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AXIAL SUPPORTERS FOR REF MIRROR

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AXIAL SUPPORTERS FOR REF MIRROR

TMT Report #80 by Budiansky (Nov 17, 1983) lists mirror deformations for a recommended support system. This system includes 3 "hard" points and 6 force pistons. A bellows system can be designed which will produce the desired forces on the mirror to the tolerances specified. The location of the force pistons can be achieved. They will be within case 1 and case 5 (see Table 1 in TMT #80). This adds about 0.4 nanometers to the nominal 5.8 nm deflection. The desired forces can be met to within case 6 and case 7. (2% of maximum or .4 lbs or 181 grams) This adds about 0.6 nm to the nominal 5.8 nm deflection.

The 6 force pistons are part of a bellows system. This system is a set of 6 identical "slave" bellows all interconnected to a single "master" bellows. The master bellows is driven by a weight. The weight does two things: it compresses the air in the closed system as the telescope is pointed toward the zenith and it compresses the master bellows. The stiffness of the master bellows is not negligible. The amount of travel is directly controlled by the initial volume in the bellows system. The air must be compressed to a fraction of its initial volume. This fraction depends on the size of the slave bellows. The ideal gas law pV=nRT is used to set the final pressure. The initial pressure is 14.7 psi (horizon-looking). The final pressure is then 16.48 lbs / slave bellows area. The slave bellows area is 2.11 sq. in. The final pressure is 14.7 + 16.48/2.11 = 14.7 + 7.8 = 22.5 psi. The ratio of 14.7/22.5 or .65 is the final volume to initial volume ratio. The "dead volume" is the volume of all 6 slave bellows and the interconnecting plumbing. This is 2.68 cubic inches. Now, a suitable master bellows is selected so that the initial volume to final volume ratio is right. Several master bellows will work. Bigger diameters require larger counterweights. A size #60 was selected and an optimized weight of 41.3 lbs will be used. (Effective area is 3.63 sq in) The weight will be positioned via a swing arm. The travel of the weight is .45 inch. Figure 1 diagrams the proposed system. Figure 2 shows details of the bellows. Note the inner plug in the slaves to minimize the dead volume. Figure 3 shows the output force at the slave bellows as a function of counterweight. 28.2 lbs is the ideal weight for a zero stiffness bellows.

The controlling equations for this optimization are included here. The Basic program is listed with the optimized output. Notes: 1. The stiffness of the slave bellows is not considered here since they do not move. But, they must be initially set up to be just touching the reference mirror pads at their free length so that no initial up or down force exits at the glass. 2. The system will have a valve for initial charging. By lifting the counterweight and opening the valve, the system will be ready to operate. 3. Temperature fluctuations. The system is insensitive to changes in the working fluid temperature. The small amount of additional expansion (or contraction) of the air is accommodated by the counterweight moving up or down. 4. The master bellows does not support the counterweight radially. A swing arm does this. For a 12" long arm, the arm pivots move only .5 degree from zenith to horizon. These will be small bearings with dust seals to minimize hysteresis in this support system. 5. For a working flu id which is incompressible, this paper is useless. In such a system, all bellows are the same size and the master bellows force is 16.48 lbs. The only error in slave force is due to the manufacture tolerance between individual bellows. During calibration, we did in fact see about a 3% difference in bellows area due to manufacturing tolerance.

see I04 disc "ref force"

