DESIGN OF MAIN LANDING GEAR OF JET TRAINER

by

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Model R-550 Linear Actuator

Airborne Accessories Corporation

Hillside, New Jersey

NOTATION

$\uparrow$ Force showing direction.

$\upuparrows$ Torque or moment, direction by conventional right hand rule.

A Area

D Diameter

f_b Internal bending stress

f_{br} Internal bearing stress

f_c Internal compressive stress

f_s Internal shearing stress

F Force

F_b Bending modulus of rupture

$F_{br,u}$ Ultimate bearing stress

F_{cc} Allowable crushing or crippling stress

F_{cu} Ultimate compressive stress

F_{cy} Compressive yield stress

F_D Force along drag strut

F_H Horizontal force

F_r Resultant force

F_{st} Torsional modulus of rupture

F_{su} Ultimate shear stress

F_{ty} Tensile yield stress

F_{tu} Ultimate tensile stress

F_v Vertical forces

I Moment of inertia

ℓ Length

M Moment

M_D Moment due to drag force

M_N Moment due to normal force

M_r Resultant moment

M.S. Margin of safety

P Force

R Radius

r Radius

S Shear

S_r Resultant shear force

S_t Shear due to torque

S_v Shear due to vertical force

T Torque

T_r Resultant torque

V Volume

W Weight, unit weight

ρ Radius of gyration

CHAPTER I

INTRODUCTION

In order to design the landing gear for any airplane certain facts must first be ascertained. Is the landing gear to be retracted, and if so, will it retract into the wing, nacelle, or fuselage? What are the loads imposed upon the gear and in what direction? What kind of retracting system (hydraulic, electric, or manual) will the gear have if it is retractable?

These questions and many others have been answered in the design of the main landing gear for the Jet Trainer the writer designed in 16.71 T. The front views of the gear before and after retraction are shown in Figures 1 and 2. The side view of the extended gear is shown in Figure 3. The gear is shown with the oleo strut in the fully extended condition so that maximum outside dimensions could be ascertained, and the maximum possible bending moments calculated. In this condition, with the oleo extended, the gear has been designed to retract into a space behind the air scoop for the engine. The fairing continues past the gear and then joins the fuselage farther back. These fairing lines were worked out in 16.71 T. The landing loads also were calculated then, and these will be used as one criterion for the design of the gear.

In order to facilitate calculation of loads, torques, and bending moments, Figure 4, was drawn, giving center line lengths and true angles.

CHAPTER II

The design of the main landing gear was begun by first deciding how far out on the wing the gear was to be fastened. The wing was designed to resist a certain maximum bending moment at its center due to a distributed Z force. Knowing this wing bending moment and the maximum normal force applied to the gear on landing and applying the formula:

Moment = Force x Distance

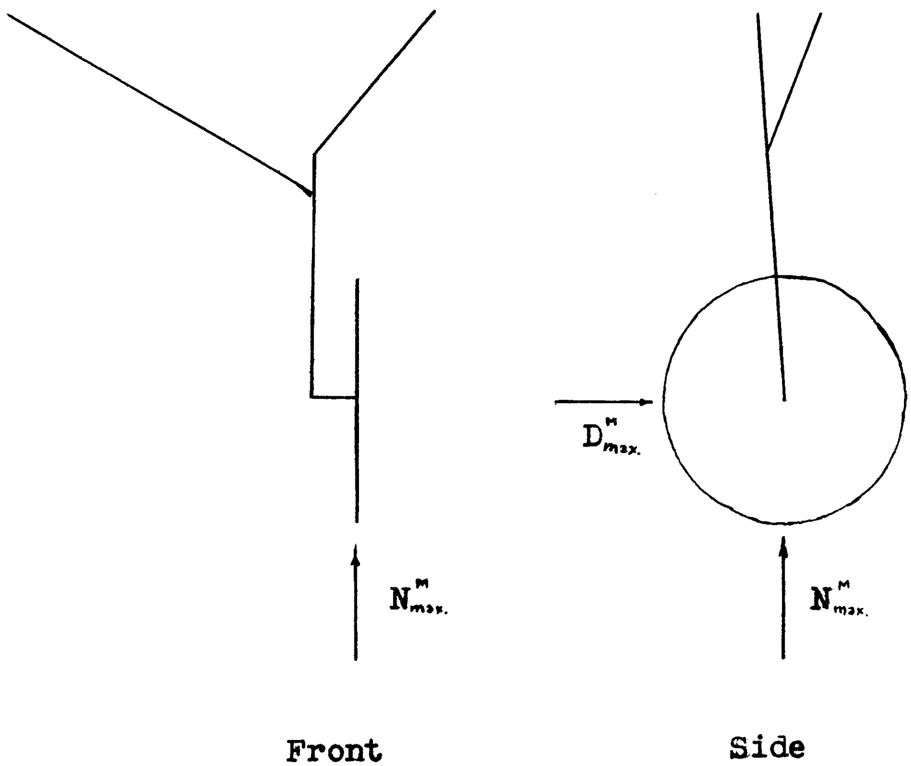
led to the result that the landing gear could be fixed approximately 47.5 inches outboard of the center of the wing without exceeding the maximum wing bending moment. Ultimate loads were used in calculating this position. The dimensions and position of the gear are such that it has a tread of about 80 inches and sufficient height so that the tail will clear the ground when the wing is rotated up to the stall angle.

