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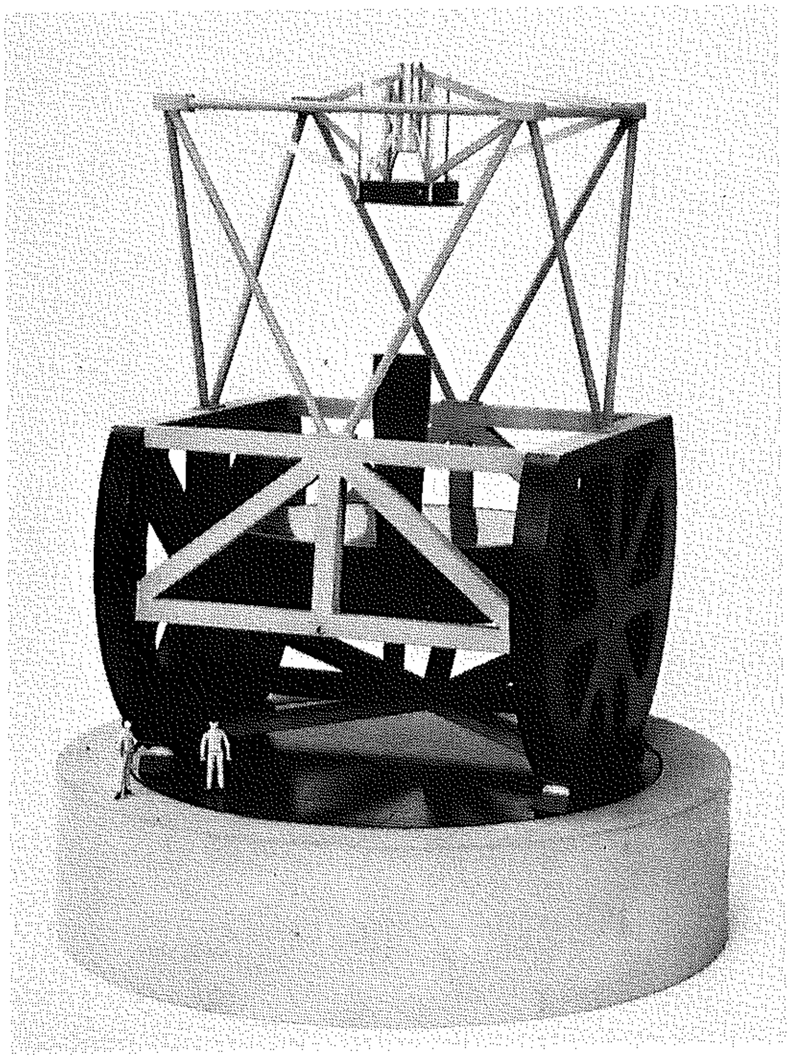

F/6.5 Wide-Field Secondary Support System

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The Magellan Project wide-field secondary support system is shown in drawing E271043 and the finite element graphics plot in Figure 1. In addition, the mirror cell structural design is shown in Figure 2, while the secondary actuator is shown in Figures 3 and 4, and drawing D271042.

The secondary support system is defined here as:

1.) Secondary mirror.

2.) Mirror support system (tentatively mercury belt/vacuum).

3.) Secondary mirror cell.

4.) Secondary actuators.

5.) Secondary support structure.

As shown, six linear actuators are used to maintain optical alignment due to thermal and gravity effects. They can also be used for fine guiding of the optical axis to relieve telescope tracking requirements (see Magellan Project Report No. 2, "Fine Guiding by Secondary Mirror Tilt in the Magellan Wide-Field f/6.50 Cassegrain"). The six actuators define the six degrees of freedom of the secondary assembly. They are shown schematically as bold lines in Figure 1, a quartering view looking down toward the system with the telescope pointed about 45° off zenith. They also appear as stubby cylinders in the drawing E271043. The thin disk is either a friction-driven disk or gear, driven at its perimeter by a DC servomotor (not shown).

Three of the actuators have axes parallel to the telescope optical axis and are referred to herein as "axial actuators." They define the piston (or focus) and two tilt motions of the assembly. The other three "transverse actuators" are in a plane perpendicular to the optical axis. The two 45° actuators define the other two translations, while the horizontal actuator defines the remaining degree of freedom, rotation about the optical axis.

The entire assembly attaches to the telescope structure at the "cantilevered" vane nodes (see Magellan Project Report No. 8, "Universal Vane System"). This attachment is made at the four solid corner blocks of the support structure shown in E271043, with eight cap screws, using a dowel pin/diamond pin for alignment with the vane system. The four planar trusses which make up the secondary support structure then provide the in-plane axial stiffness for the axial actuators, while the transverse actuators are supported directly to the corner nodes, which are virtually coincident with the vane cantilevered nodes. The small eccentricities at the ends of the planar trusses, as well as a slight eccentricity to the vane cantilevered nodes, have been included in the finite element model, and have a negligible deleterious effect on stiffness.