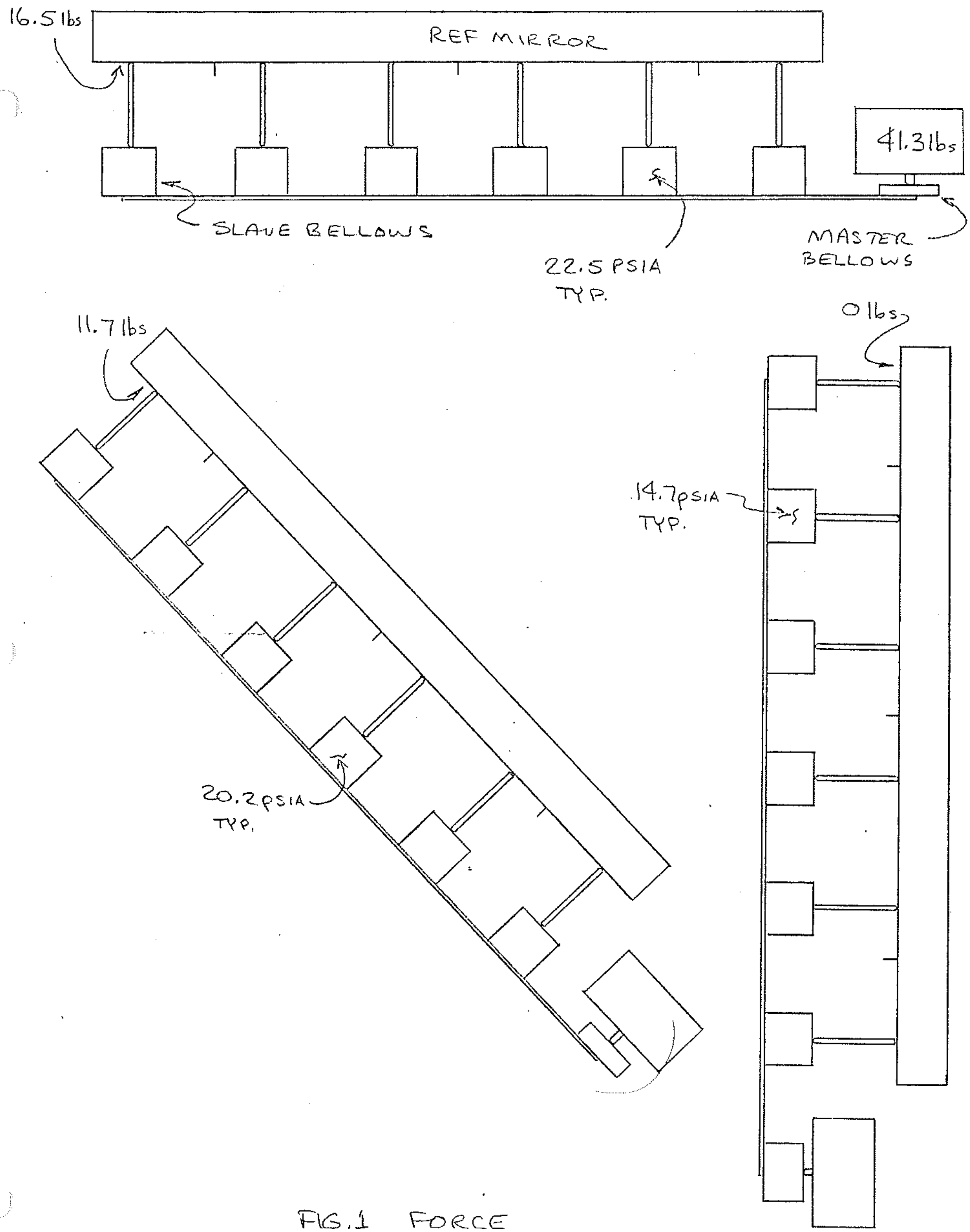
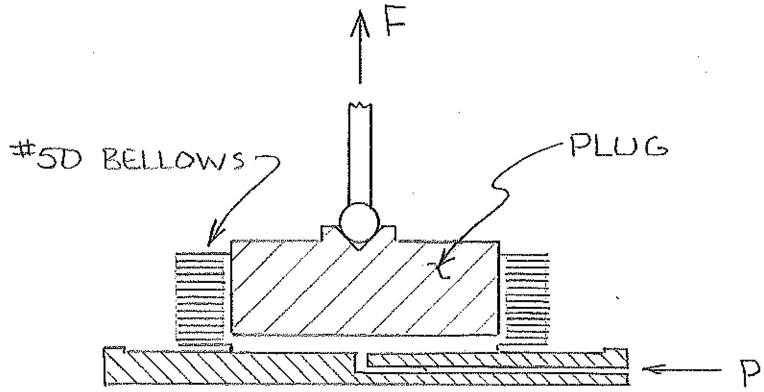
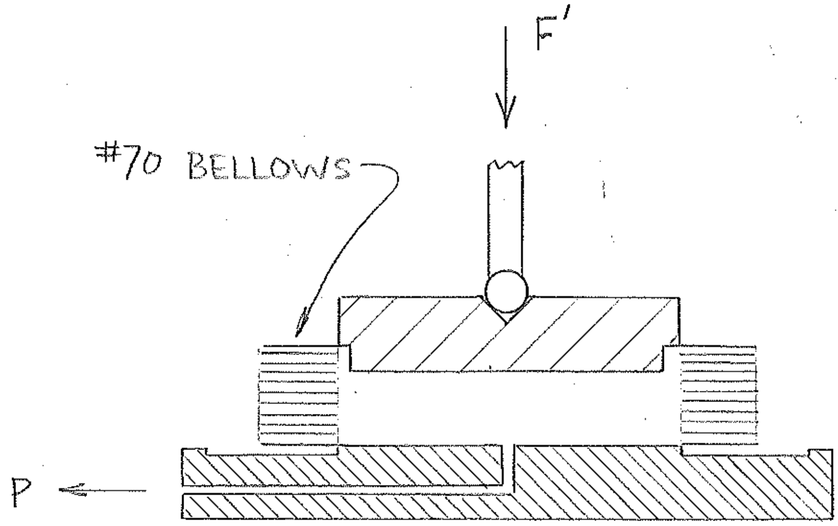


