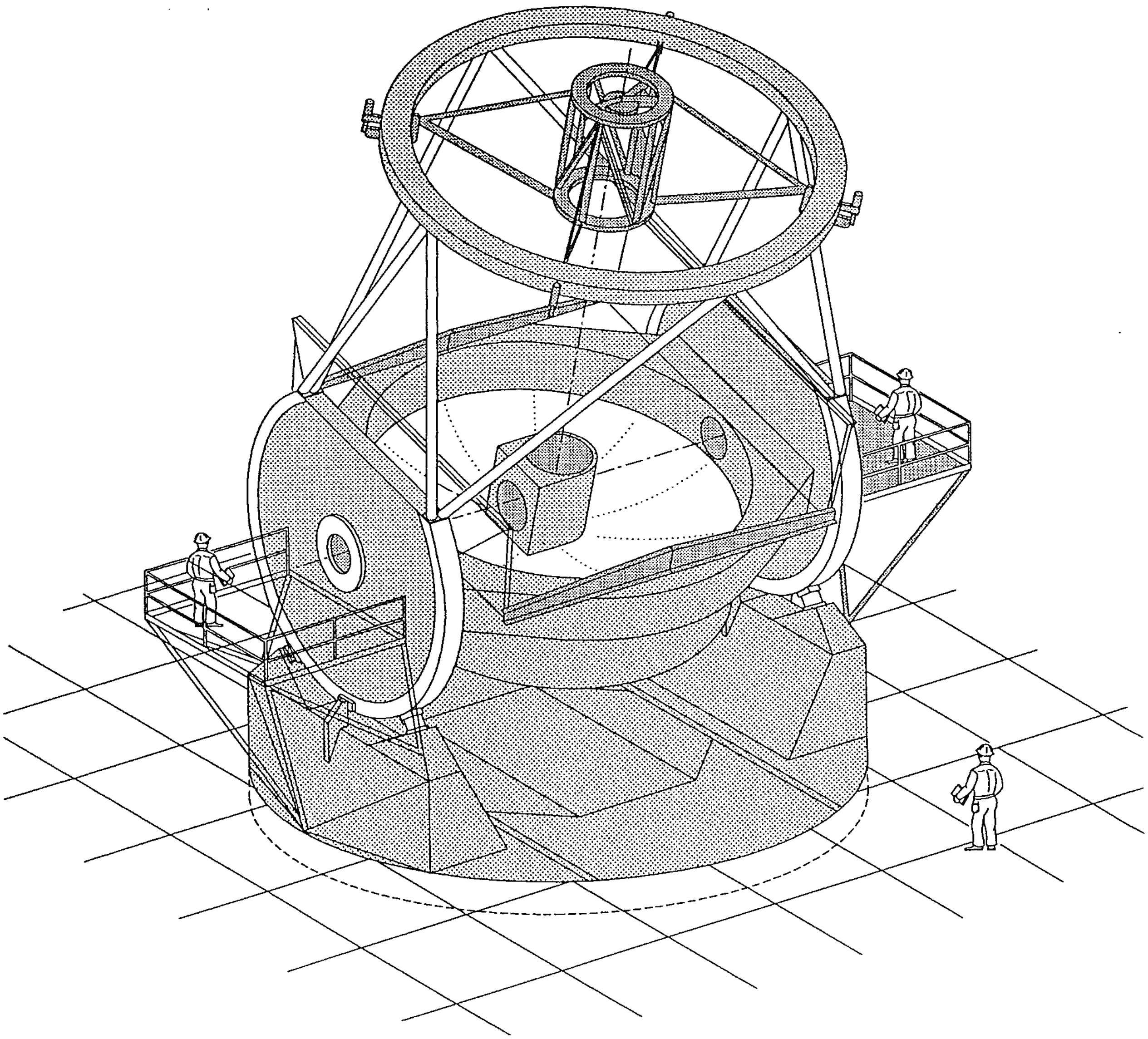
MAGELLAN PROJECT

University of Arizona

Carnegie Institution of Washington



Vane End Actuator System

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INDEX

1. Introduction

The Magellan Project 6.5 Meter Telescope is designed to accept either of four secondary mirror assemblies. Each assembly will attach to the secondary central support structure, which will in turn be supported by the four secondary vane sets. Focus and collimation adjustments will be necessary to compensate for thermal effects (growth of the telescope structure due to the 36° C ambient temperature operating range) and gravity effects (distortion of the structure due to rotating through the gravity field). These motions will be compensated on a relatively slow time scale, with focus required no more often than about each 30 seconds.

It would be quite difficult and expensive to provide for these relatively large adjustments by mechanisms mounted in or on each mirror cell. Therefore, we have elected to use one system of four actuator assemblies which connect the ends of the four vane sets to the secondary end ring. The "vane end actuators" are indicated schematically on the cover of this report and in some detail on E271100 sheet 2. The latter drawing will subsequently be referred to as "sht 2".

This report describes the design for the vane end actuators as they will be used in the telescope as well as the test apparatus which will be used to verify their performance in advance of the telescope construction.

