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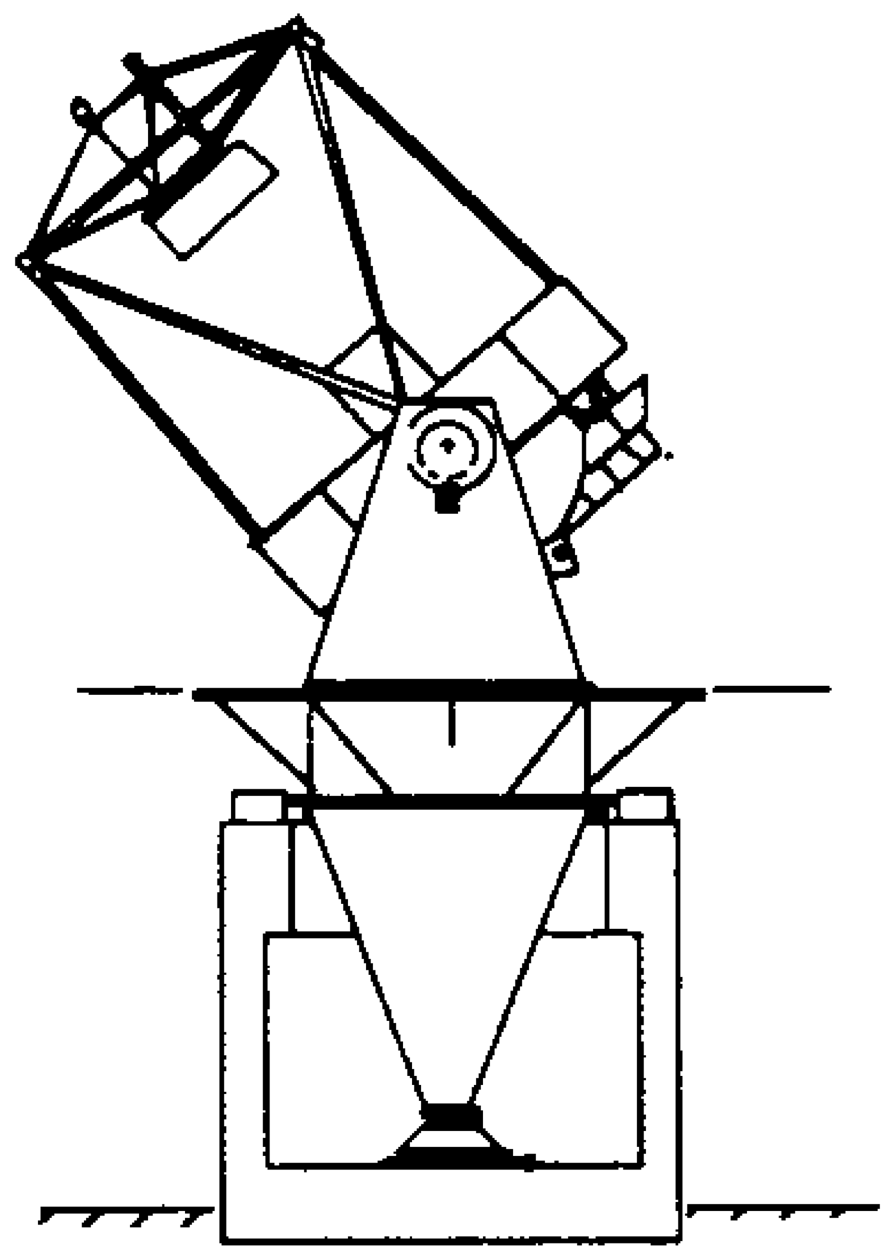

3.5 Meter Telescope

Summary Description of the NOAO 3.5M Mirror Support Design

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SUMMARY DESCRIPTION OF THE NOAO 3.5M MIRROR SUPPORT DESIGN

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I. INTRODUCTION

This summary describes the philosophy that guided the support system design, explains the functional requirements that influenced the design of the different components, and summarizes the finite-element analysis required.