The loads on the main gear as calculated in 16.71 T are as follows:

$N^{M}_{max.}$ = max. normal force = 7050 lbs.

D^{M}_{max} = max. drag force = 3670 lbs.

These loads are applied to the gear as shown below.



To facilitate calculations the applied loads are resolved into components perpendicular to, and along the strut.

Force along strut = $N^{M}_{max.} \cos 4^{\circ} - D^{M}_{max} \sin 4^{\circ}$

$(7050)(0.9976) - (3670)(0.0698) = 6774$ lbs.

limit load

Force perpendicular to strut = $D^{M}_{max} \cos 4^{\circ} + N^{M}_{max} \sin 4^{\circ}$

$(3670)(0.9976) + (7050)(0.0698) = 4152$ lbs.

limit load

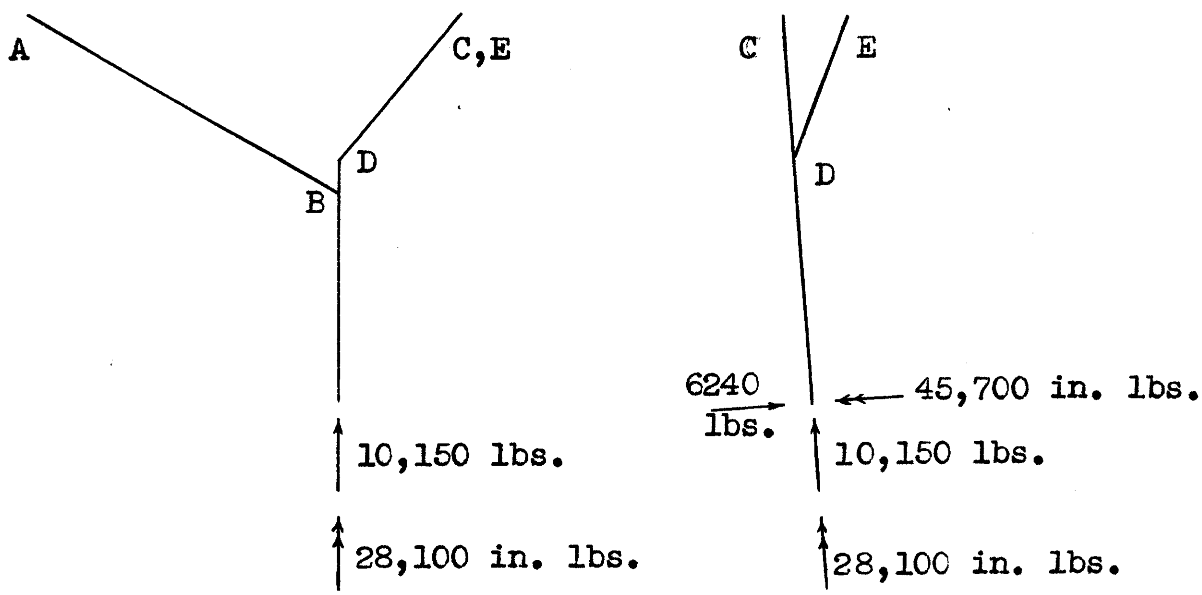
In order to provide ample strength for all landing conditions, ultimate loads will be used to design the gear.

Ultimate force along strut = (4152)(1.5) = 6240 lbs.

Ultimate force perpendicular to strut =

(6774)(1.5) = 10,150 lbs.