2. Description

2.1 Lever System

Each vaneset consists of an upper and lower vane (OSS at zenith) which intersect at a point, defined here as the "vane end node". The purpose of the vane end actuators is to position each vane end node with sufficient accuracy to define the six degrees of freedom for the secondary mirror with respect to the primary. Alignment tolerances for which the system is designed (Shectman, "Secondary Mirror Alignment Tolerances", July 8, 1992) are 3 arcsec tilt, 30 microns lateral displacement (radial position), and 3 microns focus (axial position).

At the same time the actuators must establish and maintain a (maximum 10,000 lb) preload in each vaneset. This will assure that the lowest local resonant frequency in the secondary end is above 8 Hz.

A system of levers was chosen to position the vane end node as shown on sht 2. The radial lever is pivotally mounted to the end of the axial lever so that when the radial actuator alone moves, the vane end node moves with predominantly radial motion. The axial lever is pivotally mounted to the support structure (the secondary circular end ring) so that when the axial actuator alone moves, the two-lever assembly moves so that the vane end node moves predominantly axially. The two motions will be computer controlled to compensate for the slightly arcuate path defined by either actuator alone.

The use of levers, as compared to a two axis linear system, has the following advantages in this application:

- The force of each actuator is amplified due to the mechanical advantage of the levers. This decreases the force, and increases the stroke, required from each actuator. The commercially-available actuators are therefore smaller in this short-stroke application.

- The positioning accuracy of each actuator is amplified due to the same leverage action. Stated more correctly, the positioning error of each actuator (if the open loop feedback is mounted near the actuator) is de-amplified due to the motion of the levers.

- The effective friction of the system is reduced. Although two bearings rotate for either independent motion, the bearing radius is much smaller than half the arc radius of the motion of the vane end node. Reduced friction will effect lower hysteresis under open loop servo control.

- The system is compact in the area inside the circular end ring, where space is at a premium.

2.2 Pivots

Each assembly requires three main pivots for the levers and two small pivots for each actuator. In the early preliminary design it was anticipated that all pivots might use precision bronze bushings for their low cost. However, a calculation indicated that the focus positioning error using bushings might be as high as 25 to 50 microns. The dominant effect would be hysteresis due to friction in the large pivots under the 10,000 lb vane preload. It was therefore decided that rolling element bearings would be used for the large pivots. Timken precision class "0" tapered roller bearings were finally selected for their relatively high load capacity, ease of preload, acceptable precision and relatively low cost.

Each bearing has a dynamic radial load rating of 5,700 lbs for 3000 hours at 500 rpm L_{10}. This means that less than 10% of bearings should fail under a radial load of 5,700 lbs after 90 million revolutions at 500 rpm. In our very slow, small angular motion application, the static load rating is more interesting. The bearings can withstand a static radial load of 33,500 lbs/bearing and experience only very slight brinelling of the rolling surfaces.

Although friction on the small pivots would not contribute substantially to positioning error at the vane end nodes, it could cause considerable error in the load feedback devices (the shear pin load cells discussed in section 2.6 below). Therefore, rolling element bearings were used for the small actuator pivots. Needle bearings were selected for their low cost and ease of mounting. They were used at the ends opposite the load cells as well as in the load cell clevises.

2.3 Actuators

With the lever geometry shown on sht 2, the axial actuator requires a capacity of about 1,000 lbs, the radial of about 4,000 lbs. Numerous commercially-available actuators were considered for use in the system. Most had inadequate capacity. Those remaining consisted of two types.

The ballscrew type were generally used in higher precision applications due to their low friction under active servo control. However, they are not generally self-locking. In an application such as this it would be undesirable for the motor to remain continuously active (and therefore dissipating heat) when it only needs to be updated occasionally. While a motor brake could theoretically be used (and the motor turned off most of the time) there would be a chance that the mere application of the brake might cause slight rotational motion, contributing an additional error.

The other type, using machine screws, have higher friction but are generally self-locking. That is, with the motor turned off they will not back-drive under an external axial load. However, they tend to be used in less precise applications due to their high friction under open loop servo control.

It was finally concluded that even the best ballscrew type actuators had inadequate precision (with the encoder on the motor or screw) and that we therefore must encode around the actuator. That is, that a linear encoder would be mounted in parallel with the actuator so that we could truly control the length of the actuator at or near the level of the encoder resolution. Doing this, the actuator precision is no longer critical, and its friction is only important to the extent that it contributes to hysteresis. Therefore, ordinary self-locking machine screw actuators were finally selected for their simplicity, compactness, and low cost. The hysteresis of the complete assembly including both the lever pivots and actuators will be determined by the test outlined in section 3.

The axial actuator is a rotating nut, translating screw unit. The Duff-Norton Model M-2465-2-1 consists of a 1" diameter, 1/4" pitch single lead acme screw driven by a 20:1 worm, defining an overall ratio of 80 turns of the worm (and therefore motor) per inch of stroke. The unit will include a NEMA 56C flange to which we will mount our Mycom UPS52-5913(B) 5-phase stepping motor. At 500 (full) steps/revolution, the motor provides a theoretical resolution of 0.26 microns/step at the vane end node, or 0.64 microns/step at the actuator.

