lb, 6 inch thick = 540 lb

SIDEROSTAT MIRROR

FINAL DESIGN



1.1 M SIDEROSTAT MIRROR

SUPPORT LOCATIONS

MIRROR ORIENTATION : +10 TO +85 DEGRESS

RADIAL SUPPORT -- 3 INTERNAL SOCKETS

-- LOCATED FROM MIRROR BACK SURFACE

AT R =.589

AXIAL SUPPORT -- WHIFFLE TREE SUPPORT

-- LOCATED AGAINST MIRROR BACK SURFACE

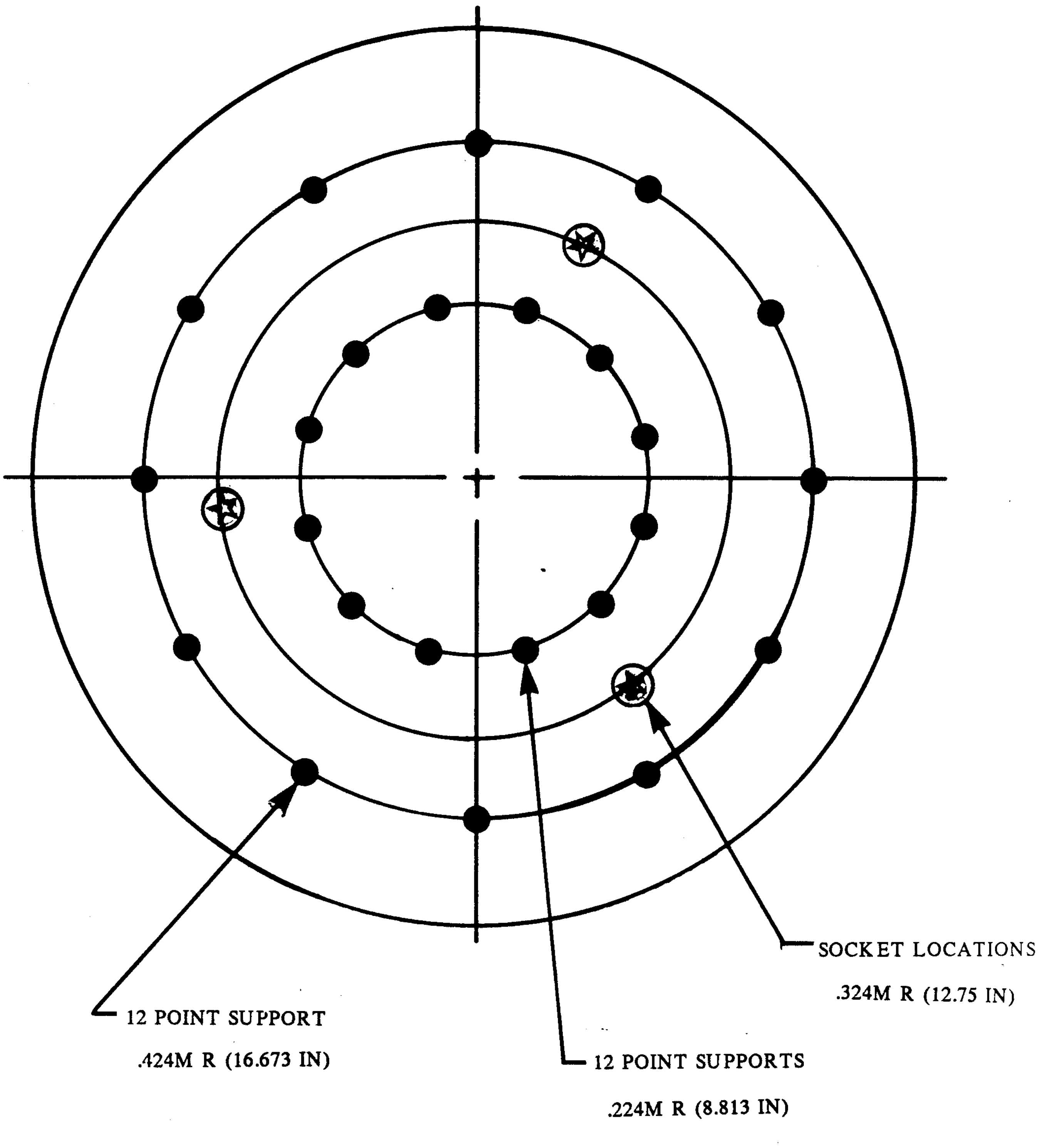
-- 24 POINTS OF SUPPORT

12 POINTS AT R =.407

12 POINTS AT R =.770

SIDEROSTAT MIRROR -- SUPPORT LOCATIONS

1.1 M (43.307 IN) DIAMETER SIDEROSTAT MIRROR



SIDEROSTAT PRIMARY MIRROR MOUNT

MOUNTING SCHEMES

COUNTERWEIGHT LEVER MECHANISM

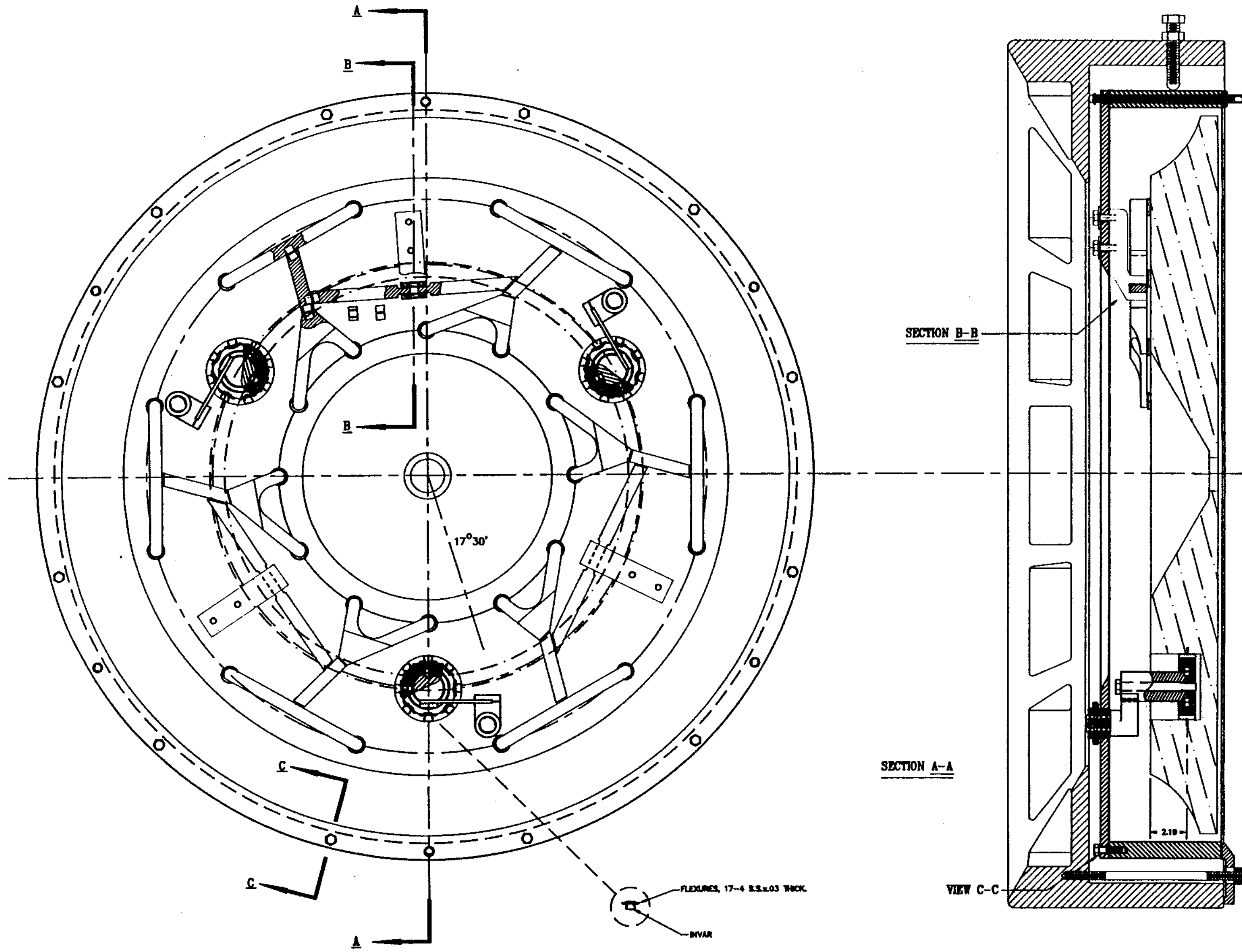
- -- LOW NATURAL FREQUENCY (PENDULUM MECHANISM)
- -- COMPLEX
- -- HIGH WEIGHT

AIR BAG SUPPORT

- -- COMPLEX AIR HANDLING SYSTEM
- -- SLOW TO RESPOND

WHIFFLE TREE WITH RADIAL SOCKET
(BIJAN IRANINEJAD DESIGN)

- -- VERY STIFF
- -- HIGH NATURAL FREQUENCY
- -- LOW IN WEIGHT


A
B
B
17°30'
C
C
A
FLEXURES, 17-4 S.S. x .03 THICK.
INVAR
SECTION B-B
SECTION A-A
VIEW C-C
2.19

SIDEROSTAT SOCKET DESIGN

- **VERTICAL HEIGHT ADJUSTMENT PROVIDED TO INSURE ZERO AXIAL LOAD IN RADIAL SOCKETS (3 PLACES)**

- **LEAF SPRINGS (.03 " THICK) ALLOW FOR THERMAL CONTRACTION AND EXPANSION OF INVAR RING TO KEEP STRESS IN THE SOCKET TO A MINIMUM.**

	MIRROR ON BACK	MIRROR ON EDGE
ΔT =(72 to -31 degrees)	δ =-5.8E-5 in	δ =-7.5E-5 in
	σ =380 psi	σ =490 psi
ΔT =(72 to 122 degrees)	δ =2.8E-5 in	δ =1.1E-5 in
	σ =185 psi	σ =75.2 psi

Where:

δ is the CTE mismatch effect between the invar ring and socket including gravit effects (leaf spring deflection)

σ is the stress in the socket due to δ

- **THREE SINGLE BLADE FLEXURES (17-4 PH SS) PROVIDED TO COUNTERACT MOMENTS DUE TO RADIAL CONTRACTION AND EXPANSION OF THE CELL**

  FLEXURE SIZE: 1" high X .1875" thick X 2" long

  σ MAX = 40,000 psi (Endurance limit/2)

  LATERAL LOAD CARRIED BY FLEXURE = 230 lbs

  MAX LATERAL DEFLECTION (THERMAL) = 9.5E-3 in

Mirror Deformations Due to Thermal Expansion of Inserts Bonded to Glass

Bijan Iraninejad, Jacob Lubliner, Terry Mast, and Jerry Nelson

Department of Civil Engineering, Space Sciences Laboratory,
Astronomy Department, and Lawrence Berkeley Laboratory,
University of California, Berkeley, California 94720

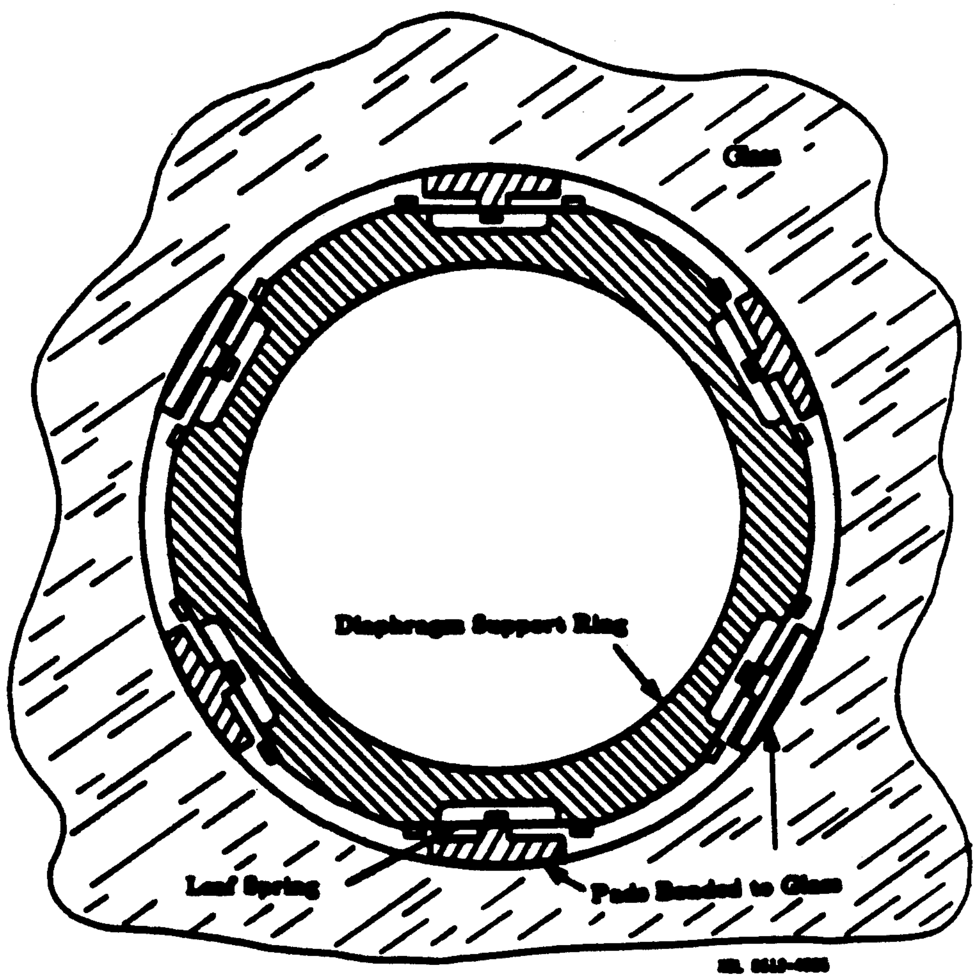
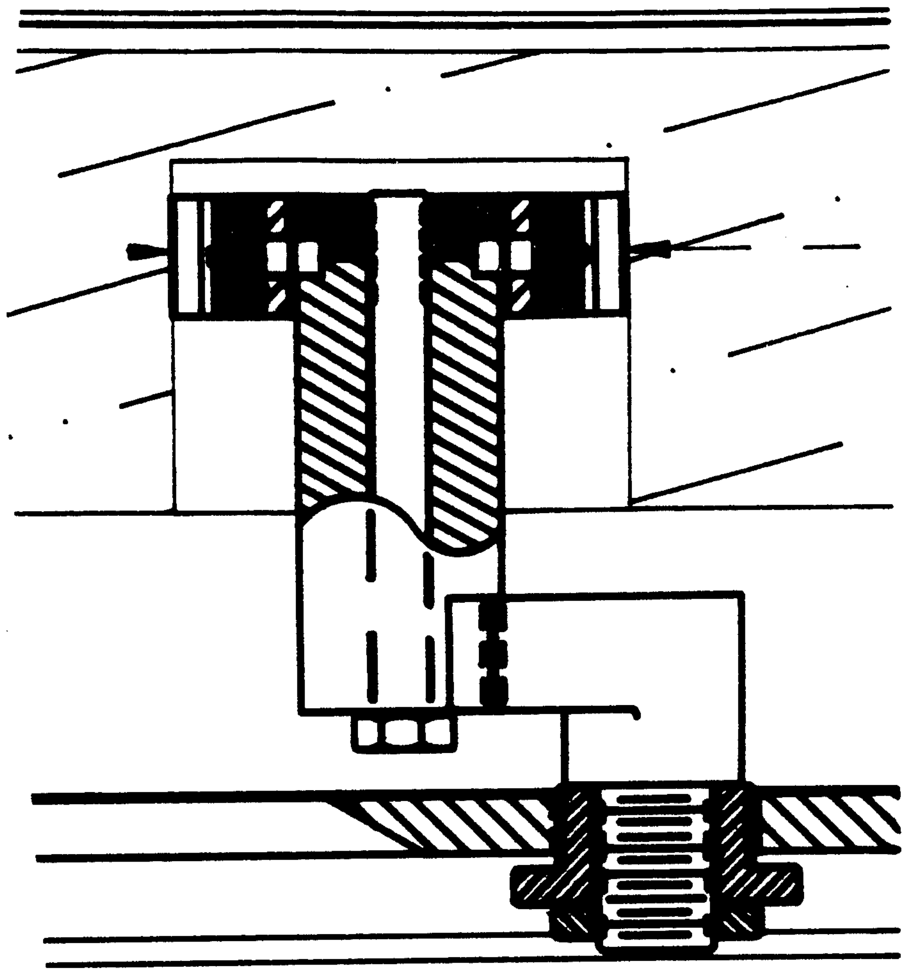
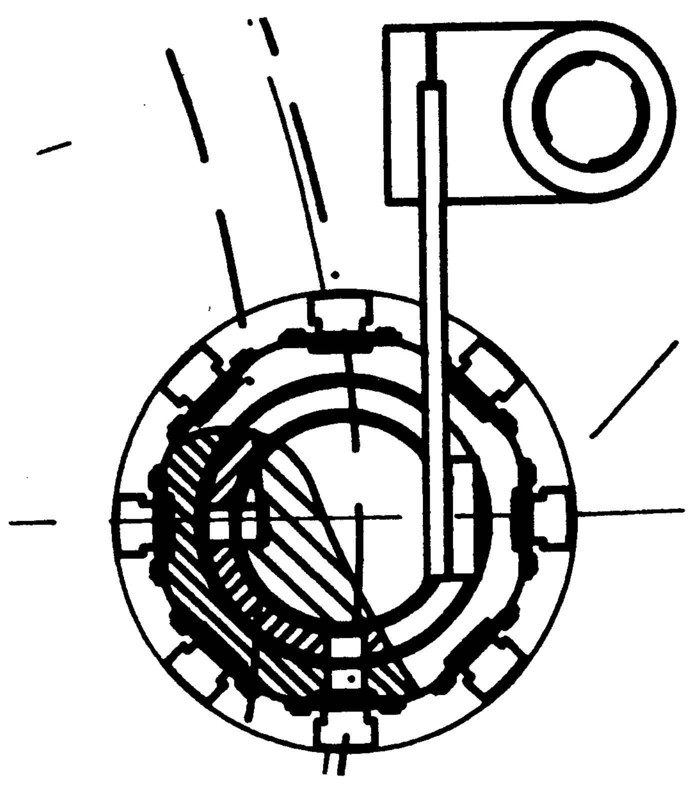


Figure 7. Radial Support Assembly, Final Design.