If theforces are moved from their position on the axle to the lower end of the strut, moments must be added. The results (with reactions at A, C, and E not shown) are:



The forces along AB and CD are found by applying the laws of statics:

$\Sigma F_v = 0$

1) AB sin 30° + CD sin 50° - 10,150 lbs. = 0

$\Sigma F_H = 0$

$$2)\ AB \cos 30^\circ - CD \cos 50^\circ = 0$$

From 2)

$$CD = AB \frac{\cos 30^\circ}{\cos 50^\circ} = AB \frac{(0.8660)}{(0.6428)} = 1.345\ AB$$

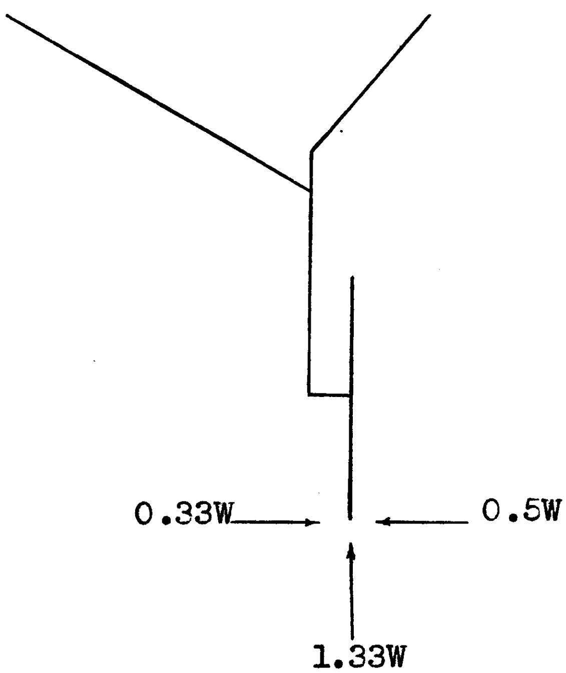
From 1)

$$AB\ (0.5000) + 1.345\ AB\ (0.7660) = 10{,}150$$

$\therefore AB = 6630$ lbs. ultimate

$\therefore CD = 8910$ lbs. ultimate

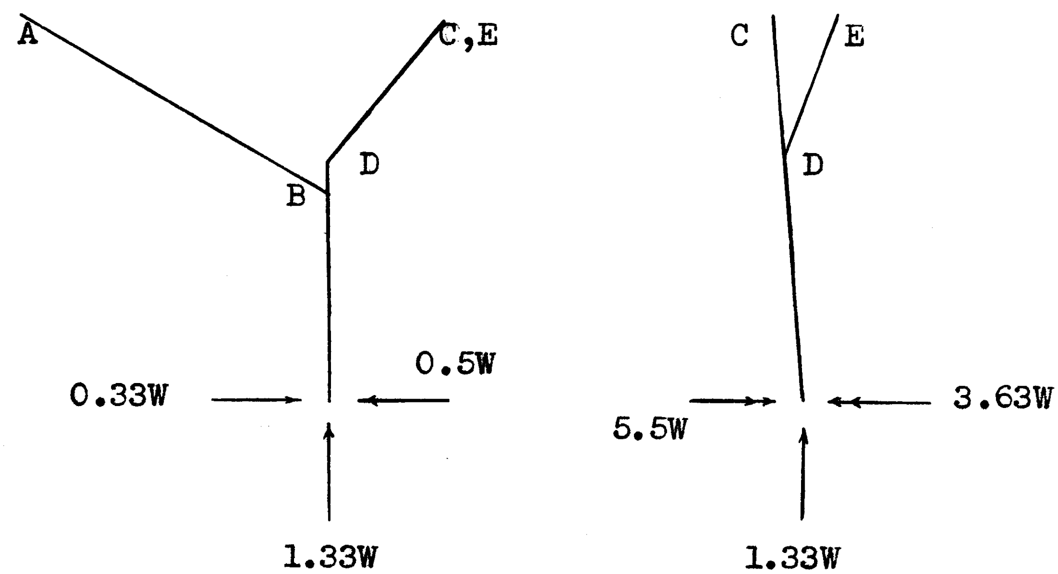
The C.A.A. specifies that the landing gear must resist certain side loads due to ground handling. These regulations can be summarized in a diagram.



Note:

These are limit loads.

