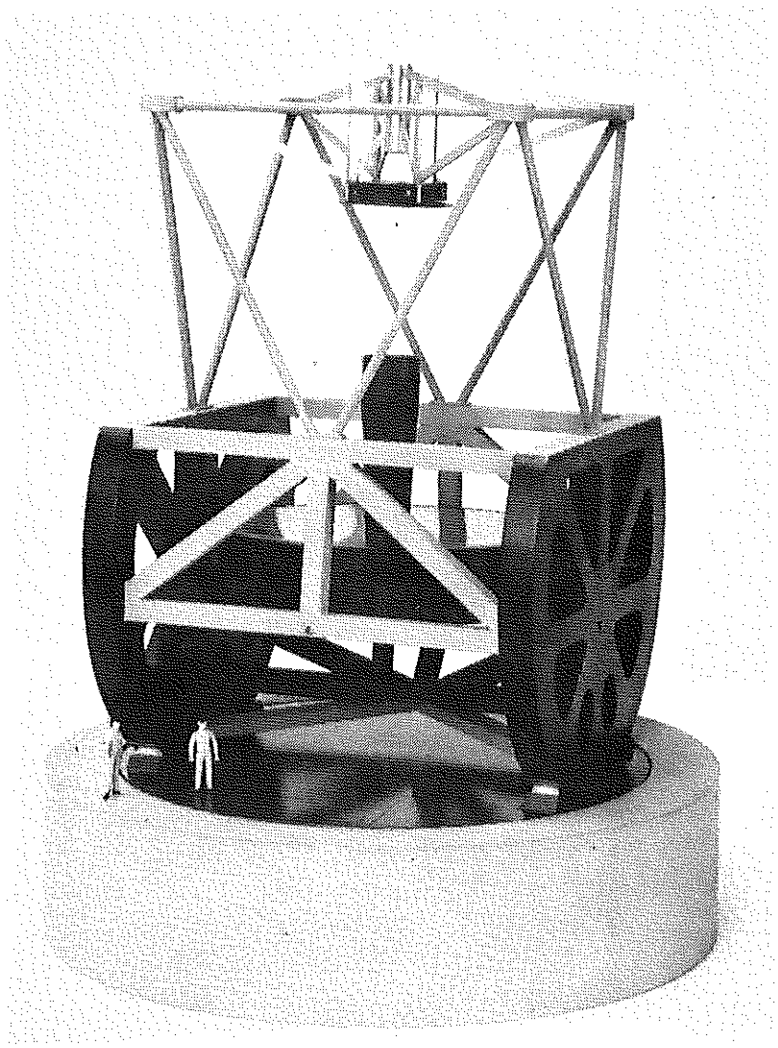

# Direct Friction Drives

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No. 18

CONTENTS

# 1. INTRODUCTION

The Magellan Project direct friction drive is defined here as a system in which a friction drive roller is driven by a servomotor which is directly connected to the shaft of the roller. This system is shown on drawing E271052 and the finite element graphics plots in Figures 1 through 4. The only motion reduction then results from the ratio of the diameter of the driven component (the azimuth or altitude drive disk) to the diameter of the friction drive roller. The system then provides for one stage of drive reduction to the telescope axis of about 45:1.

# 2. DESCRIPTION

As shown on E271052, one drive assembly consists of a drive housing, drive roller assembly, and idler roller assembly. The unit is radially preloaded against the drive disk via a spring plunger (or alternatively, hydraulic cylinder) connected to the preload lug shown. In addition, an axially stiff member will connect to the tangent arm lug to react the drive tangential load that will be developed by the drive motor. The radial preload is intentionally applied between the drive and idler roller shafts so that the idler shaft effectively stabilizes the unit by registering against the drive disk.

As shown in Sections A-A and B-B (E271052) the design will accommodate larger drive and idler rollers so that the drive disk need not have a raised contact area which projects into the drive box. For example, should it be determined to be feasible to drive directly against the machined outer diameter of the altitude disks (in an area immediately adjacent to the area of the hydrostatic bearing) then the 7" diameter drive roller can be changed to a 10" diameter roller, with no other changes necessary. As summarized below, the torque capacity of the motor and amplifier would be more than adequate (with the larger drive roller), due to lower inertia about the altitude axis, in combination with a larger drive disk radius.

Preloaded duplex pairs of angular contact ball bearings have been used to support the roller shafts. One pair on each shaft is retained in thrust to the drive housing, as shown.

The frameless motor connects to the system in a direct manner. As shown in Sec. A-A, the stator connects to the drive housing via the stator adapter, while the rotor adapter connects the motor rotor to the drive roller shaft.

The ventilation cover should provide an inexpensive and effective means of removing heat dissipated by the motor. The cover has numerous small holes to atmosphere located around its periphery near the left side of the motor (Sec. A-A). In addition, it is internally baffled by the cylindrical shell which is slightly smaller than the rotor adapter. With the 3" diameter tube fitting connected to a low pressure area of the structure (via a flexible duct), air is then drawn in the small holes, flows around and flushes the exterior surfaces of the motor (and stator and rotor adapters) passes through the flexible duct and is ultimately exhausted remote from the enclosure.

Two drives are used for each telescope rotation axis. One unit drives against each altitude disk (ref. E271029), while two units at 180° to each other drive against the azimuth drive disk (ref. E271045, Pintle Azimuth System). Torque requirements were calculated for each axis for each of four component categories:

1. Acceleration torque to achieve 0.3°/sec$^2$ during slewing.

2. Wind torque under 15 mph (6.6 m/sec) wind *acting on structure.* From recent testing at Apache Point Observatory (a similarly ventilated enclosure) this should be equivalent to about a 45 mph ambient wind remote from the enclosure.