It is assumed that the mirror will be supported within the cell by a mercury belt/vacuum (or other similar) system. Mercury belt stiffness is quite difficult to determine. Therefore, two "hard points" were used for stiffness in the modal analyses, while actual cell distortion due to gravity loading was approximated in the static analysis by manually calculated load vectors. It is anticipated that, during detail design, mercury belt stiffness will be determined and accurately modelled, or an alternate support method used, such as a roller chain.

The mirror cell structure is shown in Figure 2. It consists of front and back plates (3/16"), central tube, outer ring, six radial ribs (all 1/4"), and 3/8" tube cap at the back of the cell. All materials, unless otherwise noted, are carbon steel.

The actuators must not only support the weight of the mirror and cell (1400 lbs.), but also position it precisely while accommodating a range of motion determined by structural deflections due to thermal and gravity effects. Exaggerated actuator motions are shown in Figures 3 and 4.

While presently a preliminary design, considerable analysis has been performed in an effort to achieve high stiffness and low friction. As can be seen in D271042, the unit consists of two assemblies (the body assembly and screw assembly) connected to each other by a nut subassembly and a disk flexure subassembly.

The body assembly is made of four cylindrical (or conical) rings and incorporates a flexure at one end to attach to the secondary support structure. The screw assembly includes a recirculating roller screw with a tube assembly at one end and incorporates a flexure at the other end that attaches to the secondary mirror cell.

The screw is guided relative to the body by the disk flexure subassembly. This method stiffly defines the screw to the body for all three rotation degrees of freedom and two translations while allowing "free" motion longitudinally. Although the longitudinal motion has a considerable force/deflection curve associated with it, it offers essentially zero friction. The nut subassembly then defines the sixth degree of freedom (longitudinal motion) of the screw relative to the body. It also incorporates the friction disk (or alternatively, gear) which is ultimately driven by the DC servomotor as shown.

The disk flexure subassembly actually consists of two parallel sets of disk flexures. Each disk within a set is then connected to its mating disk in series. Both the series and parallel connections are accomplished by means of the stiff "disk coupler" as shown. This allows for a much greater range of motion (stroke) within the same space. There is a spring between the body and screw to assure a minimum load on the nut and bearing. This establishes a minimum spring rate and assures that the load through the nut and bearing does not reverse, and therefore eliminates axial backlash.

In addition, the single external spring preload assembly (E271043) would be required to fully eliminate axial backlash at the two bottom axial actuators when operating near horizon. Without this feature, a slight discontinuity and slightly degraded performance would occur only near horizon. Therefore, this may be considered an optional feature.

Some of the expected performance data and design parameters for the actuators are summarized below:

The screw/nut assembly is an SKF Transrol recirculating type roller screw, PVK 25x1, preloaded to 200 lbs. (25 mm screw diameter x 1 mm lead). According to catalogue information, this type of screw has "achieved resolutions of 25 nm." Keck has tested this type screw (12 mm x 1 mm) and gotten accuracies on the 2 nm level. Our requirement (final positioning accuracy) is only 600 nm. This screw/nut assembly has a dynamic load rating of greater than 4000 lbs. and static rating of over 7000 lbs. (irrespective of the flexure which is integral to the screw). Our maximum design load on one screw (the transverse actuators) is 1,300 lbs. This includes 848 lbs. external load on the actuator plus 319 lbs. maximum force from the preload spring plus 128 lbs. maximum load from the disk flexures.

The bearing between the nut and the actuator body is a Kaydon type X No. KB030 four-point contact ball bearing, precision class 6. It has a 3" bore and 5/16" cross-section. It has a thrust rating of 1410 lbs. dynamic and 4600 lbs. static. It also sees a maximum load of 1300 lbs. thrust, with negligible radial and moment load.

The spring is a Century No. D-1342 heavy pressure chrome vanadium die spring. Its load range in this application is 205 lbs. to 319 lbs., representing 43% to 65% compression.

The feedback unit is an LVDT, Shaevitz No. 250MHR. Its nominal linear range is +/- 6.25 mm. Our stroke requirement is +/- 5 mm. According to the catalogue, these units have "truly infinite resolution." That is, the "readability of the external electronics represents the only limitation on resolution." When asked what resolution one might expect (with "real" electronics available today), the manufacturer responded that our 2.4 microinch goal (600 nm/10 = 60 nm = 2.4 microinch) is near the lower limit; and that the 8 microinch requirement (600 nm/3) should offer no problem. The core is supported through a connecting rod (AISI 300 stainless steel) to prevent distortion of the LVDT's magnetic field. The LVDT body is supported by a phenolic cylindrical wedge preloaded by a hollow threaded fitting.

Although the resolution of the LVDT is excellent, it has some time/temperature instability. That is, if everything else is held constant (core position, external electronics, etc.), the reading will vary over time, and if the temperature is varied, over a relatively short time. Therefore, it is possible that the LVDT cannot be used as an absolute encoder, but might be used in combination with an absolute encoder on the servomotor. This would require that the drive be a gear-type drive or have a separate gear drive for a separate absolute encoder.

Since the screw assembly is well guided to the body, this makes for the ideal location to mount the feedback unit. It is as close to the target (wide-field mirror) as possible while sensing no motion from any other actuator. It therefore does not contribute "cross-talk" and should be a very stable mounting. Of course, there is still a relatively soft spring between the actuator and mirror (the flexure on the end of the screw), but this just becomes one more spring in the chain which must be compensated for by look-up table or loop closure.