The inboard and outboard loads are not meant to be applied simultaneously. These loads are to be applied with the landing gear in its static position — partially deflected oleo strut and a static design tire deflection. The oleo is considered to be deflected three inches and the tire, one inch. These side limit loads, as applied to the lower end of the strut, can be shown in a diagram similar to that drawn for the ultimate loads (Page 5).



The maximum forces along AB and CD due to side loads are found by laws of statics:

$$\Sigma F_v = 0$$

$$1)\ AB \sin 30^\circ + CD \sin 50^\circ - 1.33W = 0$$

$$\Sigma F_H = 0$$

$$2)\ AB \cos 30^\circ - CD \cos 50^\circ - 0.5W = 0$$

From 2)

$$AB\ (0.8660) - CD\ (0.6428) = 0.5W$$

$$AB = \frac{0.5W \quad CD\ (0.6428)}{(0.8660)} = (0.577)W + (0.742)\ CD$$

From 1)

$$(0.577W + 0.742\ CD)(0.5000) + CD\ (0.7660) = 1.33W$$

$$0.2885W + 0.371\ CD + 0.766\ CD = 1.33W$$

$$\therefore CD = 0.915W$$

$$\therefore AB = 1.257W$$

Since

$$W = 2110 \text{ lbs.}$$

the limit loads are

$$AB = 2660 \text{ lbs.}$$

$$CD = 1930 \text{ lbs.}$$

and so the ultimate loads are

$$AB = (2660)(1.5) = 3990 \text{ lbs.}$$

$$CD = (1930)(1.5) = 2900 \text{ lbs.}$$

The maximum moment on the strut due to these side loads is

$$5.5W \cos 4^\circ = (5.5)(2110)(0.998) = 11{,}600 \text{ in. lbs.}$$

The forces on the members due to side loads are much less than those imposed during landing. Therefore, the side loads will be ignored and the gear designed to withstand landing loads.

CHAPTER III

Now that we have the major forces, moments, and torques calculated, we can start designing and dimensioning the individual components of the gear. The major components such as the axle, oleo piston and cylinder tubes, etc., will be designed first, and the small fittings last. All the components will be designed to have a margin of safety in bearing equal to a minimum of one.

For this gear, heat treated 4140 steel will be used and heat treatment will take place after all welding has been completed. Since the components will all be less than two inches thick, the steel will have the following properties after heat treatment:

$F_{tu} = 180$ ksi

$F_{ty} = 165$ ksi

$F_{cy} = 165$ ksi

$F_{su} = 105$ ksi

$F_{bru} = 200$ ksi

$W = 0.283$ lbs./in^3

For the design of the landing gear, margins of safety based on ultimate unit stresses will be used. Sometimes the limit load might cause stresses greater than the yield point of the material and permanent set would then result from the application of the limit load. To prevent this, the ultimate stress must exceed the maximum stress due to calculated ultimate loads and the yield point must exceed the maximum due to limit loads. The latter situation need not be investigated if the yield point is greater than two-thirds of the corresponding ultimate stress. For the 4140 steel to be used for the fabrication of this gear, the yield point is approximately 92% of the ultimate stress.

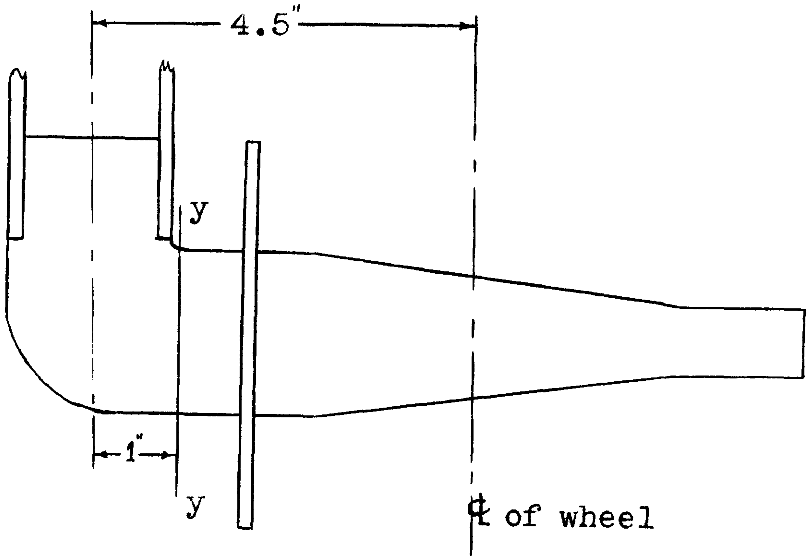
The thickness of the wing at the point of attachment of the main strut is approximately 3.3 inches. If a three inch tube is selected, there will be ample room for structure and clearance for rotation. The thickness at the point of attachment of the drag brace is approximately two inches and therefore, to provide room for structure and clearance for rotation here, a 1.5 inch tube will be used.