FIG.1 FORCE
SUPPORT SYSTEM



SLAVE BELLOWS, 6 REQUIRED
FULL SIZE



MASTER BELLOWS, 1 REQUIRED

FIGURE 2: BELLOWS DETAIL

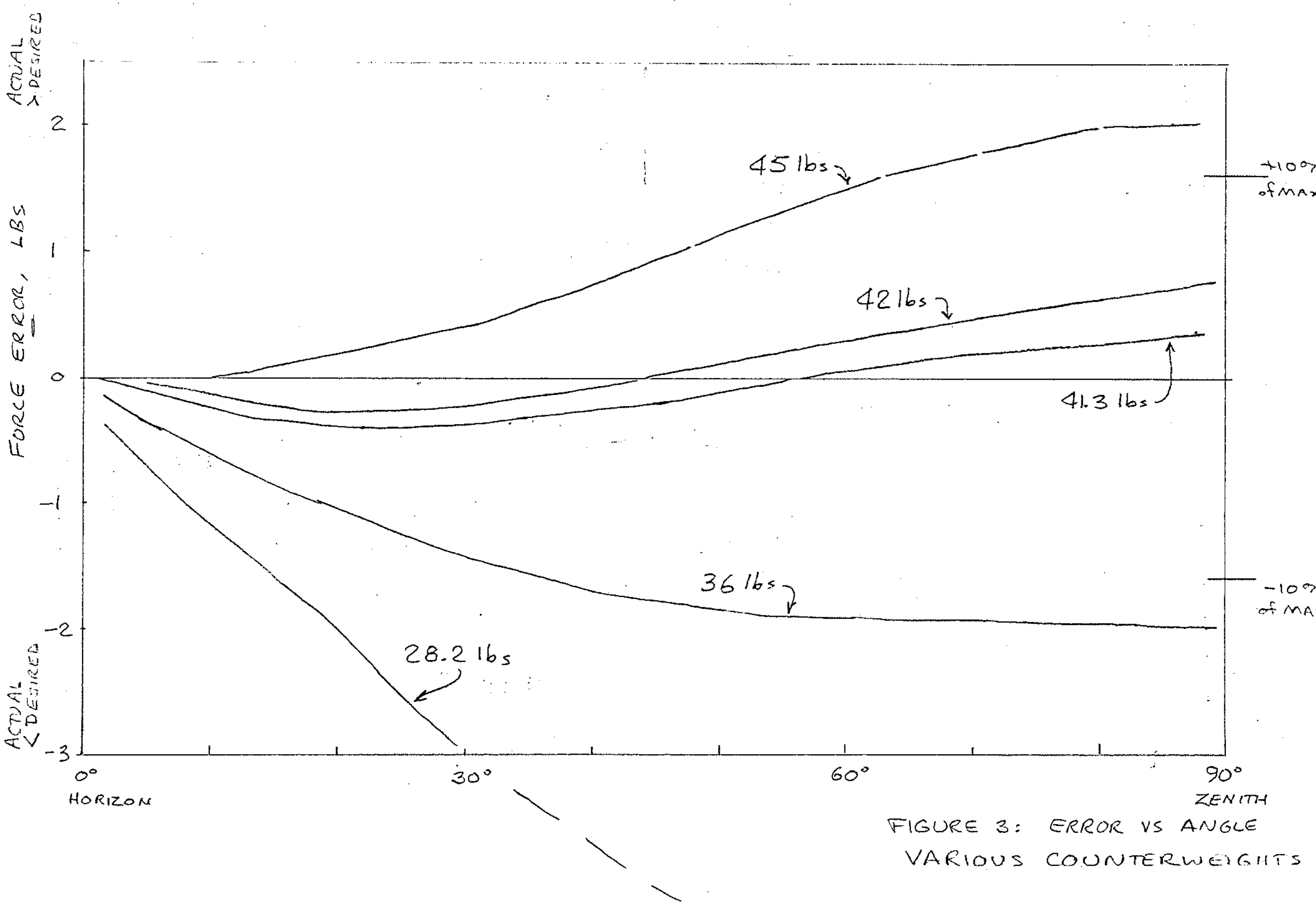
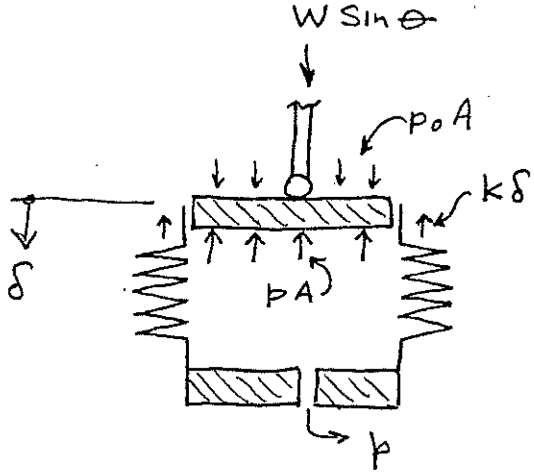