The disk flexures are .010 thick high strength stainless steel. Under the +/- 5 mm motion required of the axial actuators, they develop a maximum stress of 88 ksi. The material has a strength of 280 ksi. The disk flexures are in a parallel load path to the structurally-critically load path for the actuator. That is, a failure of the disk flexures would not cause a loss of the load, but would necessitate replacement or repair of the unit.

The complete disk flexure assembly exerts a maximum force of +/- 128 lbs. at full stroke. Under axial motion of the disk flexures, out-of-plane membrane stiffness varies geometrically with the motion and predominates over out-of-plane bending stiffness. Therefore, the force curve for the flexures is very nonlinear. However, the force-deflection curve should be very smooth and repeatable, with very low hysteresis. This, in combination with the nonlinearity being enclosed within the encoder end reference points, should negate any ill effects of the nonlinearity.

Each disk flexure set (that is, one set toward the front of the actuator, and one toward the rear) has a radial stiffness of 330,000 lbs./in. The torsional stiffness (rotation about the longitudinal axis of the actuator) is 7,300 in-lbs./deg. This torsional stiffness represents a negligible windup of the system under the maximum torque loading (when considering the effect on positioning accuracy).

The axial actuators are flexured at their ends to accommodate the transverse motion of the cell caused by the transverse actuators. Conversely, the transverse actuators are flexured to accommodate the axial motion of the cell caused by the axial actuators. The total axial motion required is +/- 5 mm. While the actual requirement for transverse motion is +/- 1 mm, +/- 3 mm was used in the design to incorporate some additional margin of safety. The design currently does not allow for large tilts of the secondary. That is, if one axial actuator were extended fully and the other two retracted fully with the transverse motion at its 3 mm limit, then the axial actuator stress would exceed its current maximum working stress of 55 ksi (against approximately 120 ksi strength) by a considerable amount. It is therefore assumed that the maximum tilt of the secondary can be limited by the control system to 5 arcmin or so. If this is not the case, the actuator end flexures would have to be lengthened to accommodate greater angular motion.

A structural failure of the actuators is unusually critical since it would mean loss of the load with a high probability of total destruction of both the secondary and primary mirrors.

This being the case, it is intended that backup safety cables will be provided between the secondary support structure and mirror cell. They will have some slack in them at all times with their size and mounting designed to minimize forces on the cell.

As can be seen from drawing D271042, the roller nut (and subsequently, ball bearing) creates a load path to the actuator body which is redundant to the disk flexure load path for side loading on the screw. For low friction and long life it is desirable to minimize radial and moment loads on roller screw assemblies. Therefore, a relatively detailed finite element analysis was performed to determine the nut/bearing reactions.

Prior to this, however, an analysis was performed to see what the hypothetical relative motion (between the nut and screw) would be if the nut were not there at all. The "motion" was .0006 inches (15 microns) radially and .00647° (23 arcsec) rotation. The corresponding reactions (conservatively assuming the nut/screw interface were infinitely stiff for radial and moment load) were 23 lbs. radial load and 40 in-lbs. moment. When asked what friction torque we could expect in this application (with the given axial, radial and moment loads), the manufacturer's response was 10-20 in-lbs. and that we should buy a unit and test it ourselves. (The value calculated from equations in the old Transrol catalogue due to just the axial load was 13.4 in-lbs. All of these equations have been deleted in the new catalogue.)

In addition to the above radial load, a maximum of 43 lbs. radial load is expected due to bearing and roller screw/nut radial runouts. The bearing axial runout can be as high as 350 microinches, and therefore must be cancelled out by the control system by look-up table or by enclosing the runout by an encoder such as the LVDT.

The total axial spring rate for one actuator assembly is currently estimated at 200,000 lbs./in. The compliance represented by this (1/spring rate, or inches/lb) is due 1/3 to the body, 1/3 to the bearing, and 1/3 to other sources. Of the "other sources", approximately 1/2 is the nut/screw axial compliance and 1/4 due to each of the two actuator end flexures.

The total calculated friction torque under the worst case of maximum spring loads and maximum external load is 23 in-lbs., referenced to the screw axis. Of this, 1.6 in-lbs. is due to nut internal preload (200 lbs.), 13.4 in-lbs. is due to maximum external and spring loads, 5 in-lbs. is due to bearing torque under maximum thrust load, and 2.1 in-lbs. is due to friction drive preload. This is sufficient friction that the complete assembly should be self-locking (35% direct efficiency).

It is anticipated that precision shimming will be done (including grinding of shims at assembly) in order to assure that very little of the actuator strokes are used up by manufacturing tolerances. A procedure can be developed to accomplish the following:

With the telescope (wide-field secondary installed) pointed to 45 degrees off zenith and thermally stabilized on a mean temperature night, the vanes are accurately aligned (rear vanes hidden behind front vanes), the actuators are at mid-stroke, and the secondary is perfectly collimated with the primary, with precision ground shims in place at one end of each actuator. The vane alignment is accomplished by precision turnbuckle adjustment at the vane ends, while the collimation (with actuators at mid-stroke) is subsequently accomplished by precision grinding of the shims.