The screw is enclosed by a tube with a wiper-scraper seal to keep dust out and grease lubrication in. It will include a threaded end on the moving tube to which we will mount our special clevis with load-sensing pin. The opposite end includes mounting provisions for a similar special clevis. The unit will include a limit switch assembly which can be used to limit both ends of travel.

The actuator has a load rating of 2,000 lbs limited by the worm and nut. In this application the 54 lb-in. peak torque rating of the motor will be limited to about 20 lb-in. by a current limitation in the control system. The calculated maximum output force we require from the actuator is about 1,000 lbs, which would require about 14 lb-in. input torque. The actual current will be determined using the shear pin load cells for force feedback during calibration of the telescope control system. The actuator manufacturer states that due to internal friction the required input torque can vary by something like 15% to 40% over the life of the actuator.

The radial actuator is also a rotating nut, translating screw unit. The Duff-Norton Model LTM-9705-4-B-C consists of a 1 1/2" diameter, 3/8" pitch single lead acme screw driven by a 6:1 worm plus a 12:1 input gearbox. This defines an overall ratio of 192 turns of the motor per inch of stoke. The unit will also include a NEMA 56C flange to which we will mount our Mycom UPS52-5913(B) 5-phase stepping motor. At 500 (full) steps/revolution, the motor provides a theoretical resolution of 0.08 microns/step at the vane end node, or 0.26 microns/step at the actuator.

The screw is enclosed by a flexible bellows-type boot. The screw will include a threaded end to which we will mount a special load-pin clevis identical to that used for the axial actuator. We will adapt the base-mounting of the body of the actuator to trunnions using a special trunnion plate. This unit will also include a limit switch assembly which can be used to limit both ends of travel.

This actuator has a structural rating as a jack of 5 tons but is de-rated to 5,000 lbs due to the input gearbox. The 54 lb-in. parallel peak torque rating for the motor will be limited to about 32 lb-in. by a current limitation in the control system. This will limit the force from the actuator to a level slightly above the 3,900 lb maximum force which we require. The actual current will be determined using the shear pin load cells for force feedback during calibration of the telescope control system, and again the required input torque can vary by something like 15% to 40% over the life of the actuator.

2.4 Axial Actuator Preload Plungers

The axial actuator requires external preloading to accommodate load reversal at low altitude angles. The load reversal (compression to tension at lowering altitude angles) would otherwise cause a sudden and uncontrollable focus and tilt motion of the secondary due to clearance on the clevis bearings and acme threads.

Two spring plunger assemblies straddling each axial actuator have been included to assure that the actuator is in compression for all secondary assemblies and at all altitude angles. Each plunger houses two Century D51 die springs in series preloaded to about 250 lbs per plunger in the minimally compressed condition. The maximum force in each plunger is about 330 lbs at maximum compression. The plungers are used to house the springs (so that they are contained should one ever structurally fail) and to provide a convenient means of preloading. All-thread can be screwed into the back end cap until the plunger rod nut can be installed. After installing the nut, the all-thread can be removed, the spherical rod end installed, and the length adjusted with the rod nut until it can be installed on the support pins on the lever and support structure.

2.5 Position Feedback System

Two modes of operation will be used for the vane end actuator system. The focus and collimation will occasionally be reset under closed loop control of both actuators. That is, the secondary will be realigned with the primary using sensors which are directly in the focal plane.

While tracking or slewing between those resets, the actuators will be under open loop control to compensate for gravity and thermal effects. That is, the computer will monitor the telescope altitude angle and structure temperature, determine the expected relative motion of the secondary with respect to the primary and continuously calculate the desired actuator positions and rates necessary to keep the secondary focused and collimated. The actuators will be controlled as dictated by computer calculation using only their local feedback devices (encoders).

Three possible locations for the encoders were considered. If located on the motor or leadscrew, mechanical errors between the encoder and vane end node would contribute to positioning error which would be additive to that error caused by system hysteresis. These mechanical errors would include actuator error (such as leadscrew pitch error and stepper step size or servo deadband) and pivot bearing runout.

If located near the actuator, the mechanical errors of the actuator would be eliminated with pivot bearing runout error (and system hysteresis) remaining.

Finally, If located at the vane end node, all mechanical errors would be eliminated, leaving only error due to system hysteresis.

It was decided that

- Even the best commercially-available actuator would contribute significant error due to mechanical tolerances. Therefore, the encoders should be across the actuator (i.e. in parallel with the actuator length) or at the vane end node.

- It would be difficult to mount the encoders at the vane end node, since the axial encoder would require a significant physical projection forward of the circular end ring and a protective device would be required.

Therefore, it was decided that the best place to put the encoders in this application is across the actuator. As shown on sht 2, both encoders, which are somewhat delicate by their construction and mounting, are naturally protected by adjacent structure (in the case of the radial unit) and the spring plungers (in the case of the axial unit).

Mitutoyo 542-325 linear (glass) scales, with 1 micron resolution, have been tentatively chosen. The axial unit is mounted very close to the axial actuator. As located it will provide maximum control resolution and adequate stroke (2 in. stroke available, 1.67 in. actual required). The radial unit is mounted about midway between the actuator and its pivot, where 1.91 in. of stroke is required. The lost accuracy is not a problem here since the radial precision required is much less demanding (30 microns at the vane end node).