3. Drive friction torque, including contact and bearing friction.

4. Hydrostatic friction torque.

A summary of these calculated torque requirements follows:

| **Category** | **Altitude torque on axis** | | **Azimuth torque on axis** | |
|---|---|---|---|---|
| | ft-lbs. | % | ft-lbs. | % |
| Acceleration | 11,600 | 73 | 17,500 | 75 |
| Wind | 3,000 | 19 | 3,030 | 13 |
| Drive friction | 1,070 | 7 | 2,540 | 11 |
| Hydrostatic friction | 120 | 1 | 180 | 1 |
| TOTAL | 15,800 | 100 | 23,300 | 100 |
| Torque required, each motor: | 180 ft-lbs. | | 256 ft-lbs. | |

The peak torque rating for the motors is 550 ft-lbs. each, although the amplifier selected will limit the maximum actual torque available to each motor to 330 ft-lbs. It would seem that the 330 ft-lbs. amplifier rating would provide a reasonable margin to the 256 ft-lbs. azimuth motor torque requirement (330/256 = 129%). Therefore, it is suggested that, with the reduction ratios planned (438/10 = 43.8:1 altitude, 318/7 = 45.4:1 azimuth), the torque be limited on the altitude drive motors to 1.29 x 180 = 232 ft-lbs. This can be accomplished by limiting the current to the motor.

It should be noted that the precision of the torque calculation for each of the above categories varied considerably, but in a favorable way. It is expected that the acceleration torque calculation is quite accurate, while the wind and friction torques are harder to predict and therefore less accurate. However, they constitute such a small fraction of the total torque that they are unlikely to cause the amplifier capacity to be exceeded.

While a formal trade study of other drive methods was not undertaken, it is felt that the friction drive is superior to two other methods currently being considered on other large telescope projects. For example, as compared to a gear drive, the friction drive should offer much lower cost (due to the elimination of large high-precision gears) while outperforming the gear drive in terms of both "stiffness" (no tooth bending) and precision (reduced noise). The only area in which a gear drive may beat the friction drive is in lower friction (in the case of a helical or spur gear drive), but this should certainly not be weighted heavily enough to overcome the other advantages of the friction drive.

When compared to a direct drive (one in which the motor is connected directly to the axis of the telescope), the friction drive has the benefits of lower cost, smaller torque perturbations (cogging and ripple) and reduced (and much more easily scavenged) heat dissipation.

It is argued that the direct drive is "stiffer." However, in a reduced drive the important criterion is not stiffness alone, but *drive stiffness as compared to drive inertia* and this is more directly measured by the drive first resonant frequency, or its near-reciprocal, the drive time constant. If this is adequate to allow operating at the control bandwidth defined by the telescope resonances (about 1/4 of the locked rotor frequencies), then any additional "stiffness" is of no value. It is believed that a drive first resonant frequency of about 10 times the telescope control bandwidth (currently estimated at about 3 Hz), or 30 Hz, would satisfy this requirement. The first resonant frequency for the direct friction drive mechanical assembly is estimated by finite element analysis to be about 325 Hz. However, this may degrade somewhat when combined with the motor electrical time constant. A test is therefore planned to empirically determine these and other drive characteristics. The test goals, procedures, and apparatus are shown in the appendix of this report.

## 3. FINITE ELEMENT ANALYSIS

The structural performance of a drive assembly has been predicted using finite element analysis. Graphics plots of the model are shown in Figures 1 through 4. The model consisted of 396 nodes, 300 plate elements, 134 beam elements, and 2355 degrees of freedom. All significant effects involving mass and/or compliance were considered, including the housing, rotor (inertia), rotor adapter, drive roller, and bearings. While an even more detailed model (using brick elements) might have been used, it is felt that the model detail was appropriate to the level of accuracy required.

It is required that the actual output torque (or tangential force) for the system be measured during testing. As shown on drawing E271053 (Test Arrangement), this requires the use of the (static or dynamic) test bar and load cell as shown. These masses and compliances were included in the finite element models, as shown in Figures 1 and 4. They cause the first resonant frequency to lower considerably (below the frequency that would exist if the drive were simply mounted to the real telescope), from about 325 Hz to 125 Hz. However, the 125 Hz is still much higher than necessary for good control performance, if the motor electrical time constant is negligible.

A summary of the analyses run (not including optimizing runs) and the results follows:

| Model | Frequency | Description |
|---|---|---|
| DFD3 | 110 | System with real static test bar and real load cell as shown on E271053; housing is rigidly constrained to ground. |
| DFD8 | 125 | Same as DFD3 but mass of housing is free to vibrate against mass of bar (not rigidly constrained to ground). This represents the best estimate of the frequency we should be able to measure in this configuration (with the real load cell and unit unconstrained). |
| DFD5 | 135 | Same as DFD3 but with "rigid load cell." |
| DFD9 | 155 | Same as DFD5 but not rigidly constrained to ground. This represents the highest frequency that we should be able to measure for the unit. |
| DFD2 | 315 | System with rigid, weightless bar and load cell, housing rigidly constrained to ground. |
| DFD7 | 325 | Same as DFD2 but not rigidly constrained to ground. This represents the best estimate of the first structural frequency for the drive box mounted in the real telescope. |

It should be noted that these frequencies are very high and may be difficult to achieve considering some practical considerations which were not included in the finite element modelling. For example, it was assumed that all mechanical joints (such as rotor to rotor adapter and rotor adapter to drive roller shaft) were rigid. While these effects may prove to be significant, it is hard to imagine that they could cause the first resonant frequency to be below the (approximate) 30 Hz requirement for good telescope control.