A DC servomotor has been tentatively selected, Inertial Motors No. A30L. Since the friction torque is not accurately defined by the roller screw manufacturer, it has been sized conservatively, and is a little larger in diameter than that shown on drawing D271042.

The calculated heat dissipation rate for this motor is 6.1 watts for each axial actuator at zenith while doing an actual 2.2 watts (maximum) of useful work (raising the secondary assembly during slewing). A slew rate of full-stroke in 1 minute was assumed, or 10 mm/min. The heat dissipation rate for the two diagonal transverse actuators is 8.6 watts when at horizon if raising or holding the assembly. At any reasonable tracking, thermal compensation, or slew rates, the power dissipation is essentially constant and varies downward from the above two values.

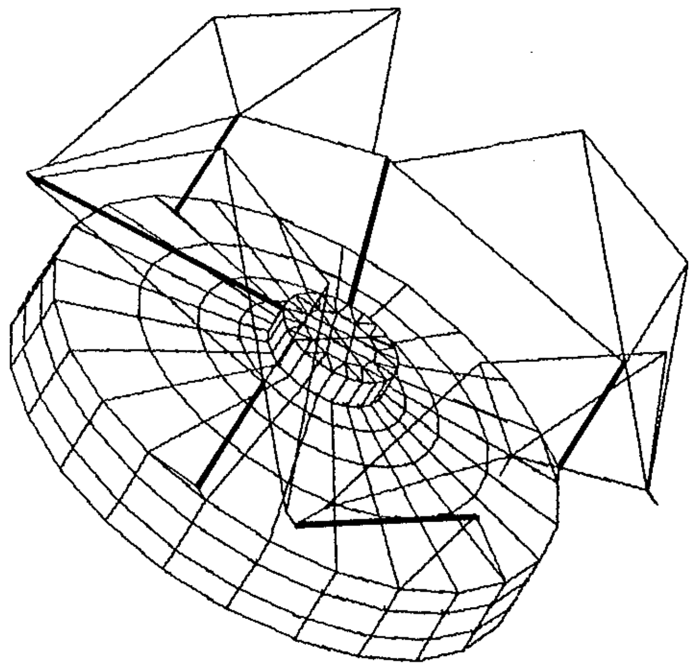
The maximum estimated heat dissipation rate to the air from the six actuators, based upon the above, would be about 25 watts. Actually, the ill-defined friction torque could cause this to increase to about 40 watts, but it is believed that a motor can be made with a more favorable winding (and therefore torque constant or oz.-in./amps), reducing the zero-speed power. Therefore, the 25 watts should be a reasonable maximum to anticipate.

It should be noted that if the control were designed to take advantage of the self-locking screw feature, and thus turned off most of the time (it is believed that thermal and tracking rates would allow this), this value could decrease dramatically. However, this would not be possible if fine guiding with a 2-3 Hz bandwidth.

However, the 25 watts will be insignificant as compared to the heat dissipated by the telescope structural mass during ambient temperature changes. For example, it is estimated that, even with the main truss tubes ventilated, the heat dissipation rate by the secondary end structure alone is 200 watts during a maximum ambient thermal rate of 0.5 °C per hour.

