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FINAL REPORT ON DESIGN DOCUMENTATION AND TEST HARDWARE FOR AN ENGINEERING MODEL OF A SERPENTUATOR

CONTRACT NAS8-30166

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FINAL REPORT ON DESIGN DOCUMENTATION AND TEST HARDWARE FOR AN ENGINEERING MODEL OF A SERPENTUATOR

CONTRACT NAS8-30166

Prepared by:

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INTRODUCTION

Under National Aeronautics and Space Administration, George C. Marshall Space Flight Center, Alabama, Contract Number NAS8-30166, and under the direction of the M&E Laboratory, Astro-Space Laboratories, Inc., (ASL) designed, fabricated, and tested a Serpentuator for intervehicular performance.

The Serpentuator so designed, fabricated, and tested is an engineering model of controllable, mechanical linkage to be used to determine the feasibility of using such a device for: 1) Material handling with the Orbital Workshop, and, 2) as an extension of the forty (40) foot ATM Serpentuator to locate tip sensors for inspection of the exterior of the OWS cluster, or to position earth looking or cluster looking experiments.

The unique linkage used in this engineering model Serpentuator was developed by ASL prior to the initiation of this contract. This development was due to the obvious need for a foldable, movable (plus or minus 180°) linkage for intervehicular Serpentuator tasks.

The rather complex mechanical linkage can best be described as an "angle multiplier" arrangement by which the input angle applied to the first member of the linkage is multiplied by a factor of approximately six (6) through the linkage.

Since the input and output work must be the same, (less friction) the penalty paid for the "angle multiplier" effect is the necessity of applying quite high forces to the input linkage. The load carrying structural members were stressed, and, where necessary, high strength steel or aluminum alloy was selected.

Shortly after design approval, the projected fabrication and assembly costs were estimated from approved shop drawings and found to exceed the value of the above cost-plus-fixed fee contract. The decision was made to reduce the hardware to stay within the original contract value.

Briefly, the number of links was reduced from four to two, and the redundant sealing inside the drive mechanism was eliminated. When building the two-link version, care was taken to make the necessary provisions to allow the system to be expanded to the original four link version, or even more, at a later date.

GENERAL DESCRIPTION OF SERPENTUATOR

Presented below are pertinent performance and design criteria as stipulated in Attachment A to Contract NAS8-30166.

1. Hinge Movement - ± 180°
2. Length, tip-to-base - 18 feet
3. Length, tip to Link #1 - 15 feet
4. Tip Force, max. - 10 lb
5. Number of Links - 4
6. Internal Pressure, max. - 10 psig
7. Specific Gravity of System - 1
8. Minimum Static Load Safety Factor - 3

The unique hinge design, as shown on Astro-Space Drawing #500253, can be described as one converting an input of low angular movement - high forces into an output of high angular movement - low forces.

Therefore, to attain the output performance set forth in the preceding specification list, while restricting the diametrical size would be a formidable design problem.

In order to select the worst case approach, as far as stress is concerned, the link nearest the base was analyzed throughout as having sufficient power capability to move a 10 pound tip force on a 15 foot lever arm, or to sustain an externally applied force of 10 pounds at the tip.

The concept of attaining ± 180° maximum mobility in each hinge joint necessitated placing the main lever hinge points (see Dwg. #500253 again) on the centerline. This is advantageous from the standpoint of mobility and immunity to internal pressurization, but it is a disadvantage from the leverage standpoint. For instance; the lever arm for the initial force imparted to the hinge joint is restricted to 1.375 inches, by the size of the tubing as is shown on ASL Drawing #500254, Find Numbers 49 and 46. This leverage situation is applicable to all the individual hinge joint members.

The net output torque generated by the leverage is

$$T_o = (15 \text{ ft})(10 \text{ lb}) = 150 \text{ ft-lbs } (1800 \text{ in-lb})$$

If the linkage being moved and the attached Serpentuator were a rigid member, the input force, based on a simple ratio of the lever arms is:

$$F_{in} = 1800/1.375 = 1309 \text{ lbs}$$

However, the ratio of forces through the linkage is reduced by the same amount as the angular movement increases. The ratio of input-to-output angles averages out to be approximately 6. This will be discussed in more detail later in this report.

Therefore, for the maximum input-to-output angle ratio

1 to 6.5, the force must be increased 6.5 to 1, making F_{in} = 8508.5 lbs. Assuming 20% must be added for frictional losses in the hinge bushings, the total gross input force must be

$$F_{in_{tot}} = (8508.5)(1.2) = 10{,}210 \text{ lbs}$$

The actual force transmitted along the axis of the ball screw must be slightly greater due to the output force being transmitted through an angle of approximately 8° with the ball screw axis. The total force which must be generated along the axis of the ball screw to yield an horizontal component of 10,210 pounds is:

$$F_1 = 10{,}210/\cos 8°$$

$$F_1 = 10{,}300 \text{ lb}$$

This is the load assumed for drive train stress calculations and sizing of other drive train components. This angular correction was not made in the analysis included in the First Monthly Progress Report.

The input torque to the ball screw to achieve an output of F_1 is

$$T_{in} = \frac{(\text{Lead})(\text{Load})}{2\pi\ e}$$

where

Lead = Screw lead, 0.2 in

Load = F_1, 10,300 lb

e = Screw efficiency, 90%

then,

$$T_{in} = \frac{(0.2)(10,300)}{(6.28)(0.90)} = 365 \text{ in-lb}$$

From Drawing #500254, it can be seen that the ball screw input torque is first passed through a 200 to 1 harmonic drive, so that T_{in} is 365/200 = 1.82 in-lb. The total input torque,

$$T_{in_{tot}} = (1.82)(e_{B.S.})(16)$$

$$= 40.9 \text{ in-oz}$$

The power source is an Inland torque motor #NT2173 capable of 54 in-oz torque at 12VDC.

STRESS ANALYSIS

A typical Serpentuator link is divided into the drive train section (Dwg. #500254), and the hinge section (Dwg. #500253). The stress analysis will cover these two areas in the above order.

Drive Train

The method of loading on the structural members of the drive train section deserves some explanation.

When the screw is rotated clockwise when looking forward along the Serpentuator, the maximum load is transmitted to the Collar (Dwg. #301559), then to the roller thrust bearing, then to the Housing (Dwg. #301558), then to the six No. 8-32 screws (Find No. 20), then to the Ball Screw Housing (Dwg. #400981), and finally to the Sleeve (Dwg. #400985), which transmits the load back to the preceding link. It should be noted here, that neither the thin outer tube (Dwg. #400984) nor the inner tube (Dwg. #400985) contribute to the load carrying capability of the drive train unit. The primary function of these tubes is to effect a seal.

Initially, it was anticipated that the Housing (Dwg. #400981) would need to be made of some type of steel to utilize the best possible material strength. However, the following stress analysis of this part has indicated that a high strength aluminum alloy, 2219, would be strong enough without paying the weight penalty of steel.

The properties of 2219, heat treated to condition H87, are presented by the following from the Alcoa Handbook.

Tensile Ultimate (ksi)	- 68
Tensile Yield (ksi)	- 56
Elongation - % in 2"	- 10
Compressive Yield (ksi)	- 59
Shear Ultimate (ksi)	- 40
Brinell Hardness	
500 Kg Load, 10mm Ball	- 126
Bearing Strength	
Ultimate (ksi), e/D = 1.5	- 106
Ultimate (ksi), e/D = 2.0	- 134
Bearing Strength	
Yield (ksi), e/D = 1.5	- 91
Yield (ksi), e/D = 2.0	- 106
Endurance Limit (Rotating Beam Fatigue Tests, R = 1.0)	
ksi at 500×10^6 cycles	- 15
Modulus of Elasticity x 10^6, psi	- 10.6

In order to get an approximation of the cross sectional area of the housing in the central area, Section A-A, Page 2 of Dwg. #400981, will be used as a basis of computation, and is illustrated by Figure 1.

Approximating the areas 1, 2, and 3 with rectangles,

$$A_1 = (0.78 + 0.62)(0.15)(2) = 0.42 \text{ in}^2$$

$$X_1 = + 0.16$$

$$A_2 = (1.4)(0.25)(2) = 0.70 \text{ in}^2$$

$$X_2 = - 1.20$$

$$A_3 = (1.95)(0.24) = 0.468 \text{ in}^2$$

$$X_3 = - 1.74$$

$$\bar{X} = \frac{A_1X_1 + A_2X_2 + A_3X_3}{A_1 + A_2 + A_3}$$

$$= \frac{(0.42)(+0.16) + (0.70)(-1.20) + (0.468)(-1.74)}{0.42 + 0.70 + 0.468}$$

$$\bar{X} = -1.00 \text{ inch}$$

To insure a conservative approach, it is assumed that A_1 takes all the ball screw load. Since the value of A_1 is somewhat critical, it will be computed more exactly. To expedite the area calculation, it is assumed that the vertical chord 1.94 inches from the vertical centerline is extended until it intersects the 2.110 radius, thus leaving A_1 as a circular segment.

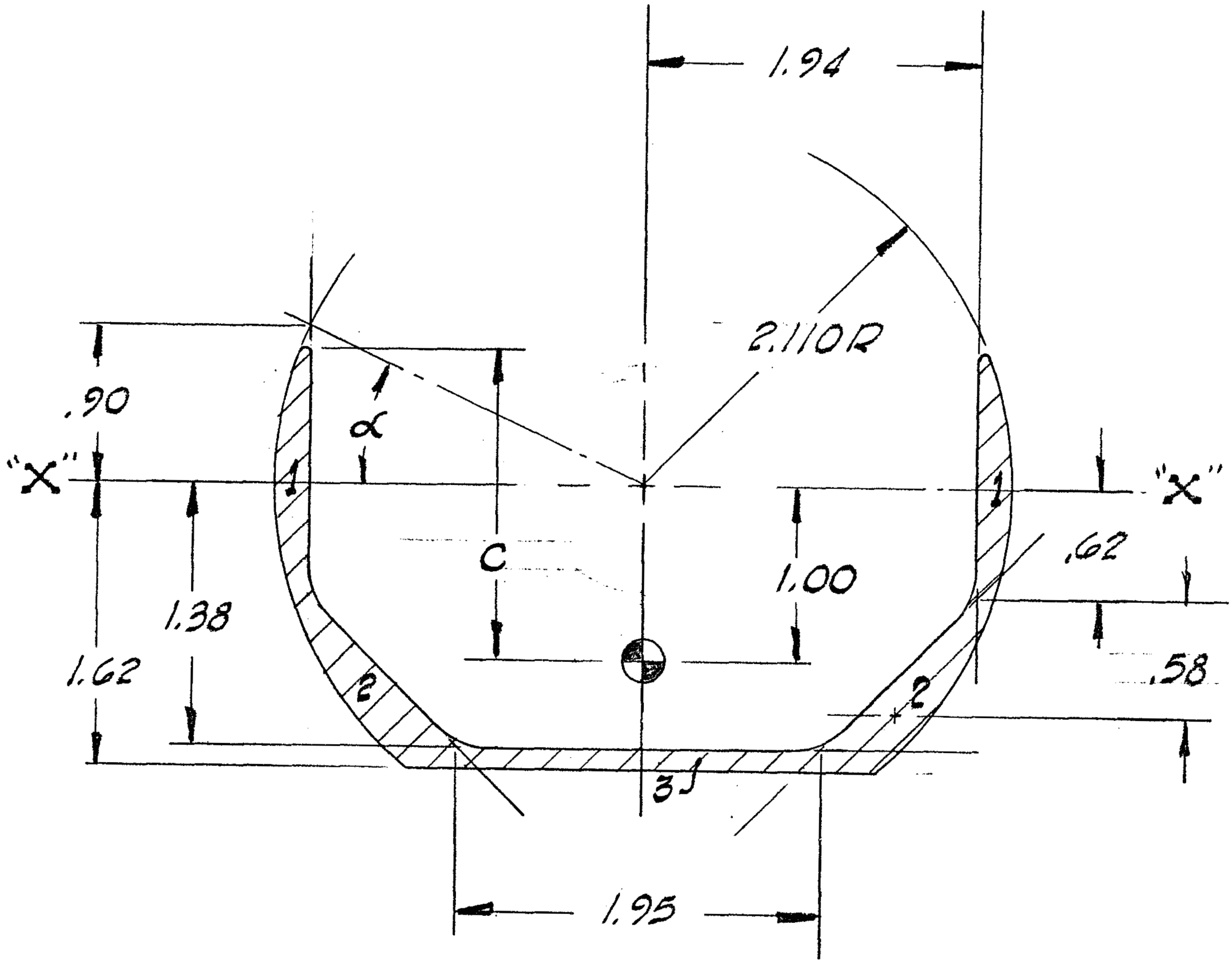

Figure 1. - Cross-Sectional Area of Housing

$$\alpha = \tan^{-1} 0.90/1.94 = 24°54'$$

$$\theta = 2\alpha = 49°48'$$

$$A_1 = \frac{\pi R^2 \theta}{360} - \frac{R^2 \sin\theta}{2}$$

$$A_1 = 0.468 \text{ in}^2 \quad ,$$

then

$$S = F/A_1$$

$$S_t = 10{,}300/0.468 = 22{,}000 \text{ psi}$$

The tensile yield strength of the material is 56,000 psi; therefore, the stress in A_1 is acceptable.

The deformation can be computed as a simple ratio since the elastic limit of the material is not approached.

$$\delta = S_t/E$$

where

δ is the deflection, inch

S_t = tensile stress, psi

E = modulus of elasticity, psi

$$\delta = \frac{22{,}000}{10.9 \times 10^6} = 0.0023 \text{ inch}$$

To compute the bending stress, the moments of inertia of each of the areas must be computed around the C.G.

$$I_1 = 2R^4 \left[\frac{1}{8}(2\alpha - \sin 2\alpha) - \frac{1}{12} \frac{(2\alpha - \sin 2\alpha)\sin 3\alpha \cos\alpha}{\alpha - \sin\alpha\cos\alpha} \right] + Ad^2$$

$$I_1 = 0.135 \text{ in}^4 + (0.468)(1.16)^2 = 0.765 \text{ in}^4$$

$$I_3 = \frac{1}{12}bh^3 + Ad^2$$

$$= (1.95)(0.24)^3 + (0.42)(0.74)^2$$

$$= 0.257 \text{ in}^4$$

The A_2 is not considered in the bending stress calculation.

$$S = \frac{Mc}{I}$$

where

S = bending stress, psi

c = distance from neutral axis to point of maximum fiber stress, in.