## 4. CONCLUSIONS

The mechanical response of the direct friction drive assembly has been predicted by analysis to far exceed that required for good telescope control. If the effective time constant for the motor can be sufficiently reduced by the control system (below the catalogue value) this combination should provide excellent control at minimal cost and with negligible effects on seeing from heat generation. The testing planned should empirically determine these and other characteristics.

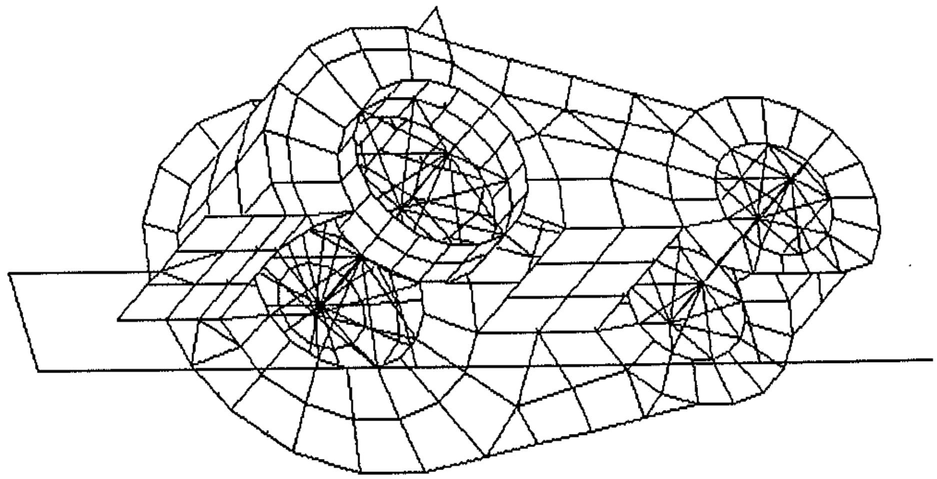


Fig. 1 - DIRECT FRICTION DRIVE FINITE ELEMENT MODEL

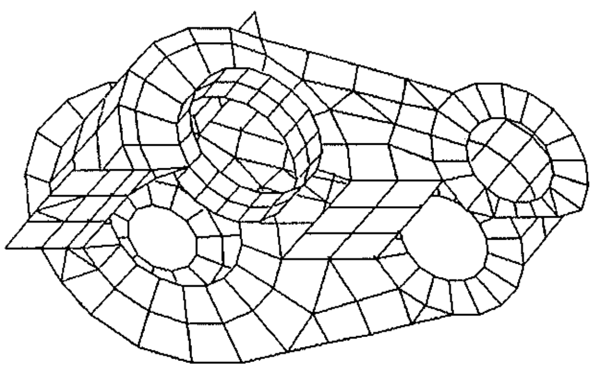


Fig. 2 - PLATE ELEMENTS, FRONT QUARTERING VIEW

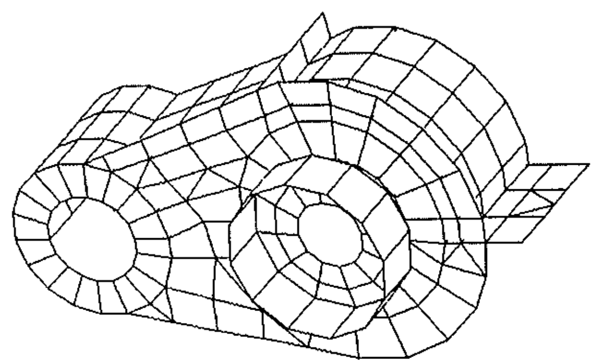


Fig. 3 - PLATE ELEMENTS, REAR QUARTERING VIEW

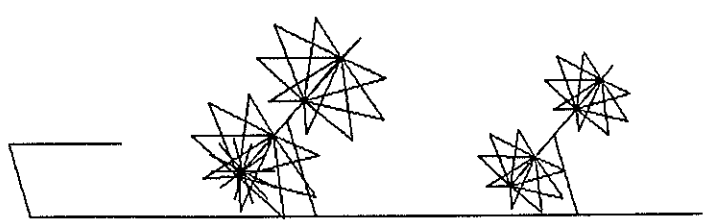