Since the oleo cylinder tube is to have an outside diameter of three inches, the oleo piston tube will have a two inch outside diameter to allow for the shock strut's internal components. With these restrictions in mind, the design of the components will begin with the axle forging.

The resulting force due to the drag and normal loads can be calculated:

$$F_r = \sqrt{(D^{n}_{max.})^2 + (N^{m}_{max.})^2} = \sqrt{(6240)^2 + (10,150)^2}$$

$$= 11,900 \text{ lbs.}$$



The maximum moment on the axle will occur at yy making the moment arm 3.5 inches.

$$M = (F)(d) = (11{,}900)(3.5) = 41{,}700 \text{ in. lbs.}$$

$$f_c = -\frac{My}{I}$$

$M = 41{,}700$ in. lbs.

$y = r =$ unknown radius of cross section

$$I = \frac{\pi r^4}{4}$$

$f_c = F_{cu} = F_{tu} = 180$ ksi (M.S. = 1)

$$180{,}000 = -\frac{(41{,}700)(y)}{\frac{\pi r^4}{4}} \quad -\frac{(41{,}700)(4)}{\pi r^3}$$

$$r^3 = \frac{(41{,}700)(4)}{(\pi)(180{,}000)} = 0.295$$

$$r = 0.669$$

$$D = 2r = 1.338 \text{ in. minimum}$$

In order to provide a positive margin of safety, this diameter will be 1.375" so that standard bearings may be used for the wheel.

$$f_c = -\frac{My}{I} = -\frac{(41,700)(0.688)}{\frac{(\pi)(0.688)^4}{4}} = -\frac{(41,700)(4)}{(\pi)(0.688)^3}$$

$$= \frac{-53,100}{(0.688)^3} = \frac{-53,100}{0.325} = -163,000 \text{ lbs.}$$

$$M.S. = \frac{F_{cu}}{f_c} - 1 = \frac{-180,000}{-163,000} - 1 = 0.11$$

The axle will be welded to the oleo piston tube in an approved manner. For the design of this tube, we will assume the following:

1) D = 2 inches

2) l = 11 inches (6 inch stroke + 3 inches inside piston + 2 inches outside for welding to the axle forging).

The maximum moment, at the point of entrance of the piston tube into the cylinder tube, will occur when the strut is fully extended. The drag force will have a moment arm of 8.69 inches and the normal force an arm of 4.5 inches.

$$M_D = (6240)(8.69) = 54,300 \text{ in. lbs.}$$

$$M_N = (10,150)(4.5) = 45,700 \text{ in. lbs.}$$

$$M_r = \sqrt{(54,300)^2 + (45,700)^2}$$

$$= 10^4\sqrt{29.5 + 20.9} = 10^4\sqrt{50.40} = 71,000 \text{ in. lbs.}$$

By trial and error, using c (end restraint coefficient) equal to 1:

$t = 0.156$ in.

$$\frac{D}{t} = \frac{2}{0.156} = 12.80$$

$$\frac{l}{\rho} = \frac{l'}{\rho} = \frac{11}{0.6542} = 16.8$$

$$\frac{I}{y} = 0.3873 \text{ in.}^3$$

$$A = 0.9050 \text{ in.}^2$$

$$F_{tu} = 180 \text{ ksi}$$

$F_{cc} = -\ 144$ ksi (Figure 2.23 c, ANC - 5)

$F_b = -\ (1.38)(180{,}000) = -\ 248{,}000$ psi

(Figure 2.321, ANC - 5)

$$f_b = -\frac{M}{I/y} = -\frac{71{,}000}{0.3873} = -\ 184{,}000 \text{ psi}$$

$$f_c = -\frac{F}{A} = -\frac{10{,}150}{0.9050} = -\ 11{,}200 \text{ psi}$$

$$R_b = \frac{f_b}{F_b} = \frac{184{,}000}{248{,}000} = 0.743$$

$$R_c \quad \frac{f_c}{F_{cc}} = \frac{11{,}200}{144{,}000} = 0.078$$

$$M.S. = \frac{1}{R_b + R_c} - 1 = 0.22$$

Using t = 0.120 in. leads to M.S. = - 0.25

In the design of the piston tube, all torques have been neglected since a scissors attachment will be added later to tansmit the torques from the axle to the oleo cylinder tube. For this tube, the maximum moment due to the drag load will be at the point where the strut makes its 40° bend.