I = total moment of areas 1 and 2

$$S_B = \frac{(10{,}300)(1.00)(1.78)}{0.765 + 0.257}$$

$$S_B = 17{,}939 \text{ psi}$$

The next areas to be checked are Points B and C as identified in Figure 2. The shaft bearing surface at Point C is quite long (2.312 inches) so that the bearing stress even for the entire 10,200 lbs is:

$$S_{B_C} \cong 10{,}200/(0.5)(2.312)(2) = 4{,}400 \text{ psi}$$

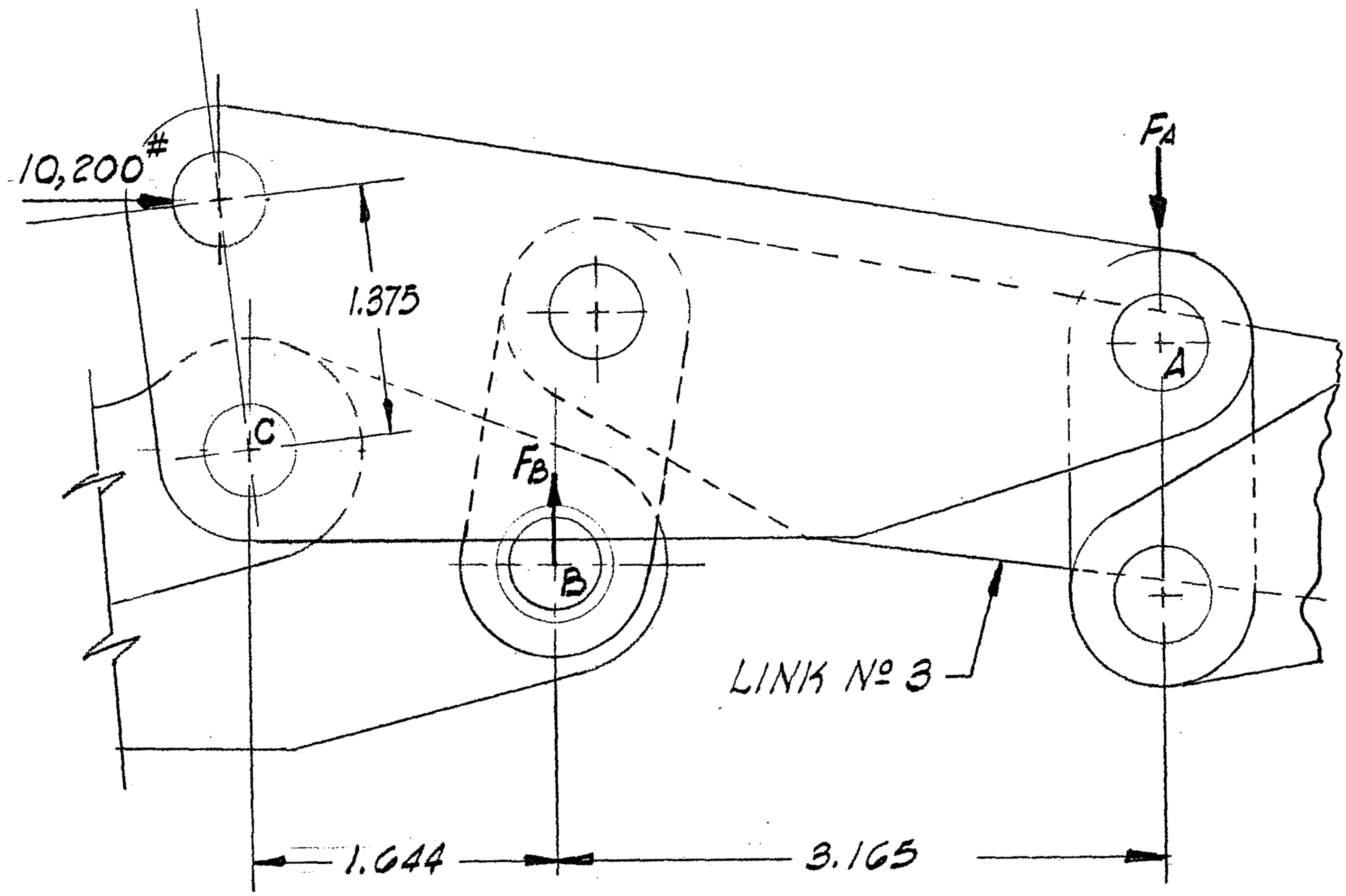

Figure 2. - Moment of Areas

The linkage, due to its unique construction, was not completely analyzed statically for every load and reaction under this contract. Most of the members are statically indeterminate, and would have required a disproportionally large number of the engineering hours to solve. More will be mentioned later; however, the reaction force at Point B on the housing is not known. By neglecting the interactions beyond Point A, the force F_A can be approximated by ratioing the applicable lever arms:

$$F_A = 10{,}200\left(\frac{1.375}{3.165}\right) = 4{,}431 \text{ lb}$$

The estimated load at Point B (neglecting the reaction at C) is 4,431 lb.

The bearing stress at B,

$$S_{B_B} = 4431/(0.75)(0.5)(2) = 5{,}908 \text{ psi}$$

The compressive stress on the hole is:

$$S_{c_B} = 4431/(0.625)(0.75) = 9{,}453 \text{ psi}$$

The thrust bearing transmits the ball screw generated axial force directly into the housing, as shown by Figure 3, thus creating a compressive stress:

$$A_c = 0.785(1.35^2 - 1.125^2) = 0.437 \text{ in}^2$$

$$S_c = 10{,}300/0.437 = 23{,}570 \text{ psi}$$

Compressive yield strength of 2219 aluminum is 59,000 psi.

The entire Housing (Dwg. #400981) is tied to the preceding

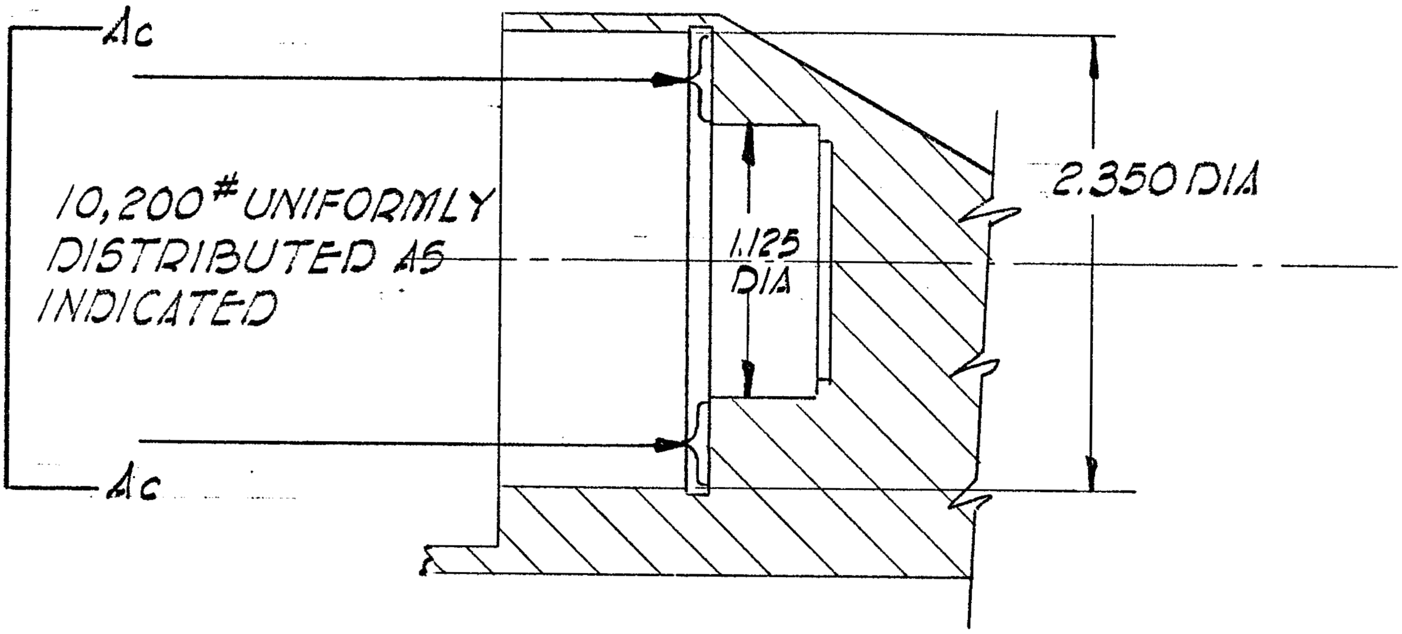

Figure 3. - Ball Screw Axial Force

load carrying member by means of a row of six No. 8 screws on a 3.810 dia. The aluminum housing is tapped to accommodate helicoils for additional strength.

A tip moment of 1800 in-lbs must be converted to an axial force on the screws by ratioing the tip-to-base length by one-half the bolt circle diameter.

$$F = 10 \text{ lbs } \left(\frac{180}{1.905}\right) = 945 \text{ lbs}$$

The single thread shear area is:

$$S_n = 3.1416nL_eD_s\min\left[\frac{1}{2n} + 0.57735(D_s\min - E_n\max)\right] ^{①} \qquad \text{Eq. 1}$$

where

n = number of threads per inch

L_e = length of engagement

$K_n\max$ = maximum minor diameter of internal thread

$E_s\min$ = minimum pitch diameter of external thread

$D_s\min$ = minimum major diameter of external thread

$E_n\max$ = maximum pitch diameter of internal thread

Solving Eq. 1:

$$S_n = \pi(32)(0.25)(0.1571)\left[\frac{1}{2(32)} + 0.577(0.1571-0.1475)\right]$$

$$= 0.0833 \text{ in}^2$$

If one bolt must carry all the load, (actually two to three are always loaded), the thread stress is:

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① Screw-Thread Standards for Federal Services, 1957 - U.S. Department of Commerce.

$$S_s = F/S_n = 945/0.0833 \text{ in}^2$$

$$S_s = 11{,}343 \text{ psi} \quad .$$

Thus, one thread is sufficient to carry the load. Now it is desired to compute the bolt cross-sectional strength in tension.

The thread root area of one bolt is:

$$A_R = \frac{\pi}{4}(0.130)^2 = 0.0133 \text{ in}^2$$

The tensile stress on one bolt is:

$$S_T = 945/0.0133$$

$$S_T = 71{,}052 \text{ psi}$$

Throughout, the bolts are "Nylock" type with a minimum tensile strength of 170,000 psi; therefore, a single bolt could carry the maximum system loads.

Ball Screw (Dwg. #301568)

The following parameters describe, in part, the ball screw:

Lead	- 0.20 in.
Pitch dia.	- 1.25 in.
Root dia.	- 1.12 in.
Length between bearing supports	- 9.50 in.

The push rod connecting the ball nut to the first linkage point is inclined at an angle of approximately 9°43' to the Serpentuator centerline, thus causing a normal force, F_n, of

$$F_n = 10{,}300\ (\sin 9°43')$$
$$= 1{,}738\ \text{lb}$$

to be applied to the ball screw. If it is assumed that this bending load is equally distributed across the ball nut, as shown in Figure 4, the simply supported screw deflection can then be calculated as:

$$y = \frac{1}{48EI}\left\{8R_1(X^3 - \ell^2 X) + W_X\left[\frac{8d^3}{\ell} - \frac{2bc^2}{\ell} + \frac{c^3}{\ell} + 2c^2\right] - \frac{2W(X-a)^4}{c}\right\}$$

Solution of the above for values in Figure 4 yields:

$$y = 0.0099\ \text{inch}$$

This is a fairly large deflection, which could cause binding in the ball screw; therefore, the manufacturer, Scully-Jones (Chicago), was contacted. After their engineering department went over the ball screw application, they stated that normal ball screw operation could be expected, even with the rather large side load and resulting deflections.

The axial load on the ball screw is taken in compression on a shoulder at each end between the 1.12 inch root diameter of the ball grooves and the 0.875 inch diameter. In computing the compressive stress S_s on these shoulders,

$$A_s = \frac{\pi}{4}\ (1.12^2 - 0.875^2)$$

$$A_s = 0.822\ \text{in}^2$$

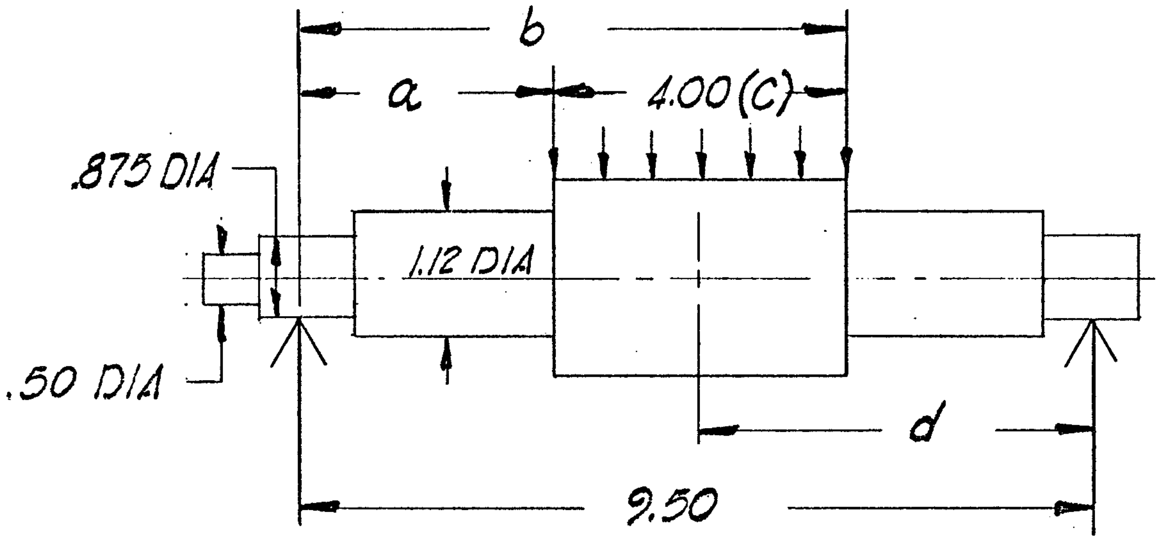


Figure 4. - Bending Load Distribution

$$S_s = 10,300/0.822\ in^2$$

$$S_s = 12,530\ psi$$

The ball screw is hardened steel with a compressive stress of approximately 175,000 psi.

The keyway on the ball screw locking the screw to the harmonic drive output is 0.50 inch long and 0.125 inch wide, thus creating a shear area:

$$A_s = (0.50)(0.125)$$

$$A_s = 0.0625\ in^2$$

The input torque to the ball screw is 382 in-lb, and converting to a tangential force on the key: $F_T = 382/0.25 =$ 1528 lb. The shear stress in the key is:

$$S_s = 1528/0.0625\ in^2$$

$$S_s = 24,448\ psi$$

This stress level is acceptable since the shear strength of 303 stainless is approximately 60,000 psi.

The ball screw capacity is 17,000 pounds static load and 11,000 pounds dynamic load with six turns of the balls. However, the Serpentuator screws are specially designed with seven turns to provide additional capacity.

<u>Link (Dwg. #400983)</u>

The push-pull link transfers the linear motion of the ball screw to the initial member in the multi-lever hinge joint. A side view and a sectional view of this link are shown by Figure 5.

Since the load is applied along a line between the two bearing diameters, the main concern with this part is the stress in the "backbone" of the part. In order to determine this, the C.G., cross-sectional area, and moment of inertia must be determined.

The centroid distance, y_1, for the outer radius in Figure 5 is:

$$y_{1_2} = R_2(1 - \frac{2 \sin \alpha}{3\alpha}) = 0.897 \text{ inch}$$

$$y_{1_3} = R_3(1 - \frac{2 \sin \alpha}{3\alpha}) + 0.38 = 1.107 \text{ inch}$$

$$y_1 = \frac{y_{1_2} + y_{1_3}}{2} = 1.002 \text{ inch}$$

The cross sectional area is $A = 2\alpha\ R_1 t$ where t is the thickness.