Both encoders will be flexure mounted with piano wire flexures mounted to sleeves at each end of the unit and at the structure. The flexures must have relatively low off-axis stiffness so as not to cause binding of the encoder plunger. However, axial stiffness of the flexures is not particularly critical, so this should be achievable.

Other types of feedback devices, especially ones providing absolute position information (that is, irrespective of power loss) are being considered for permanent use in the telescope.

2.6 Force Feedback System

We are defining 6 degrees of freedom for the secondary central support structure with 8 actuators. Therefore, it is desirable to monitor the force in the system to verify that the actuators are not unnecessarily straining the structure as well as to know the vane pretension at a very low level of precision. It is intended that the load be known and maintained to an accuracy of about 5-10% of full scale (200-400 lbs for each radial actuator, 50-100 lbs for each axial actuator).

It was initially assumed that load cells would also be necessary to protect the system from structural damage, should a control system malfunction occur in which the actuators misloaded the vanes due to overconstraint. However, a later calculation verified that the vanes and central support structure are sufficiently robust to preclude this. That is, the structure is designed such that the four axial actuators can twist the structure with their stall forces (conservatively assumed as twice their required capacity loads) simultaneous with stall forces in the radial actuators (assumed as 20,000 lbs/vaneset), and the specification safety factor of 3:1 is preserved. Later discussion with the actuator manufacturer indicates that internal drive friction in each actuator will vary by no more than 40%. Therefore, the current in all actuators can be limited such that their forces should never exceed about 140% of their required capacity loads, and the design is quite conservative.

Shear pin type load cells were selected as the simplest and least expensive means of monitoring the actuator forces. As shown on sht 2, the load pins are retained to the central support lug by a cap screw against a slotted flat on the outside diameter of the pin. Needle bearings mounted in the mating special clevis accommodate the small rotation angles established by the pivot motions. The error due to the friction torque from the needle bearings was estimated to be less than 1%. This mounting (as opposed to having the bearing in the central lug) makes for reduced looseness under the slight torsional loading created by the acme screw/nut friction (the supporting levers and structure must react these small torques).

In theory, only two load cells are necessary, one axial and one radial. However, two load cells will be used in each subsystem for redundancy (for comparison with each other, should either fail). The shear pin type load cells are commercially available from at least two sources.

3.0 Test Apparatus

3.1 Description

A prototype of one assembly will be built and tested using the test apparatus shown on E271100 sheet 1. The apparatus consists of four components:

The support frame is an L-shaped weldment made from 12 x 6 x 1/4 structural tubing with two support legs and an integral mounting structure (at its right end as shown on sht 1). The main supporting pivot and axial actuator mount in the mounting structure. The support frame serves to transfer the vane loads back to the pivot flexure, at its far end.

The vane provides a means of pulling with the rated 10,000 lb pretension while accommodating the axial and radial motions with integral flexures at its left end.

The weighted lever provides a means of loading the vane with a virtually constant force due to gravity acting on the 1,350 lb integral weight at its right end.

The pivot flexure provides a means of rotation for the weighted lever. The use of flexures for the pivot and vane end, rather than rolling element bearings, assures virtually zero hysteresis will be contributed by the test apparatus. The pivot flexure must transfer both the 10,000 lb horizontal vane load and 1,400+ lb vertical load from the weighted lever back to the support frame. It therefore is mounted at the shallow angle so that it is nominally in compression, with an added bending moment defined by the small rotation angle of the weighted lever.

Mitutoyo linear gages have been used to encode the position of the vane end node as well as the actuator positions. Actually, the gages are mounted as closely to the vane end node as possible, but cannot indicate the node directly due to its inaccessibility. Therefore, the gages will indicate to machined surfaces as shown and the test apparatus will be used primarily to measure position repeatability irrespective of the actual vane end node absolute position. The main contributors to nonrepeatability are expected to be:

- Hysteresis due to friction in the large pivots in combination with bending and shear compliance of the levers.

- Hysteresis due to friction in the actuators in combination with their internal (drive component) compliance.

- Nonrepeatable runout (noise) in the large pivot bearings.

- Actuator control due to encoder resolution and motor step size.

3.2 Test Procedure

A detailed test procedure has not been developed as yet. The purpose of the test is to determine whether the system open loop position repeatability meets or exceeds the requirements (30 microns radial position, 3 microns axial position) at the vane end node. These data should be determined over the full range of node motion (20 mm axial, 25 mm radial).

This can be accomplished by various means, such as:

- A rough idea of the system hysteresis (likely to be the dominant source of positioning error) can be had by simply stepping an actuator motor slowly and monitoring the output (node position feedback). The node position is likely to move in abrubpt increments at the level of the system hysteresis. This simple test will not include effects of mechanical errors (such as nonrepeatable bearing runout), which would be included by further tests such as those described below.

- Commanding a cyclical motion of the actuator while continuously monitoring both its position (using the actuator local feedback) and the output (using the node position feedback). If the output position is then plotted as a function of input position, a series of curves will be established. The maximum vertical separation occuring anywhere (divided by 2) will be the nonrepeatable positioning error over the range of motion represented by that particular cyclical motion. If this is done for numerous cyclical motions which cover the full range of node motion, the test could be considered complete.