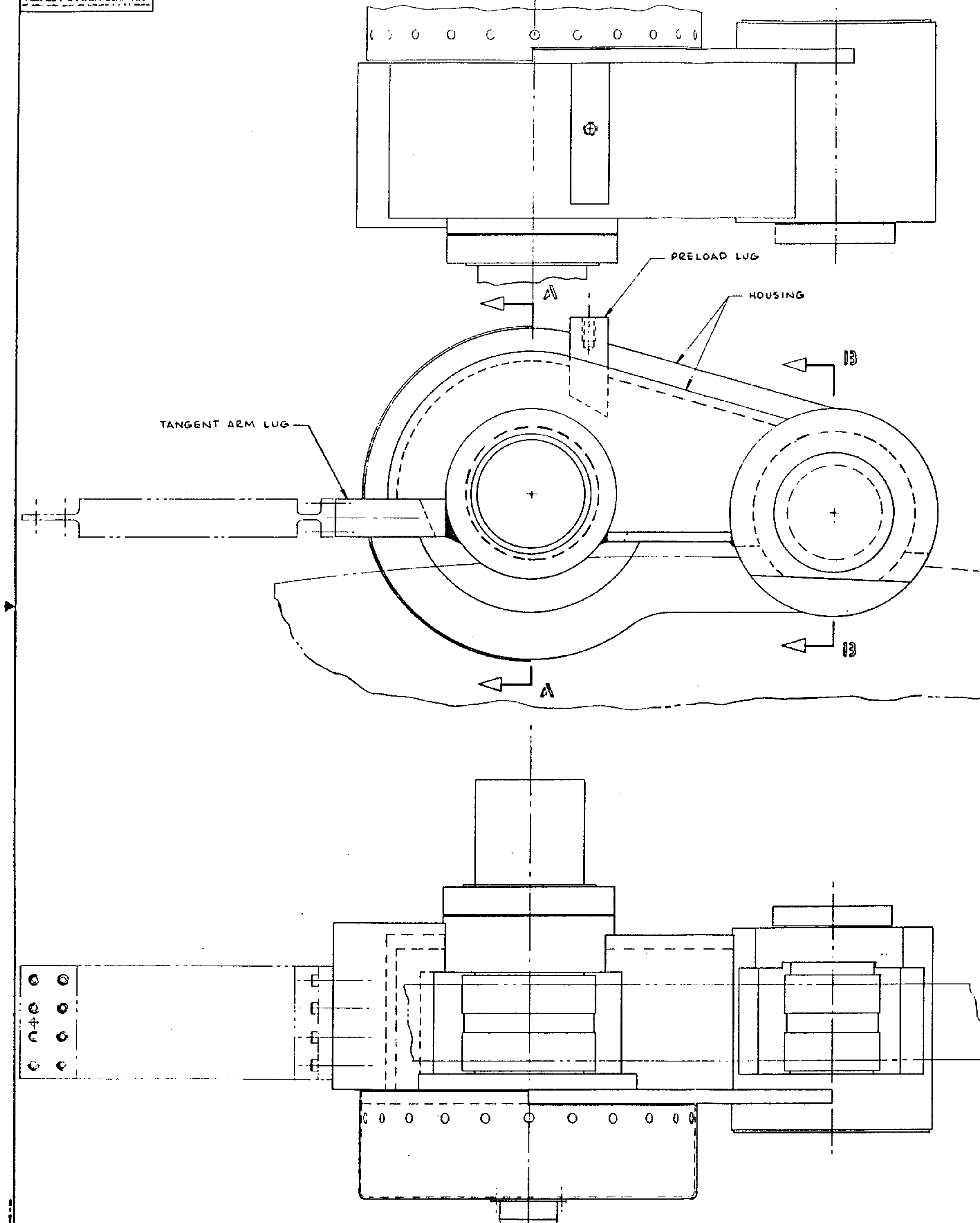
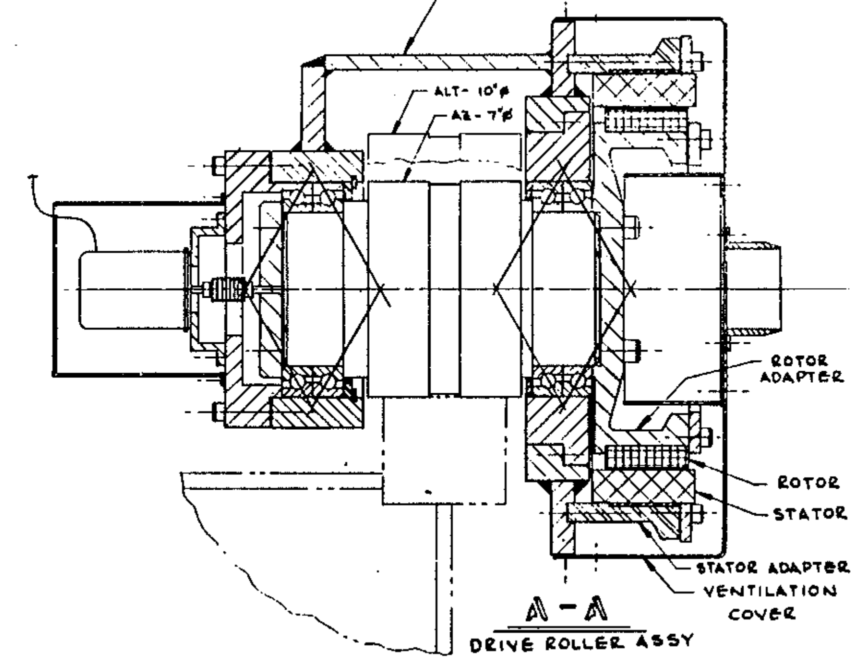


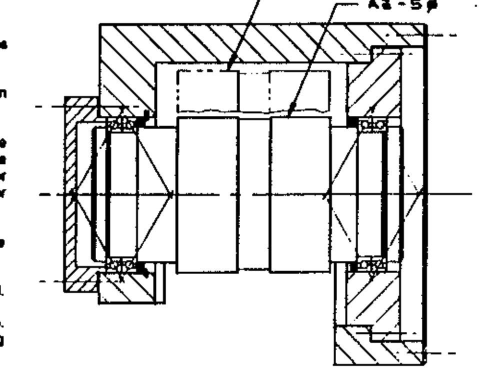
Fig. 4 - BEAM ELEMENTS, FRONT QUARTERING VIEW

| | |
|---|---|
| A | GENERAL REVISION SMG 3-22-90 |




A - A
DRIVE ROLLER ASSY




B - B
IDLER ROLLER ASSY

**MAGELLAN PROJECT 8 METER TELESCOPE**

**NOTES - LAYOUT NO. E271052**

1. Drive box housing fabricated from ASTM A36, welded, stress relieved, then machined.

2. Azimuth drive roller O.D. is slightly smaller (6.960 dia.) than bearing O.D. for installation and removal.

3. Altitude units are made with screwed-and-dowelled slug on one side for installation and removal of larger (10") rollers. It may be better to make even the azimuth units with the removable slug for commonality. These comments apply to both drive and idler shafts.