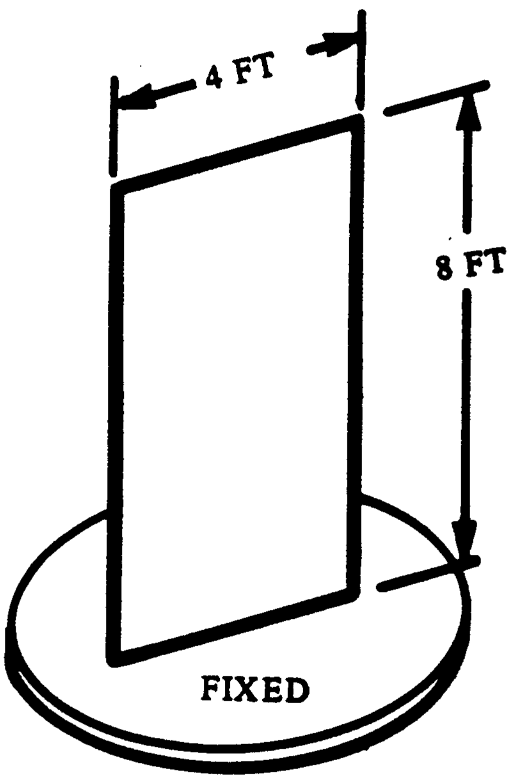
SIDEROSTAT BEARINGS -- DESIGN SPECIFICATIONS

- **MOTION:** AZIMUTH +/- 60 DEGREES
  ELEVATION +10 TO +85 DEGREES
  (RELATIVE TO HORIZON)
  (OSC RECOMMENDATION:
  ELEVATION 0 TO +90 DEGREES)

- **ACCURACY:** 2.5 μm REPEATABILITY
  1 μm TO 3 μm RUN OUT ERROR
  1 μm TO 3 μm PISTON ERROR

- **ESTIMATED STIFFNESS REQUIRED TO OVERCOME DRAG:**

ASSUME:



DRAG FORCE : $D = Q \times A$

DRAG MOMENT : $Q = \frac{1}{2} C_D \rho V^2$

REQUIRED STIFFNESS :

$$K = 81.3 \times 10^3 \ \frac{LB}{IN}$$

$$M = 10.6 \times 10^6 \ \frac{FT}{LB\text{-}RAD}$$

$C_D \approx 2$

$\rho = 0.002378 \ \frac{SLUGS}{FT^3}$

$V = 20 \text{ MPH} = 29 \ \frac{FT}{SEC}$

Siderostat Bearing Study

I) Dover Instrument Company (air bearings)
Contact: Stephen L. Hero
Declined study opportunity

II) Professional Instrument Incorporated (air bearings)
Contact: Dan Oss
Declined study opportunity

III) Rank Taylor Hobson, Incorporated (oil bearings)
Contact: William M. Mroz
Declined study opportunity

IV) A.G. Davis Gage & Engineering Company (air bearings)
Contact: Dan (Pie) Pieczulewski
Performed design study

V) ITI Bearing Division (rolling element bearings)
Contact: Jeff Chang
Can not meet runout tolerance

VI) Kaydon Corporation (rolling element bearings)
Contact: Kurt Sheridan
Can not meet runout tolerance

VII) Rotek Incorporated
Contact: Bob Hersko
Can not meet runout tolerance

Current Rolling Element Bearing Technology

I) Runout limit on large bearings is 300 - 100 x 10^{-6} inches

II) Moment, axial and radial stiffness can easily be met

III) For limited (90°) rotation, selective assembly and matching of runout can improve runout to about 50 x 10^{-6} inches

IV) Load capacity easily met

V) Highly non-linear friction for small incremental motions

Current Air Bearing Technology

I) OSC 60in. diameter pneumo rotary table (test)

A) Moment stiffness: 50 ft-lb/arc-sec
B) Axial stiffness: 5.08 x 10^6 lb/in
C) Radial runout: less than one micron
D) Axial runout: 0.5 micron
E) Coning error: 0.8 arc-sec

II) Professional Instruments model 10R-606 (catalog)

A) Moment stiffness: 24 ft-lb/arc-sec
B) Axial stiffness: 5 x 10^6 lb/in
C) Radial stiffness: 3 x 10^6 lb/in
D) Radial runout: less than 1 microinch
E) Axial runout: less than 1 microinch
F) Coning error: 0.02 arc-sec
G) Load capacity: 300 lbs
H) Air consumption: 4 SCFM at 150 PSI

III) USNO Air Bearing Test (report)

A) Axial runout: less than 0.1 micron
B) Radial runout: less than 0.1 micron

IV) Diamond turning spindle (SPIE paper)

A) Load capacity: 350 lbs
B) "Rotational" error: less than 0.25 micron

Prior Art In Telescope Air Bearings

I) NASA 48 In. Telescope (1974)

A) Azimuth air bearing
B) Load capacity of 30,000 lbs
C) Air film 200×10^{-6} in. thick
D) Air consumption of 200 SCFM at 45 PSIG
E) Moment Stiffness: 900 ft. lbs./arc-sec
F) Run-out: ISO $\times 10^{-6}$ in.
G) Built by Kollmorgen

II) NASA Kulper Airborne Observatory (1974)

A) Spherical 16 in. diameter air bearing
B) Load capacity of 4300 lb
C) Air film 750×10^{-6} in thick
D) Air consumption of 13 SCFM at 265 PSI
E) Radial stiffness: 13.5×10^{6} lb/in
F) Axial stiffness: 5.5×10^{6} lb/in
G) Built by Owens-Illinois

Air Bearing Problems and Solutions

I) Problem: Corrosion of air bearing surfaces requiring periodic re-surfacing

Solutions:

A) Use dimensionally stable stainless steel (17-4 PH, 440C) bearing surface material

B) Coat bearing surfaces with bonded dry film anti-corrosion lubricant

II) Problem: Air jet erosion reduces bearing stiffness

Solutions: Use replaceable low wear sapphire inlet jets

III) Problem: Bearing surfaces gall when moved in contact without air film

Solution: Coat bearing surfaces with bonded dry film anti-galling lubricant

Elevation Bearing

Specifications

Air Requirements:
- Air pressure .... 80 P.S.I.
- Air flow .... 3.6 CU. FT/MIN.

Air Film Thickness:
- Thrust (axial) .... 0.000400 INCH
- Radial (journal) .... 0.000800 INCH

Load Capacities:
- Vertical (weight) .... 5200 LBS.
- Axial .... 500 LBS.

Stiffness:
- Radial .... 14.0 LBS/MICROINCH
- Axial .... 5.8 LBS/MICROINCH
- Moment .... 2000 LB. FT/ARC SECOND

Bearing:*
- Radial .... 4 MICROINCHES
- Axial .... 2 MICROINCHES

*Note: Air bearing contribution only. Does not include structure flexure.

Azimuth Bearing

Specifications

Air Requirements:
- Air pressure .... 80 P.S.I.
- Air flow .... 5 CU. FT/MIN.

Air Film Thickness:
- Thrust (axial) .... 0.000500 INCH
- Radial (journal) .... 0.000560 INCH

Load Capacities:
- Thrust (weight) .... 11000 LBS.
- Axial .... 2500 LBS.

Stiffness:
- Radial .... 10.0 LBS/MICROINCH
- Axial .... 72.0 LBS/MICROINCH
- Moment .... 4550 LB. FT/ARC SECOND

Bearing:*
- Radial .... 8 MICROINCHES
- Axial .... 4 MICROINCHES
- Tilt .... 0.1 ARC SECOND

*Note: Air bearing contribution only. Does not include structure flexure.

2. Servo System Parameters

Maximum Slew Speeds: ≥ 2 degrees/second

Acceleration Time to Maximum Slew Speed: ≤ 5 seconds

Settling Time: ≤ 3 seconds

Absolute Pointing Accuracy: ≤ 15 arcseconds (open loop operation using pointing model, after large slews)

Pointing Repeatability: ≤ 15 arcseconds (open loop operation using pointing model, after large slews)

Fastest Tracking Speed: 15 arcseconds/second (sidereal rate)

Slowest Tracking Speed: 0.015 arcseconds/second (approximate)

Encoder Accuracy: 20 bits = 1.24 arcseconds

Encoder Precision: 24 bits/360° = 12.95 counts/arcsecond

Velocity Update Rate: 0.500 Hz

Cumulative Tracking Errors: 0.015 arcseconds

Minimum Drive Torque: (OSC estimate 17 ft - lb)

Drive Study

I) Direct drive DC torque motor

II) Lead screw drive

III) Traction drive

IV) Gear drive

Drive Recommendation

I) Direct DC Torque motor drive for both altitude and azimuth axis

II) Provide back-up brake and auxiliary manual drive

III) Use counter weights to reduce torque requirement for altitude axis

IV) If counterweights can not be used, use lead screw drive for altitude axis

Direct Drive DC Torque Motor

Advantages

I) Simple

II) Compact

III) Economical

IV) No backlash

V) Prior development

VI) No wear (except motor brushes)

Disadvantages

I) Limited torque

II) Not self-locking if power fails

III) Motor cogging may limit control at low speeds and small incremental motions

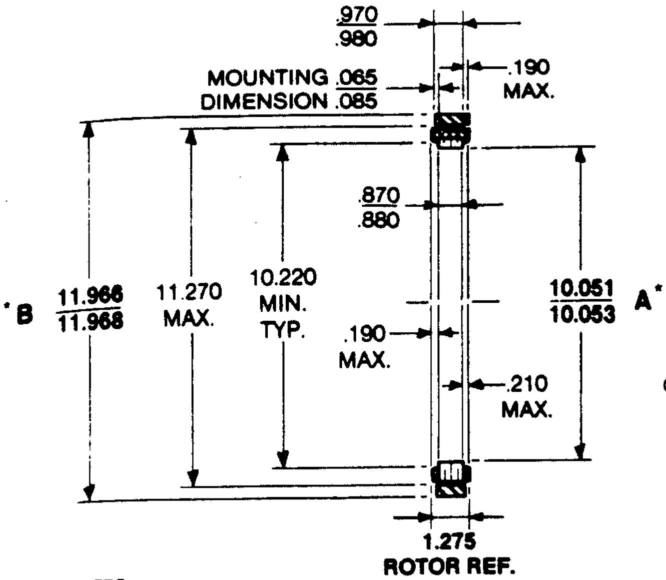
IV) No manual drive option

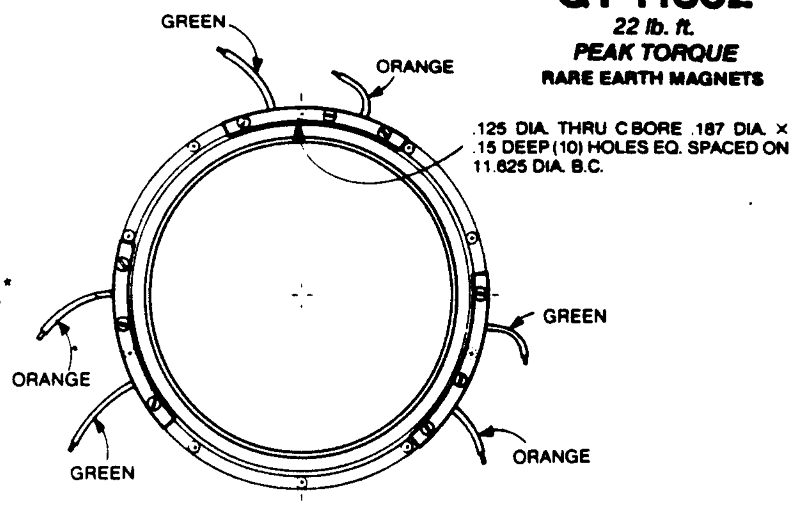
V) Los stiffness

Design Example: NASA 48 In. Telescope

QT-11302

22 lb. ft.
PEAK TORQUE
RARE EARTH MAGNETS





NOTES:

1. — MOTOR TO BE SHIPPED AS FIVE (5) SEPARATE COMPONENTS: STATOR ASSEMBLY, ROTOR ASSEMBLY, AND (3) THREE BRUSH SEGMENT ASSEMBLIES.
2. — MOUNTING REQUIREMENTS: DIAMETERS "A" AND "B" TO BE CONCENTRIC WITHIN .004 (.008 T.I.R.) WHEN MOUNTED.
3. — WITH POSITIVE CURRENT APPLIED TO GREEN LEADS, WITH RESPECT TO ORANGE LEADS, ROTATION SHALL BE C.C.W. FACING BRUSH RING END.
4. — CONNCET (3) GREEN LEADS TOGETHER AND (3) ORANGE LEADS TOGETHER FOR PROPER OPERATION.
5. — DIAMETERS MARKED "*" ARE AVERAGE OF FREE STATE.
6. — TYPICAL BRUSH LIFE > 10^7 REVS.

LEADS:
#22 AWG TEFLON COATED PER MIL W-16878, 36" MIN. LENGTH.

SIZE CONSTANTS

		Value	Units
Peak Torque Rating - T_P		22	LB. FT.
Power Input, Stalled at T_P(25°C) - P_P		232	WATTS
Motor Constant - K_M		1.44	LB.FT./ $\sqrt{\text{WATT}}$
No Load Speed, Theoretical @ V_P - ω_{NL}		7.8	RAD/S
Electrical Time Constant - τ_E		0.93	MS
Static Friction (Max.) - T_F		1.0	LB. FT.
Viscous Damping Coefficients	Zero Impedance - F_0	2.83	LB. FT. PER RAD/S
	Infinite Impedance - F_I	0.02	LB. FT. PER RAD/S
Maximum Winding Temperature		155	°C
Temperature Rise per Watt - TPR		0.5	°C/WATT
Ripple Torque (Average to Peak) - T_R		4	PERCENT
Ripple Frequency (Fundamental)		181	CYCLES/REV.
Number of Poles		40	
Rotor Inertia - J_M		0.03	LB.FT.S^2
Motor Weight		8.7	LB.

WINDING CONSTANTS

Winding Designation

	UNITS	TOLERANCES	A	B	C	D	E	F	G
Voltage, Stalled at T_P(25°C) - V_P	VOLTS	Nom.	26.4						
Peak Current - I_P	AMPERES	Rated	8.80						
Torque Sensitivity - K_T	LB.FT./AMPS	±10%	2.50						
Back EMF Constant - K_B	V per RAD/S	±10%	3.39						
DC Resistance (25°C) - R_M	OHMS	±12.5%	3.00						
Inductance - L_M	mH	±30%	2.8						

2. Servo System Parameters

Maximum Slew Speeds: $\geq$ 2 degrees/second

Acceleration Time to Maximum Slew Speed: $\leq$ 5 seconds

Settling Time: $\leq$ 3 seconds

Absolute Pointing Accuracy: $\leq$ 15 arcseconds (open loop operation using pointing model, after large slews)

Pointing Repeatability: $\leq$ 15 arcseconds (open loop operation using pointing model, after large slews)

Fastest Tracking Speed: 15 arcseconds/second (sidereal rate)

Slowest Tracking Speed: 0.015 arcseconds/second (approximate)

Encoder Accuracy: 20 bits = 1.24 arcseconds

Encoder Precision: 24 bits/360° = 12.95 counts/arcsecond

Velocity Update Rate: 0.500 Hz

Cumulative Tracking Errors: 0.015 arcseconds

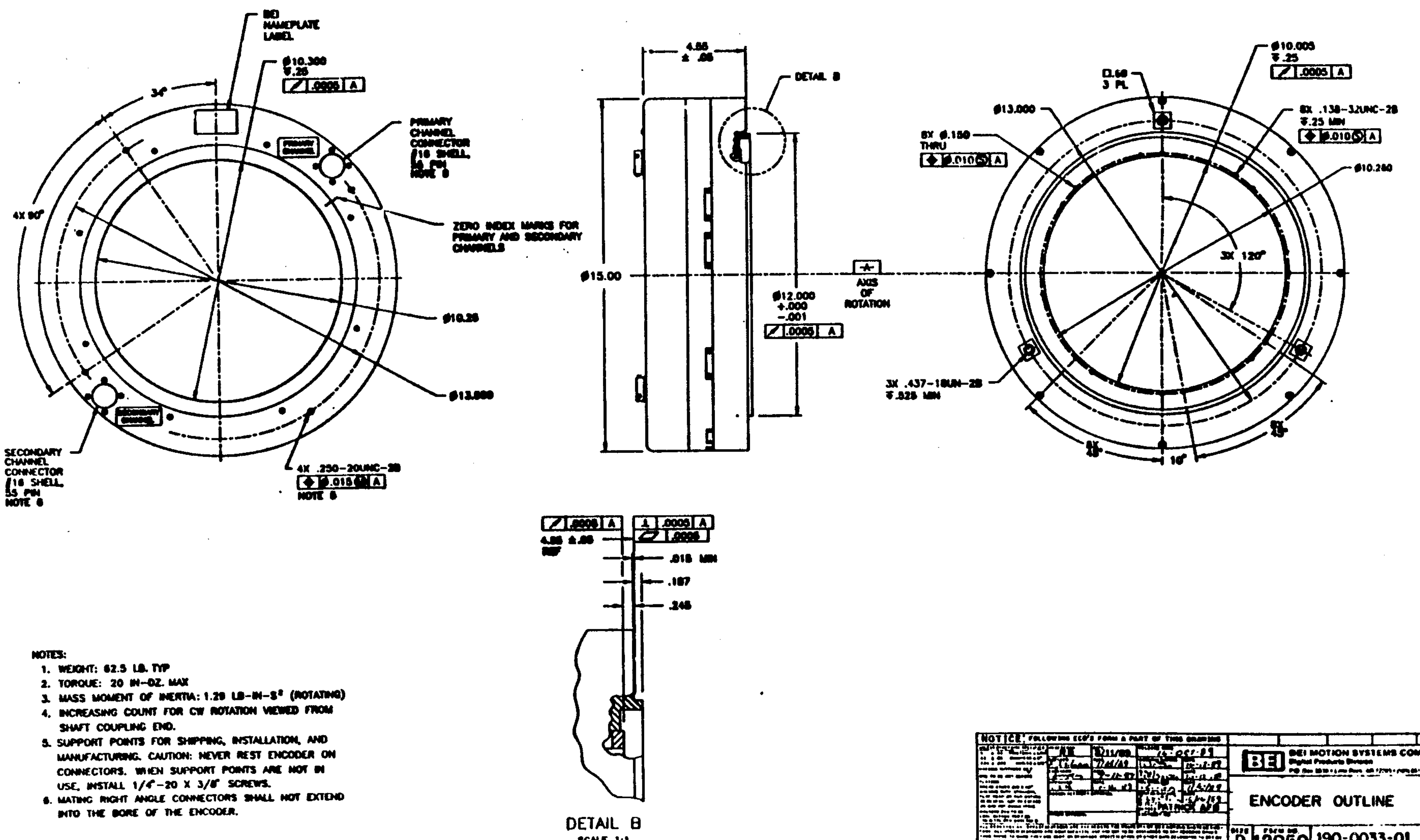
Minimum Drive Torque: (OSC estimate 17 ft - lb)

Encoder Recommendation

I) BEI 24-BIT absolute position encoder recommended

II) BEI encoders previously employed on air bearing gimbals

III) BEI potential source of motor and encoder with associated Servo electronics



NOTES:

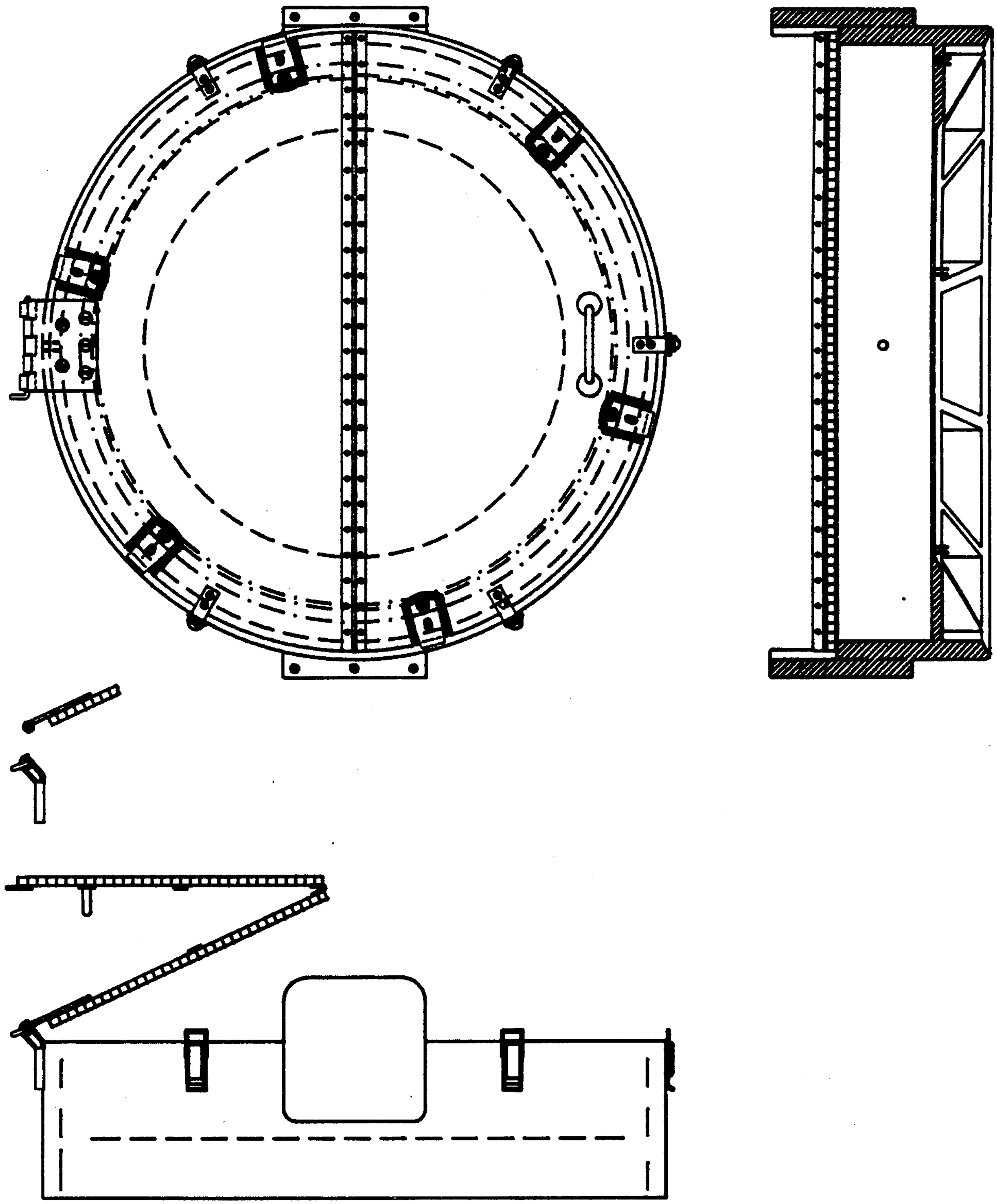
1. WEIGHT: 62.5 LB. TYP
2. TORQUE: 20 IN-OZ. MAX
3. MASS MOMENT OF INERTIA: 1.29 LB-IN-S² (ROTATING)
4. INCREASING COUNT FOR CW ROTATION VIEWED FROM SHAFT COUPLING END.
5. SUPPORT POINTS FOR SHIPPING, INSTALLATION, AND MANUFACTURING. CAUTION: NEVER REST ENCODER ON CONNECTORS. WHEN SUPPORT POINTS ARE NOT IN USE, INSTALL 1/4"-20 X 3/8" SCREWS.
6. MATING RIGHT ANGLE CONNECTORS SHALL NOT EXTEND INTO THE BORE OF THE ENCODER.