II. PHILOSOPHY

The 3.5-meter mirror has always been considered a test bed for development of technology for larger borosilicate mirrors.[1] It is intended that the 3.5-meter mirror support provide a test of design features considered necessary for the NOAO 8-meter mirrors. The NOAO philosophy for 8m mirror support design developed by Pearson, Stepp and Keppel has been described in a previous paper.[2] This philosophy is based on the following assumptions:

1. Thermal distortion of the mirror will be a significant concern; therefore the support system should not interfere with the proposed ventilation system.
2. To minimize local distortions of the optical surface, all support forces should be applied at the back surface of the mirror.
3. For 8-meter mirrors, it will not be satisfactory to react wind loading and other external disturbances on only three hard defining points; therefore, a distributed, statically-determinant defining system will be incorporated into the support system design. Disturbing forces will be reacted at all the support points simultaneously.
4. It will be difficult to meet the 8-meter mirror thermal control specification, and it will be difficult to design (in advance) a passive support system that meets the specification for support force accuracy at all zenith angles; therefore we believe active optics will be needed.
5. However, because the mirrors will be relatively stiff compared to their own weight, a passive support system should be able to provide a good first order approximation to the needed forces. Therefore to reduce the number of potential failure modes, and to minimize the dynamic range required in the active optics system, the support will be designed to perform in a passive mode as much as possible.
6. The mirror support system must be designed to work reliably for years without extensive maintenance. It should be possible to service the support system without removing the mirror or mirror cell from the telescope.
7. The support system should be compatible with the telescope environment. This means the support mechanisms should withstand mountaintop conditions without failure, and the supports should not do anything to degrade the environment around the telescope.

III. DETAILED DESIGN REQUIREMENTS

The general assumptions stated above have specific ramifications for the design of support system components. Some of these are listed below:

1. Because the lateral support forces are applied at the back surface of the mirror, the resultant of the lateral forces will not pass through the center of mass, creating an overturning moment that must be compensated by forces from the axial supports. We call these axial forces needed by the lateral support "auxiliary forces". The precise distribution of these auxiliary forces was chosen to minimize optical surface distortions, as described in section IV.
2. To achieve static equilibrium, some of these auxiliary forces must be negative. For this reason, and to allow the active optics system to work all the way to the horizon, the support mechanisms are design to both push and pull.
2. To avoid interfering with the thermal control system, all support mechanisms are external to the mirror structure, and they are not allowed to block any ventilation nozzles.
3. We need to provide distributed defining of the mirror position and orientation, and this defining system must have high stiffness to resist wind loading. At the same time we want each individual support mechanism to be astatic, that is, to apply the correct force (in magnitude and direction) to the mirror regardless of small shifts between the mirror and cell. The spring constant of the system should be high, while the spring constant of each individual support should be as low as possible. To accomplish these goals, the axial support system has been designed as a hydraulic whiffle-tree. Each support will incorporate a liquid-filled diaphragm cylinder; these will be connected together in three groups, forming a statically determinant system.
4. All support mechanisms are designed to be serviced from the back of the mirror cell.
5. To avoid degradation of the telescope environment, heat sources must be minimized. Also, any liquids that could potentially leak out must be non-toxic and non-corrosive.
6. Where possible, the support mechanisms will have covers and seals to make them "moth-proof".

Design of the axial support mechanisms

The axial support mechanism design is shown in NOAO drawing 3500.0000930E, and its relation to the mirror, the cell and the lateral supports is shown in the untitled layout drawing.

The support is attached to the mirror by an Invar pad bonded to the back of the mirror at a junction of ribs. This point in the mirror structure has good local stiffness , and is as far as possible from the holes in the back of the mirror that are used for ventilation. The Invar pad is connected to the support by a lightweight tubular column. A cross-blade flexure next to the mirror, and the compliance of the Bellofram cylinder at the other end, minimize the transmission of transverse forces and bending moments to the mirror. A load cell mounted in the column measures the force applied to the mirror. The column can be divided at a coupling, to allow the upper portion to be unscrewed from the Invar pad, or to allow the rest of the mechanism to be removed for servicing.

Active alignment may be required in the telescope; we plan to tilt the primary mirror, if necessary, by adjusting the fluid level in each zone of supports under computer control.

We want the load path to be as stiff as possible, to resist wind loading, so the Bellofram cylinder is directly below the support column. Tests show the spring constant of the support mechanism is 4200 newtons per mm (24,000 pounds per inch).

Most of the weight of the column is in the coupling and the load cell, which are close to the Bellofram cylinder. At horizon-pointing, most of this weight will be carried by the Bellofram, and only a small transverse load will be exerted on the mirror.

The fluid chosen for the hydraulic system is a 50-50 solution (by weight) of glycerine and water. This is the same mix used in the new support for the Kitt Peak 2.1m telescope. It was chosen because it has a low freezing point (-23 ° C) and is relatively non-toxic and non-corrosive.

The Bellofram cylinders cannot pull, they can only push. As the zenith angle increases the upper cylinders in each zone tend to develop a negative gage pressure because of the hydrostatic head. To keep the system working as calculated at all zenith angles (up to 75 degrees), it is necessary to prevent the fluid pressure from falling below zero gage. Therefore, a preload pressure is needed, and in the axial supports this preload is provided by the constant-force spring (item 38 in the drawing). The spring force is chosen to be proportional to the area of the cylinder. This preload also reduces hysteresis in the Belloframs at low load levels.