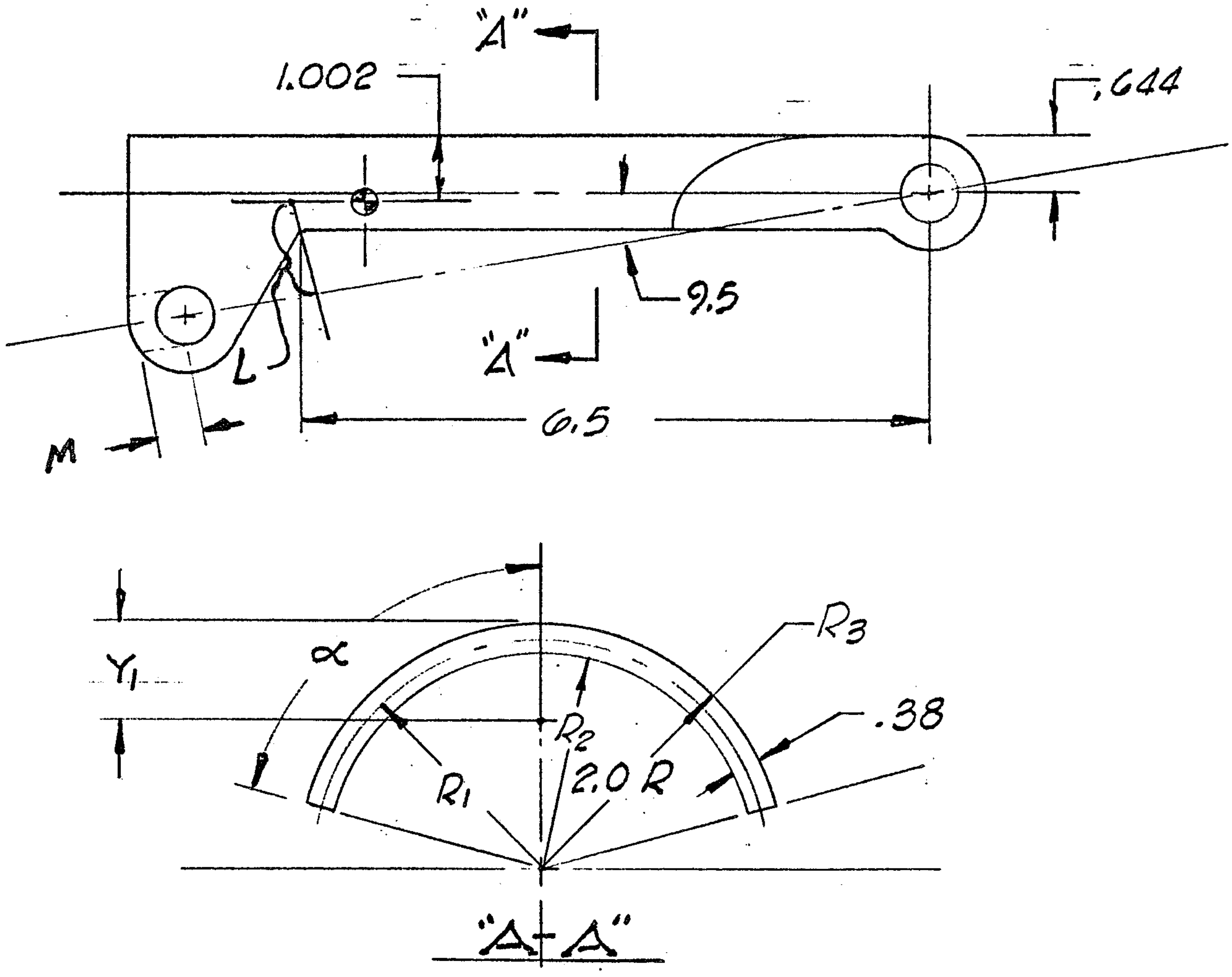
$$A = 1.44 \text{ in}^2$$

The moment of inertia is:

$$\begin{aligned} I_1 &= I_{R_2} - I_{R_3} \\ &= 1/4\ (R_2^4 - R_3^4)\left[\alpha + \sin\alpha\cos\alpha - \frac{16\sin^2\alpha}{9\alpha}\right] \\ &= 0.471 \text{ in}^4 \end{aligned}$$

The part is under combined loading so that a typical section (A-A in Figure 5) is subjected to compressive (or tensile) loads in addition to a bending moment.

Because of the fairly large cross-sectional area, the compressive stress is low; i.e., 7,000 psi.



R_1 = 1.80 IN.
R_2 = 1.62 IN.
R_3 = 2.0 IN.

Figure 5. - Side &Sectional View, Push-Pull Link

To compute the bending stress about the C.G., located 6.5 inches from the right hand side (see Figure 5), the moment with a force of 10,300 cos 9.5° and a lever arm, L, must first be determined, where

$$L = \frac{6.5 \sin 9.5° - (1.002-0.644)}{\cos 9.5°}$$

$$= 0.725 \text{ inch}$$

Then the bending stress is

$$S_B = \frac{Mc}{I} = \frac{(10{,}300 \cos 9.5)(0.725)(1.002)}{0.471} \qquad c = y_1$$

$$S_B = 15{,}663 \text{ psi.}$$

Next, the combined stress in the small end of the cradle will be computed (see Fig. 6):

$$I = \frac{1}{12} bh^3 = \frac{1}{12}(0.75)(1)^3$$

$$I = 0.0625 \text{ in}^4$$

$$c = 0.50 \text{ inch}$$

$$A = (0.75)(1.00) = 0.75 \text{ in}^2$$

$$F_c = 10{,}157 \text{ lb} = \frac{F_c}{A} + \frac{Mc}{I}$$

$$M = 10{,}157(L)$$

$$L = 0.81 \sin 9.5° + (0.500 - 0.356)$$

$$= 0.278 \text{ inch}$$

$$= \frac{10{,}157}{0.75} + \frac{(0.271)(10{,}157)(0.50)}{0.0625}$$

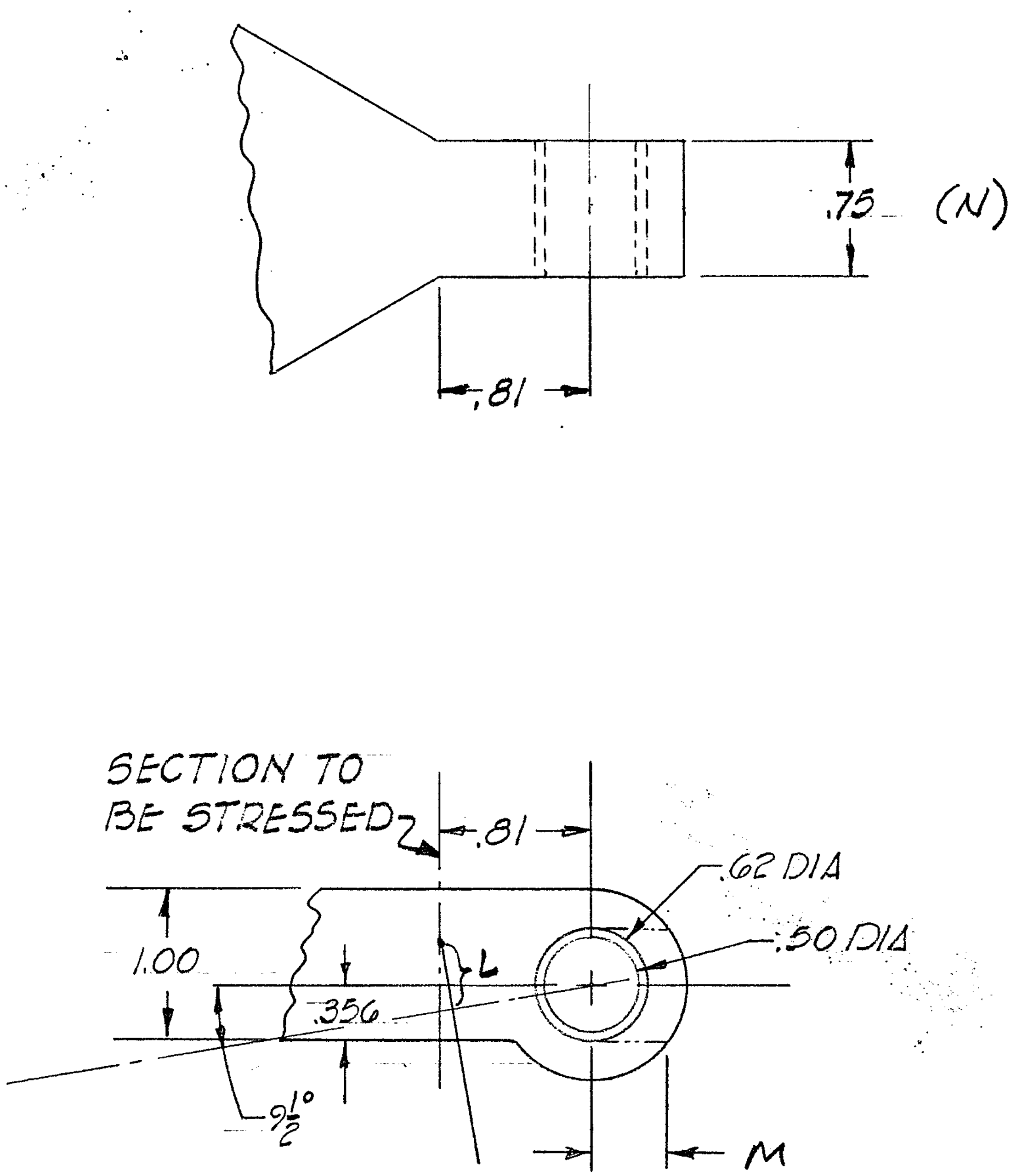


Figure 6. - Bearing Stress Area

$$S = 35{,}740 \text{ psi}$$

The cradle, like the ball screw housing, is made of 2219 aluminum with a minimum yield of 56,000 in Condition H87, therefore, the stress level is acceptable.

The loads on the ball screw end must be checked for bearing stress in the metal outside the hole and compressive stress in the 0.625 diameter hole itself. There are two identical lugs on the link that straddle the ballnut.

The bearing area (see Figure 6 and Dwg. #400983) is:

$$A_B = 2MN$$

$$A_B = (0.625)(0.625)(2)(2)$$

$$= 1.560 \text{ in}^2$$

where

M = bearing length, and

N = bearing thickness

$$S_B = 10{,}200/1.560 \text{ in}^2$$

$$S_B = 6{,}538 \text{ psi}$$

In compression, the compressive area being considered is the diameter of the hole times its length:

$$A_c = (0.656)(2)(0.500)$$

$$A_c = 0.656 \text{ in}^2$$

The compressive stress in the hole is then

$$S_c = 10,200/0.656$$

$$S_c = 15,549 \text{ psi}$$

On the hinge end, the same stress considerations apply; however, only one larger support is involved. Referring to Figure 6 and Dwg. #400983, the bearing stress area is:

$$A_B = 2MN$$

$$A_B = (2)(0.40)(0.75)$$

$$A_B = 0.600 \text{ in}^2$$

$$S_B = 10,200/0.60$$

$$S_B = 17,000 \text{ psi}$$

The compressive stress area is:

$$A_c = (0.625)(0.75)$$

$$A_c = 0.469 \text{ in}^2$$

$$S_c = 10,200/0.469 \text{ in}^2$$

$$S_c = 21,748 \text{ psi}$$

Collar (Dwg. #301559)

The stress on the collar is very important to the life of the drive train because, in effect, it transfers the large axial loads from the ball screw to the thrust bearings at each end. Ideally, the 0.25 inch thick flange on the collar will remain perfectly vertical under loading, so that the loads will be uniformly distributed in a radial direction on the thrust bearing roller. In reality, however, the method of loading as shown in Figure 7(a) indicates that there will be deflections, the magnitude of which can be calculated employing flat plate equations.

Assuming that the load was equally distributed in a radial and circumferential direction on the thrust bearing, Case 16 on Page 221 of Roark's Stress and Strain, Fourth Edition, McGraw-Hill, New York, 1965 most nearly fits the application. This loading is depicted in Figure 7(a).

The maximum deflection occurs at the outer edge and is given by

$$\text{Max } y = \frac{3w(m-1)}{16Em^2t^3}\Bigg[a^4(7m+3)+b^4(5m+1) - a^2b^2(12m+4) - \frac{4a^2b^2(3m+1)(m+1)}{(m-1)} \log\frac{a}{b} + \frac{16a^4b^2(m+1)^2}{(a^2-b^2)(m-1)}\left(\log\frac{a}{b}\right)^2\Bigg] \qquad \text{Eq. 3}$$

where

$w = W/\pi(a^2-b^2)$

a = inner radius, 0.875/2 = 0.4375 inch

b = outer radius, 2.23/2 = 1.115 inch

t = thickness, 0.25 inch

m = reciprocal of Poissoins ratio, 3.704

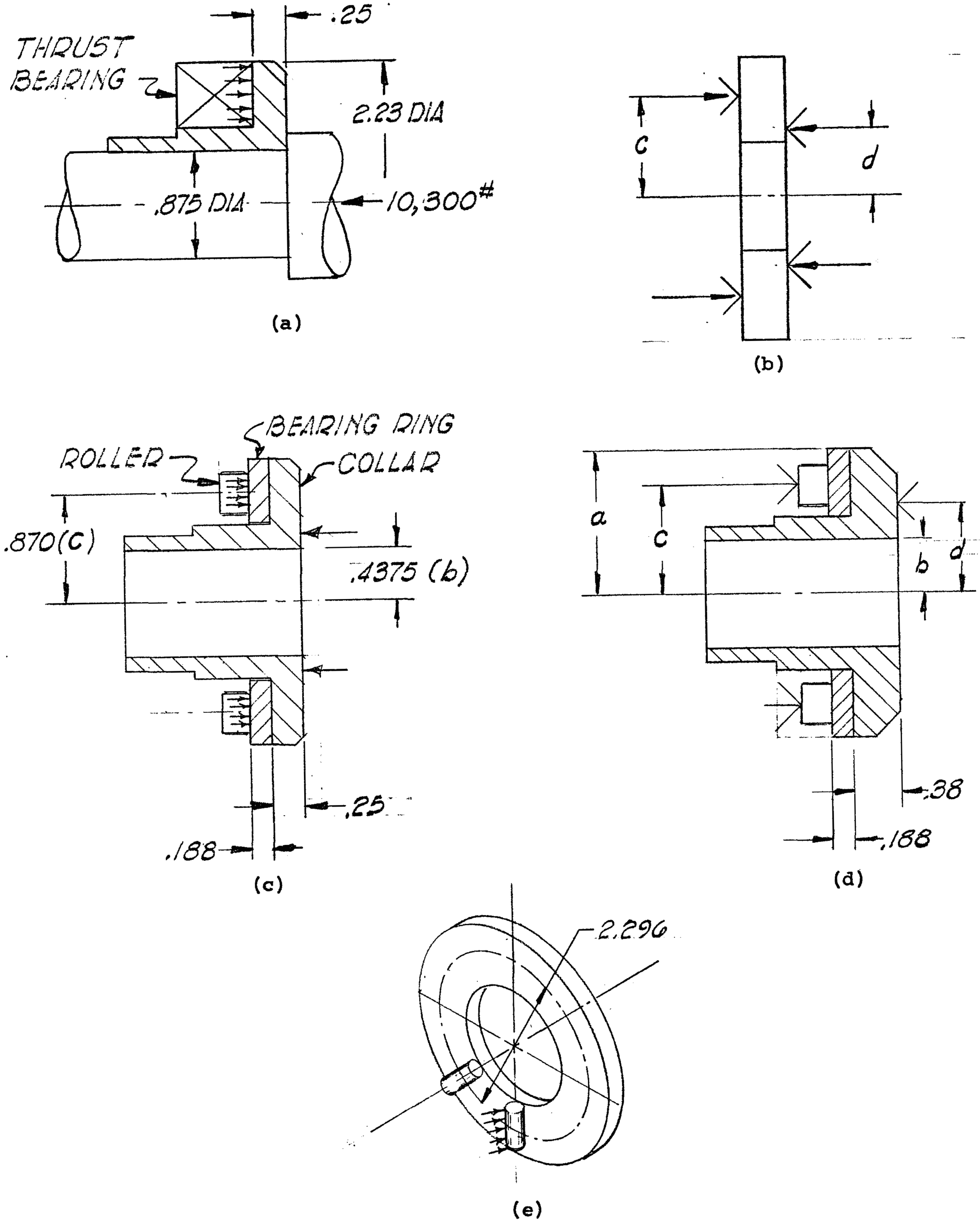

.25
THRUST BEARING
2.23 DIA
.875 DIA
10,300#
(a)
c
d
(b)
BEARING RING
ROLLER
COLLAR
.870(C)
.4375 (b)
.25
.188
(c)
a
c
b
d
.38
.188
(d)
2.296
(e)

Figure 7.