4. Purchased parts at drive shaft: *(Note - some parts may be purchased by customer).*

   1. Encoder is "Parameter 1, Option G", data sheet attached.

   2. Coupling is Servometer SHFC-12/SHSC-12 (either part no. is acceptable), data sheet attached. With CL6SS clamping collar each end.

   3. Bearings are Kaydon KG060B (seperator, precision, fit TBD). Two duplex pairs required. Preloaded so that no clearance exists under 15,000 lb. radial load on roller (total both pairs). Maximum load expected is 11,000 lbs.

   4. Retaining ring is Waldes Truarc No. N5000-700 or equal.

   5. Motor is per Inland Dwg. No. 96887, revised 3-15-90.

5. Purchased parts at idler shaft:

   1. Bearings are Kaydon KD040B (seperator, precision, fit TBD). Two duplex pairs required. Preloaded so that no clearance exists under 3,500 lb. radial load on roller (total both pairs). Actual maximum load expected is 2,500 lbs.

   2. Retaining ring is Waldes Truarc Co. No. N5000-500 or equal.

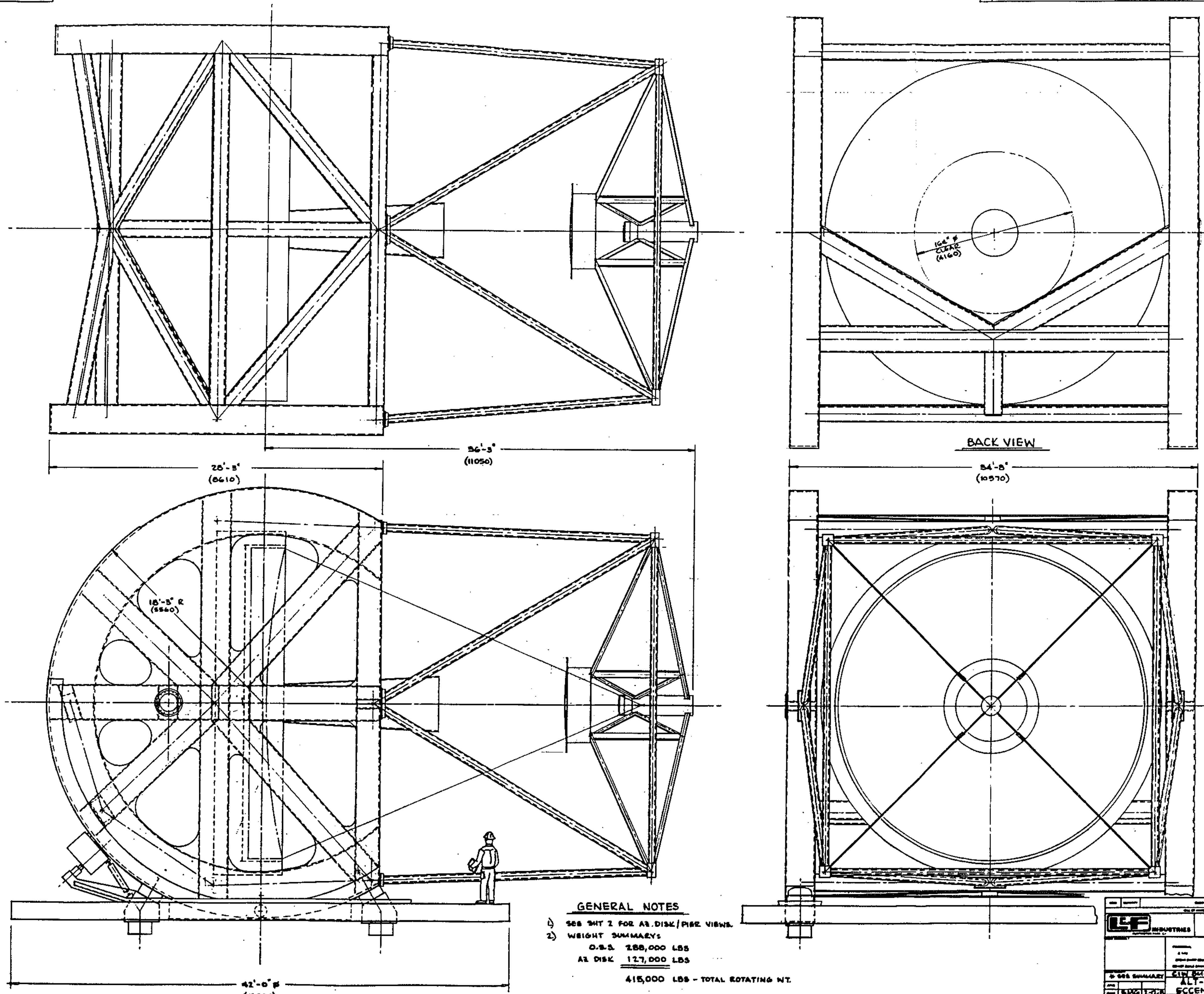
L&F INDUSTRIES

MAGELLAN PROJECT 8M
DIRECT FRICTION DRIVE
-INLAND MOTOR-

| | | | |
|---|---|---|---|
| SMG 3-10-0 | | | |
| SMG 3-10-0 | 1/2 | E271052 | A |

E271052 A

| | | |
|---|---|---|
| A | REVISED VANE SYS. REF E271052 | SMG 7-22-88 |




## GENERAL NOTES

1) SEE SHT 2 FOR AZ. DISK/PIER VIEWS.
2) WEIGHT SUMMARY:

O.S.S. 288,000 LBS
AZ DISK 127,000 LBS

415,000 LBS - TOTAL ROTATING WT.

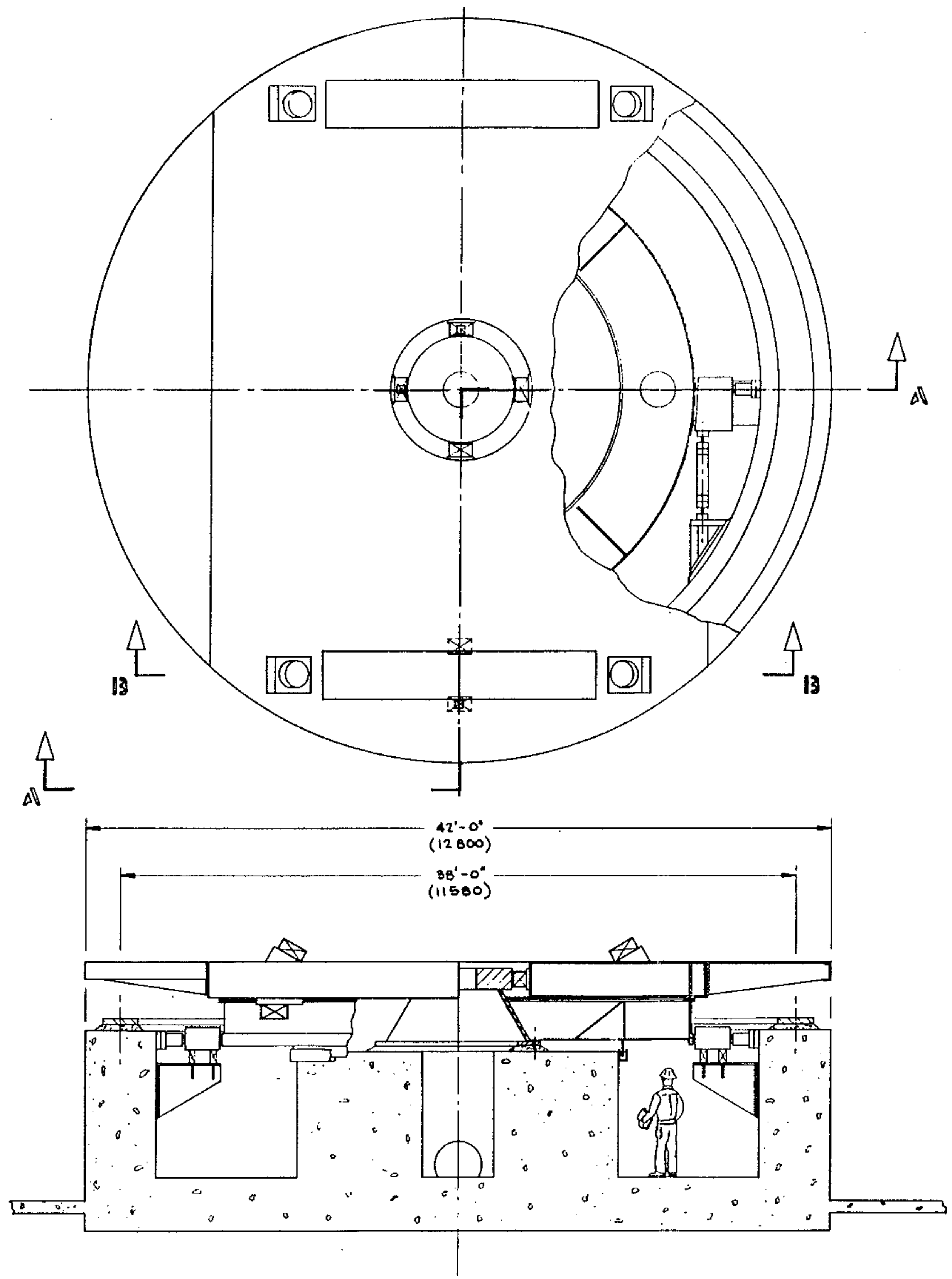
L&F INDUSTRIES

CIW 8M - MAGELLAN PROJECT
ALT-AZ DISK - ECCENTRIC NASMYTH.

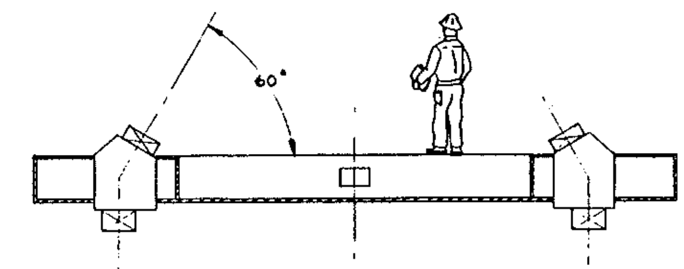
WGT SUMMARY

SMG 7-7-88

SMG




A - A




B - B

☒ INDICATES RIGIDLY MOUNTED HYDROSTATIC PAD. (STIFFNESS LOAD PATH)

☒ INDICATES PRELOAD HYDROSTATIC PAD.