The preload force, any active force, and the auxiliary force are applied to the mirror by means of the lever (item 6). This lever couples into the loading column through a ball that is a close fit between two flats. This design minimizes introduction of lateral loads into the column. The active force is applied by deflecting a flat spring rigidly attached to the lever. The auxiliary force is applied by means of an auxiliary cylinder (item 10), which can be located above or below the lever as required.

The auxiliary cylinder is coupled to the lateral support hydraulic system, so as the load increases on the lateral supports the auxiliary force increases proportionately. For fine adjustment, this cylinder can be slid along the lever to change the leverage.

The active optics mechanism is designed to provide fine force adjustments, and to hold the force setting without being continuously energized. A stepper motor having 200 steps per revolution drives a ball screw through a 25:1 gearbox. This ball screw drives a nut attached to the end of the flat spring, applying a force to the mirror. To prevent back driving at high force levels, a slip clutch (item 39) is attached to the end of the motor.

Some recent design changes have not yet been documented on the assembly drawing. One change is the installation of three microswitches on the stiffener plate (item 30). Two of these serve as limit switches, and the third provides an indication of the center of the active force range. A second change is the incorporation of a slot for a printed circuit card that reads the load cell, and controls the stepper motor. There have also been clearance holes added for cabling.

The axial support assembly is sealed by sheet metal panels at the ends. There is a flexible rubber seal around the support column, and the holes for wires and tubes leading into the mirror cell will be sealed with foam rubber grommets.

Design of the lateral support mechanisms

The lateral support mechanism design is shown in NOAO drawing 3500.0001941D. The lateral forces will be applied to the mirror at the holes in the back plate, with clearance allowed for ventilation.

The lateral supports are entirely passive. The weight of the mirror is reacted through a lever mechanism to a Bellofram cylinder. Although the design originally proposed for the 8-meter lateral support was divided into two hydraulic zones to define the mirror against rotations about the optical axis, the design for the 3.5-meter mirror has all of the lateral supports connected together in one system. This was done to simplify adjustments of fluid level for centering purposes.

We want the lateral support to apply the correct force to the mirror, even if there is a small shift between the mirror and the cell. Motions of half a millimeter in any direction can be accommodated, with minimal transverse forces introduced. Axial motions are accommodated by swinging of the lever mechanism, and transverse motions are accommodated by the parallelogram flexures built into the links. The Delrin pad that contacts the mirror is crowned slightly to maintain good contact with the wall of the hole in the mirror.

It is important that the mechanism be balanced, so that it does not exert any axial force at zenith-pointing. The adjustable counterweight (item 9) is used to balance the mechanism before installation in the mirror cell.

The lateral supports are preloaded by a compression spring (item 19).

A loose flap (item 15) minimizes air flow past the lever.

A transfer plate is bolted to the machined front surface of the mirror cell, and the lateral support mechanism is bolted to this transfer plate. This allows the mechanism to be unbolted and removed from the back of the mirror cell for servicing.

Because of the flexibility of the parallelogram flexures, the Bendix flex pivots, and the main lever (which is loaded in bending), the lateral supports are not as stiff as the axial supports. The measured spring constant is 1750 newtons per mm (10,000 pounds per inch). The decentration of the mirror between zenith and horizon caused by its own weight would be about half a millimeter, if not compensated by an adjustment of the fluid level.

The lateral position of the mirror, in the elevation direction, will be defined by the fluid level in the lateral support system. The position in the transverse direction will be defined by tangent arm linkages, similar to those used in the ARC Telescope.

Design of the fluid level control mechanisms

It is necessary to have an accumulator unit on each hydraulic system to control the system level by controlling the fluid volume. The design of these accumulator units is shown in NOAO drawing 3500.0003029E.

The volume of fluid in each support system will be controlled by computer to ensure that the mirror pointing and position will remain correct even in the presence of thermal expansion of the system, slow leaks, or loads changing with zenith angle. Feedback for control of these units will come from four LVDT's built into the mirror cell. Low frequency defining of the mirror position will be done by adjustment of the fluid levels, while high frequency defining (to reject wind loading) will be handled by the stiffness of the supports.

The accumulator units were designed to use many of the same components as the active optics mechanisms. The mechanism works in the same way except the ball screw drives a master cylinder, adding or subtracting fluid from the support system, instead of moving the end of a spring.