FIGURE 3: ERROR VS ANGLE
VARIOUS COUNTERWEIGHTS



$\theta \equiv 0°$ Horizon-Looking
$\equiv 90°$ Zenith-Looking

Force balance:

$$W \sin\theta + p_0 A = pA + k\delta$$

And p is found from $pV = p'V'$ & $p = \frac{p_0 V_0}{V}$

And $V = V_0 - \delta A$

What we want is $p(\theta)$. This looks hard so we increment δ and see what θ comes out. We will plot $p(\delta)$ and $\theta(\delta)$ for various values of W. See Figure 3 for results... ERROR vs. θ

Definitions:

δ = motion of bellows from "Free" position

k = Spring constant of bellows.

p = pressure in system (absolute)

p_0 = atmospheric pressure, 14.7 psi

W = counterweight

θ = inclination angle of mirror
0°, mirror is "on edge"
90°, mirror is "horizontal", looking at zenith

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2 REM THIS PROGRAM IS SIMILAR TO I08"FINAL SUP"EXCEPT IT ALLOWS FOR THE COUNTER-
4 REM WEIGHT TO BE ARTICULATED. R1 IS THE SHORT AND R2 IS THE LONG ARM. 4-17-84
6 REM J. OSBORNE, LICK
8 REM STORED ON I08 AS "BELLOWS9"
10 PARALLEL
12 OPEN1,4
14 POKE 53281,0
16 POKE 53280,12
20 PRINT""
22 REM THAT CHARACTER IS CTRL&2. CURSER=WHITE
24 PI=3.14159
26 REM SUB 1 REFERS TO SLAVE CYLINDER (50)
28 REM SUB 2 REFERS TO MASTER CYLINDER (70)
100 D=0
150 A2=4.89
152 K=29
154 PO=14.7
162 A1=2.11
165 L1=.885
167 ID=1.36
169 L3=.845
170 L2=1.1
172 MP=.54
174 VT=.2
176 REM:VT IS TUBING VOLUME, CU. IN.
178 VO=(A1*L1-ID^2*L3*PI/4)*6+A2*(L2-MP)+VT
180 R1=8.46
190 R2=16.46
200 INPUT"W,LBS=";W
202 IFW<OTHENGOTO10000
205 PRINT"W=";W
300 PRINT#1,"BELLOWS OPTIMIZATION, CONTINUED, 4-24-84, J. OSBORNE."
306 PRINT#1,"A1=";A1;"L1=";L1
308 PRINT#1,"A2=";A2;"L2=";L2
310 PRINT#1,"INITIAL VOLUME, VO=";VO;"CUBIC INCHES"
312 PRINT#1,"COUNTERWEIGHT= ";W;" LBS"
315 PRINT#1,"R1= ";R1;" IN. R2= ";R2;" IN."
320 PRINT#1
338 PRINT#1,"                                        SLAVE FORCE"
340 PRINT#1,"TH    AL       DISPLACEMENT,          ACTUAL IDEAL          ERROR"
344 PRINT#1,"DEG DEG                    IN.          LB      LB             LB     "
345 PRINT#1,"................................................................"
346 PRINT#1
500 INPUT"DEL, IN. =";DEL
720 D=D+DEL
750 X=D/R1
755 IF X^2>1THEN GOTO3000
760 AL=ATN(X/SQR(-X^2+1))
780 V=VO-D*A2
800 P=PO*VO/V
850 X=(P*A2+K*D-PO*A2)/(W*(R2/R1))
890 IF X^2>1THEN GOTO 3000
895 REM U=90-TH+AL SO THAT ZENITH IS 90, AND HORIZON IS O.