$$M_D = (6240)(21.9) = 137{,}000 \text{ in. lbs.}$$

A constant moment also exists due to the normal force on the axle:

$$M_N = 45{,}700 \text{ in. lbs.}$$

The resultant moment is the vector sum of these two moments.

$$M_r = \sqrt{(137{,}000)^2 + (45{,}700)^2} = 144{,}500 \text{ in. lbs.}$$

Using the same technique as was used on the piston tube:

$$t = 0.156 \text{ in.}$$

$$\frac{D}{t} = \frac{3}{0.156} = 19.2$$

$$\frac{l'}{\rho} = \frac{12}{1.007} = 12$$

$$\frac{I}{y} = 0.9436 \text{ in.}^3$$

$$A = 1.3959 \text{ in.}^2$$

$$F_{tu} = 180 \text{ ksi}$$

$$F_{cc} = -\ 144 \text{ ksi (Figure 2.23 c, ANC - 5)}$$

$$F_b = -\ (1.21)(180{,}000) = -\ 218{,}000 \text{ psi}$$

(Figure 2.321, ANC - 5)

$$F_{st} = (0.585)(180{,}000) = 111{,}000 \text{ psi}$$

$$f_c = -\ \frac{10{,}150}{1.3959} = -\ 7300 \text{ psi}$$

$$f_b = -\ \frac{M}{I/y} = -\ \frac{144{,}500}{0.9436} = -\ 153{,}500 \text{ psi}$$

$$f_s = \frac{T}{2(I/y)} = \frac{28{,}100}{2(0.9436)} = 15{,}400 \text{ psi}$$

$$R_c = \frac{f_c}{F_{cc}} = \frac{7300}{144,000} = 0.051$$

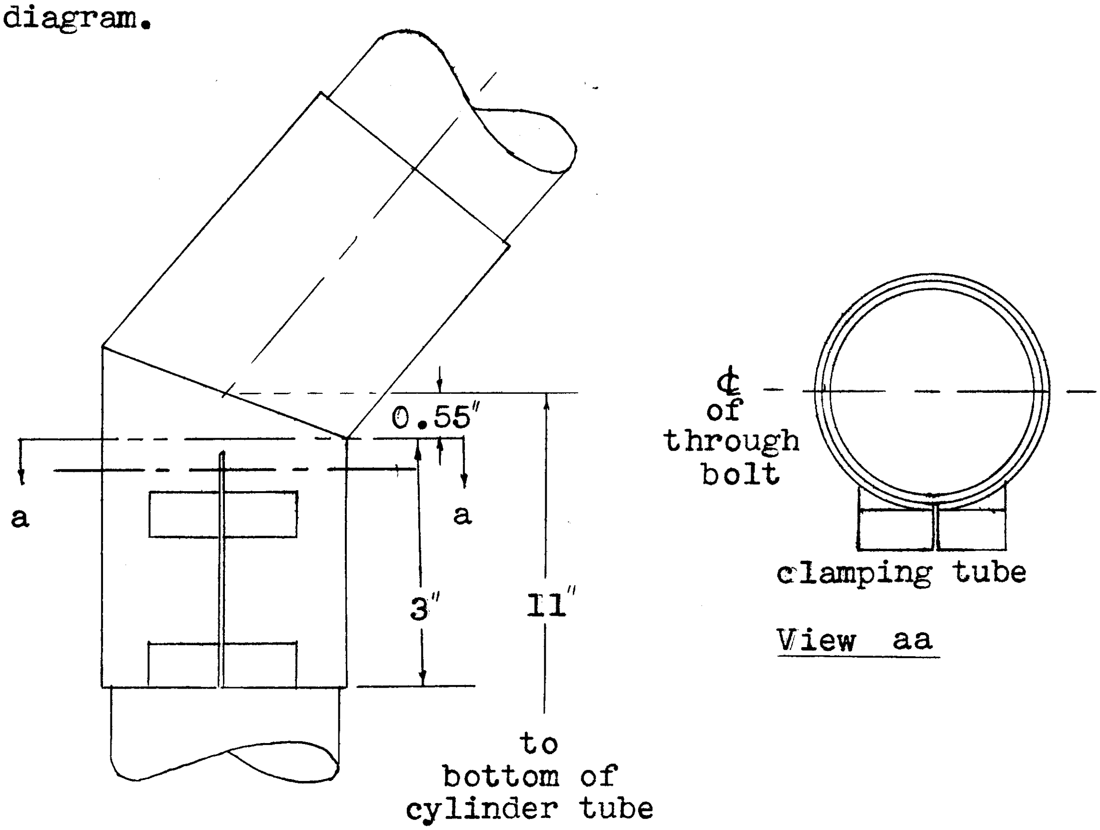
$$R_b = \frac{f_b}{F_b} = \frac{153,500}{218,000} = 0.705$$