- Establishing a reference position, noting the input and output positions (actuator local feedback and node position feedback), then commanding a series of moves away from, then back to, the reference position. If this is done for many alternate positions for each of numerous reference positions, a scattering of points will be established around each reference position. The radius of each scattering of points would be the local nonrepeatable positioning error, the radius of the largest scattering the maximum nonrepeatable positioning error.

3.3 Test Setup Notes

- Approximately 0.16 in. of node travel will occur before the test apparatus is fully strained. Therefore, compensate for this in the initial alignment (set up with axial actuator 0.6 in. extended beyond mid-stroke with system unstrained but at "neutral" position).

- Adjust limit switches carefully so that Mitutoyo gages (especially the one on the radial actuator) aren't over stroked (damaged). Don't install gages until system is strained and limit switches set. Then position @ 1 limit and install gage safely within travel range. Make sure limit switch is always in circuit to motor. Don't unstrain system (at lower support stop) with Mitutoyo installed, *or set up system for slightly reduced radial travel just for the test.*

- Prior to and during torquing of pivot pins, measure rotation friction, compare with that predicted from Timken estimate.

GENERAL NOTES

UNLESS OTHERWISE NOTED

1. ALIGN SYSTEM AS FOLLOWS:

   AFTER ASSEMBLING ACTUATOR/LEVER SYSTEM, POSITION EACH ACTUATOR IN ITS NEUTRAL POSITION (CENTER OF ACTUATOR TRAVEL RANGE). ASSEMBLE VANE TO PIVOT PIN AND LOOSELY TO D101 WEIGHTED LEVER. ASSEMBLE D101 LOOSELY TO E102 SUPPORT FRAME. WITH D101 IN NEUTRAL POSITION (NOTE 5" REF DIM, D101 AND E102 COPLANAR NOTE) AND ALIGNED WITH E102 (6" WIDTH OF D101 CENTERED IN 12" WIDTH OF E102), DRILL OUT (6) DOWEL HOLES FOR 3/8" DOWELS (ITEMS 36), INSTALL DOWELS AND TORQUE ALL CLAMP PLATE BOLTS.

   DURING ABOVE ALIGNMENT PROCEDURE, SUPPORT D101 WEIGHTED LEVER WITH BLOCKING TO FLOOR (5" DIMENSION).

2. TORQUE ITEM 5 LOCKNUT TO 125 FT–LBS PRIOR TO TIGHTENING (3) SET SCREWS.

3. (2) INBOARD SPRING PLUNGERS TO HAVE (2) –7 DIE SPRINGS EACH, OUTBOARD PLUNGERS TO HAVE (1) –7 AND (1) –8 EACH. USE 1/2–13 NC ALLTHREAD THRU B123 CAP TO COMPRESS SPRING ASSEMBLY. NOTE 1 3/4 DIMENSION APPLIES IN NEUTRAL POSITION (AXIAL ACTUATOR AT MID–STROKE).

4. LIGHTLY SNUG (1) ITEM 32 SET SCREW AGAINST ITEM 40 PIANO WIRE, THEN INSTALL OPPOSITE SET SCREW TO PINCH PIANO WIRE BETWEEN 2 SET SCREWS. TORQUE TO 15 IN–LB. ALL DIMENSIONS IN VIEW C APPLY AT NEUTRAL POSITION (ACTUATORS AT MID–STROKE). TO ACHIEVE THESE DIMENSIONS, ADJUST DEPTH POSITION OF C126 SLEEVES IN THEIR SUPPORTING HOLES (IN 1" PLATES). TORQUE ALL #8–32 SET SCREWS TO 15 IN–LBS.

5. TORQUE –31 TO 15 IN–LB.

6. PRIOR TO DRILLING AND DOWELING B113 TRUNNION SUPPORT BAR, INSTALL –3 RADIAL ACTUATOR AND ALIGN SO THAT B113 BARS ARE PERPENDICULAR TO SIDE PLATES OF E112 RADIAL LEVER.