Solution of (3) for the above values yields:

$$\text{Max } y = 0.0095 \text{ inch}$$

Using the companion equation, the maximum stress, which occurs at the inner radius, was calculated:

$$\text{Max } s = 57{,}000 \text{ psi}$$

The above deflection is unacceptably high, since it would cause all the load to be carried on the inner portion of the rollers, resulting in brinelling of the races at the inner diameter.

Since this first computation indicated unsatisfactory operating conditions, the deflection was calculated again using a similar loading assumption, with the only difference being that the thrust bearing is loaded by a single concentrated force at a ring of ΠC, with C as shown in Figure 7(b). These loading assumptions fit Case 15 of the above Roark reference. The deflection equation was not given, so only the Max s, occurring at the inner edge was computed.

$$\text{Max } s = \frac{3W}{2\Pi m t^2}\left[\frac{2a^2(m+1)}{a^2-b^2}\log\frac{c}{d} + (m-1)\frac{c^2-d^2}{a^2-b^2}\right] \qquad \text{Eq. 4}$$

where

c = mean bearing radius, 0.870 inch

d = mean ball screw shoulder radius, 0.499 inch

a,b,t,m same as for Eq. (3)

Solving Equation (4)

$$\text{Max } s = 165{,}420 \text{ psi}$$

Since this is within the elastic limit of the material, the deflection at the outer edge of the collar will be even greater than that computed in Equation (3).

At this point, it was apparent that with the analytical methods used, the deflections in the collar were excessive, in spite of the collar being fabricated of 4340 steel heat treated to Rockwell 40 (S_T = 200,000 psi).

Further discussion disclosed one possible solution: Namely, that the bearing race adjacent to the collar will also add stiffness to the system, through essentially increasing the collar thickness by the bearing race thickness. This concept is shown in Figure 7(c).

Using Case 16 (Roark) again for a composite thickness of the collar and race (0.25 + 0.188), the deflection is:

$$\text{Max } y = 0.0021 \text{ inch}$$

To determine if this deflection is acceptable, the deformation caused by the rollers on the race is calculated.

By using Case 4 on page 320 of the Roark reference for Bodies Under Direct Bearing, the maximum compressive stress, S_c, is determined:

$$S_c = 3190\sqrt{\frac{\rho}{D}} \qquad \text{Eq. 5}$$

where

ρ = load per linear inch

D = ball diameter, inch

To determine ρ, the Auburn Thrust Bearing has 14 rollers, each 0.25 inch diameter and 0.5 inch long. The effective length is simply (14)(0.5) = 7 inches, and

ρ = 10,300/7 = 1471 lb/inch
ν = 0.27
E = 30 x 10^6 psi
D = ball diameter, 0.25 inch

Solving Eq. (5),

S_c = 244,600 psi which is very close to the ultimate compressive strength of the material.

The deformation of the rollers is:

$$\Delta D = 4\rho\left(\frac{1-\nu^2}{\Pi E}\right)\left(\frac{1}{3} + \log\frac{2D}{b}\right) \qquad \text{Eq. 6}$$

where

ν = Poissoin's ratio

b = width of rectangular contact area = $0.0004\sqrt{\rho D}$ = 0.008 inch

Solving Eq. (6)

$$\Delta D = (4)(1471)\left(\frac{1-0.25^2}{3.14(30\times10^6)}\right)\left(\frac{1}{3} + \log\frac{2(0.25)}{0.008}\right)$$

$$\Delta D = 2750 \times 10^{-6} \text{ inch, or } 0.0028 \text{ inch}$$

Therefore; in order to load the balls evenly across their length (see Figure 7(e)), the collar must deflect less than 0.0028 inch.

Previous calculations showed that the collar deflected 0.0021 inch, but, due to the loading assumptions, etc., this was not considered an adequate margin. Since there was room, it was decided to increase the collar thickness from 0.25 inch to 0.38 inch as shown in Figure 7(d). Using Case 15 (Roark) again,

$$\text{Max } y = 502\text{x}10^{-6} \text{ inch, or}$$

$$0.0005 \text{ inch, and}$$

$$\text{Max } s = 31{,}340 \text{ psi}$$

This deformation of the collar, added to the unaccounted for amount of stiffness realized from the collar I.D. being fixed rather than simply supported, makes the thicker collar structurally and operationally sound.

Housing (Dwg. #301558)

As discussed earlier, this Housing actually transmits the main ball screw thrust load back to the main ball screw housing via the thrust bearing. This can be seen by review of the Drive Mechanism (Dwg. #500524).

The primary loading of this member, shown in Figure 8, is again a flat plate type situation where the input load is applied thru the thrust bearing and is being restrained by the 3.00 inch diameter bolt circle.

Referring to Case 17 (Roark),

$$\text{Max } s = \frac{3w}{4t^2}\left[a^2-2b^2+\frac{b^4(m-1)-4b^4(m+1)\log\frac{a}{b}+a^2b^2(m+1)}{a^2(m-1)+b^2(m+1)}\right] \qquad \text{Eq. 7}$$

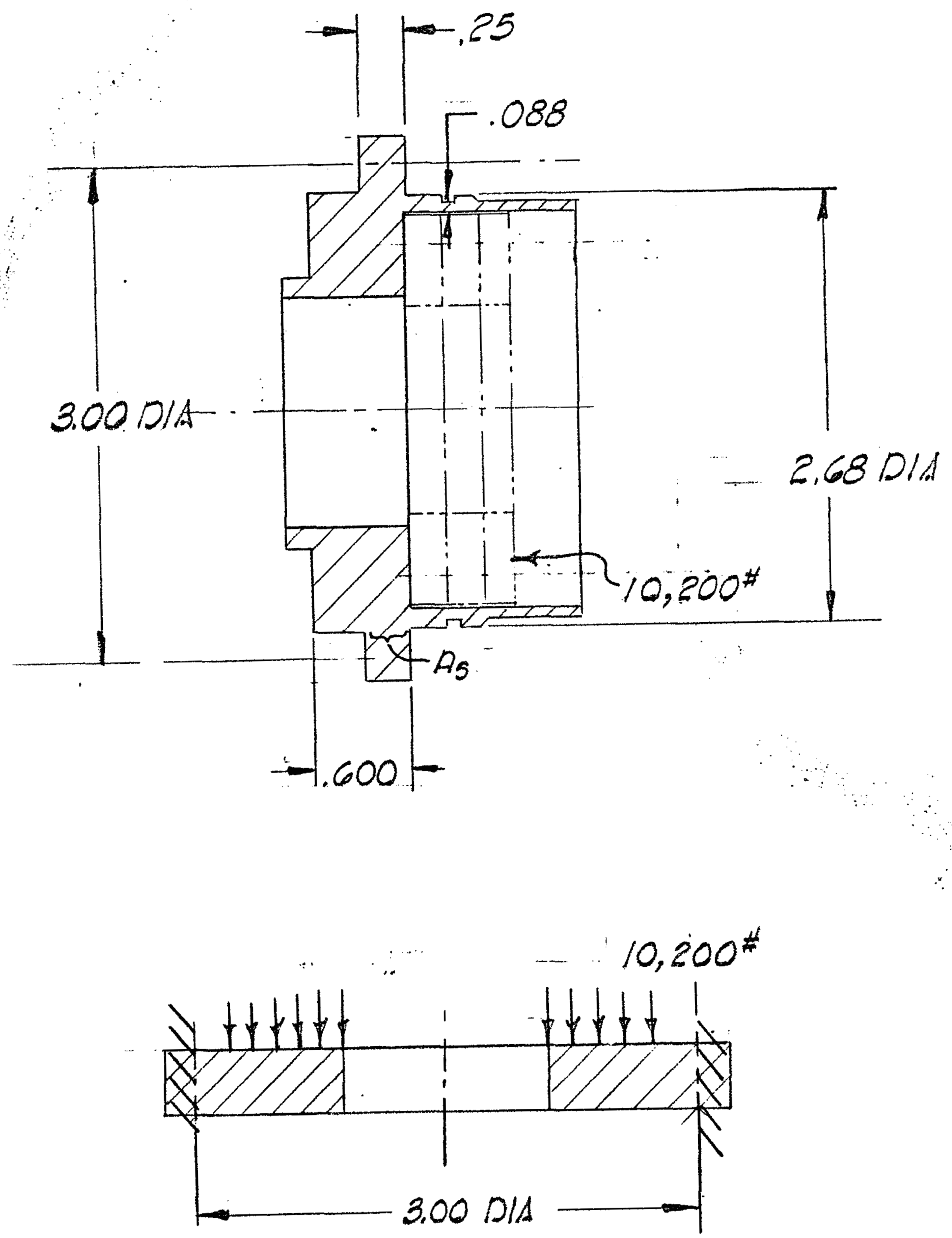


Figure 8. - Shear Stress

where

$w = W/\pi(a^2-b^2)$

t = plate thickness, 0.25 inch

a = plate outer radius, 3.00/2 = 1.5 inches

b = plate inner radius, 1.375/2 = 0.688 inch

m = reciprocal of Poissoin's ratio, 3.70

Solving Eq. (7) for the above values:

$$\text{Max } s = 35{,}089 \text{ psi}$$

Another consideration related to this part is the shear stress in an area marked A_s in Figure 8.

$$A_s = 2.68(\pi)(0.25) = 2.105 \text{ in}^2$$

$$S_s = F/A_s = 10{,}300/2.105 \text{ in}^2$$

$$S_s = 4893 \text{ psi}$$

The bolts holding the Ball Screw Housing to the Sleeve (Dwg. #400985) have already been stressed, and since the loading on the Housing (Dwg. #301558) is not as severe, this calculation will not be repeated.

Trunnion (Dwg. #400999)

This trunnion transmits the axial force from the ballnut to the Link (Dwg. #400983). The trunnion is attached to the ball screw nut by means of four (4) 1/4-20 high strength Nylock bolts, with the threaded portion in the trunnion. A light press fit between the trunnion I.D. and the ball nut O.D. insures that no radial play between the parts exists.

To attain maximum strength, the trunnion was machined out of solid stock, thus eliminating possible loss of strength had welding been used to attach the lugs to the body.

As with the ball screw housing and the link, the trunnion is subject to the maximum output force generated by the drive train, and must, therefore, be carefully checked for structural integrity.

The first check will be for bending stress in the lugs. The maximum diametral clearance between the lugs and the link is 0.0014 inch, with a nominal clearance of 0.001 inch. Bending can take place within this clearance space.

From Figure 9, it can be seen that the total ball screw output of 10,300 pounds is assumed to act in the center of each lug for an individual force of 5,150 pounds. The situation is a cantilever beam with the bending stress

$$S_B = \frac{Mc}{I}$$

Bending moment:

where $M = (5150)\left(\frac{3.688-2.436}{4}\right) = 1612$ in-lb

c = Distance to point of max. fiber stress, 0.25

I = Section modulus $= 1/4\pi R^4 = 0.0031$ in^4

Using the above values and definitions, the bending stress is:

$$S_B = \frac{(1612)(0.25)}{(0.0031)}$$

$$S_B = 130{,}000 \text{ psi}$$

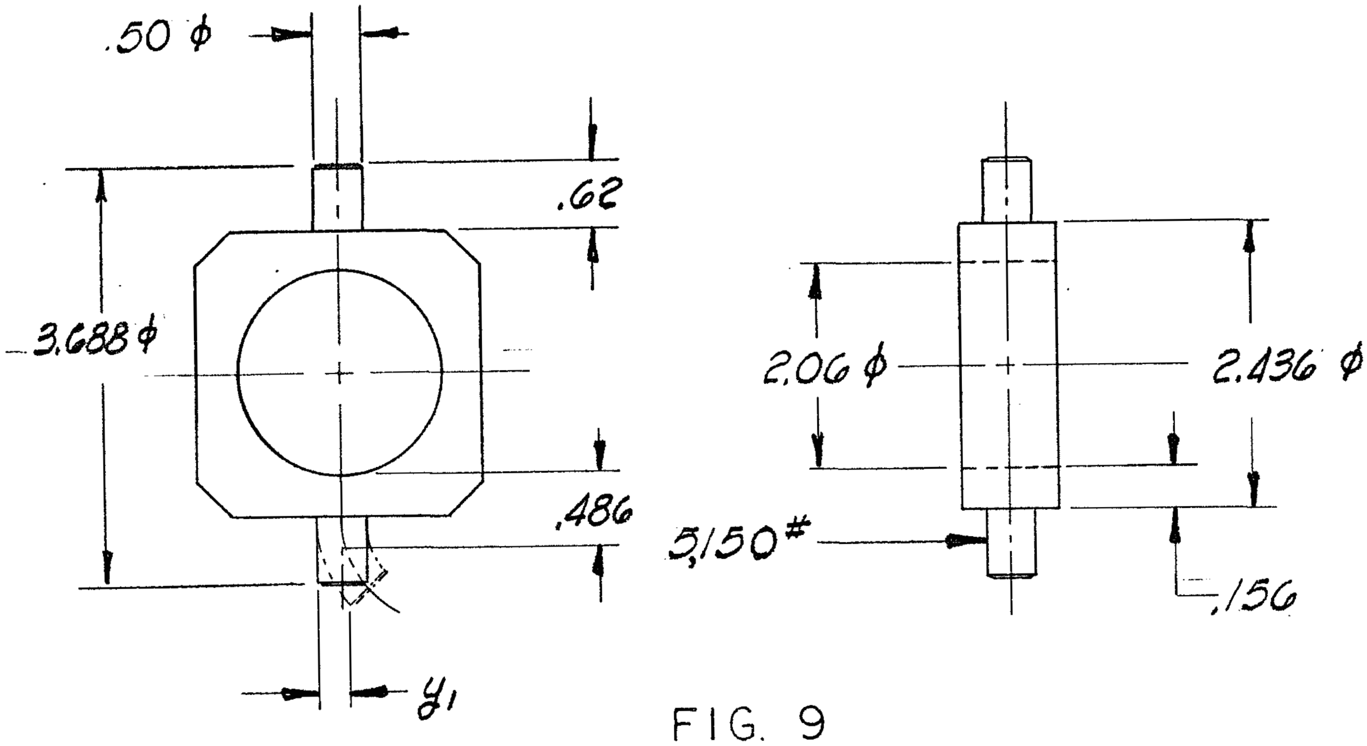


FIG. 9

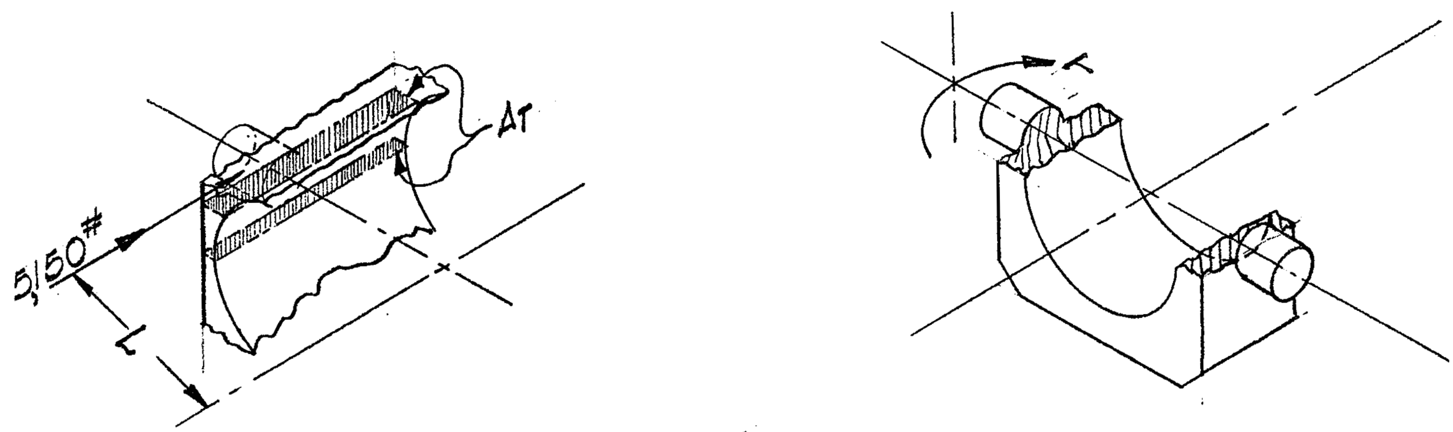


FIG. 10

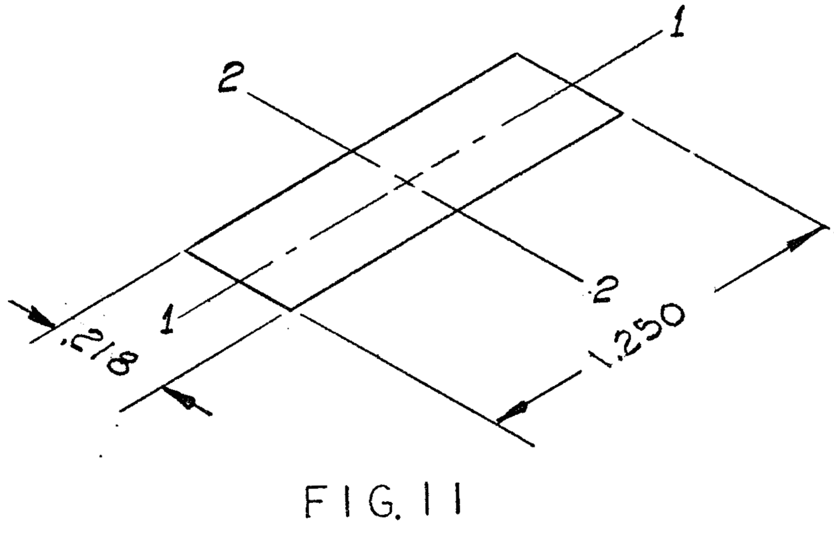


FIG. 11

This part is 4340 steel, heat treated to R/C 45, which is equivalent to a tensile strength of 225,000 psi.

The deflection of the lug at its outer edge will be

$$y = \frac{W}{6EI}(3a^2 \ell - a^3)$$

where

W = applied load, lb

a = distance from support to point of applied load, inch

ℓ = total lug length, inch

E = Modulus of Elasticity, psi

I = Section Modulus, in^4

$$y = \frac{(5,150)}{(6)(30\times10^6)(0.0031)}\left[(3)(0.33)^2(0.66)-(0.33)^3\right]$$

$$y = 0.00166 \text{ inch}$$

Next, if it is assumed that the load is carried as pure shear, as would be the case if the fit on the lugs was sufficiently close to prevent bending, the shear area A_s, is

$$A_s = (\frac{\pi}{4})(0.50)^2 = 0.196 \text{ in}^2$$

and the shear stress, S_s, is

$$S_s = 5150/0.196 \text{ in}^2$$

$$S_s = 26,275 \text{ psi}$$

The next stress condition to be investigated is the torsion on the lugs. Under a bending moment, M, of (5150)(L), the torsional stress is induced in the shaded area, A_T, of Figure 10. The shaded area is again shown in Figure 11, with the dimensions and the major axes designated 1-1, and 2-2.

$$I_1 = \frac{1}{12} bh^3 = \frac{(1.250)(0.218)^3}{12} = 0.0011 \text{ in}^4$$

$$I_2 = \frac{1}{12} bh^3 = \frac{(0.218)(1.250)^3}{12} = 0.0355 \text{ in}^4$$

The polar moment of inertia for both top and bottom areas is

$$J = (I_1 + I_2)2 = 0.0732 \text{ in}^4$$

$$T = (5150)(0.528) = 2719 \text{ in-lb}$$

$$r = 1.250/2 = 0.625 \text{ inch}$$

$$\text{Max. } S_s = \frac{Tr}{J} = \frac{(2719)(0.625)}{0.0732}$$

$$\text{Max. } S_s = 23{,}215 \text{ psi}$$

In summary, the part is considered structurally adequate.

<u>Sleeve (Dwg. #400987)</u>

This sleeve must transmit the full load of the Serpentuator, which will be in the form of a moment. The magnitude of the moment is referred back to the specification tip force of 10 pounds, and the overall length of the Serpentuator:

$$M = (17 \text{ ft})(12 \text{ in/ft})(10 \text{ lb}) = 2040 \text{ in-lb}$$

The bending stress is again

$$S = \frac{Mc}{I}$$

From Figure 12:

$$c = 2.19 \text{ inches}$$

$$I = \frac{\Pi}{4}(R_1^4 - R_o^4) = 3.081 \text{ in}^4$$

$$S_B = \frac{(2040)(2.19)}{3.081} = 1450 \text{ psi}$$

The deflection (using Case 9 on page 106, (Roark)) is:

$$y_{max} = \frac{M\ell^2}{2EI}$$

$$y_{max} = 0.004 \text{ inch}$$

Linkage

Simplified sketches of the linkage mechanism are shown by Figures 13-a through 13-e. In these figures, the joint is progressively rotated counterclockwise from a folded position through 360° to the other folded position.

It is apparent that each member has multiple reaction points, all of which must be solved to properly analyze the stresses in the linkage. The manner in which the links are pinned together would require solution of a statics problem in which numerous equations must be solved simultaneously for the x and y reactions at each pin.

Under the funding and schedule limitations for delivering the Serpentuator hardware, the linkage could not be thoroughly analyzed to insure that there were no structural failures in