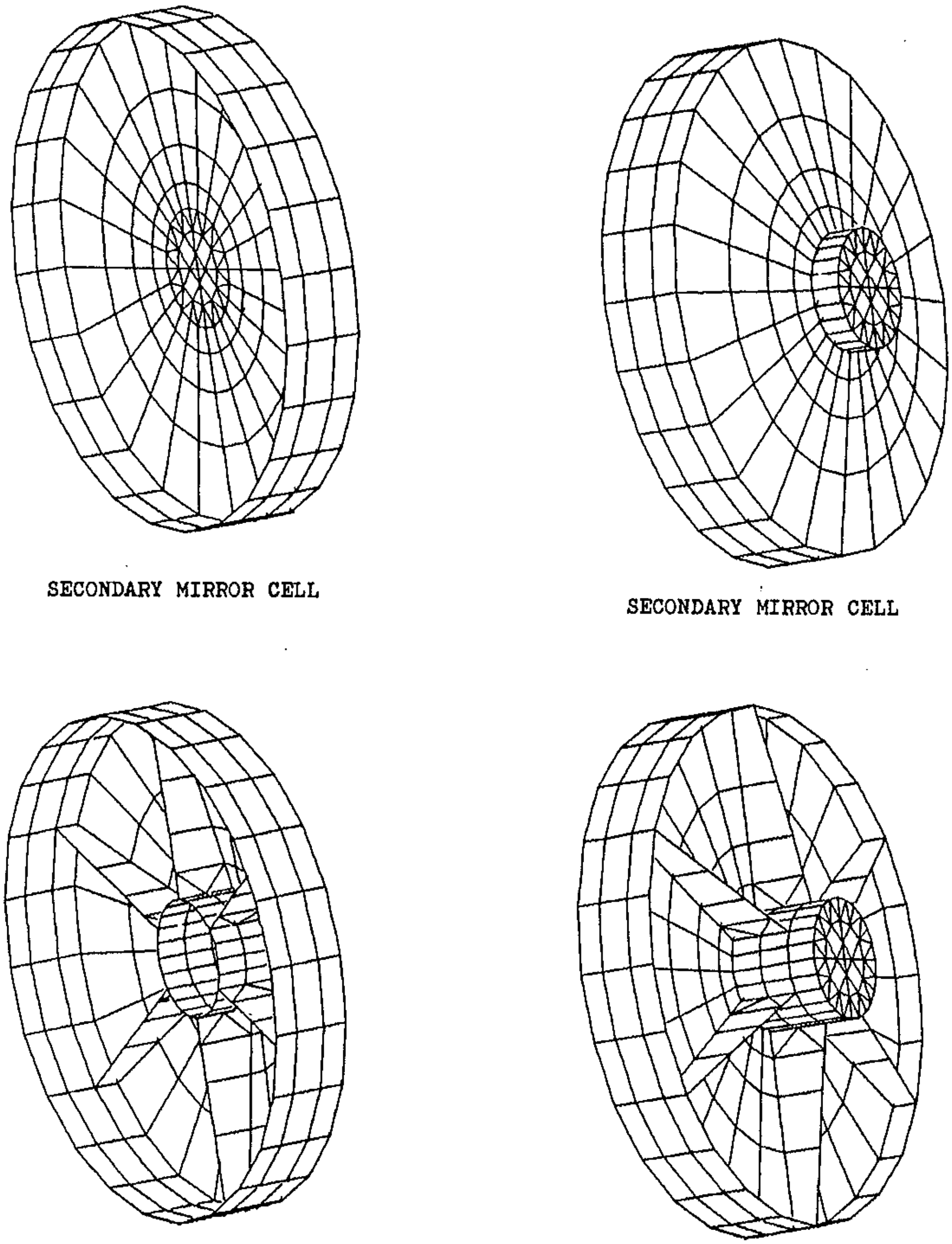
In summary, it is believed that the above-described secondary support and actuator system should have good performance and is appropriate for its intended use. Its stiffness is good. For example, the first resonant frequency at the secondary end with an infinitely rigid system (mirror, support, cell, actuators, and support structure) was 7.9 Hz, and 7.2 Hz with all of the "real" components described above. Other modes degraded by a smaller percentage yet. Heat dissipation rates are low as compared to other effects in the telescope system, and positioning accuracy and control should be excellent due to high stiffness, low friction, and especially low hysteresis.

However, the actuator position feedback/encoder devices are not yet fully defined. Considering this, inaccurate friction torque data, and lack of specific experience with this type of disk flexure system, the development and testing of a prototype actuator may be warranted.