Design of the LVDT assemblies

NOAO drawing 3500.0002663D shows the design for the three axial support LVDT sensor mechanisms. The LVDT mechanism for sensing the lateral position of the mirror is currently being designed.

The axial LVDT mechanisms are designed as adjustable pads onto which the mirror can be placed for installation of the axial support connecting columns. There are three of these pads in the mirror cell 120 degrees apart. Each is located close to the centroid of one support zone. The design incorporates a differential screw to adjust the height of the pad. The pad has a universal joint flexure to ensure even support of the mirror weight. The spring-loaded LVDT probe will be compressed flush with the top of the pad during the installation of the mirror, then the pads will be retracted about a millimeter.

The entire unit can be retracted from the back of the mirror cell for servicing.

IV. SUMMARY OF FINITE-ELEMENT ANALYSIS

The finite element model of the mirror blank used six degree of freedom plate bending elements. Three levels of nodes were used to obtain an acceptable aspect ratio in the rib elements at the outer edge. The one-half symmetric model has a total of 994 nodes and 5802 active degrees of freedom.

To account for the decrease in stiffness due to the cast holes the bottom plate thickness in the model was reduced from 26.7 mm to 17.1 mm. The material density of the bottom plate was increased to compensate for the loss of weight.

Axial support optimization

We started the support design effort by studying several possible patterns of axial support locations. Several likely looking support patterns were selected, ranging from as many as 90 supports to as few as 66 supports. Finite-element analysis of load cases with unit loads applied at each support gave us displacement information that we combined into an influence coefficient matrix. The magnitudes of the support forces were optimized by a least-squares fit that minimized the distortion of the optical surface. Five different support patterns were optimized in this manner. The WIN telescope error budget allows 0.09 arc seconds image spread for the mirror support. The pattern with the minimum number of supports (66) produced a distortion resulting in 0.08 arc seconds FWHM of image spread. This was judged to be acceptable performance, so the 66-point support was selected.

This support pattern is composed of three equivalent hydraulic zones, each contained within a 120 degree sector. Since the pattern includes supports along the symmetry lines of the mirror structure, and since it is not possible for different hydraulic zones to share supports, the 120 degree sectors are skewed relative to the structural symmetry of the mirror. This means each group of supports does not have left-right symmetry. However, the overall support pattern does have six-fold symmetry. Thirteen unique support types occur in one 60 degree sector.

The optimized forces for this axial support design range from 211 to 425 newtons. For simplicity we wanted to use catalog sizes of diaphragm cylinders if possible. We chose a series of diaphragm sizes from the Bellofram catalog and fit the required forces to the areas of these diaphragms in much the same way lens radii in an optical design are fit to test plates. After eleven reoptimizations with increasing numbers of constraints we had fitted all the forces to catalog sizes. These modified forces were applied to the finite-element model and the resulting optical surface distortions were still quite acceptable. It may be necessary to make minor changes in the support forces after two-position testing.

Lateral support optimization

After selecting the axial support locations, we selected locations for the lateral supports. We used a series of least squares fits to guide us in choosing the particular holes at which to apply the lateral forces. Forty potential positions, distributed over the back of the mirror, were studied. Individual unit force load cases were analyzed for each position. Displacement information from the 66 axial and 40 lateral unit load cases was combined into an influence coefficient matrix, and a least-squares fit determined the optimum forces for horizon-pointing support. Axial force load cases were included because the resultant of the lateral forces does not pass through the center of gravity of the mirror, and the overturning moment must be balanced by auxiliary forces at the axial supports.

Some of the lateral forces determined by the least-squares fit were negative, in other words, the support forces would be downward. The supports at these locations were eliminated. The optimization was repeated with fewer lateral locations included, and this time any lateral supports with negative or small forces were eliminated. After several iterations of this type the number of lateral supports was reduced to 24. It could have been reduced further (the distortions were acceptable with as few as 18 lateral supports) but we chose to retain 24 lateral supports to prevent the force applied at each support from getting too large.

For ease of fabrication we decided to make all the lateral support mechanisms identical. The only variation in force between individual supports is caused by hydrostatic pressure variations in the system. At the horizon, the forces at the bottom of the mirror (886 newtons) are larger than those at the top (754 newtons). With the lateral force magnitudes determined by the assumption of identicals mechanisms, the horizon-pointing least-squares fit optimization was redone to determine the auxiliary forces required at the axial supports. The optical surface resulting from this reoptimized system was still quite acceptable.

The auxiliary forces perform two functions. First, they maintain static equilibrium by counter-balancing the overturning moment of the off center lateral support. In addition, they help correct the figure produced by the lateral support.