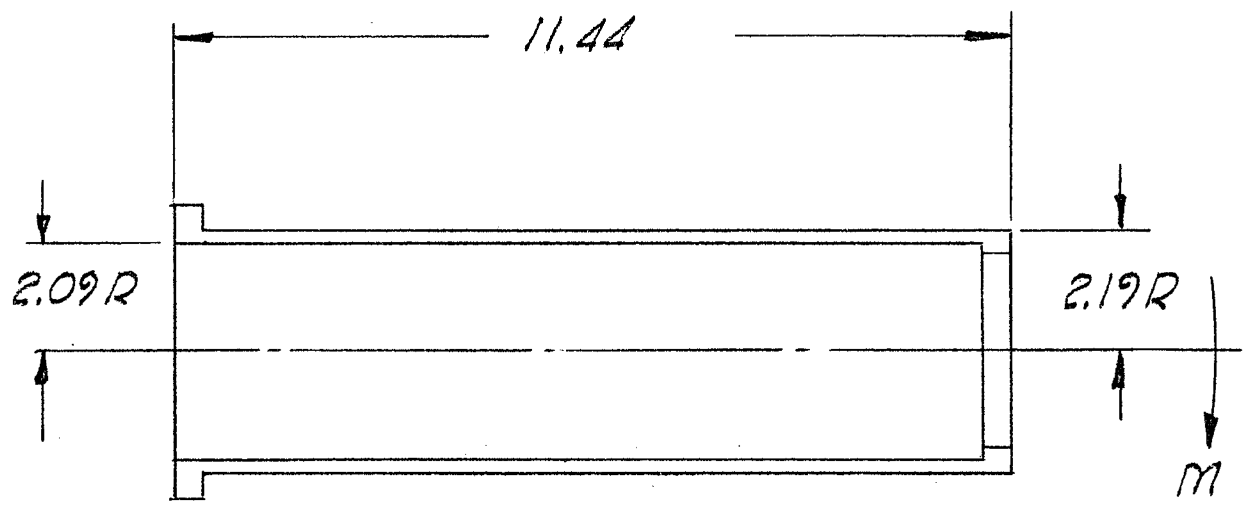


FIG. 12

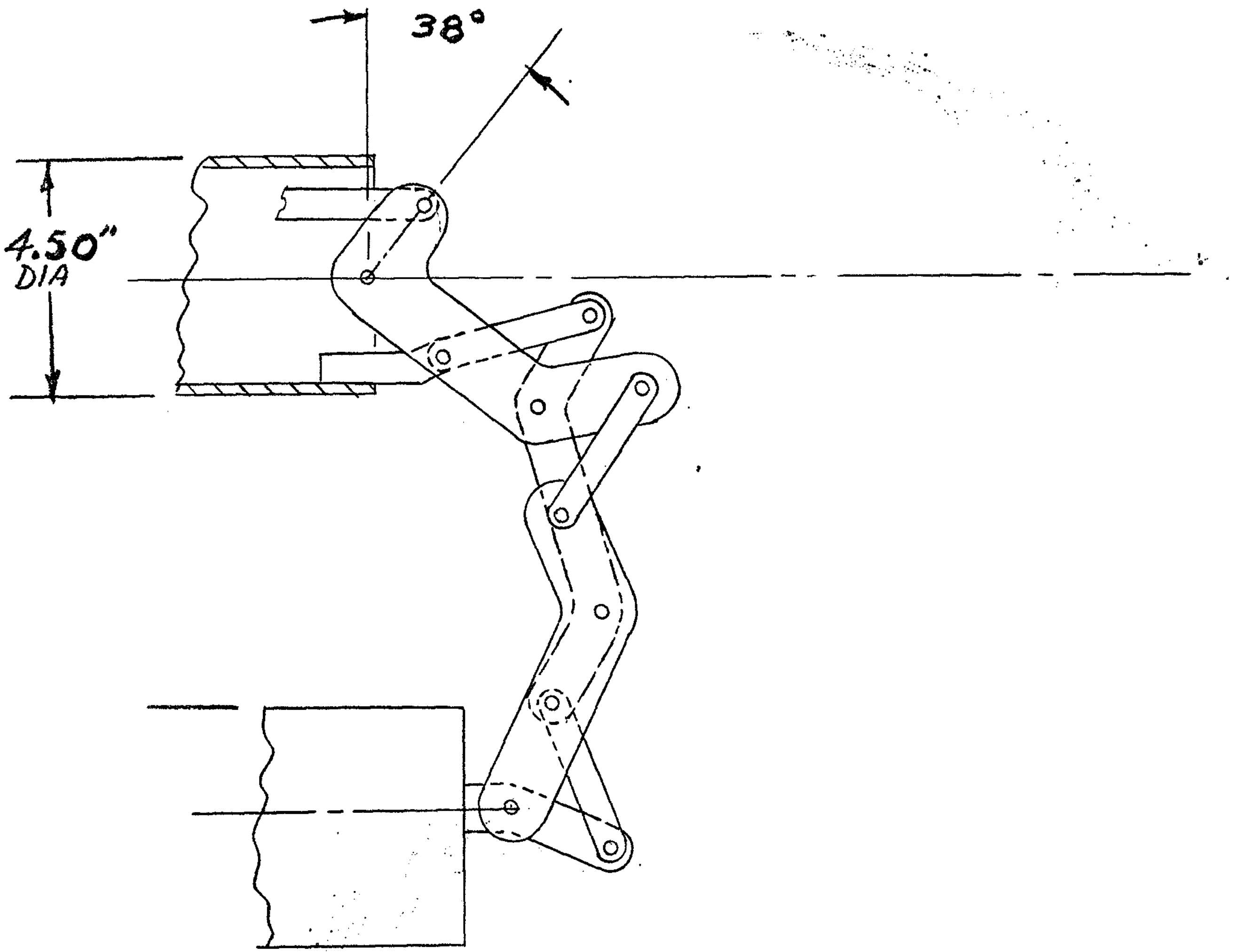


FIG. 13a

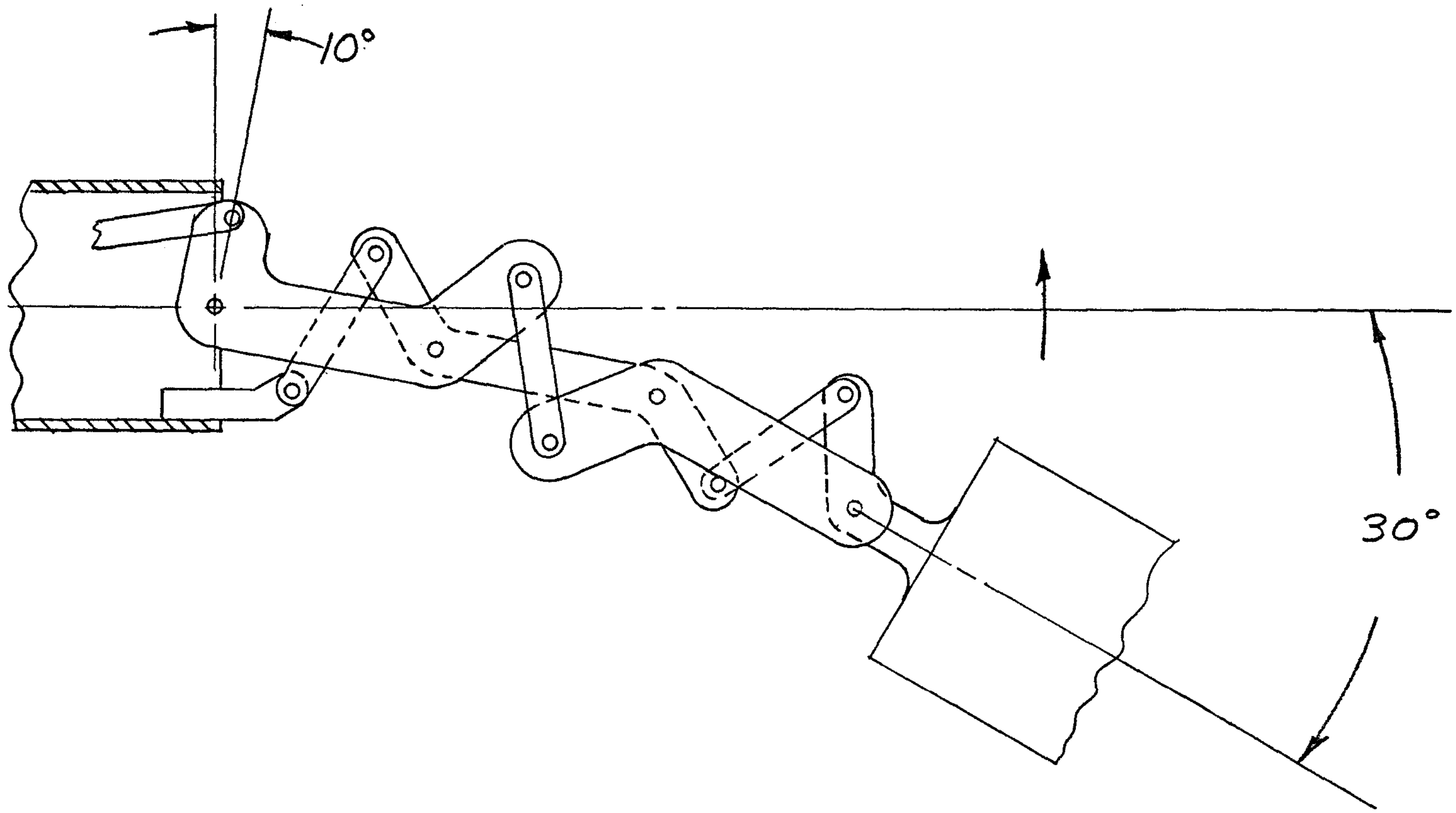


FIG. 13w

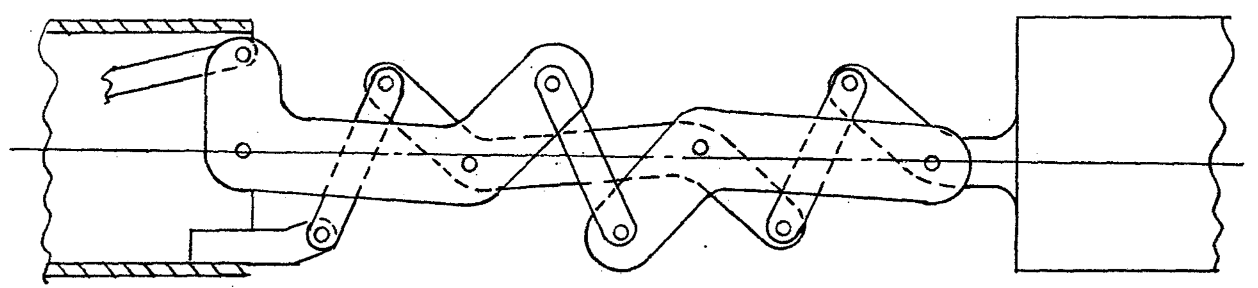


FIG. 13N

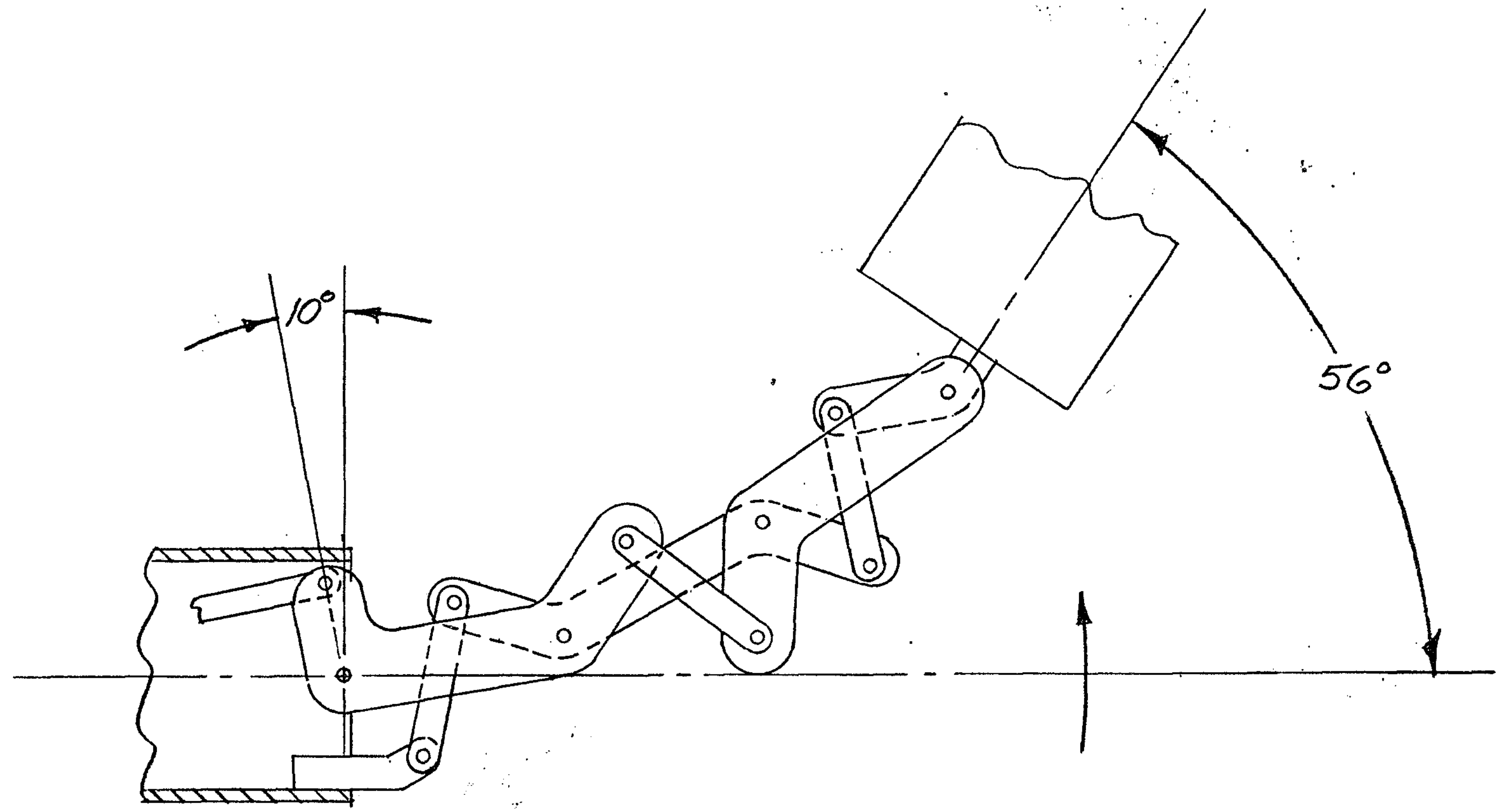


FIG. 13d

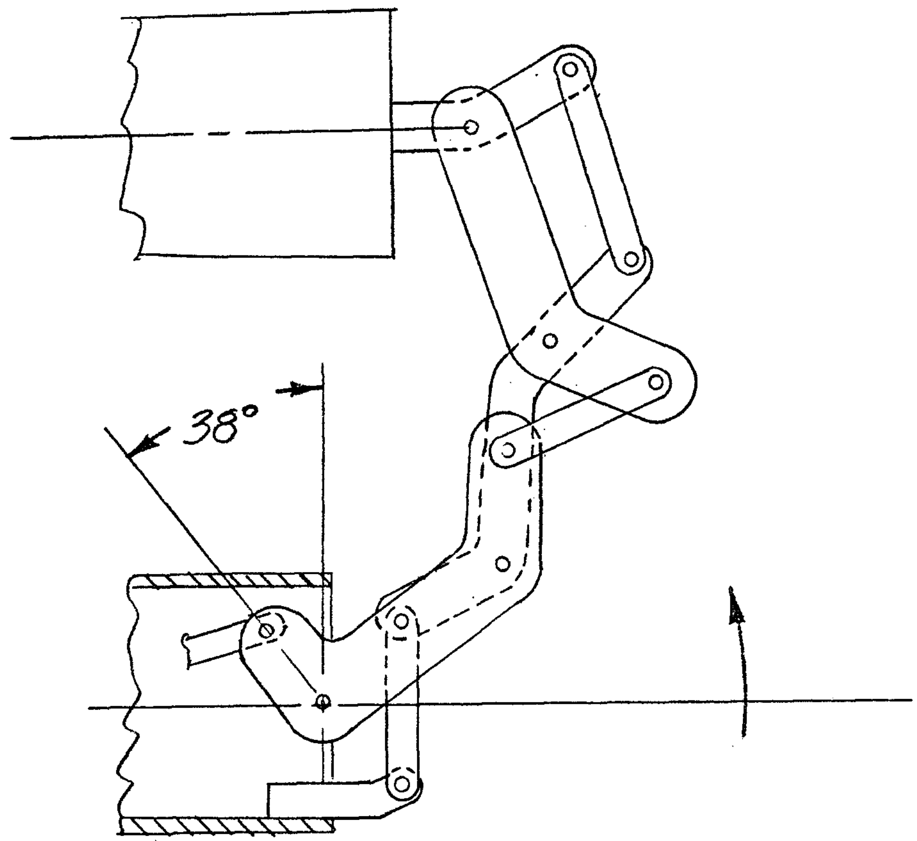


FIG. 13l

the linkage members. A conservative approach was taken, however, as to the size of the linkage members, the pivot points, and material selection. Specifically, the single links were 0.75 thick 4340 steel, heat treated to Rockwell C40, which is equivalent to 200,000 psi tensile strength. The split linkages, which straddle the single linkages, are of equivalent thickness and of the same material.

Super Oilite No. 16 bushings were used in all pin joints. The shafts were all press fitted 60 Case, Class L diameter, Thompson shafting (Thompson Industries, Manhasset, N.Y.). The Oilite bushings were reamed to a range of 0.5000 to 0.5004 inch diameter, while the shafting was 0.4990 to 0.4995 inch diameter, thus providing a maximum diametral clearance of 0.0014 inch and a minimum diametral clearance of 0.0005 inch.

Brief check-out tests of the completed Serpentuator indicated that the rather massive and heavy linkage members could be optimized either in size or material selection to a smaller and lighter package without affecting performance or structural adequacy.

KINEMATIC ANALYSIS

The formal analysis of the proposed Serpentuator linkage in its present form is beyond the scope of this contract.

In order to understand the basic operation of the linkage, an analogy must be made whereby the actual linkage is replaced by a series of planetary gears as shown in Figure 14a. Unlike the actual Serpentuator, the gear system shown in Figure 14a can rotate continuously without interference between connecting members.

The overall gear ratio for a planetary system would be the sum of the ratios for the system divided by the ratio for one link, or simply:

$$\frac{2 + 2 + 2 + 2}{2} = 4$$

In such a system, an input angle of 45° would result in an accumulative angle of 45°-90°-135°-180° for links 1, 2, 3, and 4, respectively.

The actual Serpentuator, because levers are used instead of gears, is not linear as indicated above. The angular displacements are 36°-79°-127°-180° for the four links. The overall gear ratio is in the range of 5.0 to 6.5, depending on the input angle.

The connecting links in the succeeding figures are numbered 1 through 4, and the complete motion of each link will be given for one revolution of Link 1.

In Figure 14b, Link 1 is rotated 360°, and thus follows a circular path. The position of Link 1 for any input angle, α, is derived from the formula for a circle: $X = \sin \alpha$, $y = \cos \alpha$.

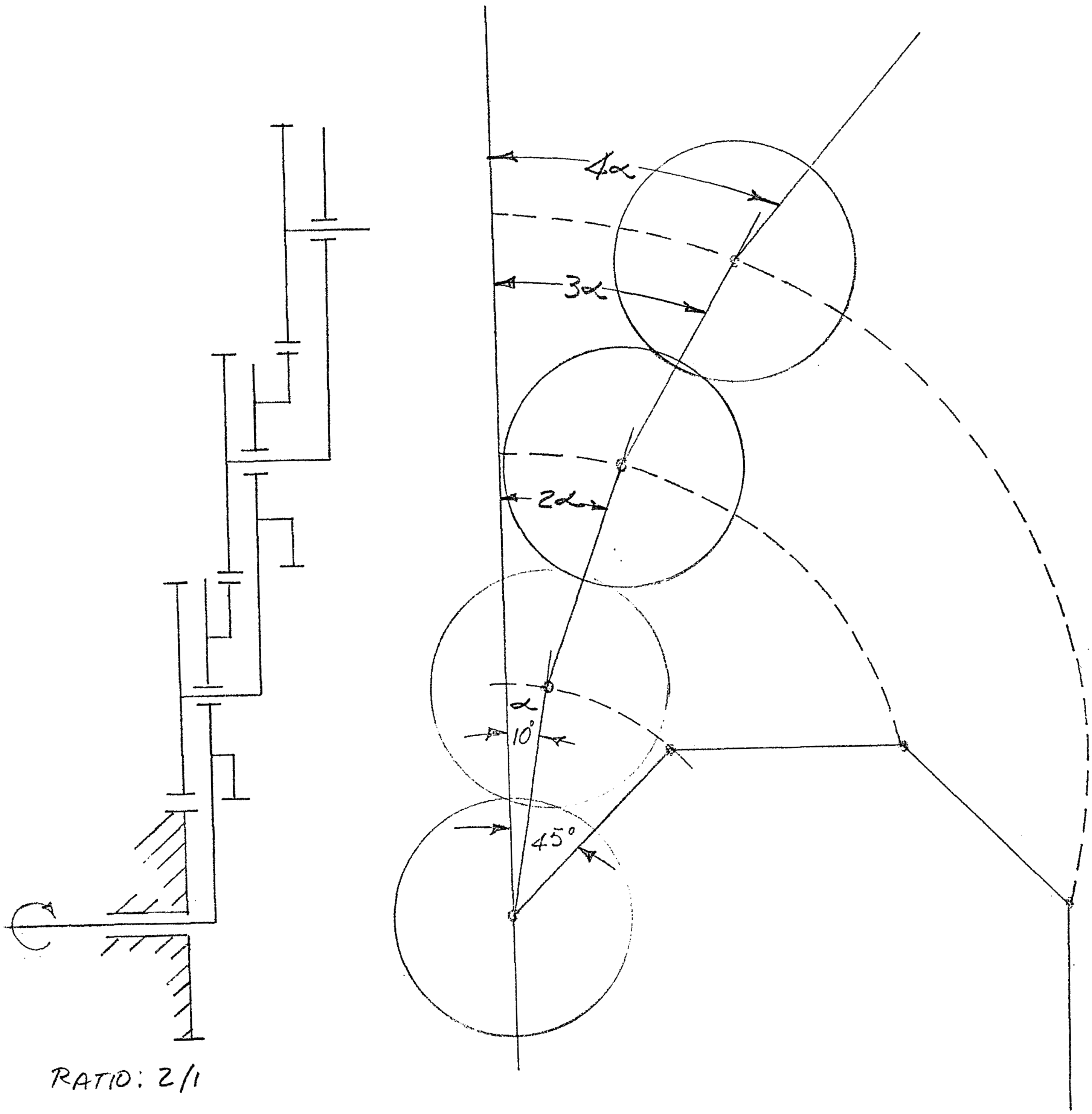
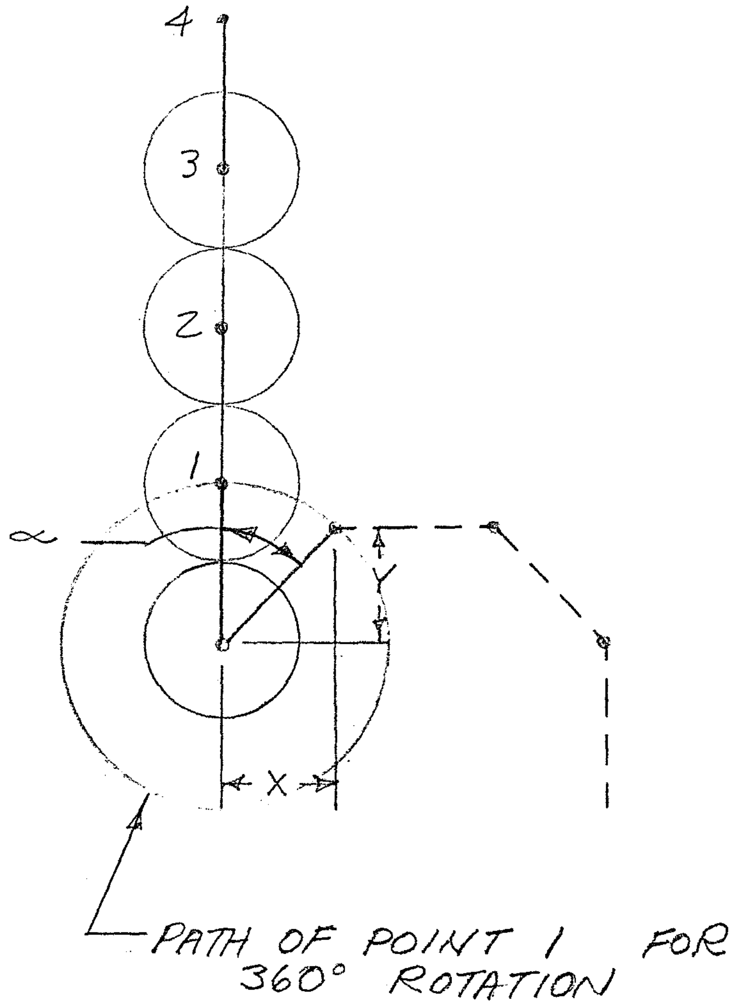


Figure 14a.



$$X = \sin\alpha$$

$$Y = \cos\alpha$$

$$X^2 + Y^2 = 1$$

Figure 14b.