BEI MOTION SYSTEMS COMPANY
Digital Products Division

ENCODER OUTLINE

D 12050 190-0033-01

SCALE 1:2



Siderostat Structural Options

I) Athermal Truss

A) Meets vertical change with temperature specifications

B) Complex to assemble

C) Meets deflection due to wind specification

D) Structural weight: 2300 lbs

E) Size limits rotations in azimuth

II) Invar Fork

A) Meets vertical change with temperature specification

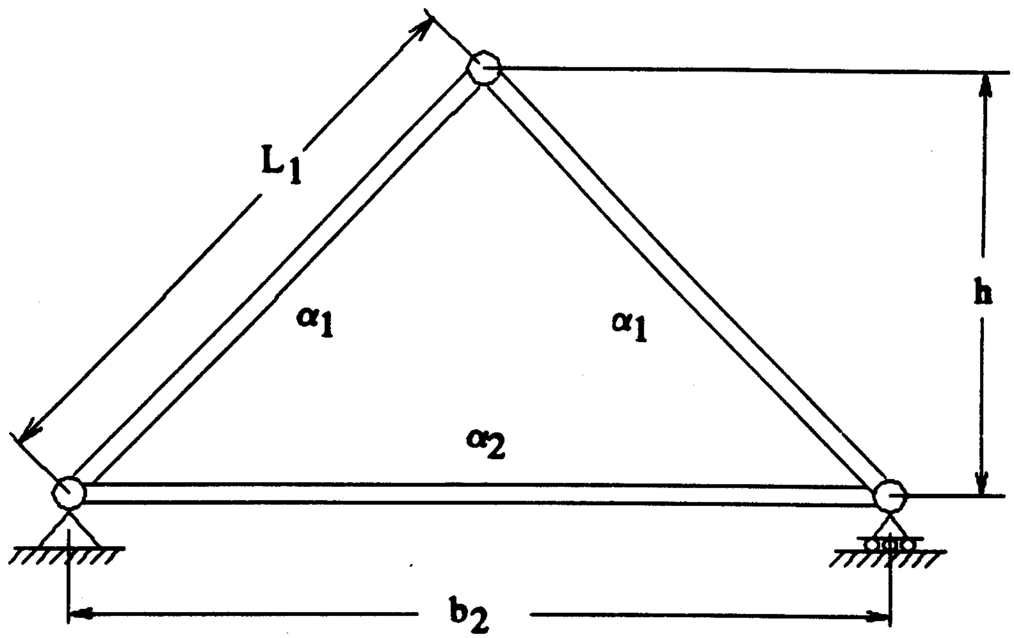
B) High material cost

C) Complex to assemble

D) Meets deflection due to wind specification

E) Structural weight: 2300 lbs

SIDEROSTAT ATHERMAL TRUSS CONCEPT



α_1, α_2 = THERMAL COEFFICIENTS OF EXPANSION

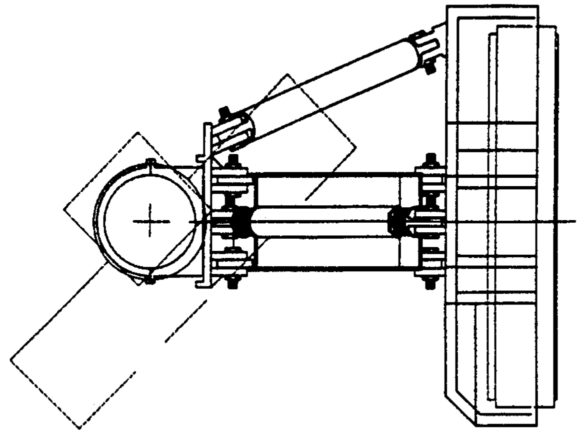
FOR h CONSTANT WITH RESPECT TO A
TEMPERATURE CHANGE AT ΔT:

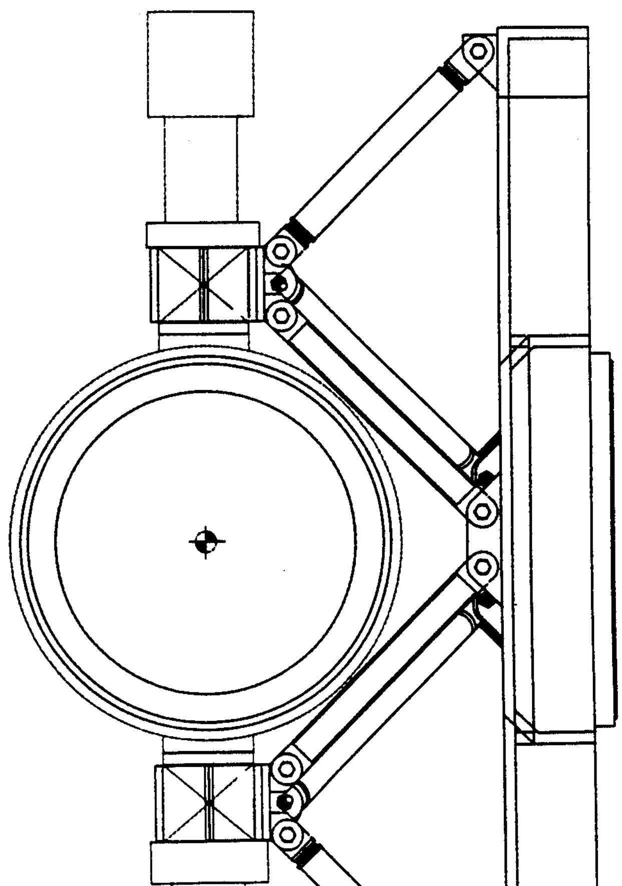
$$b_2 = \frac{2h}{\left[\frac{\alpha_2 (2 + \alpha_2 \Delta T)}{\alpha_1 (2 + \alpha_1 \Delta T)} - 1\right]^{\frac{1}{2}}} \cong \frac{2h}{\left[\frac{\alpha_2}{\alpha_1} - 1\right]^{\frac{1}{2}}}$$

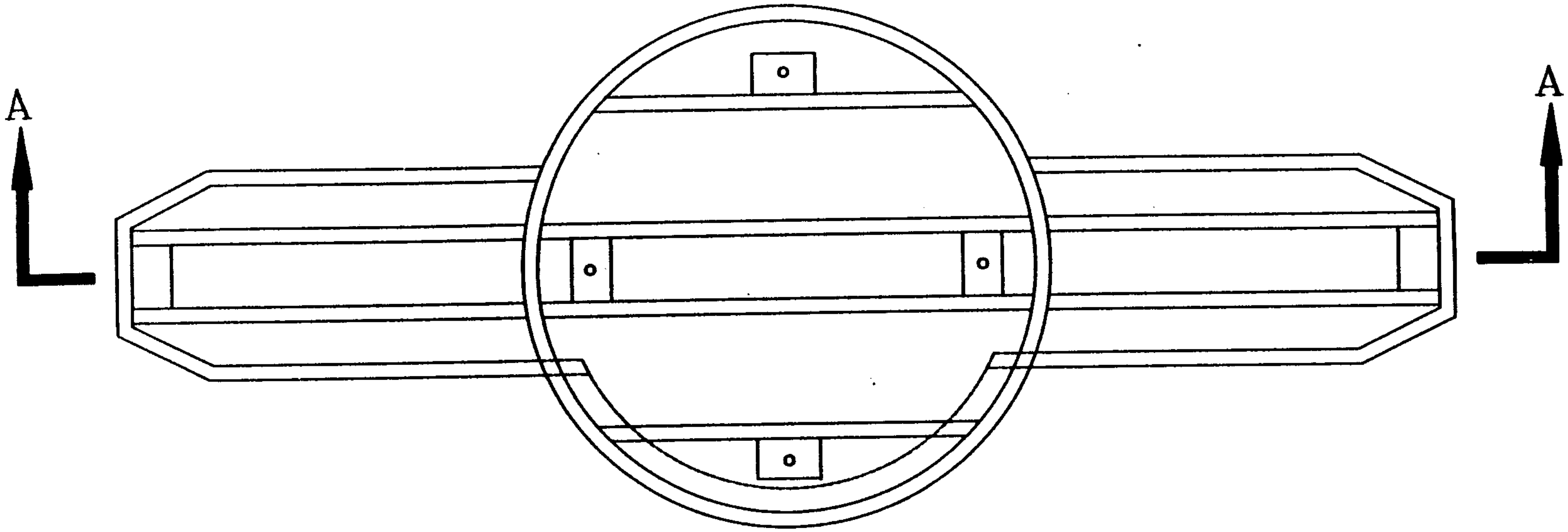
IF L_1 IS STEEL, $\alpha_1 = 11.7 \times 10^{-6} \frac{m}{m\text{-}K}$

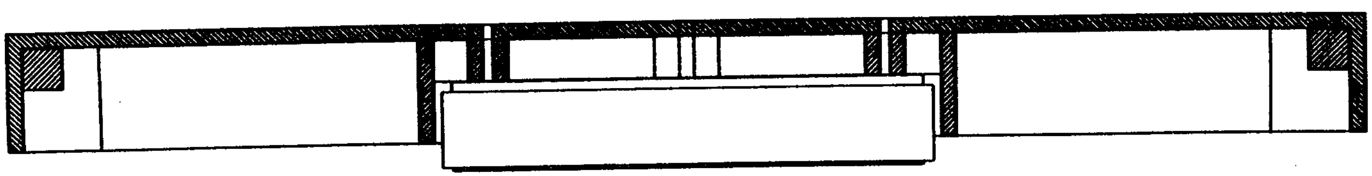
b_2 IS ALUMINUM, $\alpha_2 = 23.4 \times 10^{-6} \frac{m}{m\text{-}K}$

AND $b_2 = 2h$

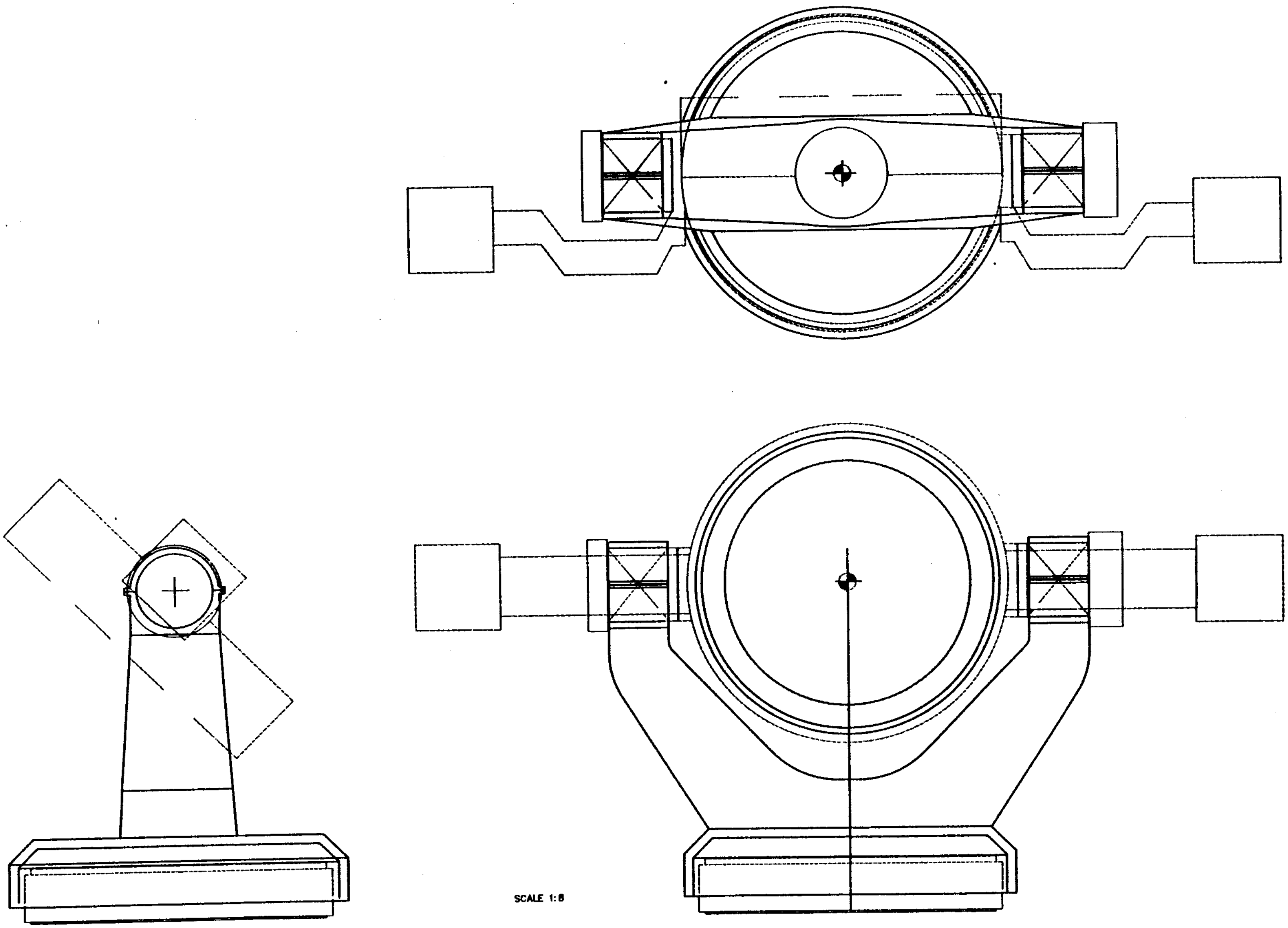




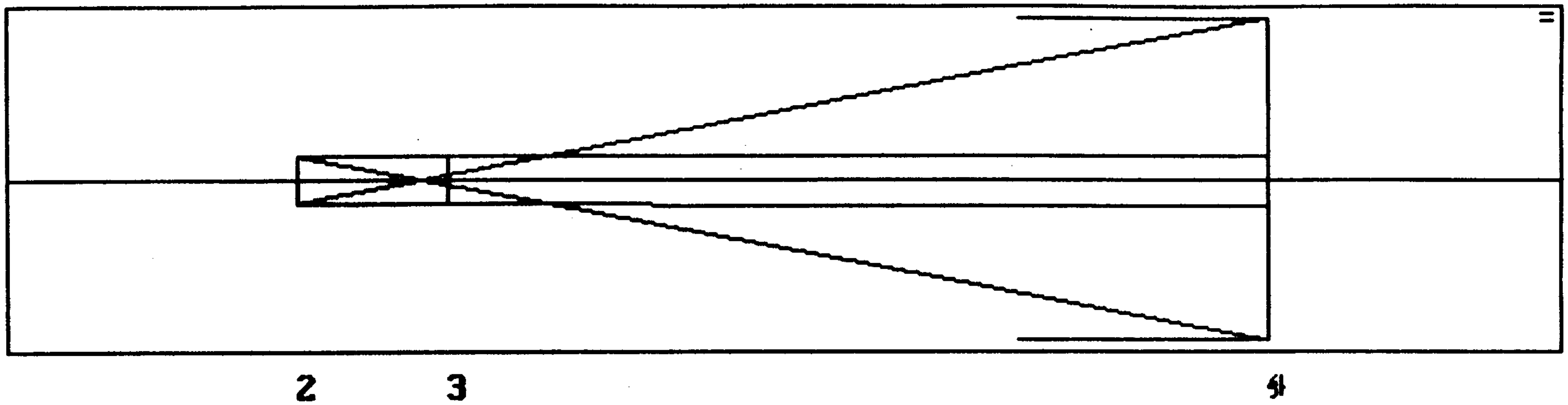




SECTION A-A

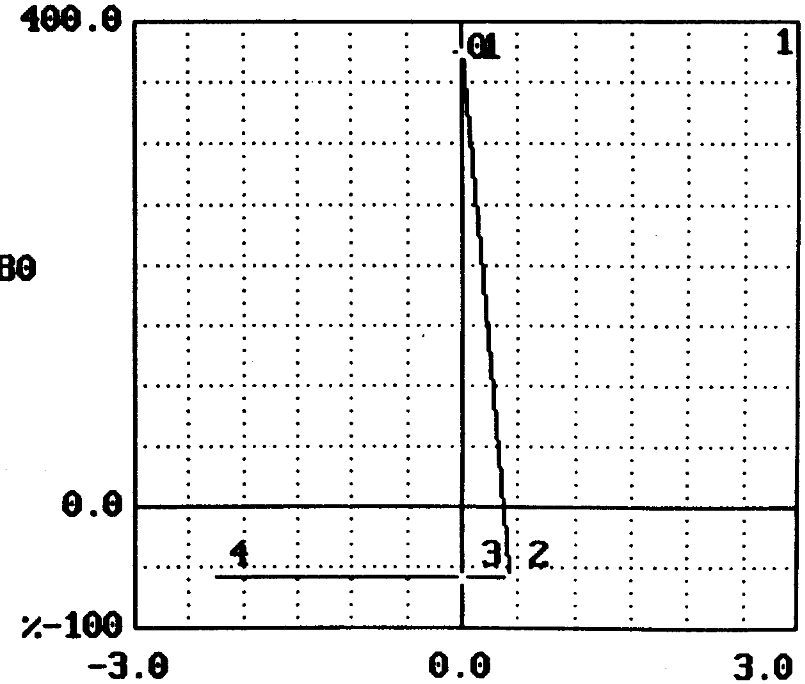

SCALE 1:8

Primary clear aperture:	0.75
Primary central hole:	0.125 m > h_p < 0.1154 m
Primary optical quality:	0.05 wave peak-to- valley (1 wave = 633 nm) over any 0.3 m diameter of the clear aperture, 0.1 wave peak-to-valley over entire clear aperture
Primary cosmetic quality:	40-20
Secondary clear aperture:	Less than 0.125 m
Secondary optical quality:	0.05 wave peak-to-valley over entire clear aperture
Secondary cosmetic quality:	20-10
Beam compression:	6.5:1
Beam diameter variation:	Less than 1% collapser to collapser
Minimum focal ratio:	f/5000 (redundant)