Two effects in the axial supports can help counter the overturning moment: overall reactions at the three zones, and hydrostatic pressure variations within each zone. A decision was made that the cylinders in the axial supports could not be large enough for the hydrostatic pressure variations to provide all of the counterbalancing moment. This meant that the overall reactions at the three zones would not be zero. In other words, if the forces applied by the supports in each zone were replaced by their resultant, acting at the centroid of the zone, the resultants from the two upper zones of supports would be pulling on the mirror at horizon-pointing, and the resultant from the lower zone would be pushing.

All of the information required for the detailed support force calculations is contained in a Lotus spreadsheet. This handles the algebra and accounting for such things as piston areas, forces and moments exerted on the mirror, spring selection, system pressures, etc. The calculations are based on the following parameters: (1) specific gravity of hydraulic fluid, (2) zenith angle limit, (3) weight of mirror, (4) position of mirror center of mass, (5) preload pressure in each system, (6) preload spring leverage, (7) thickness of mirror cell structure. When one of these is changed, the entire spreadsheet recalculates.

Support Error Analysis

Errors in the magnitudes of the axial forces may result from fabrication tolerances, hysteresis in the mechanisms, or errors in the active force system. Since the lateral support also requires axial forces, axial force errors will produce deviations from the calculated optical surface at all zenith angles.

Because the support system is designed as a hydraulic whiffle tree, an axial support force error at one location will be reacted by forces at all other support locations. In the error study conducted by Bill Keppel, loads were applied individually at each axial support. The displacement information from these load cases was synthesized to model a force error plus the 66 reactions. A force error was evaluated at each of the 13 different types of axial support.

If the force errors on the 66 supports occur randomly, the effects of the individual errors can be added in quadrature. The results of the analysis show that individual force errors should be smaller than about 2.4 newtons each.

In a more recent study, Myung Cho modeled three random distributions of force errors within the range of ± one newton per support. The average amplitude of surface error in these cases was 0.02 λ P-V, and the average spot size was 0.004 arc seconds FWHM, or 0.005 arc seconds for 75% encircled energy.

In addition, we have studied the effect on the optical surface of position errors in the axial supports. Individual loads were applied at 12.7 mm offset, which in practice is very large, in two perpendicular directions for each of the 66 axial supports. The displacement information from each of the load cases was synthesized as described above. There were only very small differences in optical surface distortion between the offset and non-offset support position load cases. This indicates that the mirror is not very sensitive to position errors in the axial support.

Active Optics Calculations

The active optics system is similar to a system developed previously for a 1.8m borosilicate mirror.[3] The active forces are applied through the axial support mechanisms. At each location the axial force can be thought of as having two component parts: a support force, which varies as a function of zenith angle, and a corrective force, which is constant with zenith angle.

As described in previous sections, the support forces have been calculated by finite-element analysis, and this has provided the starting point for the initial system design. The support forces actually used in the telescope will be calibrated by means of two-elevation testing in the optical shop, and eventually by multi-elevation testing in the telescope.

The corrective forces will be calculated on the basis of optical testing performed in the telescope. The forces required to correct different distortions of the mirror surface have been determined by finite-element analysis. Individual unit force cases were run and the results were assembled in an influence coefficient matrix. To determine the set of forces that will best correct a particular distorted surface a least-squares fit will be performed.

The values in the influence coefficient matrix will be updated after the stiffness of the mirror is calibrated in our Phase II testing.

In our current plan for operation of the active optics system, there will be a monitoring loop in the computer that will check load cell readings several times a minute. When one or more of the forces is off by more than a specified tolerance (perhaps 2 newtons), the forces will be adjusted. The corrective force components will periodically be updated. When the telescope operator wishes to update the figure correction, he will initiate an optical test. The information about figure errors will be used to calculate new corrective force components. Then, at the next cycle of the monitoring loop the force changes will be implemented.

It is important to make the force changes without affecting the pointing of the telescope. To avoid rigid body tilts, constraints will be imposed on the least-squares fit to ensure:

$$\sum F_z = 0$$

$$\sum M_x = 0$$

$$\sum M_y = 0$$

Elastic tilts and defocus can also be minimized if this becomes necessary.

To minimize image jitter during adjustments, the force changes will be divided into enough parts to ensure that the maximum force change in any one cycle is less than 8 newtons.

The control system will also check to see if the sum of the support and corrective force components prescribed at any location would exceed the safe limit for mirror loading. If it would, the corrective force component must be reduced. Smoothing factors can be applied in the least squares fit to reduce the force at the required locations while maintaining the optical correction as well as possible.

V. REFERENCES

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