L&F INDUSTRIES

MAGELLAN 8M TELESCOPE

PINTLE AZIMUTH SYSTEM

SMG 9-4-89

SMG 9-4-89

1/8"=1'-0"

E 271045

E 271045

# APPENDIX

# MAGELLAN PROJECT 8 METER TELESCOPE
# DIRECT FRICTION DRIVE TEST

Steve Gunnels
Alan Schier

## GOALS:

1. Demonstrate that the first resonant frequency of the mechanical assembly (that is, with drive electronics off) will not limit telescope performance. This would require that this frequency is about 10 times 3 Hz (the anticipated upper limit control bandwidth limited by telescope structural resonances), or 30 Hz. Finite element analysis predicts this first resonance in the test apparatus to be about 110/125[1] Hz. FEA also predicts this first resonance in the telescope configuration (that is, with the output drive roller locked at the telescope drive disk) to be about 315/325[1] Hz.

2. Demonstrate that the complete drive (that is, with drive electronics on) will not limit telescope performance. That is, with the motor electrically active, its effective time constant will cause the above 110/125 Hz. to degrade to a lower value. A negligible effective time constant should cause a negligible reduction and thus allow a control bandwidth of 30 Hz. (or higher), confirming that the complete drive, including mechanical and electrical effects, will not limit telescope performance.

3. Measure the frequency response of the entire system (drive electronics included) with the commanded torque as the input and measured torque as the output. This should be done under both small signal and large signal conditions. The small signal test will establish the frequency response of the system without the effects of amplifier saturation. This is important in designing the closed-loop system. The large signal test will identify the so-called "full-power bandwidth." This has a lot to do with how well we will be able to reject any large disturbances (such as high wind).

4. Measure torque perturbations (cogging and ripple) in the drive motor, and determine the level to which these can be corrected in an open-loop fashion.

5. Measure traction coefficient of friction (at impending slippage) dry and wet, allowing accurate determination of preload force required.

6. Determine life (to surface endurance failure) of hardened drive roller running against A36 "drive disk."

7. Measure total drive friction (starting and running) torque. Although this will include the friction from the preload track rollers, it should give a conservative estimate of the friction.

## STATIC TEST:

1. Per view shown on E271053, connect "rigid load cell." Slowly apply full torque to motor. Hold torque for a few seconds, then suddenly unload motor (full load to zero load as a step function). Determine ringing frequency using encoder output (or, if necessary, with an added accelerometer). This will give a conservative measurement for the first

mechanical resonant frequency of the drive, but will necessarily include the mass and compliance of the test apparatus bar. With the load cell temporarily replaced with the "rigid load cell", this will allow the highest possible measurement for this frequency. The FEA predicted value for this is about 135/155[1] Hz. The true drive mechanical first resonance (in the real telescope) should still be much higher than this, however, predicted by FEA to be about 315/325 Hz.

2. Replace the "rigid load cell" with the real load cell, and repeat 1. above for comparison with the 110/125 Hz. value predicted by the FEA in this configuration.

3. In this same configuration, apply a sudden torque (zero to full load as a step function). Measure the time required to sense full load at the load cell, and compare this to the mechanical time constant for the drive (which can be back-calculated from the measured frequency from 2. above). This should tell us whether the electrical time constant for the motor controlled by the real amplifier is negligible.

4. In this same configuration apply a full load torque sinusoid, gradually increasing its frequency. Monitor the input torque versus output load (load cell) with time to see at what frequency a significant phase lag develops.

## DYNAMIC TEST:

1. Per view shown on E271053, with drive roller/bar contact area dry, and hydraulic cylinder bottomed out, (full load torque applied), slowly decrease preload (from 22,000 lb. initial force at preload load cell) until slippage just occurs. Record output force and preload force. Repeat with drive contact area wetted with oil or grease.