GREG MERSENNE F/2.7 (MERSNC) 1/4/91

NO.	Y	YBAR	EFL	DIST
0	375.000	-0.107	0.000	599.697
1	375.000	0.000	2025.000	%-2336.580
2	-57.700	0.418	311.580	359.522
3	-57.700	0.000	0.000	1977.058
4	-57.700	-2.300	0.000	0.000



Beam Collapser Optical Design

1.	Both mirrors will be concave paraboloids:	2025;	+5/0	mm
2.	Primary focal length:	750		mm
3.	Primary clear aperture:	311.5;	+0.5/-0	mm
4.	Secondary focal length:	116		mm
5.	Secondary clear aperture:			
6.	Spacing tolerance, vertex-to-vertex, secondary to primary:	±2		μm
7.	Centering tolerance, primary to secondary:	±40		μm

Material	α (m/m - K)	D (m^2/sec)	ΔR/m (waves)	$\Delta\alpha$ (m/mK)	δ/m (waves)
Borosilicate Glass (OHARA E-6)	3.3×10^{-6}	604×10^{-9}	149×10^{-6} (235)	50×10^{-9}	300×10^{-9} (0.48)
Fused Silica (Corning 7940)	56×10^{-9}	840×10^{-9}	1.41×10^{-6} (2.2)	2×10^{-9}	12×10^{-9} (0.02)
ULE (Corning 7971)	3×10^{-9}	777×10^{-9}	88×10^{-9} (0.14)	10×10^{-9}*	60×10^{-9} (0.10)
Zerodur (Schoott)	5×10^{-9}	800×10^{-9}	139×10^{-9} (0.22)	20×10^{-9}**	120×10^{-9} (0.19)
Aluminum (6061-T6)	23×10^{-6}	66×10^{-6}	(0) (0)	120×10^{-9}	722×10^{-9} (1.14)

Assumptions:

1. Mirror thickness = 0.12 m
2. Mirror radius of curvature = 4 m
3. Mirror diameter = 0.75
4. ΔR computed on the basis of 10 K instantaneous change, ΔR is figure error after 1 hr.
5. δ computed on the basis of 10 K change, with $\Delta\alpha$ varying through the thickness of the mirror.

*Soabgenberg-Jolley, J., and Hobbs, T., "Mirror substrate fabrication techniques of low expansion glasses," Proc. SPIE 966, pp.284 (1988).

**Mueller, R.W., Hoeness, H.W., and Marx, T.A., "Spin-cast Zerodur mirror substrates of the 8-m class and lightweighted substrates for secondary mirror," Proc. SPIE 1236, pp. 723 (1990).

DESIGN SPECIFICATIONS

- Support structure must maintain optical elements to the following tolerances:
  - separation: ≤ 4 μm
  - decenter: ≤ 25 μm (relative)
  - primary mirror tilt: ≤ 3 arcsecs
  - secondary mirror tilt: ≤ 6 arcsecs
  - vertex-to-vertex spacing: 2.34 m → 92 inches
- Tolerances must be met under the following load conditions
  - gravity
  - wind
  - thermal
- Further design requirements
  - fundamental frequency of the structure ≥ 10 Hz
  - structural survival under 100 MPH wind
  - structural survival under Zone 4 Seismic Earthquake

Structural Analysis and Design of the Beam Collapser Support Structure

- Design of Modified Serrurier Truss
- Analysis
  - Gravity loading
  - Wind loading
  - Thermal loading
  - Frequency and modal response
  - Earthquake loading

Modified Serrurier Truss Design

- Secondary end-ring and upper truss members block siderostat f.o.v.

- Modify traditional Serrurier truss

  - remove upper part of the secondary truss and place a pair of links lower on the side

  - use a semi-circular secondary end-ring

  - same deflection characteristics as the standard Serrurier truss

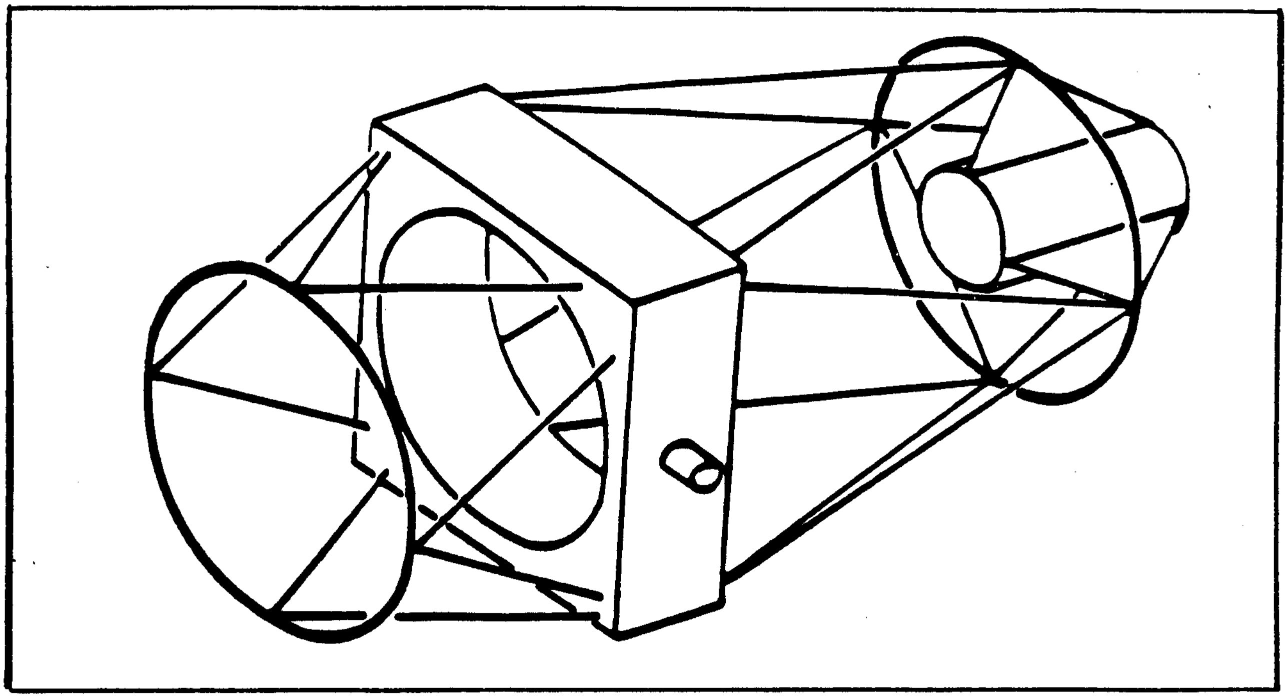


Figure 1. Serrurier Truss

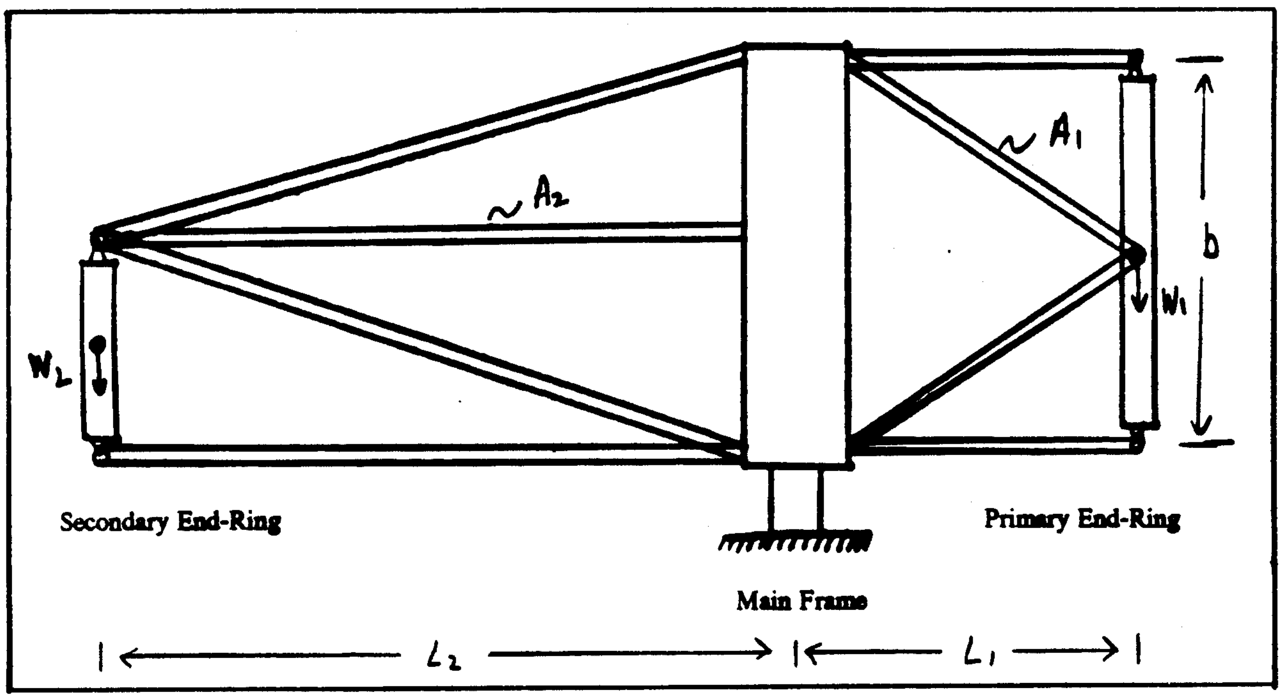
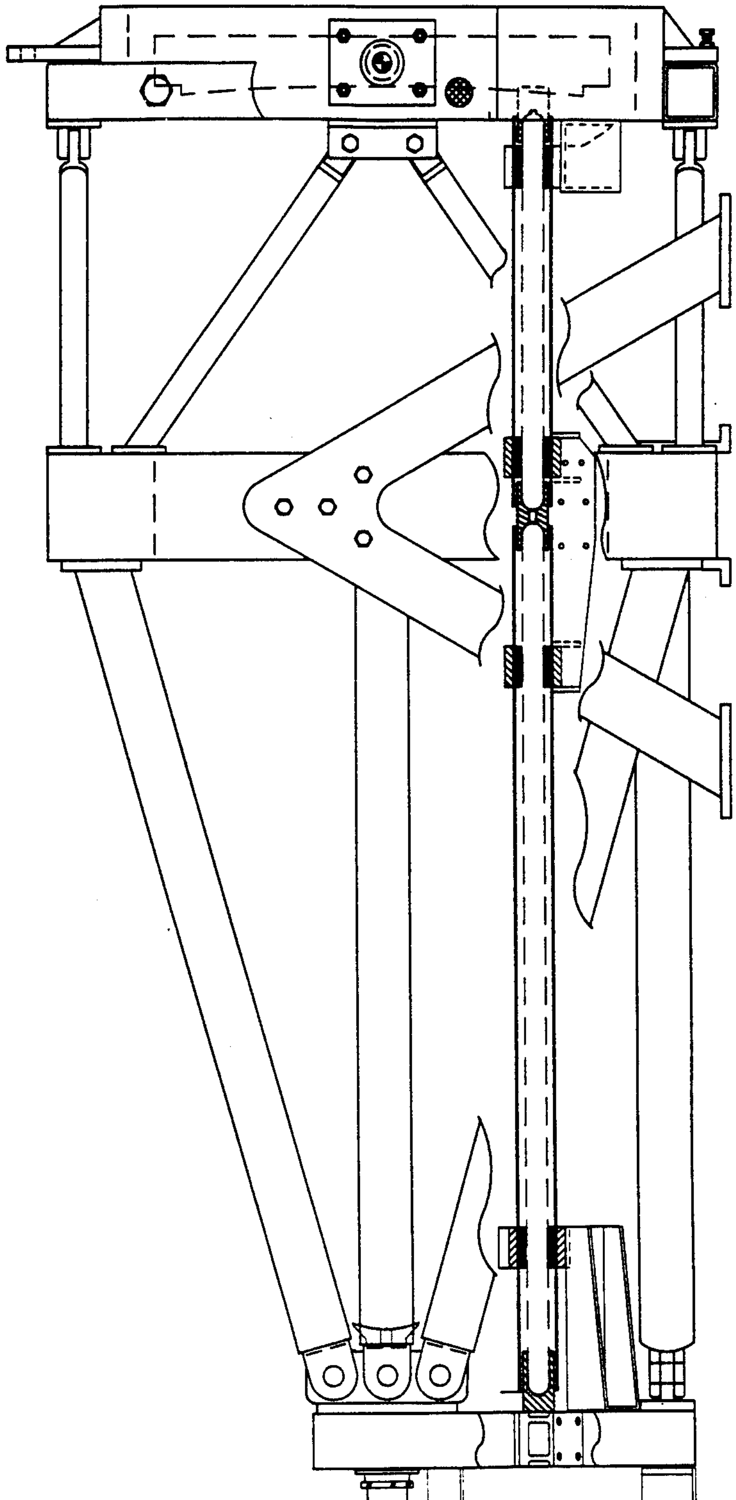


Figure 2. Modified Serrurier Truss

Modified Serrurier Truss Design

- A NASTRAN FEM was produced of the support structure
- Structural member details
    - secondary and primary end-rings
        - 4" x 4" x 1/2" steel tubing
    - main frame
        - 5" x 5" x 5/16" steel tubing
    - secondary truss members
        - 3.5" nominal diameter steel pipe
    - primary truss members
        - 2.0" nominal diameter steel pipe



Modified Serrurier Truss Design

Gravity Loading

- Rotation of structure about main frame
- Deflection of truss members
- 9.0 μm z decenter
- 11.3 μm axial separation
    - re-focused on site

Modified Serrurier Truss Design

Wind Loading

- Method of Analysis
  - applied pressure loads to NASTRAN FEM
  - $(F/L) = \frac{1}{2} C_d \rho w v^2$
  - used C_d of 1.2 for round truss members
  - used C_d of 2.05 for square tubes of end-rings and main frame
  - performed two worst case wind loadings on model
    - wind acting laterally
    - wind acting along the optical axis
  - deflection characteristics of truss due to 9 m/s wind
  - structural integrity of truss due to 100 MPH wind

Modified Serrurier Truss Design

Wind Loading

- Response of Structure to 9 m/s wind
    - lateral direction (Y--dir)
        - 1.5 μm decenter
    - optical axis direction (X--dir)
        - 1.9 μm decenter

Modified Serrurier Truss Design

Wind Loading

- Response of structure to 100 MPH wind
  - analysis conducted same as 9 m/s wind
- Total lateral wind force = 1320 lbs.
- Total optical axis wind force = 1760 lbs.
- Maximum stress = 475 psi
  - below microyield of steel

Modified Serrurier Truss Design

Thermal Response

- FE model used to perform thermal analysis
- One degree linear gradient--1°C
  - x, y, z axes
  - axial separation x: 11.73 μm, y: 10.54 μm, z: 12.19 μm
- Thermal soak--20°C
  - axial separation 561 μm
- Metering rods

Modified Serrurier Truss Design

- Dynamic Response

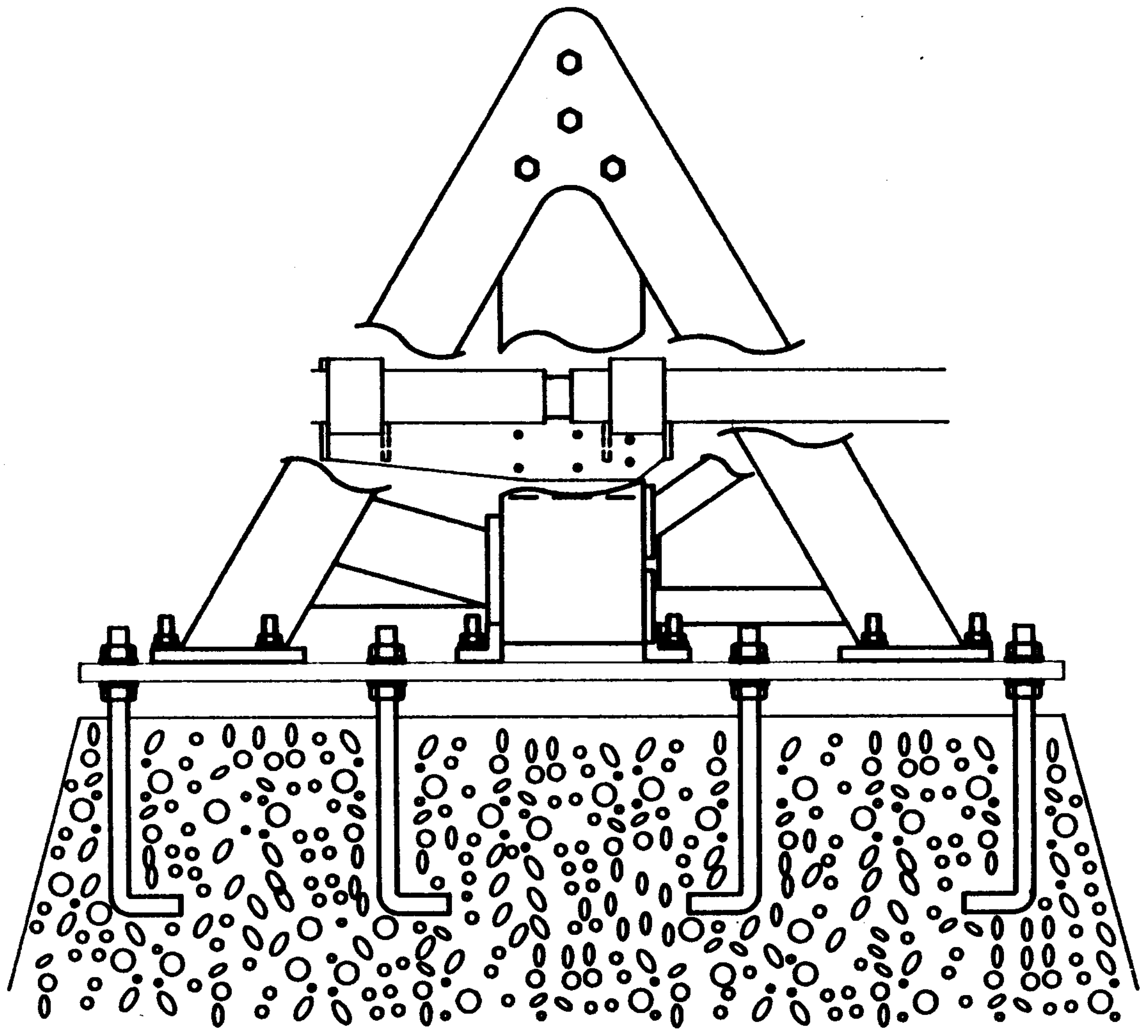
- NASTRAN computed the fundamental frequency of the structure