$$R_s = \frac{f_s}{F_{st}} = \frac{15,400}{111,000} = 0.139$$

$$M.S. = \frac{1}{R_c + \sqrt{R_b^2 + R_s^2}} - 1 = \frac{1}{0.051 + \sqrt{(0.705)^2 + (0.139)^2}} - 1$$

$$= 0.305$$

In order to be able to grind parts of the shock strut after welding, the strut will be made separate from the rest of the gear and then bolted and clamped to the 40° elbow. This would also allow for replacement of parts when necessary. The bolting and clamping process is shown in the following diagram.



The elbow will be 3.00 inches inside diameter and 3.25 inches outside diameter. It would probably be fabricated in two pieces of tubing welded together to form the 40 angle. The clamping tubes would then be welded on, and the assembly slit in the proper place.

For this elbow:

$$t = 0.125 \text{ in.}$$

$$\frac{D}{t} = \frac{3.25}{0.125} = 26$$

$$\frac{I}{y} = 0.8906 \text{ in.}^3$$

$$A = 1.180 \text{ in.}^2$$

$$M_r = 144{,}500 \text{ in. lbs.}$$

$$F_{tu} = 180 \text{ ksi}$$

$$F_{cc} = 144 \text{ ksi}$$

$$F_b = 1.12(180{,}000) = 202{,}000 \text{ psi}$$

$$F_{st} = 0.570(180{,}000) = 102{,}500 \text{ psi}$$

$$f_c = -\frac{10{,}150}{1.180} = -8600 \text{ psi}$$

$$f_b = -\frac{M}{I/y} = -\frac{144{,}500}{0.8906} = -162{,}000$$

$$f_s = \frac{T}{2(I/y)} = \frac{28{,}100}{2(0.8906)} = 15{,}800 \text{ psi}$$

$$R_c = \frac{f_c}{F_{cc}} = \frac{8600}{144{,}000} = 0.06$$

$$R_b = \frac{f_b}{F_b} = \frac{162{,}000}{202{,}000} = 0.803$$

$$R_s = \frac{f_s}{F_{st}} = \frac{15{,}800}{102{,}500} = 0.154$$

$$M.S. = \frac{1}{R_c + \sqrt{R_b^2 + R_s^2}} - 1$$

$$\frac{1}{0.06 + \sqrt{(0.803)^2 + (0.154)^2}} - 1 = 0.14$$

In order to be able to disregard the slit in the elbow, the two clamping bolts must be able to withstand the ultimate tension that the slit area could have resisted.

Area of slit = (t)(l) = (0.120)(2.5) = 0.30 in.

$F_{tu} = 180$ ksi

$f_{tu} = 0.30\ (180{,}000) = 54{,}000$ lbs.

To develpp the ultimate tensile strength necessary to resist the 54,000 lbs. ultimate load, two NAS - 149 bolts will be used, each having an ultimate tensile strength of 29,800 lbs.

The clamping tubes, welded in place before slitting, will be 0.75 outside diameter with a wall thickness of 0.120 in. After welding, this inside hole will be reamed to accomodate the 9/16 inch bolt, which will reduce its wall thickness to 0.094 in. Will it resist the ultimate compressive stress imposed on it?

$$A = \pi(r_o^2 - r_i^2)$$

r_o = outside radius

r_i = inside radius

$$A = \pi\left[\left(\frac{0.75}{2}\right)^2 - \left(\frac{0.562}{2}\right)^2\right] = 0.198 \text{ in.}$$

$$f_c = \frac{54,000/2}{0.198} = 136,000 \text{ psi}$$

$$F_{cc} = 180,000 \text{ psi}$$

$$M.S. = \frac{180,000}{136,000} - 1 = 0.32$$

The through bolt is put in to prevent relative motion between the parts. It carries a resultant shear due to the torque and vertical force. The bolt is in double shear, but the following are shears at each of the shear-carrying sections of the bolt.