The second link makes two revolutions along the path indicated in Figure 14c while Link 1 makes one circular revolution. The path taken by Link 2 appears to be a curve type called a Limacon described in polar coordinates by the equation

$$r = a - b \cos \alpha$$

$$a < b$$

Although the curve appears to be somewhat involved as far as Serpentuator motion is concerned, the problem is academic because only the portion of the curve between $2\alpha = 0°$ and $2\alpha = 90°$ will be used.

In rectangular coordinates,

$$X = \sin \alpha + \sin 2\alpha$$

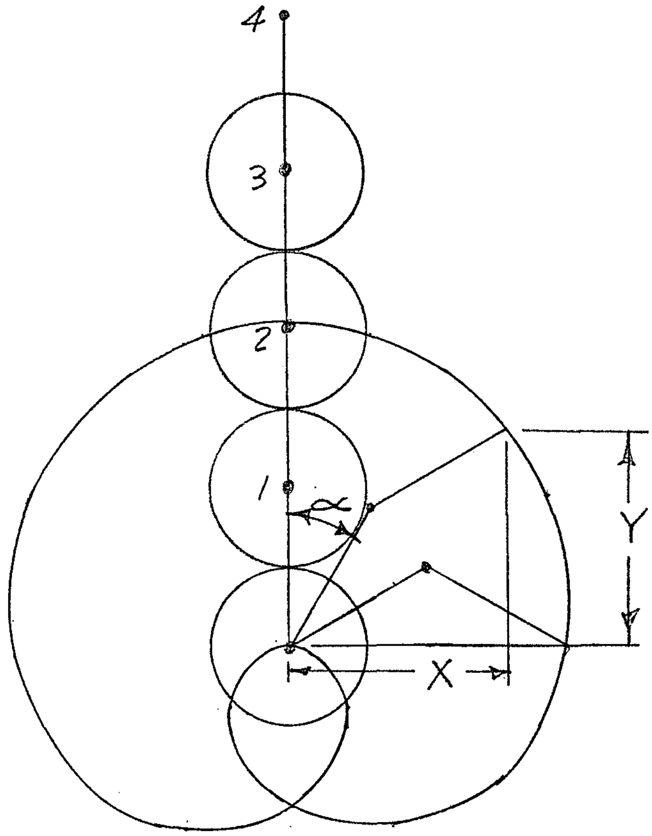
$$Y = \cos \alpha + \cos 2\alpha$$

$$X = Y \cdot 2 \sin \alpha - \sin \alpha + 4 \sin^3 \alpha$$

The third link, by carrying the same convention further, makes three revolutions along a path shown in Figure 14d for one circular revolution of Link 1. The formal type of curve for this motion is not determined, although the rectangular coordinates may be found by

$$X = \sin \alpha + \sin 2\alpha + \sin 3 \alpha$$

$$Y = \cos \alpha + \cos 2\alpha + \cos 3 \alpha$$



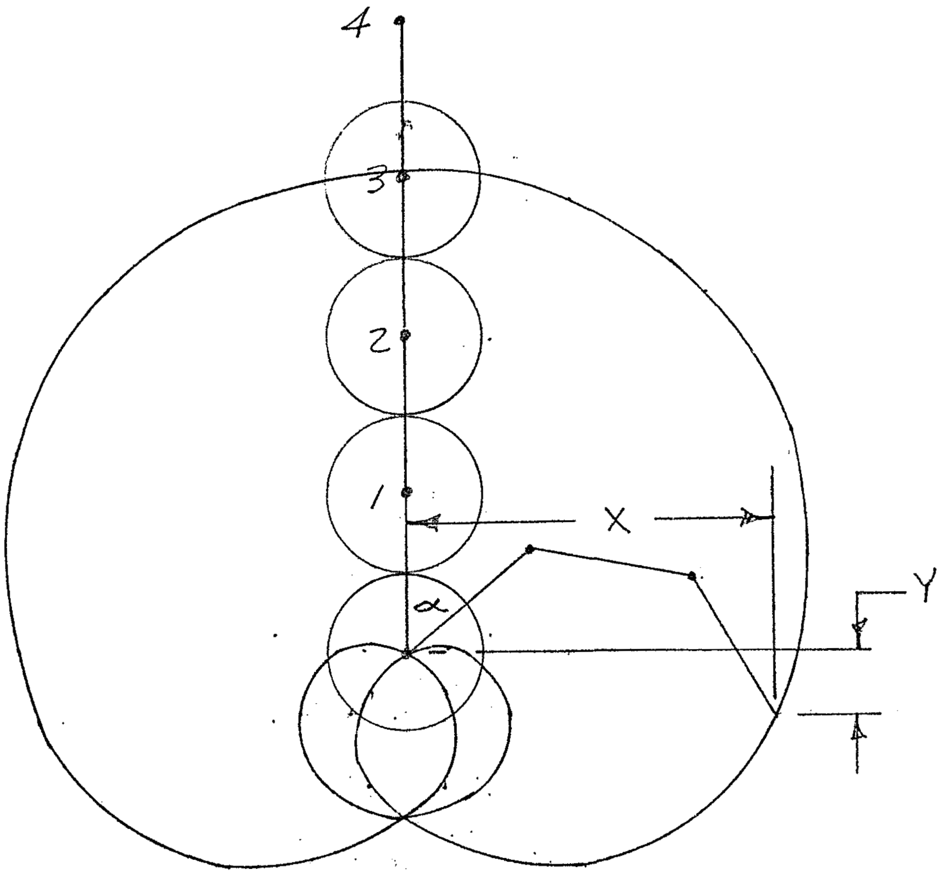
PATH OF POINT 2 FOR 360°
ROTATION OF POINT 1.

$$X = \sin\alpha + \sin 2\alpha$$

$$Y = \cos\alpha + \cos 2\alpha$$

$$X = Y \cdot 2\sin\alpha - \sin\alpha + 4\sin^3\alpha$$

Figure 14c.



PATH OF POINT 3 FOR 360°
ROTATION OF POINT 1

$$X = \sin\alpha + \sin 2\alpha + \sin 3\alpha$$
$$Y = \cos\alpha + \cos 2\alpha + \cos 3\alpha$$

Figure 14d.

As with Link 2, the full motion profile will not be applicable to the Serpentuator operation. Only the portion of the curve between $3\alpha = 0°$ to $135°$ is used.

The last link, No. 4, is the most significant of all because it determines the type of motion to be followed by an extension tube attached to the last point. The curve as shown in Figure 14e is an extension of the previous curves, but this link makes four revolutions for one revolution of Link 1.

It now becomes apparent that if the linkage depicted in the last four figures were considered to be the outboard elbow on the Inserp, an extension arm attached to the linkage would not follow a circular pattern. Figure 14f is a replot of the last link curve along with a circular arc equal to the extended length of the linkage. It also holds that; the difference between the actual curvature and a circular arc will always be constant, regardless of whether the entire 15 foot Serpentuator (from first elbow) is swept through $\pm 180°$ from the inboard link, or whether the 3 foot extension tube at the outboard link is swept.

An actual measurement with the scale model linkage shows the difference between a circular arc of 12.88 inch radius and the linkage to be 5.18 inches.

As stated earlier, the Serpentuator linkage is, in reality, a series of levers instead of gears, thus making the foregoing analysis based on linear input-to-output relations only approximate.

During the design stages of the program, the linkage pivot points were positioned by fairly involved trigonometric

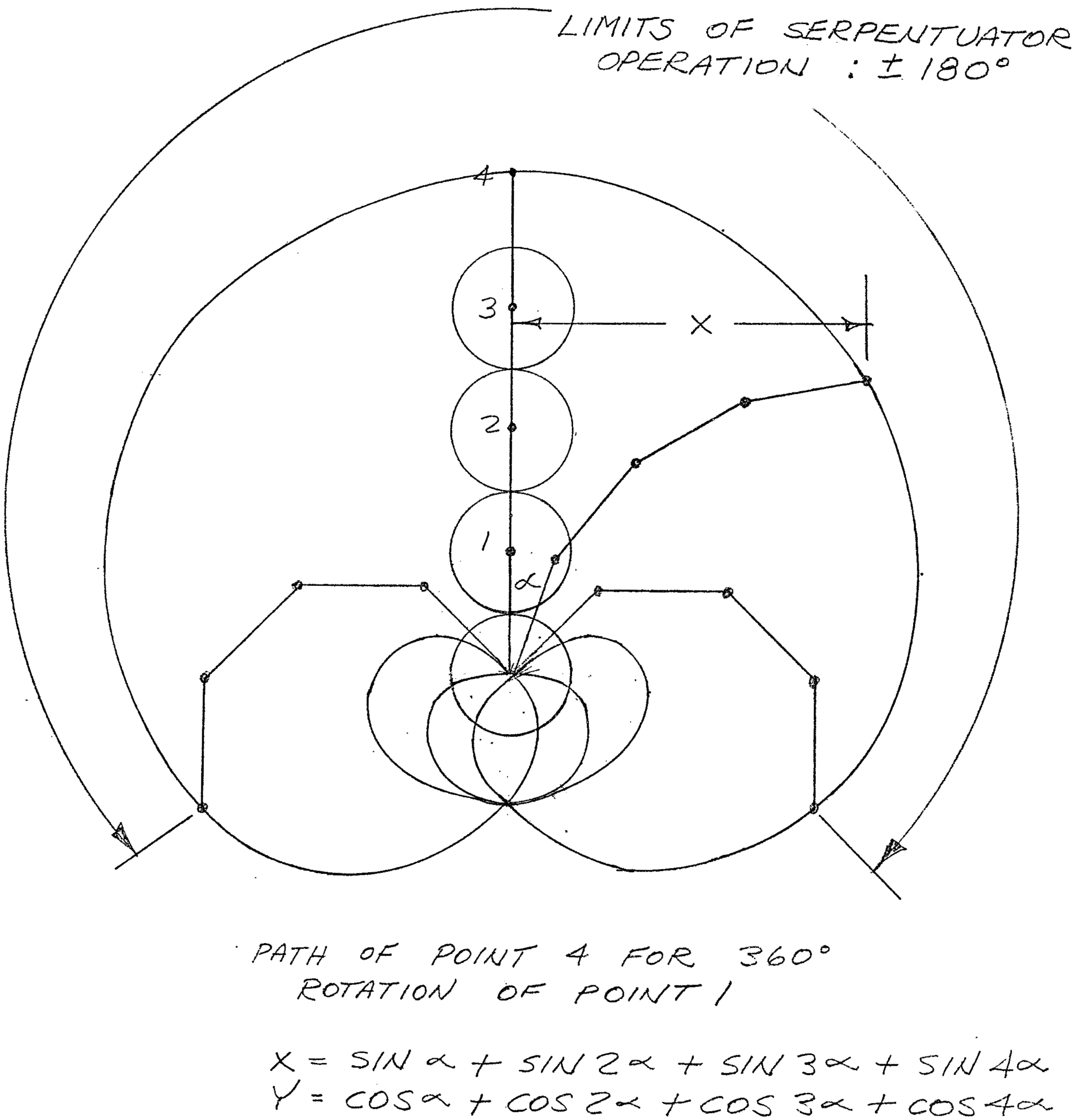

LIMITS OF SERPENTUATOR OPERATION : ± 180°
4
3
2
1
α
X
PATH OF POINT 4 FOR 360° ROTATION OF POINT 1
X = SIN α + SIN 2α + SIN 3α + SIN 4α
Y = COS α + COS 2α + COS 3α + COS 4α

Figure 14e.

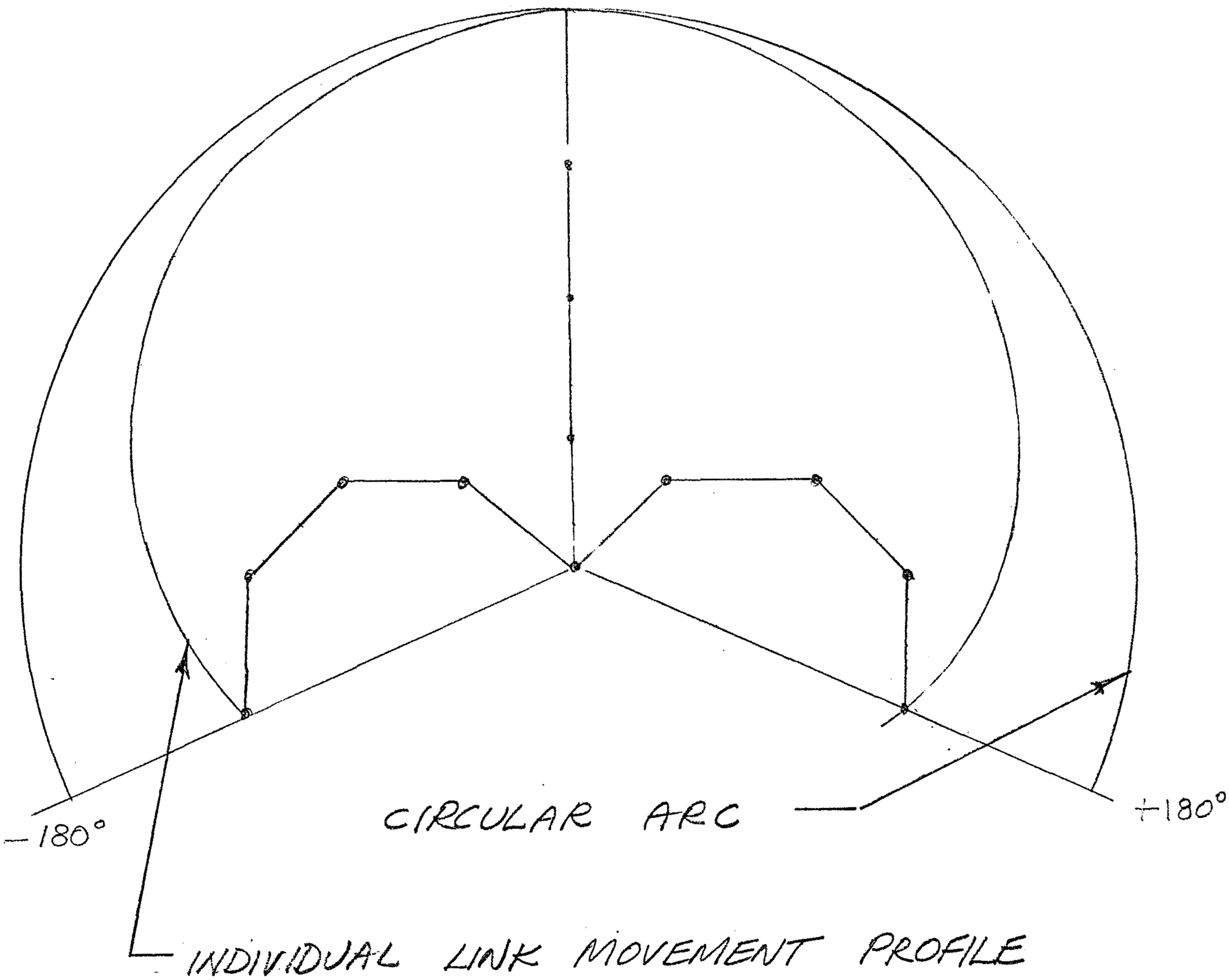


Figure 14f.

relationships to achieve nearly uniform movement of the Serpentuator. The linkage configuration was designed to: (1) Provide equal tapering of the angle ratio among the linkage members; and (2) keep the angle ratio between linkage input and output angles close to the same for both clockwise and counterclockwise operation.