READY.
```

```
900 U=-ATN(X/SQR(-X^2+1))+PI/2+7.0*PI/180
950 TH=PI/2+AL-U
980 F=W*(R2/R1)*COS(U)
1000 F1=(P-P0)*A1
1010 REM: F1 IS FORCE AT SLAVE BELLOWS
1020 REM: FS IS 15.57*SIN(THETA+5.5), IDEAL FORCE AT SLAVE BELLOWS.
1030 FS=15.57*SIN(TH+5.5*PI/180)
1031 IFTH>110*PI/180THENGOTO4000
1032 Z1=INT(TH*1800/PI)/10
1033 Z2=INT(AL*1800/PI)/10
1034 Z3=D
1035 Z4=INT(F1*100)/100
1036 Z5=INT(FS*100)/100
1037 Z6=INT((F1-FS)*1000)/1000
1038 REM F1-FS IS ERROR, LBS.
1040 PRINT#1,Z1;Z2,Z3,Z4;Z5,Z6
2000 GOTO720
3000 PRINT"X^2>1";D
3001 D=D-DEL
3002 DEL=DEL*.1
3004 IF DEL<.0001THEN GOTO4000
3005 GOTO720
4000 PRINT#1,"W=";W;"LBS"
4990 D=0
5000 GOTO200
10000 PRINT#1
10001 CLOSE1