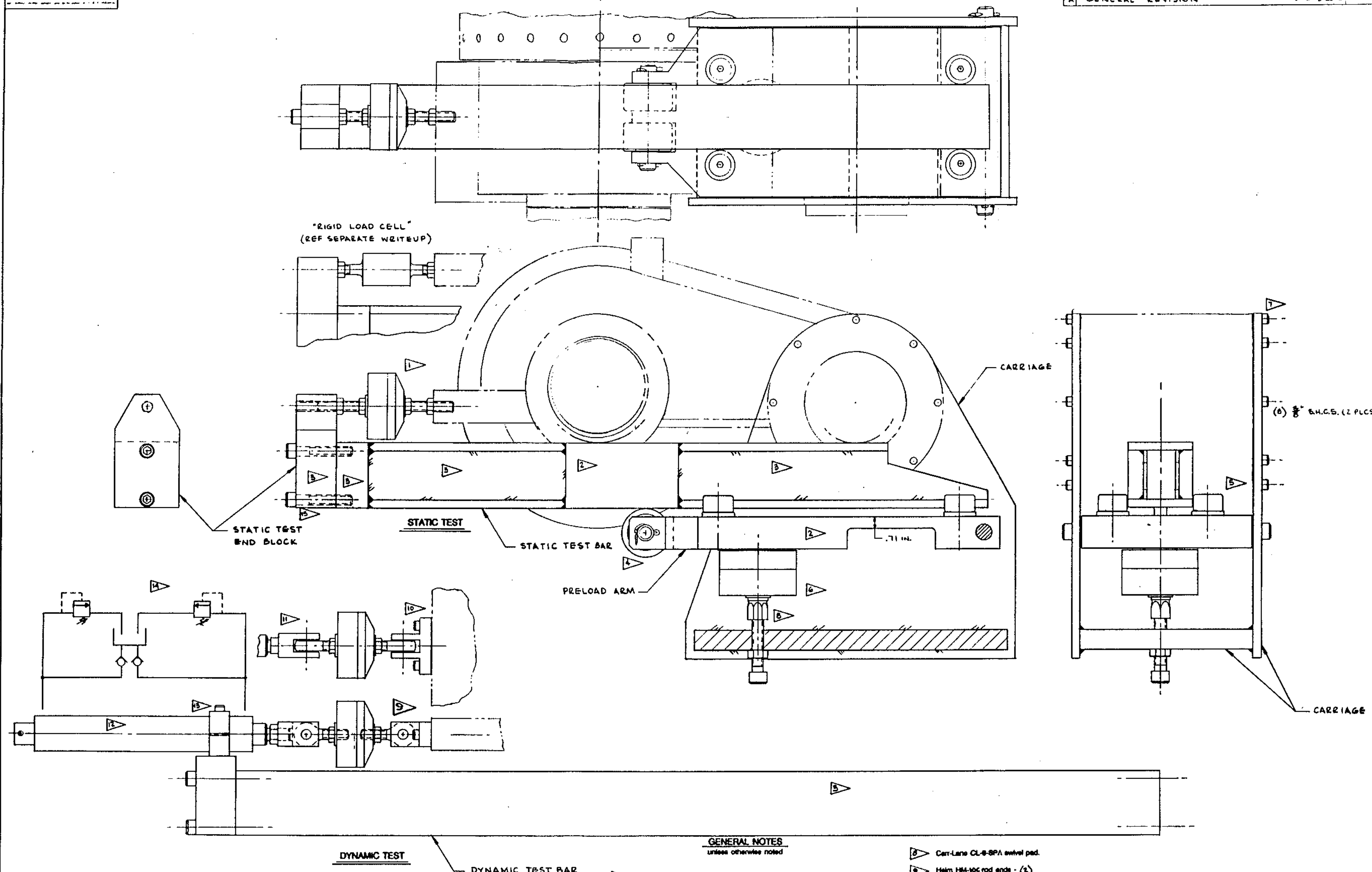
2. Adjust preload to two times the above-measured slippage preload. Perform dynamic testing to determine torque perturbations, etc. by driving against hydraulic cylinder. Force can be adjusted by relief valves shown.

3. After above test, disconnect and remove hydraulic cylinder and load cell. Attach end blocks to mechanically limit total travel of bar. Adjust preload to 22,000 lbs. (measured at pancake load cell), measure motor starting and running current (for later conversion to torque) at 8 in./sec., 2 in./sec., and .01 in./sec. Repeat same for 10,000 lb. preload and 5,000 lb. preload.

4. With preload adjusted to (TBD) lbs., cycle system back and forth over central 8" of travel area at (TBD) in./sec travel speed. Periodically check preload and inspect bar surface for signs of wear or failure. Note condition of roller and bar surfaces as a function of total cycles.

[1] The lower of each frequency set referred to in the body of the text is the value predicted if the drive box is restrained to a rigid, massive support. The higher frequency is the value predicted if the drive box is free to move as the subject vibration occurs.

| | | |
|---|---|---|
| A | GENERAL REVISION | SMG 3-20-0 |




## GENERAL NOTES

unless otherwise noted

1. 1,000 lb. capacity load cell w/threaded rods on ends to provide some flexural action (to limit side loads during static test). These threads are $\frac{5}{8}$-18 UNF-3B INTERFACE 1210 W/6101 BASE.
2. 4140 HT. (OR H.T. AFTER WELDING AS APPLICABLE).
3. ASTM A36.
4. Torrington YCR-48 track rollers, or equal. (2) ea. required.
5. Torrington CRHB-30 track rollers, or equal, with CARR-LANE CL-12-FW WASHER (4) ea. required.
6. Load cell. Interface Corp. Part no. 1221. WITH 6106 BASE.
7. Mild steel carriage weldment, hold inside dimension for mating with machined faces of drive unit. End cap on drive unit must be temporarily removed to install carriage.
8. Carr-Lane CL-6-SPA swivel pad.
9. Heim HM-10C rod ends - (2) required.
10. (Special) clevis pad.
11. (Special) rod-end clevis.
12. Allenair model E-2-10-OS air/hydraulic cylinder.
13. Allenair BM-2 block mounting bracket.
14. Hydraulic system:

    Adjustable relief valve, 0-400 psi, 10 gpm (minimums).
    Check valve, 10 gpm w/10 psi maximum pressure drop.
    Reservoir - tiny.

15. (2) REQ'D - CAN BE USED AS END TRAVEL STOPS DURING DYNAMIC TEST (SURFACE ENDURANCE TEST).

INDUSTRIES

MAGELLAN 8M TELESCOPE
DIRECT FRICTION DRIVE
TEST ARRANGEMENT

| | | |
|---|---|---|
| SMG | 3-13-0 | |
| SMG | 3-13-0 | |

| | | |
|---|---|---|
| 1/2 | E 271053 | A |

E 271053 A