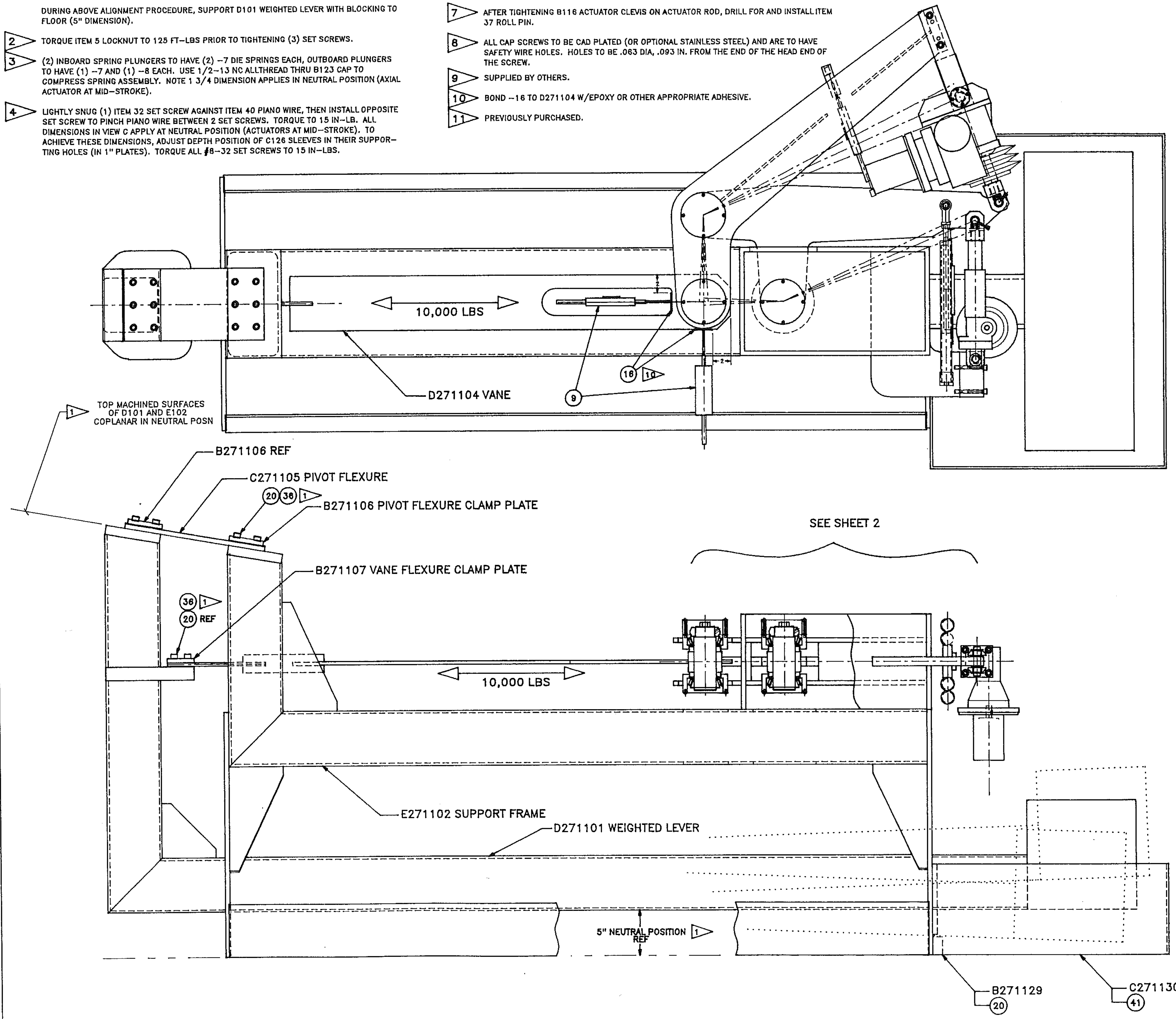
7. AFTER TIGHTENING B116 ACTUATOR CLEVIS ON ACTUATOR ROD, DRILL FOR AND INSTALL ITEM 37 ROLL PIN.

8. ALL CAP SCREWS TO BE CAD PLATED (OR OPTIONAL STAINLESS STEEL) AND ARE TO HAVE SAFETY WIRE HOLES. HOLES TO BE .063 DIA, .093 IN. FROM THE END OF THE HEAD END OF THE SCREW.