The first (1) objective was considered the more important, recognizing that the stresses induced among the members would depend on their loading, and the loading would depend on the extent of the mechanical disadvantage each member had to operate under. To keep the loads and have the stresses uniform, the portion of the angular gain achieved by each member should be tapered uniformly between 0° and 180°.

The second objective could not be attained, at least within the schedule and cost limitations of this contract.

The final angular relationship between input and output angles for both full clockwise and counterclockwise operations is shown by Figure 15. This figure demonstrates that, for given input angles, the clockwise mode of operation is slightly more sensitive (i.e., larger output angles) than the opposite direction. It is also obvious from Figure 15 that in spite of the differences in the median input angles, the extremes at 0 and 180° are again equal in both directions.

After consultation with the customer, it was decided that the slight nonlinearity and unequalness would not affect the operation of the Serpentuator since the operator would be controlling the movement manually, and would have visual contact with the tip at all times. In future versions where the tip position must be remotely operated, or even programmed, the nonlinearity would need to be compensated for electronically, or eliminated entirely from the linkage through further refinement.

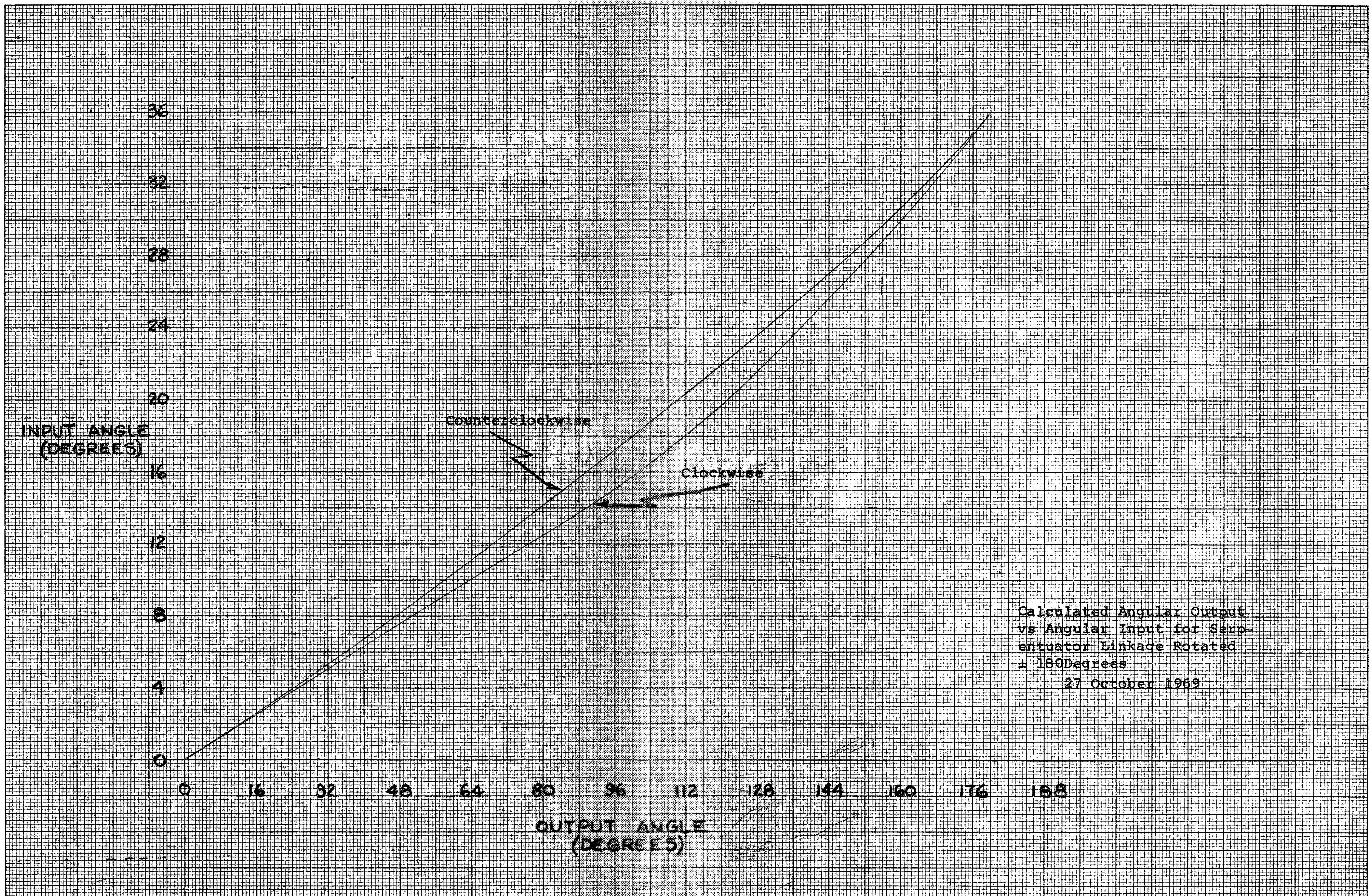

36
32
28
24
20
16
12
8
4
0
INPUT ANGLE
(DEGREES)
0
16
32
48
64
80
96
112
128
144
160
176
188
OUTPUT ANGLE
(DEGREES)
Counterclockwise
Clockwise
Calculated Angular Output
vs Angular Input for Serp-
entuator Linkage Rotated
± 180Degrees
27 October 1969

Figure 15.

FABRICATION

The intervehicular Serpentuator that was originally planned for service within the Orbital Workshop, was to have been 18 feet long and foldable into a volume which could be passed through a removable bulkhead opening in the S-IVB tank and the EVA air lock hatch.

The Serpentuator was to have been provided with four (4) powered links with controls provided at both the fixed and the free ends. Internal redundant hermetic sealing of all critical electrical and mechanical parts was also to be included.

Shortly after the design review meeting, at which the design layouts and detail drawings were presented and approved, a firm shop estimate of the fabrication cost was obtained. This estimate, including the machine shop time and the assembly time appreciably exceeded the allotted funds.

The high cost was attributed to the rather sophisticated drive train and hermetic sealing systems which evolved through several preliminary design review meetings. The decision was made to reduce the number of deliverable links, to eliminate the internal hermetic seals (the primary seals were O-rings at the flange and metal bellows over the linkage), and to provide only one control panel.

Documentation delivered to the customer described the four link version, and provisions were maintained throughout the program to retain the feature of having the two link Inserp easily convertible to the original four link model.

The bulk of the raw material used in the Serpentuator fabrication was furnished by the government, and resulted in some reduction in cost. All excess material was returned with the finished product.

The fabrication of parts proceeded smoothly, and an unusually low number of rework or reject situations arose.

Assembly also went smoothly, except for an occasional thread that needed to be chased, or a shaft lapped to get the proper fit. The mechanical stops did not line up exactly and a minor last minute modification had to be incorporated.

After the first linkage and drive train were assembled, it was found that there was approximately 3 inches of uncontrolled play in the plane of the Serpentuator movement. This play was measured 17 inches away from the outboard side of the joint, and if this lever arm were ratioed to the original length of 18 feet, the tip play would have been unacceptable.

Approximately 2 days time was spent locating the cause of the play, by using locating indicators at various points, disassembling the system and measuring the Oilite bearing clearances, etc. It was finally determined that the play was due to an accumulation of three situations: (1) Deflection in the walls of the Ball Screw Housing (Dwg. #400981); (2) play in the ball nut-to-link bushing; and (3), play in all the linkage joints.

Especially large contributors to the overall play are items (1) and (2), and the first links of (3). These points are significant because the play existing here is multiplied through the linkage just as a planned angular movement is multiplied at a ratio as shown by Figure 15.

It should suffice to say that, the addition of the individual play at each joint ratioed through the 17 inch lever arm, and also ratioed through the joint, equalled the total play very closely.

The problem was corrected to a great extent by taking the deflection out of the ball screw housing (Dwg. #400981) through shims added between the thrust bearings and the Housing (Dwg. #301558).

It was decided that the tolerance limits on the linkage bearings were consistent with good design practices, and that closer tolerances (minimum clearance of 0.0004 inch) would have imposed severe cost penalties. In retrospect, the walls of the ball screw housing could have been made more rigid, even though it is not required from a pure stress failure standpoint.

WEIGHT ANALYSIS

The contract specified that the Serpentuator should have a specific gravity of one to insure compatibility with underwater testing as will be conducted in the Marshall Neutral Buoyancy Facility.

With the tube diameter and length used, however, the buoyancy was primarily a function of the weight of the drive train and linkage components inside the tube. A breakdown of the weights of the individual components is shown by Figure 16.

The actual weight of the assembly is 117 pounds. However, to compute the buoyancy, it must be noted that the Control Link is fastened to a bulkhead or beam, thus making the Control Link, with Drive Train No. 1, and Linkage No. 1 self-supporting; i.e., it adds nothing to the buoyancy of the remaining parts. The two outboard spacer tubes, Drive Train No. 2, and Linkage No. 2 should be neutrally buoyant.

Assuming that a 4.5 inch diameter is constant along the entire Serpentuator length for simplification of calculations, the outboard volume is:

$$V = \frac{\pi}{4}(4.5)^2(72) = 1145 \text{ cubic inches}$$

Since a cubic inch of water weighs 0.0361 pounds, the buoyant force is

$$F_B = (0.0361 \frac{\text{pounds}}{\text{cubic inch}})(1145 \text{ cubic inches})$$

$$F_B = 41.3 \text{ pounds}$$

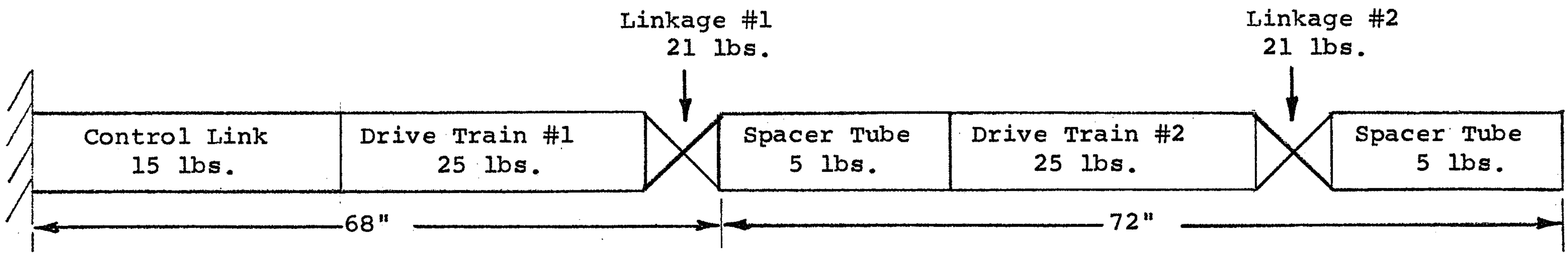


Figure 16. - Sketch of Weight per Unit/Item

The weight of the outboard components (Figure 16) is 66 pounds, which indicates the need to add flotation material to achieve buoyancy. Another 693 cubic inches of volume is required, which can be in the form of styrofoam deployed for a length of 59 inches (over two spacer tubes and Drive Train No. 2), with a maximum diameter of 5.93 inches and a minimum diameter of 4.50 inches.

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The Engineering Model Intervehicular Serpentuator was successfully designed and fabricated by Astro-Space Laboratories in accordance with Contract NAS8-30166.

During progress meetings with the customer early in the program, considerable effort was directed toward achieving a realistic back-up hermetic sealing design for the components which could be damaged by exposure to either water or hard vacuum. Evolving from these meetings was a complex design entailing a rather elaborate method of compartmentalizing the Serpentuator for internal pressurization, deploying the electrical wires between compartments, and sealing the compartments from each other and from the outside.

The fabrication and assembly costs of such a system, when coupled with a drive train required to provide quite large forces, and a complex mechanical linkage, were higher than the contract value.

The quantity of hardware was accordingly reduced to more nearly fit the contract value, then fabrication and assembly continued on the smaller system without any unexpected problems.

Astro-Space Laboratories, Inc., has participated in the design and fabrication of several Serpentuator systems; but, the system developed under this program offered the greatest engineering challenge, because the linkage in each joint was unique, and the drive section required to power the linkage had to produce quite large forces within a small envelope.

Recommendations

General

During the course of the program, several ideas presented themselves that would be worthy of consideration if similar systems are proposed in the future.

The linkage would be considered state of the art, and the solution of the loads and, hence, the stresses in the members would make possible the optimization of the links as structural members. Further analysis into the geometry of the links would eliminate or reduce the nonlinearity discussed in connection with Figure 15. Undoubtedly there is a computer program for the solution of such a problem.

The application of a metal bellows as the joint seal was, according to discussions with vendors, an unusual application, with the uncertainty resulting from the ±180° bending of the bellows, rather than only axial movement. Since the future potential of the Serpentuator depends heavily on the reliability of the metal bellows, it is recommended that fatigue tests be conducted on the present design, as well as additional analytical studies on the bellows geometry so that space-compatible reliability may be achieved.

To reduce uncontrollable play in the joint linkage, it is recommended that the tolerance limitations with respect to the unique geometry of the linkage be studied so that a realistic trade-off between cost and performance can be made on subsequent systems using the "angle multiplier" linkage.

Space Compatibility

The original design, which existed prior to the reduction in fabrication effort, incorporated features which provided for redundant sealing of all critical components, as well as provisions for permanently sealing all flange joints by either welding or wrapping with Perma Seal Tape (Union Carbide). The control switches and the electrical connectors used throughout were of the hermetic type, and would need no modification for space use. The switch bodies and connector flanges would require welding for sealing their mounting interface for space application. Even though the electric drive motors would be doubly protected against the environment, a third protection can be added in the form of a proprietary treatment to the motor components to make them capable of operation in a hard vacuum. This feature is attained, however, at a loss in the life expectancy of the motor; down to approximately 50 hours.

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