F6.5 WIDE FIELD SECONDARY SUPPORT SYSTEM

FIG. 1



SECONDARY MIRROR CELL

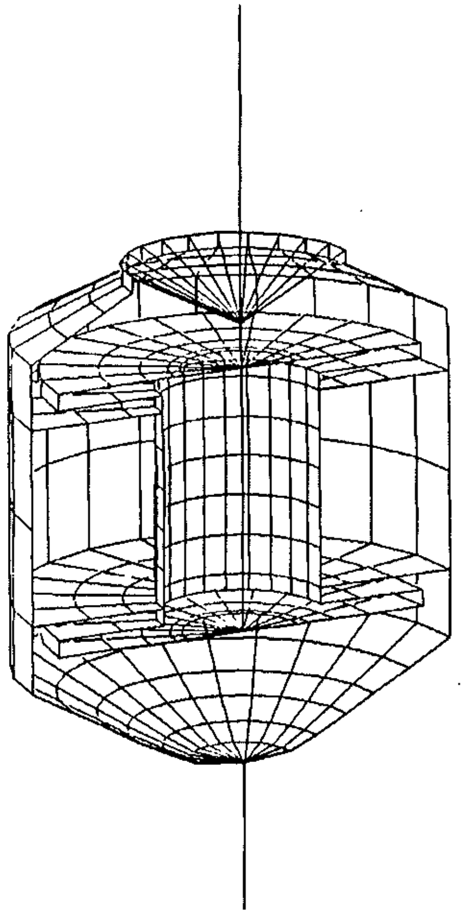
SECONDARY MIRROR CELL

FRONT PLATE REMOVED

BACK PLATE REMOVED

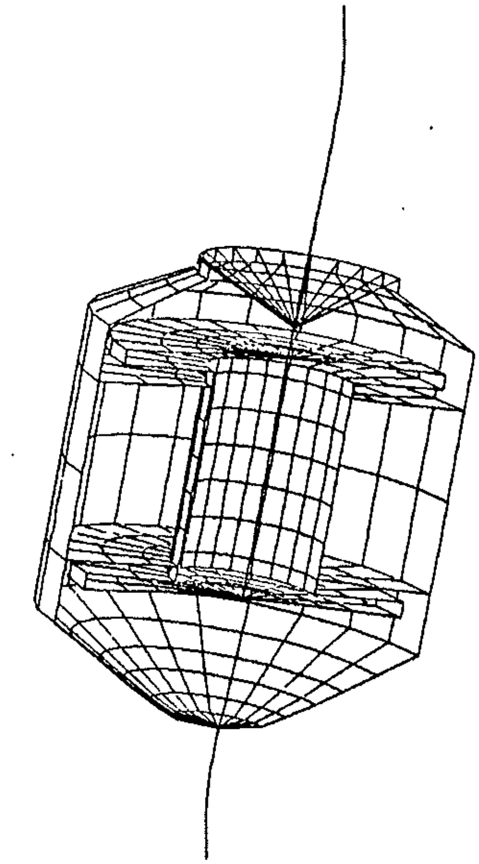
WIDE FIELD SECONDARY MIRROR CELL

FIG. 2



FINITE ELEMENT MODEL OF ACTUATOR

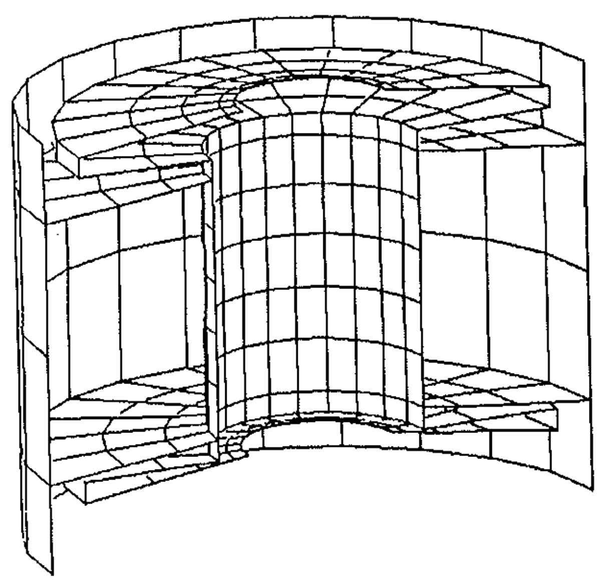