9. SUPPLIED BY OTHERS.

10. BOND –16 TO D271104 W/EPOXY OR OTHER APPROPRIATE ADHESIVE.

11. PREVIOUSLY PURCHASED.



Note	Item	Quan.	Description
	41	4	SOCKET HD CAP SCR – 1/2–13 NC X 3/4 CAD PL OR SS
	40	4	PIANO WIRE – .047 DIA X 2 LG
	39	6	COTTER PIN – 1/8 X 1 1/2
	38	4	HEX NUT – 1/2–20 UNF – CAD PL OR SS
	37	2	SPRING ROLL PIN – 1/4 X 2 1/4
	36	6	HARDENED GROUND MACHINE DOWEL PIN, 3/8 X 1 1/2
	35	4	HARDENED GROUND MACHINE DOWEL PIN, 3/8 X 1 1/2
	34	4	HARDENED WASHER – 1/2" – CARR–LANE CL–9–FW OR EQ
8	33	8	SOCKET HD CAP SCR – M6 X 20 MM LG CAD PL OR SS
	32	18	SOCKET HD SET SCR – #8–32NC X 3/16LG CAD PL OR SS
	31	2	SOCKET HD CAP SCR – #8–32NC X 5/8 LG CAD PL OR SS
	30	4	SOCKET HD CAP SCR – 1/4–20NC X 1/2 CAD PL OR SS
	29	20	SOCKET HD CAP SCR – #10–24NC X 2 CAD PL OR SS
	28	4	SOCKET HD CAP SCR – 1/2–13NC X 1 3/4 CAD PL OR SS
	27	4	SOCKET HD CAP SCR – 1/2–20NF X 1 3/8 CAD PL OR SS
	26	4	SOCKET HD CAP SCR – 5/8–11NC X 1 1/4 CAD PL OR SS
	25	8	SOCKET HD CAP SCR – #10–24NC X 5/8 CAD PL OR SS
	24	8	SOCKET HD CAP SCR – #10–24NC X 3/8 CAD PL OR SS
	23	1	SOCKET HD CAP SCR – 1/4–20NC X 3/4 CAD PL OR SS
	22	1	SOCKET HD CAP SCR – 1/4–20NC X 1 1/4 CAD PL OR SS
	21	2	SOCKET HD CAP SCR – 3/8–16NC X 3 1/4 CAD PL OR SS
8	20	22	SOCKET HD CAP SCR – 1/2–13NC X 1 1/2 LG CAD PL OR SS
	19	3	RETAINING RING – INDUSTRIAL NO. 3100–243 OR EQ
	18	4	SPHERICAL ROD END – BOSTON CFHD–8
	17	8	THRUST WASHER – BOSTON TB–1222
	16	2	PRECISION GROUND FLAT STOCK – 1/8 X 1/2 X 2 1018
	15	2	THRUST WASHER – BOSTON TB–1632
	14	4	INNER RING – TORRINGTON IR–128
	13	2	NEEDLE BEARING – TORRINGTON B–128
	12	4	NEEDLE BEARING – TORRINGTON B–168
	11	2	NEEDLE BEARING – TORRINGTON B–1612
	10	2	1 DIA X 2 1/2 HARDENED GROUND MACHINE DOWEL PIN
9	9	4	LINEAR GAGE – MITUTOYO 542–325
	8	2	DIE SPRING – CENTURY D49
	7	6	DIE SPRING – CENTURY D51
11	6	2	SHEAR PIN LOAD CELL – NO. ALD–SP–SHEAR PIN–930811 – A.L. DESIGN INC. 1411 MILITARY RD. BUFFALO, NY 14217 (716)875–6240 FAX (716)875–2404
11	5	3	LOCKNUT, SKF NO. KMT11. SDK CO. 23392 MADERA RD. SUITE L – MISSION VIEJO, CA. 92691 – (714)855–9881
11	4	6	TAPERED ROLLER BEARING – ASSY NO. 906A2 – (REF CUP 28920 CONE 28995), PRECISION CLASS "0" – THE TIMKEN CO. – 23832 ROCKFIELD BLVD. SUITE 170, LAKE FOREST, CA. 92630
11	3	1	DUFF–NORTON MAXIPAC ACTUATOR WITHOUT MOTOR. LTM–9705–4–B–C. 12:1 REDUCER RATIO. MOTOR ON RIGHT HAND SIDE, POSITION 3. LIMIT SWITCH NO. SKA–6000–C–10 ON LEFT HAND SIDE, POSITION 3. P.O. BOX 7010 – CHARLOTTE, NC. 28241–7010
11	2	1	DUFF–NORTON CO. MODULAR ACTUATOR NO. M–2465–2–1, NEMA 56C FLANGE MOUNT. 1"–8NC X 1 1/8 LG THREADED END (11.25 IN. CLOSED LENGTH TO END OF THREAD). 20:1 WORM RATIO. WITH LIMIT SWITCH SKA–6000–C–10 MOUNTED IN POSITION "2".
9	1	2	STEPPER MOTOR – MYCOM UPS52–5913(B)

Item	Quan.	Description
–	1	C271130 GUARD
–	1	B271129 SUPPORT BAR
–	2	C271126/7/8 FLEXURE SUPPORT SLEEVES
–	2	C271125 MOTOR C–FACE ADAPTER
–	2	B271124 SPRING PLUNGER SUPPORT PIN
–	4	B271123 SPRING PLUNGER CAP
–	4	B271122 SPRING PLUNGER PISTON
–	4	B271121 SPRING PLUNGER BODY
–	1	B271120 PILLOW BLOCK
–	1	B271119 CLEVIS PIN
–	1	B271118 CLEVIS PAD
–	4	B271117 CLEVIS BEARING RETAINER
–	2	B271116 ACTUATOR CLEVIS
–	1	C271115 TRUNNION ADAPTER
–	2	B271114 TRUNNION BEARING RETAINER
–	2	B271113 TRUNNION SUPPORT BAR
–	1	E271112 RADIAL LEVER
–	1	E271111 AXIAL LEVER
–	6	B271110 BEARING SPACER
–	3	B271109 COVER SPACER
–	5	B271108 DUST COVER
–	1	B271107 VANE FLEXURE CLAMP PLATE
–	2	B271106 PIVOT FLEXURE CLAMP PLATE
–	1	C271105 PIVOT FLEXURE
–	1	D271104 VANE
–	3	B271103 PIVOT PIN
–	1	E271102 SUPPORT FRAME
–	1	D271101 WEIGHTED LEVER
	1	E271100 VANE END ACTUATOR TEST APPARATUS

BILL OF MATERIAL

L&F INDUSTRIES
HUNTINGTON PARK, CA.