- $f_n = 35.0$ Hz

- Satisfies structural requirment that fn $\geq$ 10 Hz

Modified Serrurier Truss Pier Attachment

- Four-point mount
- Bolted brackets connecting main frame to steel plate
- 'A' Frame Support
  - prevents rotation about main frame
  - 5" x 2" x 5/16" box sections



PIER ATTACHMENT

MAIN FRAME TO CONCRETE PIER

Modified Serrurier Truss Results Summary (μm, arcsec)

Loading			Gravity	Wind Lateral	Wind OA	Thermal (soak, gradient) 20°	Thermal X	Thermal Y	Thermal Z	Totals
SECONDARY	Translation	X	14.63	-----	1.90	-375.92	-7.90	-8.15	-5.16	-----
		Y	-----	1.51	0.0065	-----	-----	-12.14	-----	-----
		Z	6.43	-----	0.78	161.80	-9.27	1.25	3.61	-----
	Rotation	R_Y	0.97	-----	0.11	0.37	0.29	0.057	0.45	2.25
		R_Z	-----	-----	-----	-----	-----	0.30	-----	0.30
PRIMARY	Translation	X	3.33	-----	0.36	185.67	3.84	4.52	8.08	-----
		Y	-----	1.20	-----	-----	-----	-6.58	-----	-----
		Z	-2.59	-----	-0.38	164.34	-0.54	-----	5.33	-----
	Rotation	R_Y	0.085	-----	-----	1.25	0.14	0.057	0.062	1.59
		R_Z	-----	-----	-----	-----	-----	0.15	-----	0.15
RELATIVE DEFLECTION	Translation	X	11.30*	-----	1.54	561.34*	11.73*	12.67*	13.24*	585.91
		Y	-----	0.31	0.007	-----	-----	5.56	-----	5.88
		Z	9.02	-----	1.18	2.54	8.74	1.23	1.73	24.44

*Exceeds Tolerances

Allowed Tolerances:

Axial separation X: 4μm
Decenter Y,Z: 25μm
Rotation Y,Z (primary): 3 arcsec
Rotation Y,Z (secondary): 6 arcsec

Modified Serrurier Truss Design

Earthquake Analysis

- UBC/SEAOC Seismic Design Code
- Equivalent lateral force method

  $V_B = ZICW$

  V_B - base shear

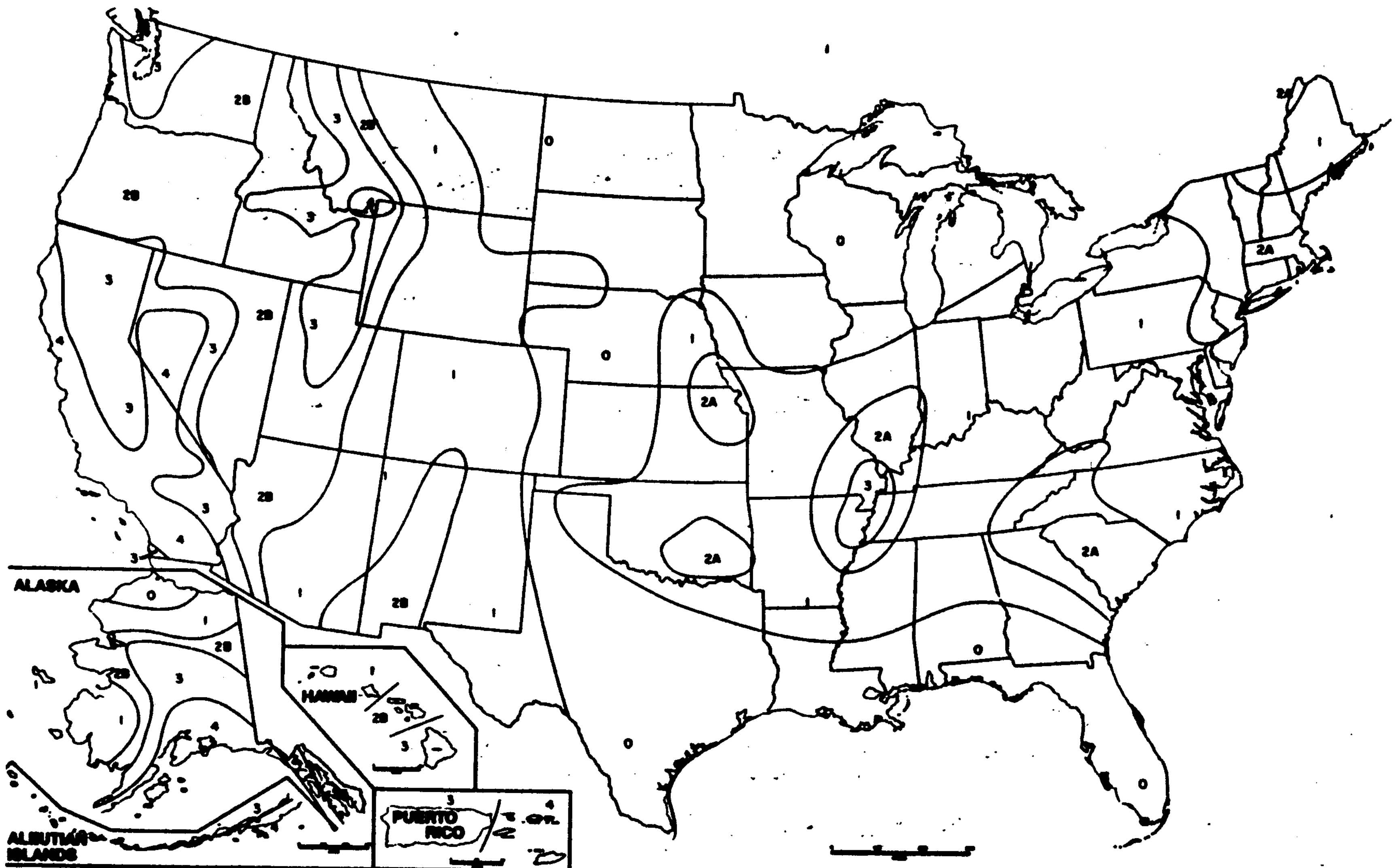
  W - weight of structure (DL + LL)

  Z - seismic zone factor

  I - importance factor

  C - seismic coefficient

- Apply load V_B using FEM
- Stress below 3000 psi



SEISMIC ZONE MAP OF THE UNITED STATES

BEAM COLLAPSER PRIMARY MIRROR

DESIGN SPECIFICATIONS

- **MIRROR MATERIAL:** CORNING ULE
- **MIRROR DIAMETER:** 0.75 M CLEAR APERTURE
- **MIRROR THICKNESS:** AS REQUIRED BY DESIGN
- **MIRROR WEIGHT:** NO RESTRICTIONS
- **ORIENTATION:** FIXED - 10 TO - 20 DEGREES
- **CENTRAL HOLE:** GREATER THAN 11.54 CM
LESS THAN 12.5 CM
- **FOCAL RATIO:** F/3 (APPROXIMATELY)
- **OPTICAL SURFACE:** TO BE DETERMINED
- **SURFACE ACCURACY:** .05 λ P-V OVER ANY 30 CM DIA.
.10 λ P-V OVER CLEAR APERUTRE
(BOTH EXCLUDE 1 CM RADIUS FROM CENTER)
- **COSMETIC:** 40 - 20 SCRATCH DIG OR BETTER

0.75 M BEAM COMPRESSOR PRIMARY February 1991

USNO INTERFEROMETER - SUMMARY OF OPTICAL DEFLECTIONS

Mirror Material = Fused Silica

$E = 9.8\text{ E }6\text{ psi}$

$\nu = 0.17$

$\rho = 0.08 \frac{\text{lb}}{\text{in}^3}$

Mirror supported with a roller chain and equally spaced support points

Optical deflections as calculated by PCFRINGE and expressed in waves, $\lambda = 0.633$ micron

MODEL	THICK. (in.)	SUPPORT PTS.	ORIENTATION (degrees)	OPTICAL DEFECTION (P-V)	(RMS)
TV02US	4	6 pts, front outer edge	10	.061	.013
TV00US	5	6 pts, front outer edge	10	.051	.009
TV03US	6	6 pts, front outer edge	10	.044	.008
TV12US	5	12 pts, front outer edge	10	.045	.009
TV07US	5	continuous ring back edge	90	.243	.048
TV08US	5	ring support at R =.65	10	.024	.004
TV09US	5	6 pts at R =.65	10	.026	.005

Tolerances:

Weight : no restrictions

Optical Deflection : .1λ P-V

Mirror Weights :

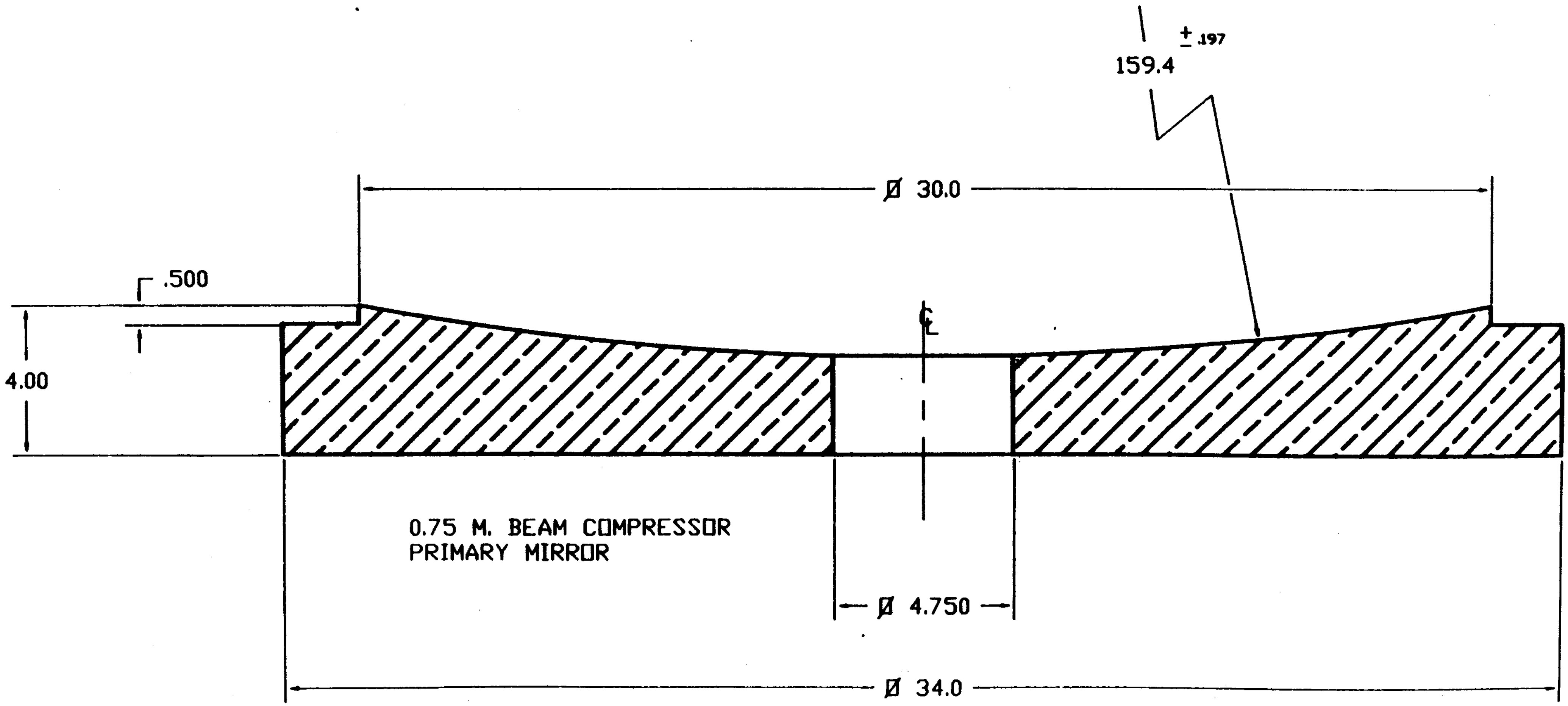
4 inch thick = 258 lb

5 inch thick = 329 lb

6 inch thick = 400 lb

BEAM COLLAPSER PRIMARY MIRROR

FINAL DESIGN



0.75 M BEAM COMPRESSOR PRIMARY

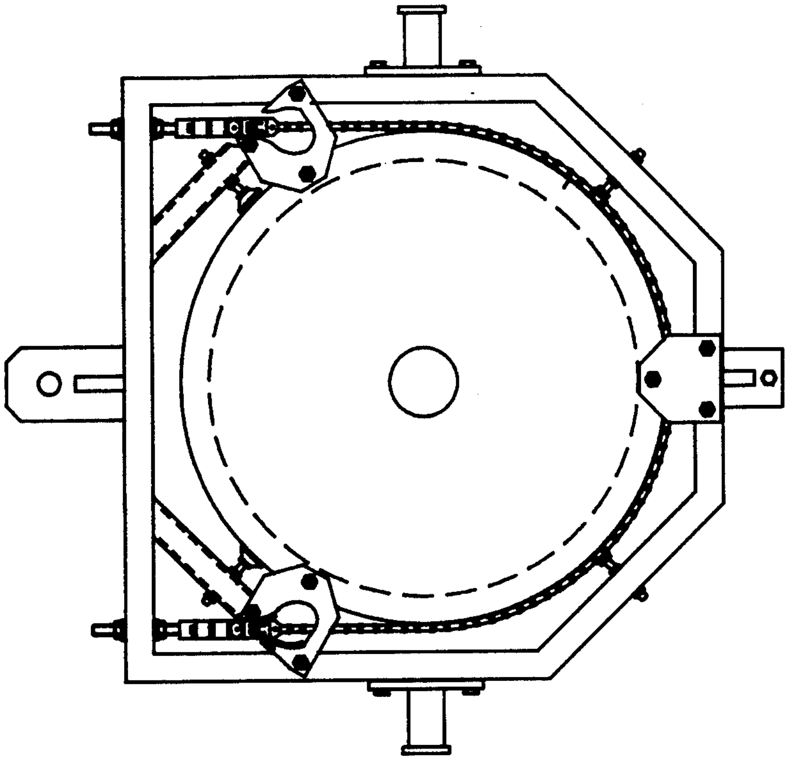
SUPPORT LOCATIONS

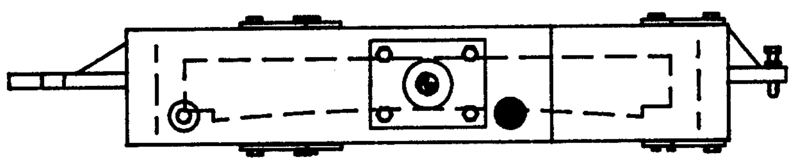
MIRROR ORIENTATION : +10 DEGREES

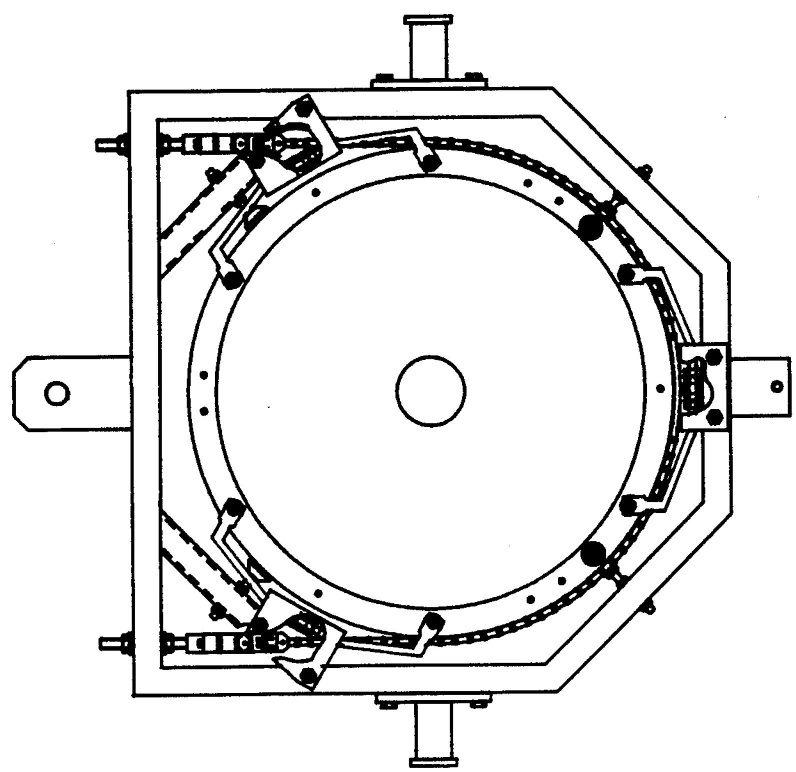
- **RADIAL SUPPORT** -- CONVENTIONAL ROLLER CHAIN

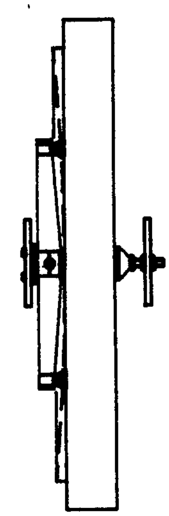
- **AXIAL SUPPORT** -- SIX POINTS EQUALLY SPACED

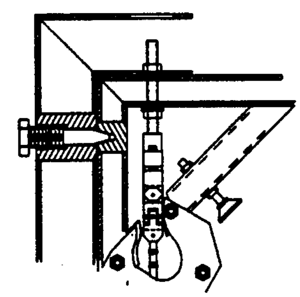
  -- LOCATED AGAINST MIRROR FRONT OUTSIDE LIP