ALIGNED



FINITE ELEMENT MODEL OF ACTUATOR

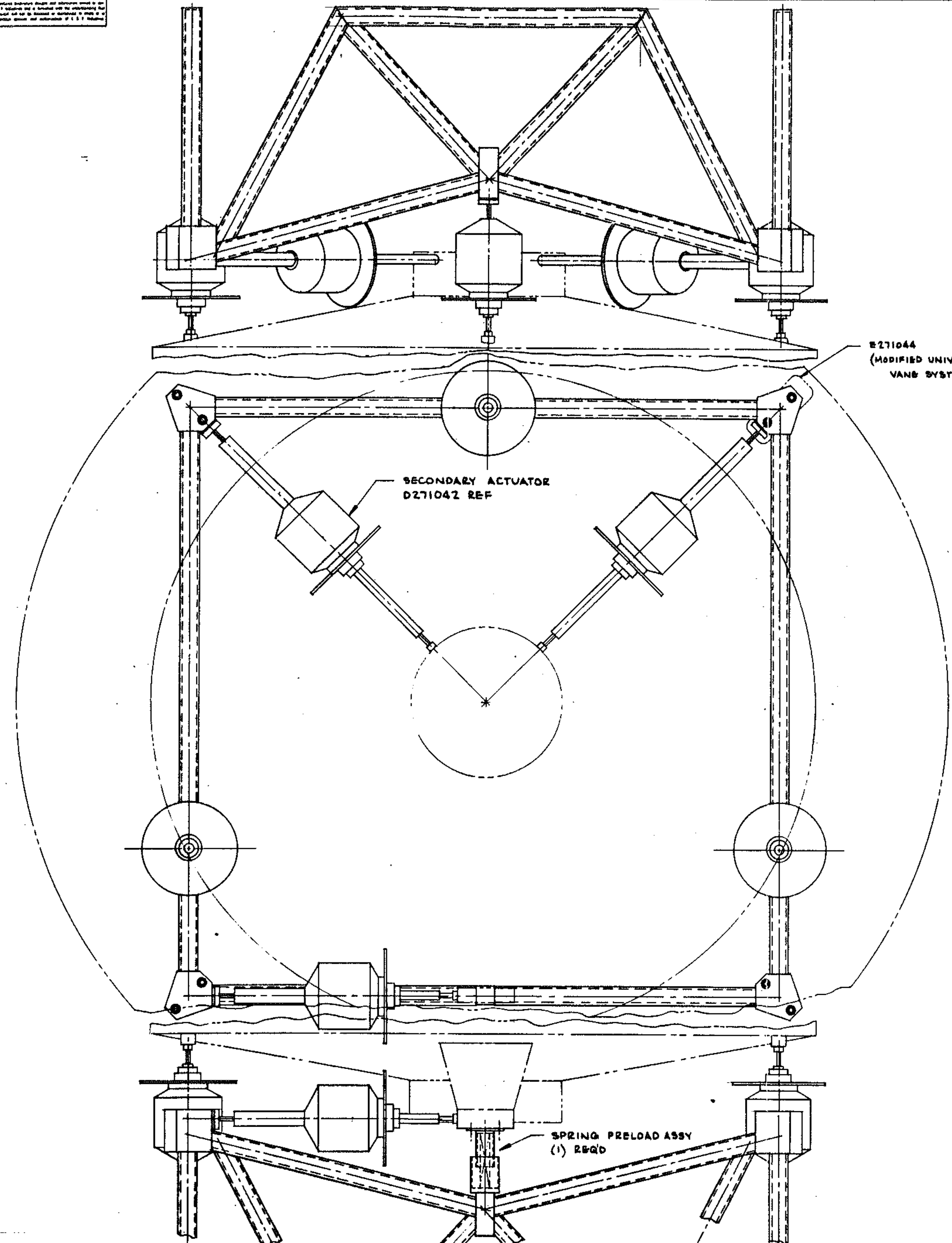
EXAGGERATED LATERAL MOTION

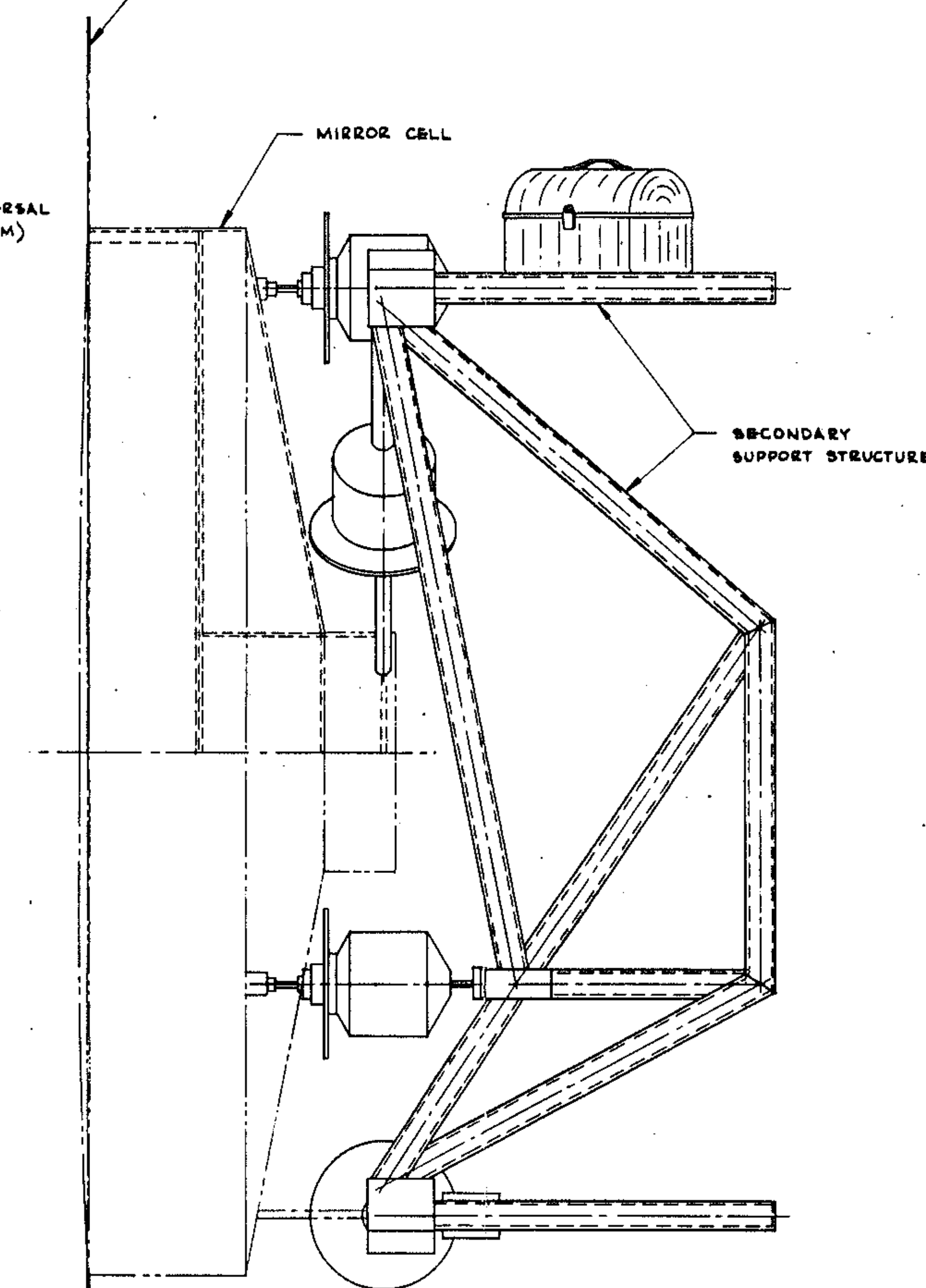
FIG. 3



DISK FLEXURES UNDER AXIAL STROKE

FIG. 4





GENERAL NOTES:

1) SEE "SECONDARY SUPPORT SYSTEM NOTES" AND "SECONDARY ACTUATOR NOTES".