NEXT ASSEMBLY: ——

UNLESS OTHERWISE SPECIFIED
FRACTIONAL +/– 1/16; DECIMAL .XX +/– .03, .XXX +/– .010; ANGULAR +/– 1/2 DEG
BREAK SHARP EDGES; DO NOT SCALE DRAWING
MACHINED SURFACES 125√; ALL DIMENSIONS IN INCHES

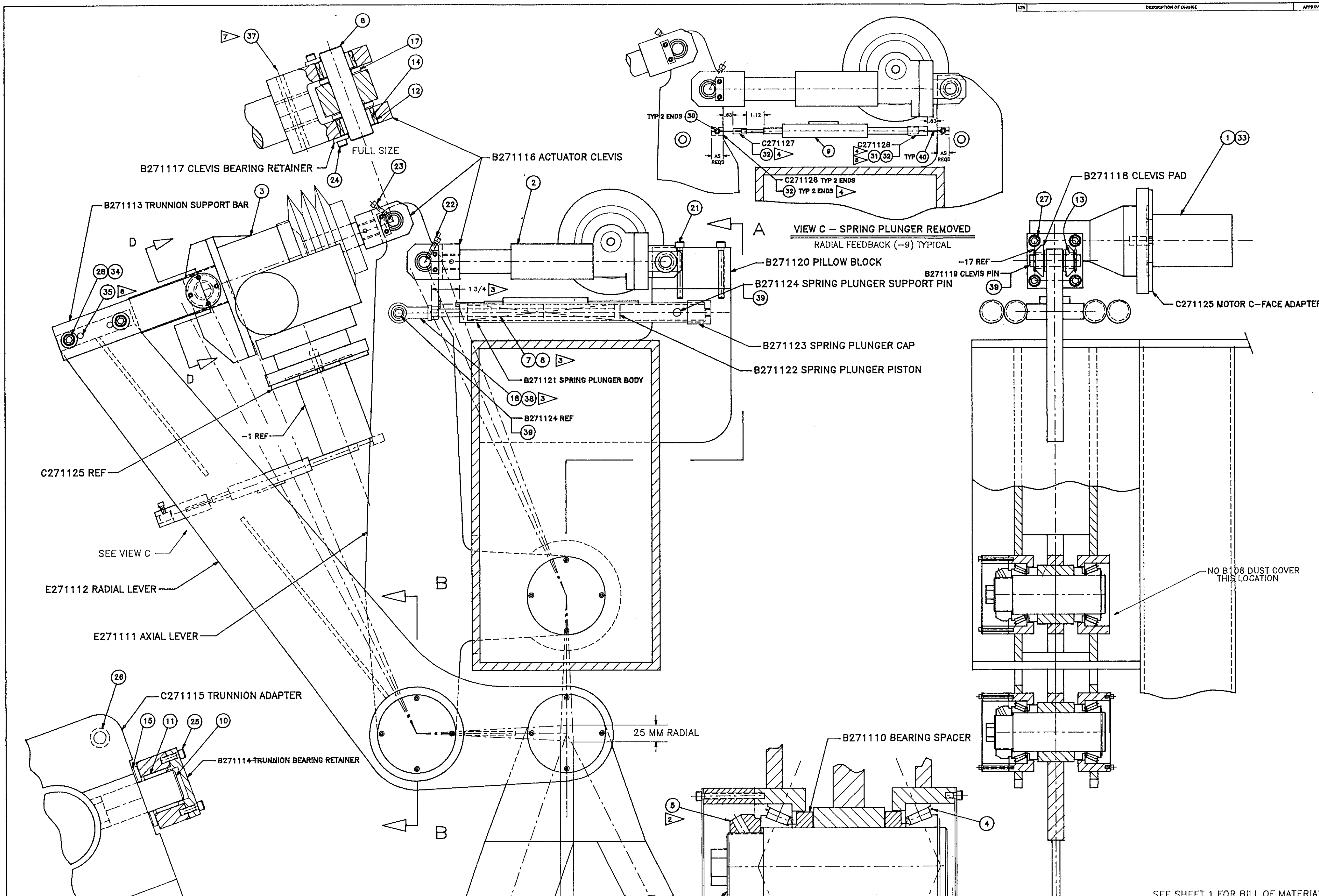
CALC WEIGHT 2800 LBS

	BY	DATE
APVD		
ENGR	SMG	8–30–93
CHK	SMG	8–30–93
DWN	SMG	8–30–93

MAGELLAN PROJECT TEST APPARATUS

VANE END ACTUATOR TEST APPARATUS ASSEMBLY

SCALE 1/4 | DRAWING NUMBER E271100 SHT 1 | REV



SEE SHEET 1 FOR BILL OF MATERIAL

ITEM	QUAN.	DESCRIPTION
BILL OF MATERIAL		

L-F INDUSTRIES
HUNTINGTON PARK, CA.

NEXT ASSEMBLY: E271100

UNLESS OTHERWISE SPECIFIED
FRACTIONAL +/- 1/16 | DECIMAL .XX +/- .03 .XXX +/- .010 | ANGULAR +/- 1/2 DEG
BREAK SHARP EDGES
DO NOT SCALE DRAWING
MACHINED SURFACES 125√
ALL DIMENSIONS IN INCHES

CALC WEIGHT ——

MAGELLAN PROJECT TEST APPARATUS

VANE END ACTUATOR TEST APPARATUS ASSEMBLY

	BY	DATE
APVD		
ENGR	SMG	6-30-93
CHK	SMG	6-30-93
DWN	SMG	6-30-93

SCALE 1/2
DRAWING NUMBER E271100 SHT 2
REV