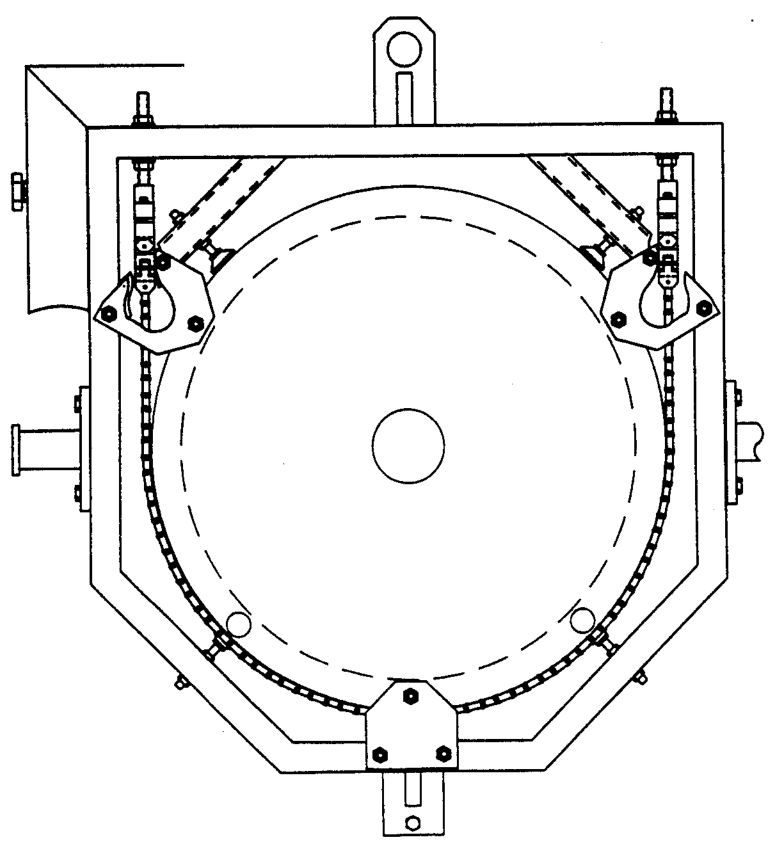








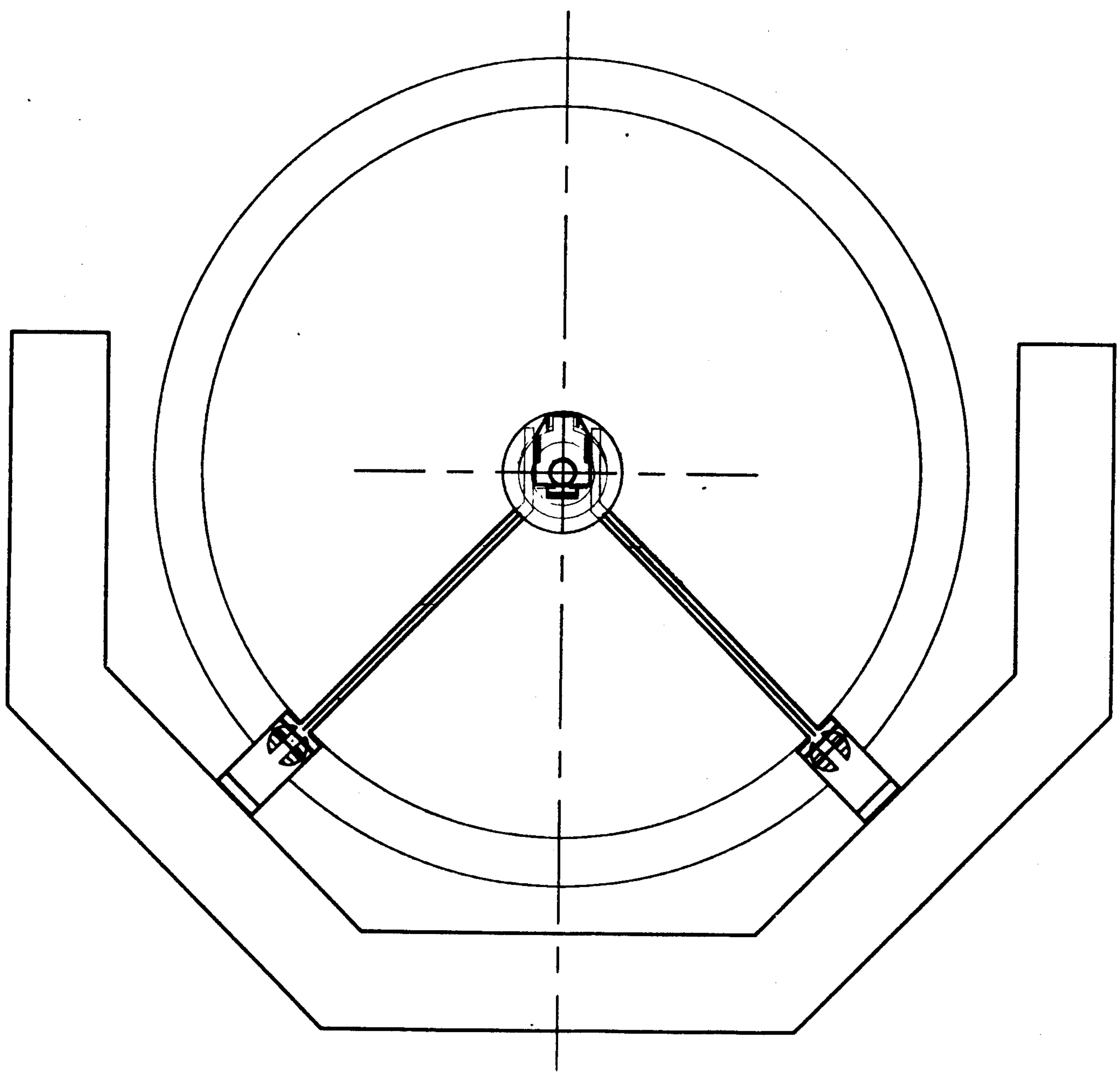




BEAM COLLAPSER SECONDARY SPIDER MOUNT

PARALLEL SPRING GUIDES IN CONTACT WITH THE METERING RODS

FORCE REQUIRED TO MOVE ONE SET OF SPRINGS .02 INCHES = 1.731 LB



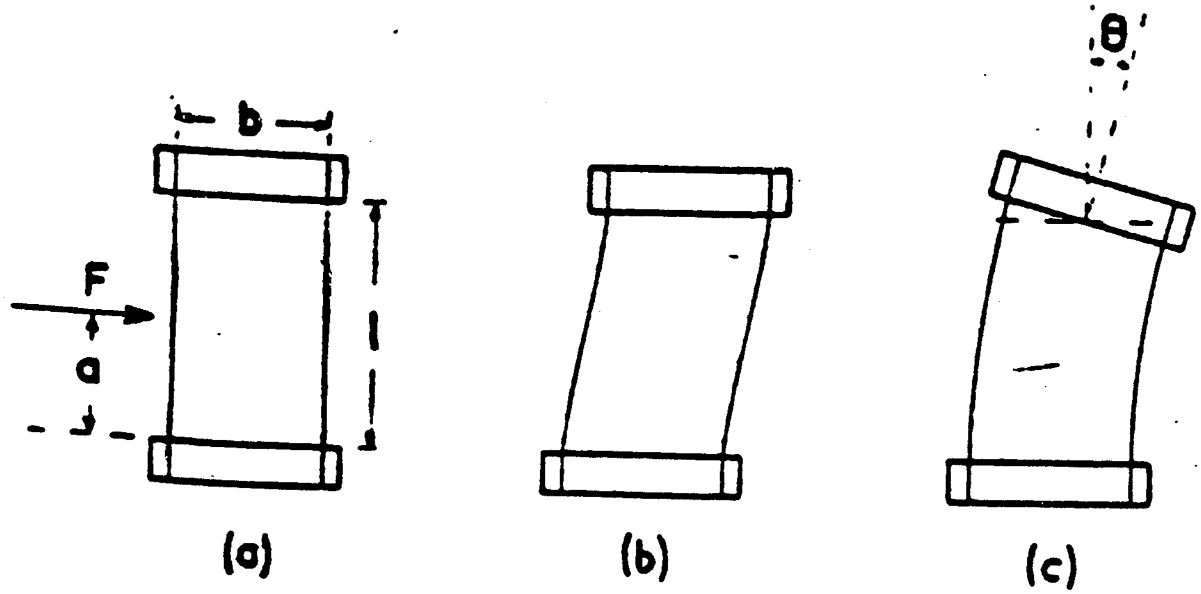


Fig. 1. Simple parallel movement: (*a*) undeflected, (*b*) intended parallel deflexion, (*c*) undesired cantilever bending

a = 2.748 in

b = 3.5 in

l = 2.248

ERRORS DUE TO PARALLEL SPRINGS

TILT =2.14905E-6 RAD

FOCUS =2.686EE-5 IN

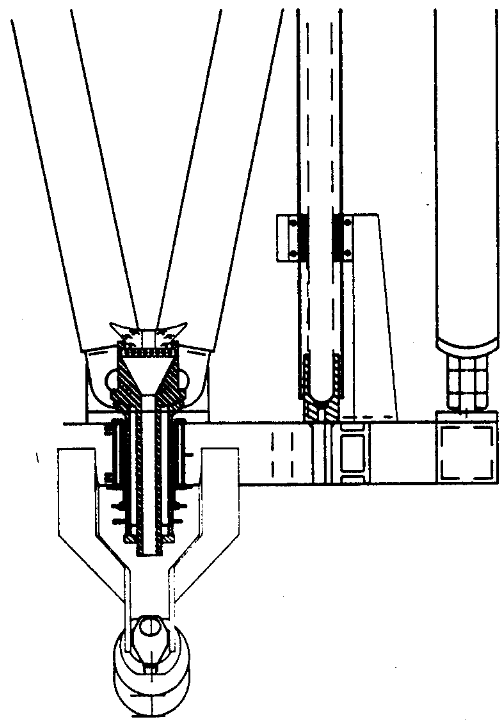
DECENTER =0 IN

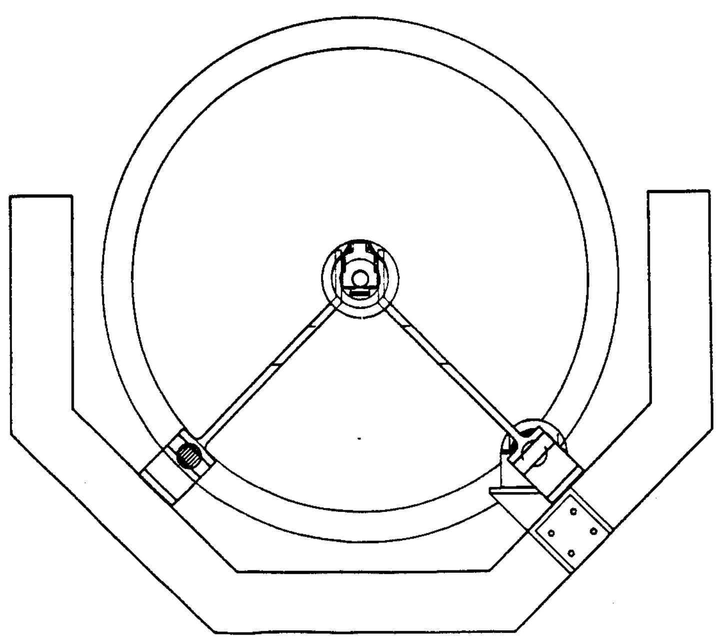
TOLERANCES

TILT =2.9088E-5 RAD

FOCUS =1.575E-4 IN

DECENTER =9.840E-4





Beam Collapser Metering Rods

I) Thermal analysis of beam collapser indicates that axial spacing between secondary and primary mirrors changes more than the design tolerance when a significant temperature shift occurs

II) Cost, stiffness-to-weight, and dimensional stability prohibit use of a low thermal coefficient of expansion material such as Invar

III) Axial spacing is maintained using low thermal coefficient of expansion metering rods

IV) Metering rods are made of same material as primary and secondary mirrors (Schott Zerodur or Corning ULE Code 7971)

V) Metering rods contact primary mirror ledge and parallel spring guide flexures at base of secondary support spider

VI) Each metering rod is contained in a steel tube and supported via bonded flexures at the airy points

VII) A joint <u>may</u> be placed at the central support ring of the beam collapser truss

Examples of Telescope Metering Rods

1948 - Palomar 48 in. Schmidt
(Invar rods control plate focus)

1972 - Orbiting Astronomical Observatory, Copernicus (OAO-C), 31.5 in. cassegrain (Fused silica rods control primary to secondary spacing)

1978 - Voyager Imaging Science Subsystem (ISS)
(Invar rods control axial spacing)

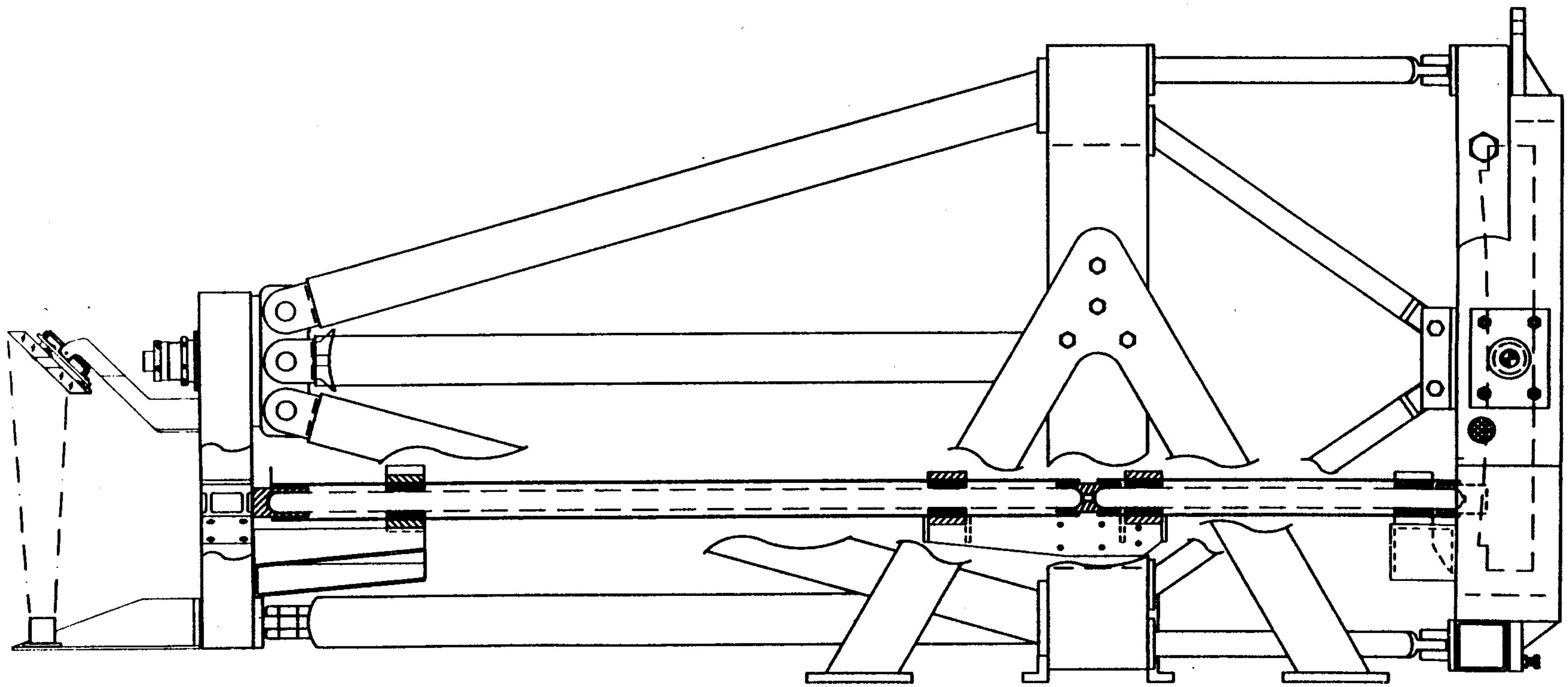
1986 - Shanghai Observatory 1.56m Astrometric Telescope (Cer-vit metering rods control primary to secondary spacing)

BEAM COLLAPSER

SYSTEM CTE -- AS PROVIDED BY METERING RODS

α ZERODUR $= 0.05 \times 10^{-6}$

α STEEL $= 10 \times 10^{-6}$



SYSTEM CTE $= -0.004 \times 10^{-6}$

BEAM COLLAPSER METERING RODS

SUPPORT LOCATIONS -- "AIRY POINTS"

THE "AIRY POINTS" ARE THE SUPPORT LOCATIONS FOR A BEA WHICH RESULT IN A ZERO SLOPE AT THE ENDS OF THE BEAM.

METERING ROD

MATERIAL = zerodur

DIAMETER = 1.5 in

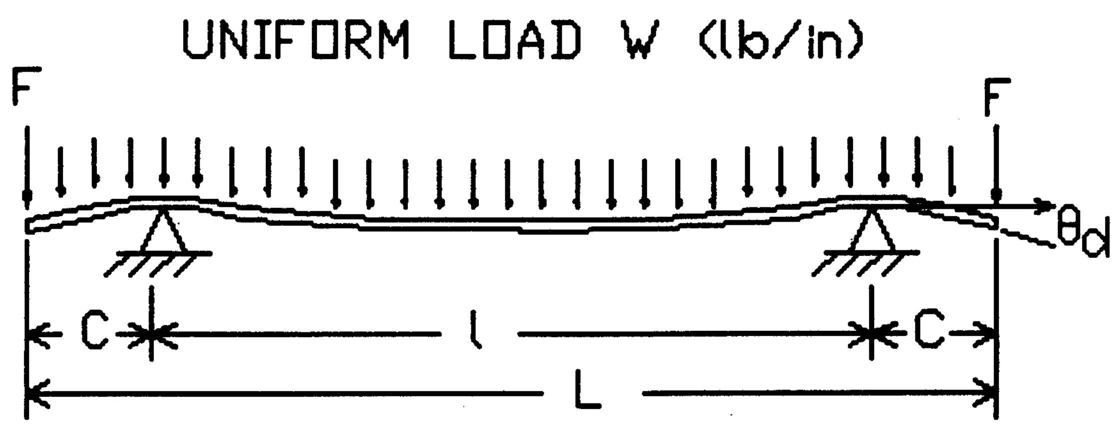
$E = 13.20E6$ psi

$\rho = 0.091 \ \frac{lb}{in^3}$

L = 64.396 in, 28.846 in

WEIGHT W = 10.403 lb, 4.660 lb

F = 1.14 lb (assumed due to steel spacers)



IF $\theta_d = 0$ (zero slope)

$$C = \frac{L}{2}\left[1 - \left(1 - \frac{W}{6\left(\frac{F}{2} + \frac{W}{4}\right)}\right)^{\frac{1}{2}}\right]$$

METERING ROD 1

L = 64.396 in

C = 10.53 in

METERING ROD 2

L = 28.846 in

C = 3.70 in

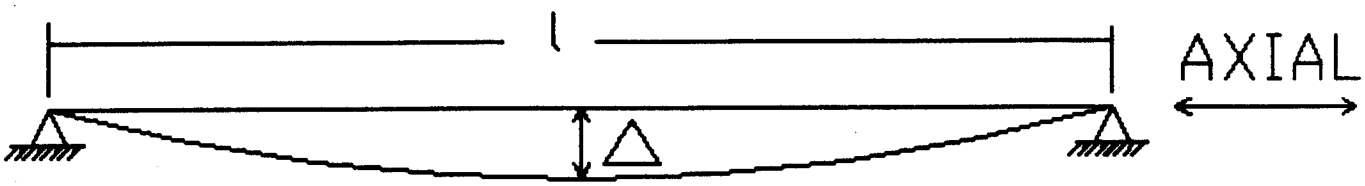
BEAM COLLPASER METERING RODS

MAXIMUM DEFLECTION OF METERING RODS BETWEEN SUPPORT POINTS

$w = .08077 \ \frac{lb}{in}$

$I = \frac{\pi r^4}{4} \ in^4$

l = distance between support points



$$\Delta = \frac{5 \, w \, l^4}{384 \, E \, I}$$

Δ = MAX LATERAL DISPACEMENT (translates to axial)