READY.
```

BELLOWS OPTIMIZATION, CONTINUED, 4-24-84, J. OSBORNE.
A1= 2.11 L1= .885
A2= 4.89 L2= 1.1
INITIAL VOLUME, V0= 6.77745693 CUBIC INCHES
COUNTERWEIGHT= 25.8 LBS
R1= 8.46 IN. R2= 16.46 IN.

TH DEG	AL DEG	DISPLACEMENT, IN.	SLAVE FORCE ACTUAL LB	IDEAL LB	ERROR LB
-2	.3	.05	1.16	.96	.191
3.4	.6	.1	2.41	2.41	-1E-03
9.1	1	.15	3.76	3.92	-.164
15.2	1.3	.2	5.23	5.52	-.292
22	1.6	.25	6.82	7.2	-.377
29.7	2	.3	8.56	8.97	-.409
38.6	2.3	.35	10.47	10.85	-.374
50	2.7	.4	12.58	12.83	-.25
67.5	3	.45	14.91	14.89	.016
70.3	3	.455	15.15	15.1	.058
73.8	3.1	.46	15.4	15.3	.106
78.7	3.1	.465	15.66	15.49	.168
79.4	3.1	.4655	15.68	15.5	.176
80.1	3.1	.466	15.71	15.52	.185
81	3.1	.4665	15.73	15.54	.194
82.1	3.1	.467	15.76	15.55	.205
83.5	3.1	.4675	15.78	15.56	.219

W= 25.8 LBS

BELLOWS OPTIMIZATION, CONTINUED, 4-24-84, J. OSBORNE.
A1= 2.11 L1= .885
A2= 4.89 L2= 1.1
INITIAL VOLUME, V0= 6.77745693 CUBIC INCHES
COUNTERWEIGHT= 26.8 LBS
R1= 8.46 IN. R2= 16.46 IN.

TH DEG	AL DEG	DISPLACEMENT, IN.	SLAVE FORCE ACTUAL LB	SLAVE FORCE IDEAL LB	ERROR LB
-2.2	.3	.05	1.16	.92	.239
3	.6	.1	2.41	2.31	.097
8.5	1	.15	3.76	3.77	-.013
14.4	1.3	.2	5.23	5.31	-.084
20.9	1.6	.25	6.82	6.93	-.11
28.2	2	.3	8.56	8.64	-.08
36.7	2.3	.35	10.47	10.45	.019
47.1	2.7	.4	12.58	12.37	.209
61.9	3	.45	14.91	14.38	.529
64	3	.455	15.15	14.58	.572
66.2	3.1	.46	15.4	14.78	.618
68.8	3.1	.465	15.66	14.99	.668
71.8	3.1	.47	15.91	15.19	.724
75.6	3.2	.475	16.17	15.38	.788
82	3.2	.48	16.43	15.55	.877
83.4	3.2	.4805	16.45	15.56	.892

W= 26.8 LBS

TMT PRIMARY MIRROR COVERS

Jack Osborne, October 1984

PURPOSE:

1. The covers should substantially reduce dust getting on the primary mirror when the telescope is not in use.
2. The covers should not degrade telescope performance while open.
3. The covers should offer a measure of protection from falling objects when closed.
4. The covers won't do the following:
   a. allow walking.
   b. allow the primary mirror to be stopped (i.e. they don't function as an iris)
   c. provide a tight thermal package around the primary mirror and cell.
   d. capture a great amount of water.
5. Further, the covers should be lightweight, simple and inexpensive.
6. Also, the cover system should include a mechanism for maintaining correct tube balance at all times.
7. The covers must be closeable during a power failure.
8. The covers will be closed or opened only when the telescope is at the zenith, and will retain their integrity as the telescope is moved to the horizon.

CONCLUSIONS:

We have invented a primary mirror cover system which will be described in this report. It meets all the above requirements. This report details the features of the system, points out areas where further work might be done, and briefly mentions alternative cover systems.

List of Figures:

1. Cover shown partially open (star view).
2. Cover shown partially open (side view).
2b. Cover shown partially open (oblique view).
3. Cover shown fully open (side view).
4. Cover shown fully open showing wind and thermal obstructions.
5. Cover shown fully closed (side view).
6. Latching mechanism.
7a. Guide rails.
7b. Guide rails, details
8. Hinge details.
9. Guide rail loads vs. cover extension.

10. Panel weights and link loads.
12. Alternate cover system.

GENERAL DESCRIPTION:

The primary mirror cover is made up of six panels. Each panel is a parallelogram linkage with a thin skin of aluminum on its top surface. See Figures 1,2,&2B. The folding panels will be stowed between the elevation ring and the mirror cell. Figures 3,4. The packaging is rather tight and part of the folded panel remains inside the elevation ring at the two Nasmyth sides of the elevation ring. This is 4 cm from a 1 degree off-axis light beam. The covers are driven by air motors. One motor drives each of the six panels (six motors). This air system must have enough stored compressed air to close the cover during a power failure. Each motor drives a shaft located just above the mirror cell. This shaft extends across the base of the panel and drives all six guide rail rollers. Figure 7. The middle two guide rail slides control the moving ends of 7-bar linkages. The next two guide rail slides (one on either side of the two central ones) control the moving ends of 5-bar linkages. The outer two guide rail slides control the moving ends of 3-bar linkages. The motion of all six panels can be simultaneous since no overlap at the joints is intended. Opening or closing time will be on the order of 1 minute. Once closed, twelve latches on the Cassegrain tower will engage the extended cover (Figure 6). Suitable limit switches and control logic will be provided. The latches will be pneumatic since power-failure mode is required. The opening and closing sequences are done while the telescope is pointed at the zenith. This reduces the loads on the drive components. Once latched to the central tower, the telescope may be moved to the horizon for storage or whatever. The mirror covers are shown with the top pivot of the parallelogram moving down (and hence the center of gravity also moves down) as the covers are closed. See Figure 5. An inversion of this scheme is possible, that is the top pivot can be fixed and the lower pivot can be moved up to close the covers. There may be thermal or other reasons for doing this. The design is flexible in this area. Each panel of the cover will weigh 330 lbs and the required guide rails and motors will weigh 500 lbs. There are six sets, so the total weight of the cover system is 5000 lbs. The moving weight 2000 lbs moves 20 inches and so a counter-moving weight must be provided. If this weight is mounted on the elevation ring, where 10 feet of travel is available, then the weight is 330lbs.(3300 ft-lb)