2) WEIGHTS:

SECONDARY SUPPORT STRUCTURE	-	402
ACTUATORS	-	182
MIRROR CELL	-	712
BAFFLE	-	65
MIRROR	-	540
MISC	-	50
		1,950#

L&F INDUSTRIES

1,950#

MAGELLAN 8M TELESCOPE

SECONDARY SUPPORT SYSTEM

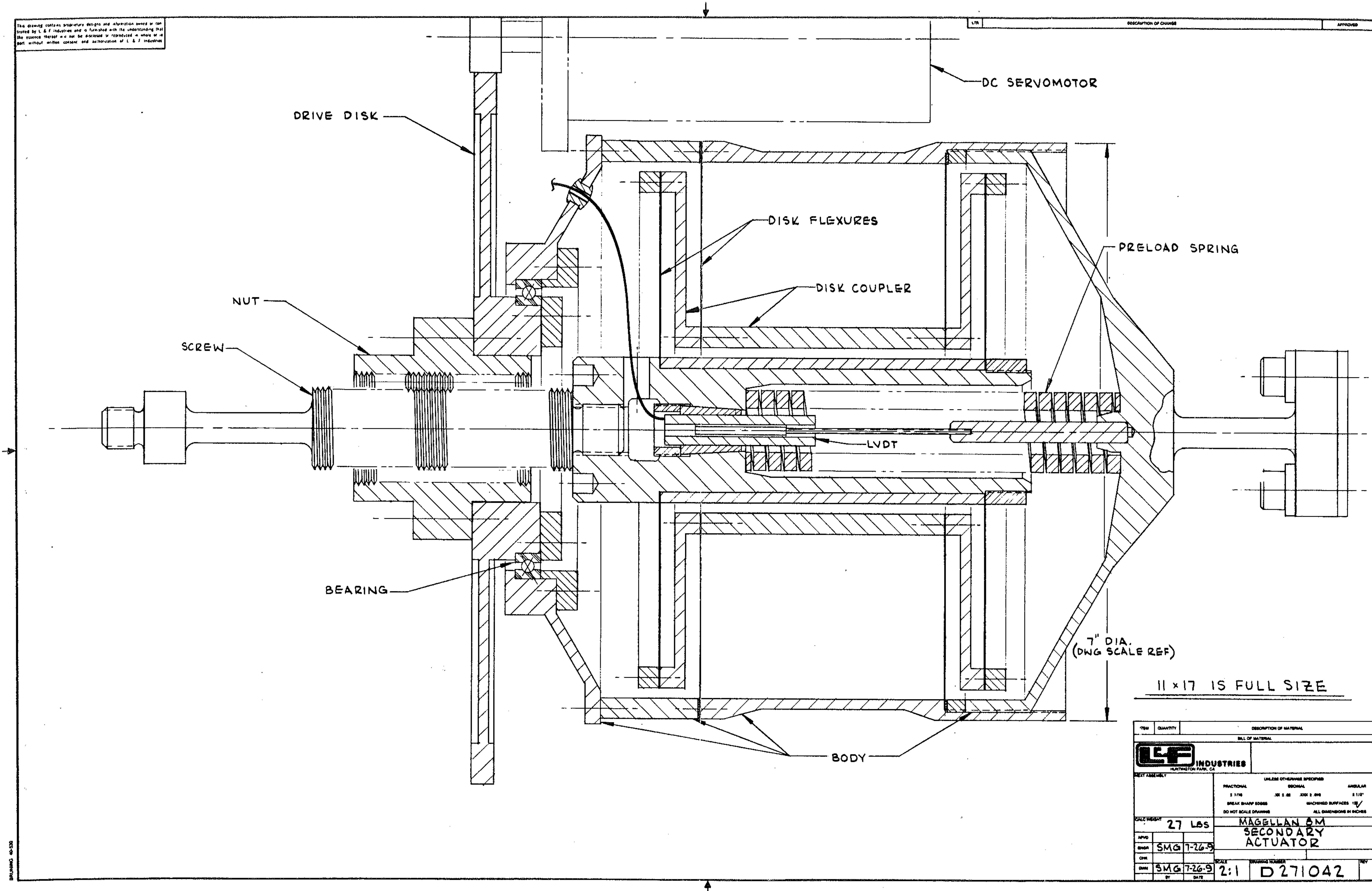
SMG 8-16-9?

SMG 8-16-9?

1/4

E 271043

E 271043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LTR
DESCRIPTION OF CHANGE
APPROVED
DC SERVOMOTOR
DRIVE DISK
DISK FLEXURES
PRELOAD SPRING
DISK COUPLER
NUT
SCREW
LVDT
BEARING
7" DIA.
(DWG SCALE REF)
11 × 17 IS FULL SIZE
BODY
ITEM
QUANTITY
DESCRIPTION OF MATERIAL
BILL OF MATERIAL
INDUSTRIES
HUNTINGTON PARK, CA
NEXT ASSEMBLY
UNLESS OTHERWISE SPECIFIED
FRACTIONAL
DECIMAL
ANGULAR
BREAK SHARP EDGES
MACHINED SURFACES
DO NOT SCALE DRAWING
ALL DIMENSIONS IN INCHES
CALC WEIGHT
27 LBS
MAGELLAN
SECONDARY
ACTUATOR
APVD
ENGR
SMG
7-26-9
CHK
DWN
SMG
7-26-9
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2:1
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D 271042
REV
BRUNING 40-530