δ = MAX AXIAL DISPLACEMENT

METERING ROD 1

$l = 43.35$ in

$\Delta = 1.13 \times 10^{-3}$ in

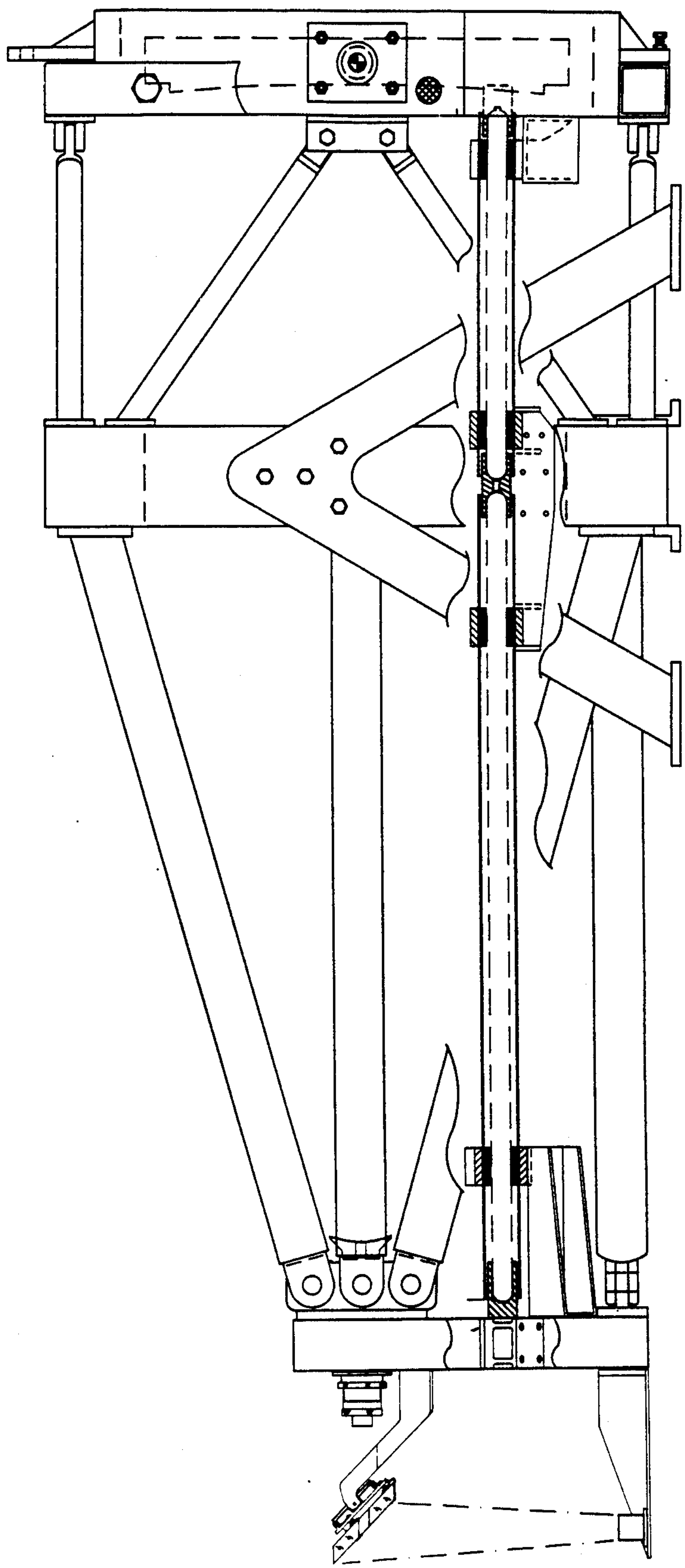
$\delta = 2.94 \times 10^{-8}$ in

METERING ROD 2

$l = 21.44$ in

$\Delta = 6.78 \times 10^{-5}$ in

$\delta = 2.50 \times 10^{-10}$ in



INTERFEROMETER OPTICS -- HANDLING CONSIDERATIONS

1) INSTALLATION AND REMOVAL OF MIRROR CELLS

   A) ALIGNMENT PINS PROVIDED

   B) SAFETY LIFTING LINKS

   C) SAFETY CLIPS

2) REMOVAL OF OPTICS FOR COATING

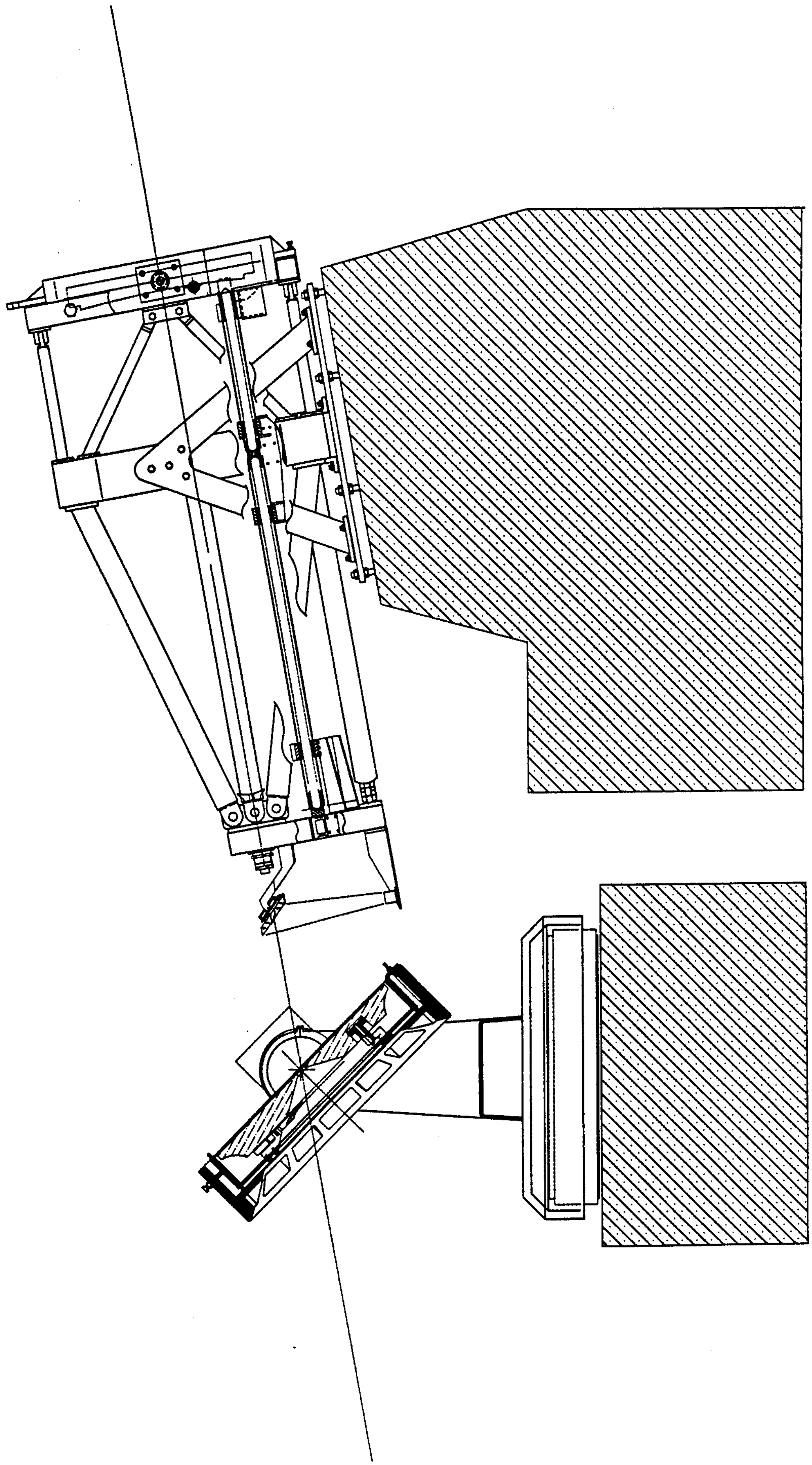
   -- CELLS CAN BE MADE VACUUM COMPATIBLE
   BUT COST WILL INCREASE

3) PROTECTIVE COVERS FOR OPTICS

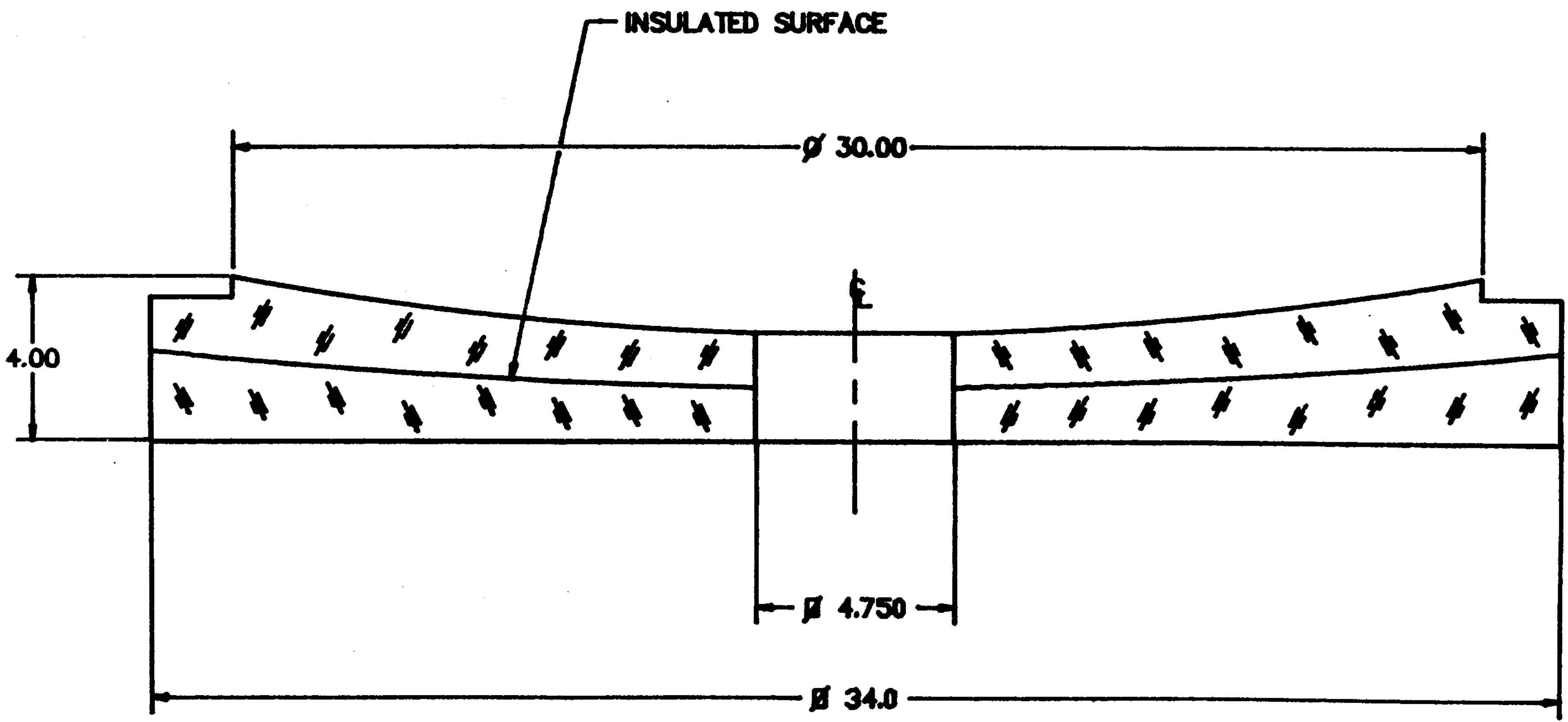
   A) MANUAL OPERATION

   B) PROVIDE PROTECTION FROM DUST

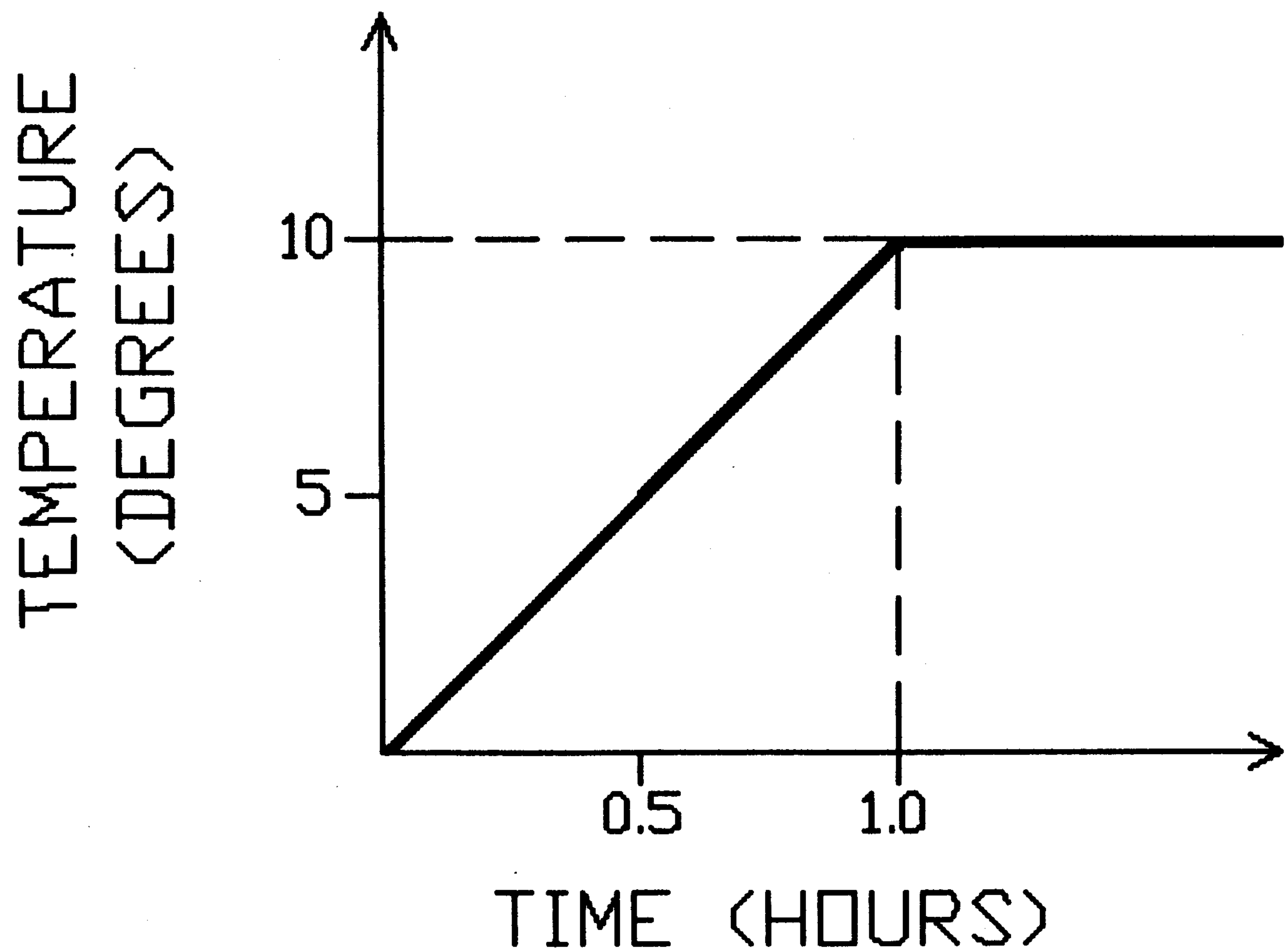
   C) NOT WATERPROOF



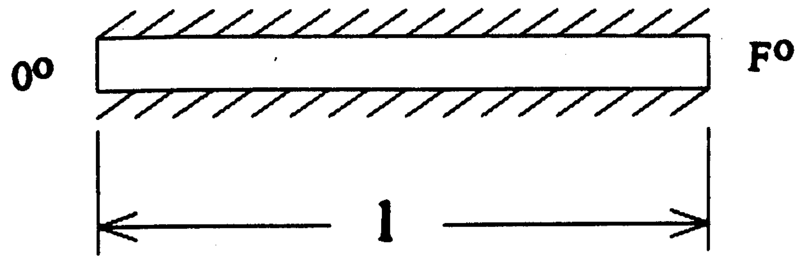
BEAM COLLAPSER PRIMARY MIRROR
THERMAL MODEL



TRANSIENT THERMAL ENVIRONMENT
SURFACE TEMPERATURE



CAL TRANSIENT DEMONSTRATION PROBLEM

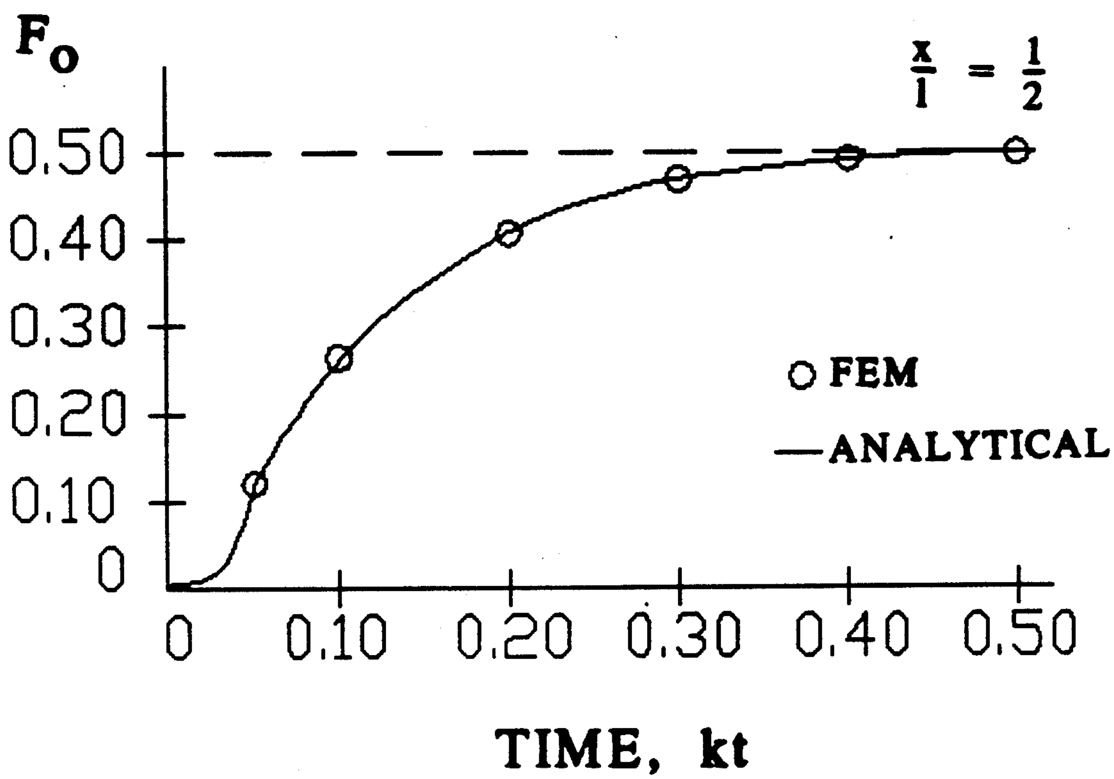


Heat Equation: $U_t\,(x,t) \;=\; k\,U_{xx}\,(x,t)$

B.C.'s $U\,(0,t) \;=\; 0, \quad u\,(l,t) \;=\; F^o$

I.C. $U\,(x,0) \;=\; 0$

$$U\,(x,t) \;=\; F_o\left[\frac{x}{l} + \frac{2}{\pi}\sum \frac{(-1)^n}{n}\, e^{-n^2\pi^2 kt} \sin\left[\frac{n\pi x}{l}\right]\right]$$



THERMAL

ZERODUR MATERIAL PROPERTIES

CTE (α)	0.10E-6 /K
Thermal Conductivity (K)	$1.46 \frac{W}{m - K}$ (@ 293K)
Specific Heat (C_p)	$0.80 \frac{J}{g - K}$ (@ 293K)
Density (ρ)	$2.53 \frac{g}{cm^3}$
Diffusivity (κ)	$0.72E\text{-}6 \frac{m^2}{sec}$