The covers never come closer than 2.5 cm to the primary mirror segments. The nominal skin thickness is .06 in (1.5 mm). A very crude drop test was done on a 30"x30" panel on a raised wooden support. A 12" crescent wrench was dropped 50 feet and did not penetrate this panel (12" crescent wrenches weigh 2 lbs). The kinetic energy was converted to accoustical energy and a slight puncture in the aluminum panel . This material was 6061-T6, a fairly brittle aluminum alloy. Further study should be done to select a softer material. Also, since the weight of the skin is much of the cover weight, and dictates the loads on the guide rail supports, a thinner skin might be investigated. .06" was picked arbitrarily. Figure 10 lists weights.

The panels use piano hinge strips for the top and middle pivots for the linkages. These have air gaps (dust gaps) and since the panels fold up when opening, provision has been made to catch any dust falling down into the three valleys. See Figure 8. These channels must be periodically cleaned.

An alternate geometry for mirror covers was proposed in July 1980 (TMT Report 39) which had 6 panels hinged inside the top of the elevation ring. See Figure 12. It is now felt that in the open position, these will have adverse effects on the telescope due to wind loading. The covers stowed just above the primary mirror cell (and only 1/3 as long when folded up) minimize this effect although they are considerably more complicated structures. Figure 4 illustrates this.

Water: The geometry of this cover design has a flaw. That is, if the joints are all water tight and the covers get rained on, the valleys will fill up with water. This added weight will damage the structure. Therefore, the covers will not protect the mirrors from rain. The mirrors will be protected from a light fog and occasional oil or cryogens falling from above. If rain water protection is essential, then the telescope should be stored horizontally when not in use.

Future work: No attempt was made to optimize the linkage elements, but instead, larger members were selected for this first design. The elements of the parallelogram are statically indeterminate. If possible, an analysis should be made to set the size of the members and the pivots. As mentioned above, the upper skin might be optimized, both material and thickness. The guide rail supports are nominally 4" x 4" built-up "I" beams. This gives a section moment of about 19 in_4. The central 7-bar linkage imposes a 300 lb lateral load to the guide rail and the deflection of the rail is .05" (.5" at the latch) The upper attachment is a bolted bracket to the underside of the elevation ring and the lower attachment is a sliding joint, see Figures 7a, 7b. These elements are a large fraction of the mirror cover system and could stand some optimizing.

Interface: The vertical elements which make up the guide rails must support lateral forces at mid span and must resist bending. The upper connection to the elevation ring can be rigid but the connection at the bottom, to the mirror cell can be a sliding joint. That is, it won't attach to the mirror cell so that loads are transmitted from the cell to the ring in a direction parallel to the optical axis of the telescope tube. They will however, transfer some of the weight of the covers directly to the mirror cell, when looking at the horizon. The structural engineers should be aware of this, since this affects optical collimation. About 3000 lbs is now added to the top set of mirror cell nodes when the telescope is horizontal.

Dynamics: While the covers are folded up during observing, there may be unwanted natural frequencies (rattles) present. It may be necessary to provide a device or system to force the covers closed against rubber stops. This is easier to handle as a retro-fit than to predict what the resonances will be before hand.

Re-aluminizing: Since the covers don't overlap each other or "nest" when closed, it may be possible to open only one panel while mirrors are removed and replaced for re-aluminizing, thus offering some protection for the other mirrors.

Cassegrain tower: It is not clear whether there will be a permanent structure outside of the tower. This awaits the design of the Cassegrain baffle design. If it is decided to have some permanent stuff outside the tower (partially shadowing the inner ring of segments) then the design of the covers becomes simpler. That is, the furthest link of the parallelogram gets shorter and stowing becomes more compact.

Optimizing the linkage: Figure 9 shows how the parallelogram linkage angle affects the guide rail loads and the cover extension. Guide rail loads (lateral) are plotted versus extension. 20 degrees (40 degrees between 2 links) was selected because the loads get very much higher for very little increase in extension. This value then determined the extra extension added to the last panel in the linkage. This extra part is what is stowed inside the elevation ring (only at the elevation axis flats).

See Disc I12 mirror cover