HEAT EQUATION AND BOUNDARY CONDITIONS

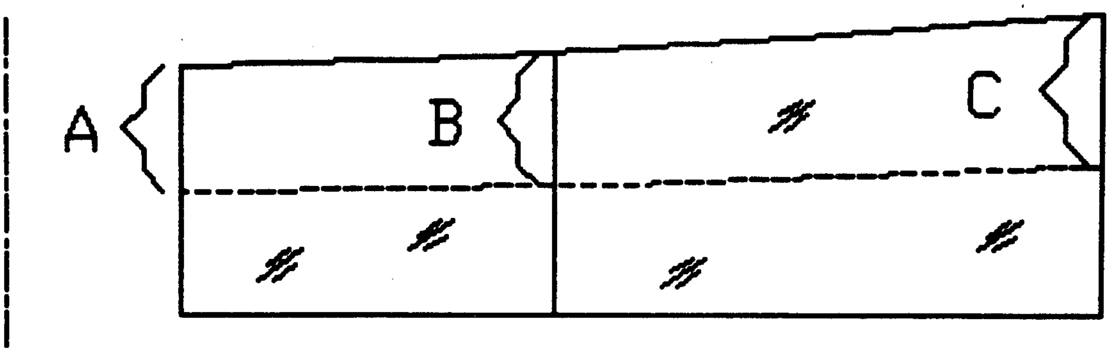
$$U_t(x,t) = \kappa U_{xx}(x,t)$$

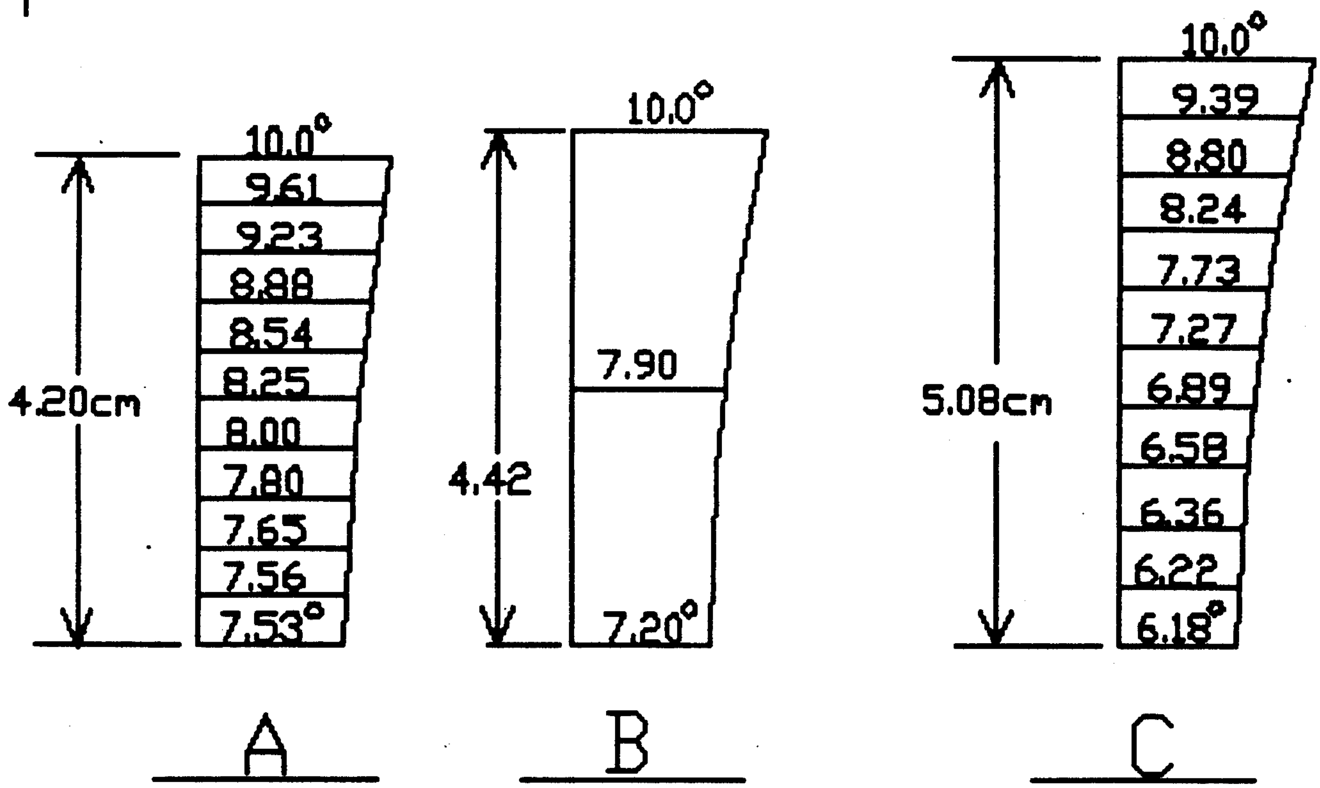
$$U_x(0,t) = 0$$

$$U(l,t) = \begin{cases} 10\,t & 0 \leq t \leq 1 \text{ hour} \\ 10 & t > 1 \text{ hour} \end{cases}$$

USNO PRIMARY MIRROR

THERMAL INERTIA STUDY





TEMPERATURE PROFILES AT TIME = 1 HOUR

$$\Delta = \int \alpha * T(x)\, dx$$

$\Delta_A = 35.4\text{nm}$ $\Delta_B = 36.15\text{nm}$ $\Delta_C = 38.42\text{nm}$

SURFACE DISPLACEMENTS AT TIME = 1 HOUR

USNO PRIMARY MIRROR THERMAL INERTIA STUDY

PCFRINGE VERSION 3.4
RMR DESIGN GROUP INC
16:24:58 2-27-1992

FRINGE VERIFICATION

							J	J	I	I	J	J							
					J	I	I	I	I	I	I	I	I	J					
			J	I	I	I	I	H	H	H	H	I	I	I	I	J			
		J	I	I	I	H	H	H	H	H	H	H	H	I	I	I	J		
		I	I	H	H	H	H	H	H	H	H	H	H	H	H	I	I		
	J	I	I	H	H	H	G	G	G	G	G	G	H	H	H	I	I	J	
	I	I	H	H	H	G	G	G	G	G	G	G	G	H	H	H	I	I	
J	I	I	H	H	G	G	G	G	G	G	G	G	G	G	H	H	I	I	J
J	I	H	H	H	G	G	G	G	G	G	G	G	G	G	H	H	H	I	J
I	I	H	H	H	G	G	G	G			G	G	G	G	H	H	H	I	I
I	I	H	H	H	G	G	G	G			G	G	G	G	H	H	H	I	I
J	I	H	H	H	G	G	G	G	G	G	G	G	G	G	H	H	H	I	J
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	J	I	I	H	H	H	G	G	G	G	G	G	H	H	H	I	I	J	
		I	I	H	H	H	H	H	H	H	H	H	H	H	H	I	I		
		J	I	I	I	H	H	H	H	H	H	H	H	I	I	I	J		
			J	I	I	I	I	H	H	H	H	I	I	I	I	J			
					J	I	I	I	I	I	I	I	I	J					
							J	J	I	I	J	J							

WAVELENGTH .633 MICRONS

	N	RMS				
RAW	0	.003				
PLANE	2	.003				
.00000		.00000				
SPHERE	3	.000				
.00000		.00000	.00481			
4TH ORDER	8	NEGATIVE VARIANCE				
.00000		.00000	.00479	.00000	.00000	.00000
.00000		.00038				
	10	NEGATIVE VARIANCE				
.00000		.00000	.00479	.00000	.00000	.00000
.00000		.00038	.00000	.00000		

STREHL RATIO 1.000 AT DIFFRACTION FOCUS

FOURTH ORDER ABERRATIONS

MAGNITUDE WAVES	ANGLE DEG	DESIGNATION
.000	167.4	TILT
.010		DEFOCUS
.000	21.6	ASTIGMATISM
.000	77.7	COMA
.002		SPHERICAL ABERRATION

RESIDUAL WAVEFRONT VARIATIONS EVALUATED AT DATA POINT LOCATIONS
USING A SURFACE DESCRIBED BY THE ACTUAL DATA

PTS	RMS	MAX	MIN	SPAN	STREHL
312	.003	.005	-.004	.009	1.000

RESIDUAL WAVEFRONT VARIATIONS EVALUATED AT UNIFORM GRID POINTS
USING THE ZERNIKE POLYNIMIAL SUFACE

PTS	RMS	MAX	MIN	SPAN	STREHL
688	.003	.005	-.004	.009	1.000

RMS CALCULATED SOLEY ON ZERNIKE COEFFICIENTS= .003

RESIDUAL WAVEFRONT VARIATIONS EVALUATED AT UNIFORM GRID POINTS
USING LOCAL INTERPOLATION OF DATA VALUES

PTS	RMS	MAX	MIN	SPAN	STREHL
684	.002	.005	-.004	.008	1.000

ZERNIKE POLYNOMIAL COEFFICIENTS

.0000	.0000	.0048	.0000	.0000	.0000
.0000	.0004	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000

GENERAL POLYNOMIAL COEFFICIENTS

.0000	.0000	.0073	.0000	.0000	.0000
.0000	.0023	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000
.0000	.0000	.0000	.0000	.0000	.0000

ASPHERIC COEFFICIENTS

FOCUS	AD	AE	AF	AG	AH
.0073	.0023	.0000	.0000	.0000	.0000

USNO PRIMARY MIRROR THERMAL INERTIA STUDY

PCFRINGE VERSIO
RMR DESIGN GROU
16:26:56 2-27·

CONTOUR STEP	WIDTH	PAGE SIZE	-M-	-N-	-P-	-Q-
.002	.300	2.000	-.002	-.001	.001	.002

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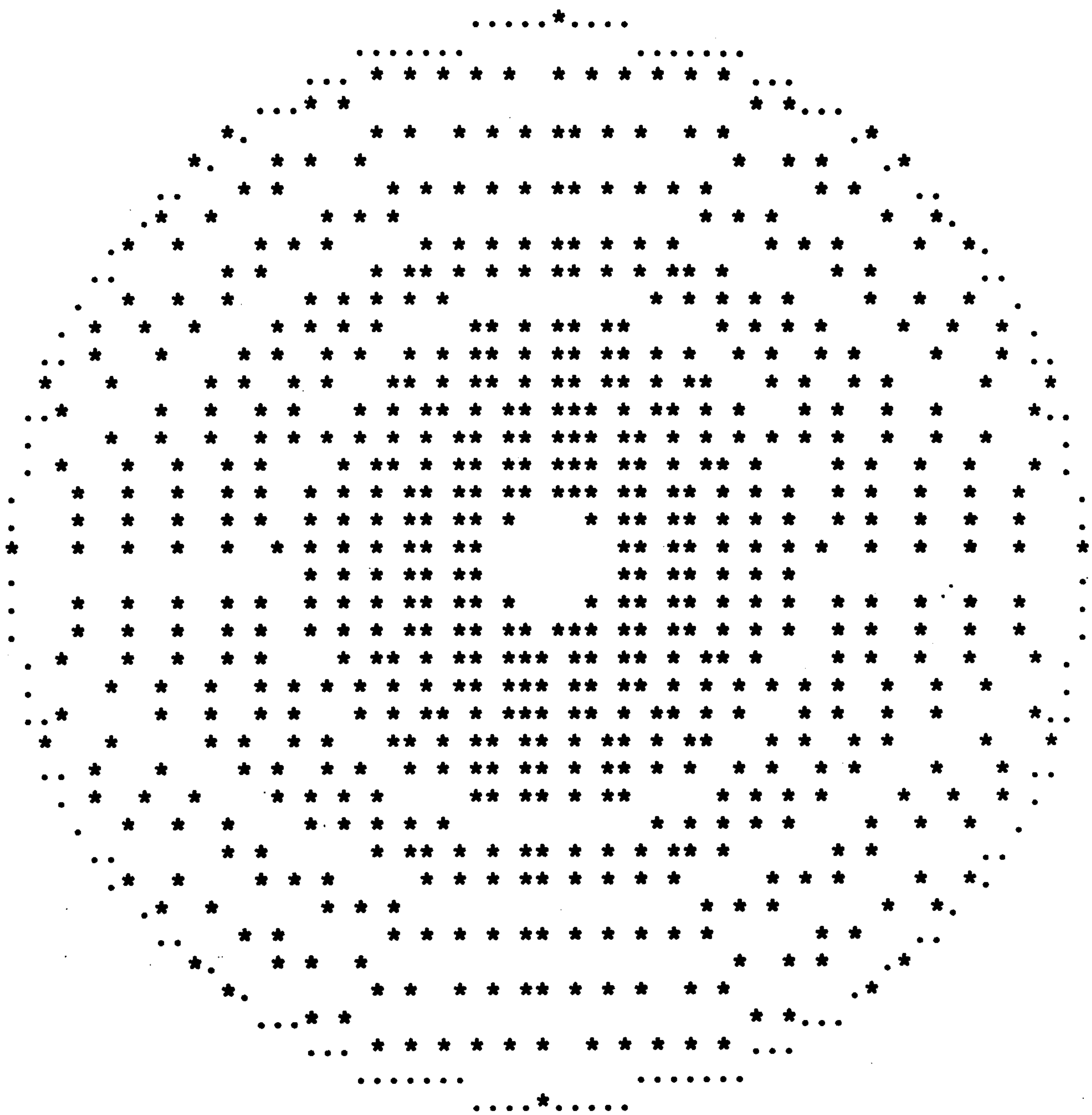
+            ++          +           +            ++          +
                                     k
                               RRRRRRRRRRRRR
                      PPPPPP                   PPPPPP
                    P QQQQ  QQQQQQQ     QQQQQQQ  QQQQ P
+                    Q   QQ                     QQ   Q
+               RR   QQ       PPPPPPPPPPPPPPP       QQ   RR
                   QQ    PPP                   PPP    QQ
          Q  Q   Q    PPP        NNNNNNNNN        PPP    Q   Q  Q
           R   Q    PP      NNNN           NNNN      PP    Q   R
          R  QQ   PP    NNN                     NNN    PP   QQ  R
+        Q  Q    P    NN          MMMMMMM          NN    P    Q  Q
      PP Q Q   PP   NN       MMMMMMMMMMMMMMMMM       NN   PP   Q Q PP
          Q   P    NN     MMMM               MMMM     NN    P   Q
         Q   P    N     MMM                     MMM     N    P   Q
       QQ   P    N     MM                         MM     N    P   QQ
+      QQ  PP   N     MM                           MM     N   PP  QQ
+   QQQQ   P   NN    MM              L              MM    NN   P   QQQQ
    R  Q  P    N    MM          L         L          MM    N    P  Q  R
          P   N    MM          LL         LL          MM    N   P
   R  Q   P   N    MM                                 MM    N   P   Q  R
   R  Q   P   N    MM        L               L        MM    N   P   Q  R
+ aR  Q   P   N    MM        L               L        MM    N   P   Q  Ra
   R  Q   P   N    MM        L               L        MM    N   P   Q  R
   R  Q   P   N    MM                                 MM    N   P   Q  R
          P   N    MM          LL         LL          MM    N   P
    R  Q  P    N    MM          L         L          MM    N    P  Q  R
+   QQQQ   P   NN    MM              L              MM    NN   P   QQQQ
+      QQ  PP   N     MM                           MM     N   PP  QQ
       QQ   P    N     MM                         MM     N    P   QQ
         Q   P    N     MMM                     MMM     N    P   Q
          Q   P    NN     MMMM               MMMM     NN    P   Q
      PP Q Q   PP   NN       MMMMMMMMMMMMMMMMM       NN   PP   Q Q PP
+        Q  Q    P    NN          MMMMMMM          NN    P    Q  Q
          R  QQ   PP    NNN                     NNN    PP   QQ  R
           R   Q    PP      NNNN           NNNN      PP    Q   R
          Q  Q   Q    PPP        NNNNNNNNN        PPP    Q   Q  Q
                   QQ    PPP                   PPP    QQ
+               RR   QQ       PPPPPPPPPPPPPPP       QQ   RR
+                    Q   QQ                     QQ   Q
                    P QQQQ  QQQQQQQ     QQQQQQQ  QQQQ P
                      PPPPPP                   PPPPPP
                               RRRRRRRRRRRRR
                                     k
+            ++          +           +            ++          +
```

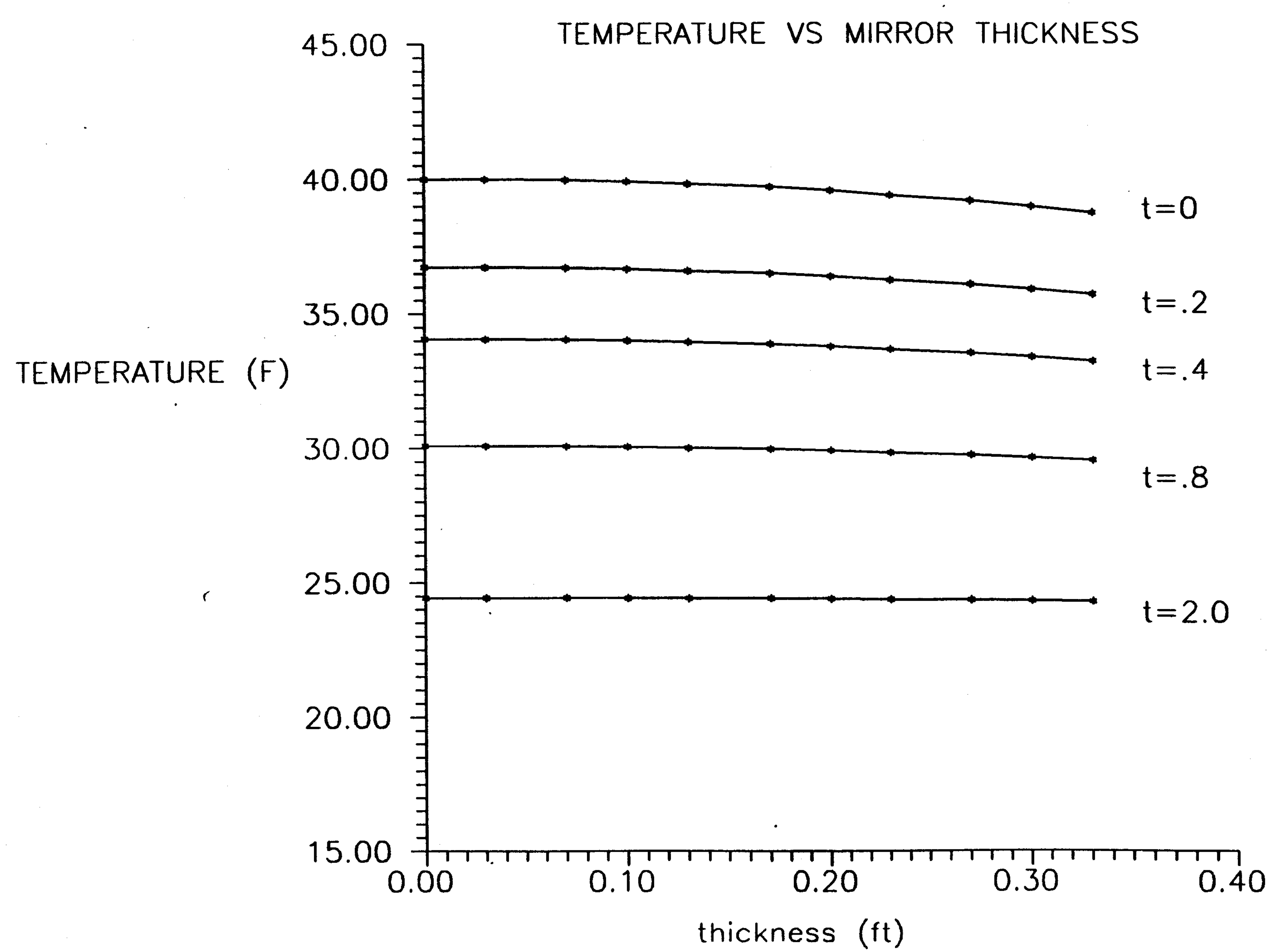
USNO PRIMARY MIRROR THERMAL INERTIA STUDY

PCFRINGE VER
RMR DESIGN G
16:27:26 2-

SPOT DIAGRAM
RADIUS = .00814931ARCSEC



--RED



TEMPERATURE VS MIRROR THICKNESS
TEMPERATURE (F)
45.00
40.00
35.00
30.00
25.00
20.00
15.00
0.00
0.10
0.20
0.30
0.40
thickness (ft)
t=0
t=.2
t=.4
t=.8
t=2.0