Shear due to torque

$$S_t = \frac{T}{d} = \frac{28,100}{3} = 9370 \text{ lbs.}$$

Shear due to vertical force at each shear surface

$$S_v = \frac{10,150}{2} = 5,075 \text{ lbs.}$$

$$S_r = \sqrt{(9370)^2 + (5,075)^2} = 10,700 \text{ lbs.}$$

Using a NAS - 148 bolt ($\frac{1}{2}$ in. diameter)

Allowable single shear = 18,650 lbs.

$$M.S. \text{ (in shear)} = \frac{18,650}{10,700} - 1 = 0.74$$

The addition of the through bolt adds a bearing stress at the holes through which it passes. An investigation must be made to determine the bearing stresses at these holes in the oleo cylinder tube and in the elbow.

In the cylinder tube:

$$f_{br} = \frac{P}{dt} = \frac{10,700}{(0.5)(0.156)} = 137,000 \text{ psi}$$

$$F_{bru} = 200,000 \text{ psi}$$

Thus the margin of safety in bearing is less than one, which was set as the minimum. In order to increase this margin, a 0.065 in. thick tube is welded inside the upper end of the cylinder tube to increase its wall thickness. Then

$$f_{br} = \frac{10,700}{(0.5)(0.156 + 0.065)} = 97,000 \text{ psi}$$

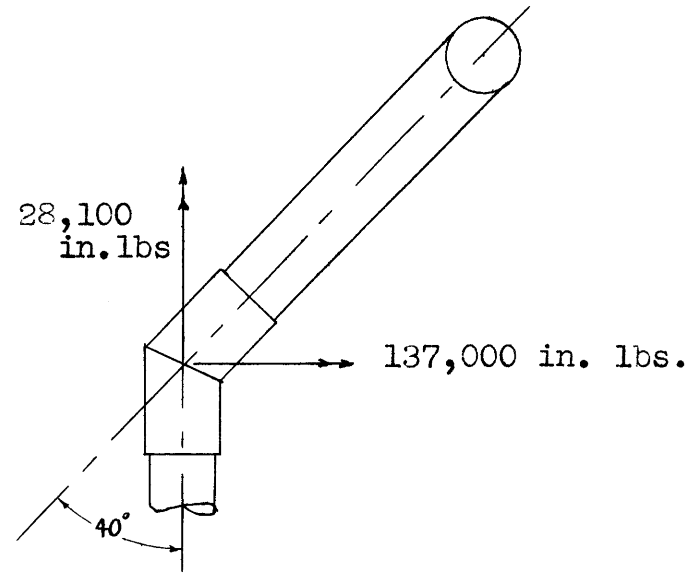
$$M.S. = \frac{200,000}{97,000} - 1 = 1.06$$

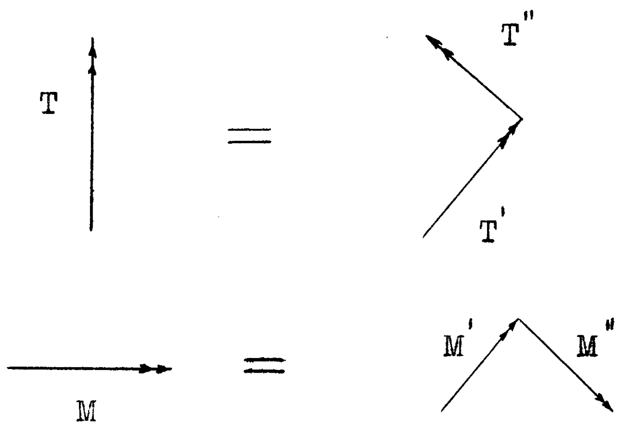
For the elbow, two circular patches will be welded on and the holes continued through them. These patches are to be 0.095 in. thick.

$$f_{br} = \frac{10,700}{(0.5)(0.120 + 0.095)} = 99,500 \text{ psi}$$

$$M.S. = \frac{200,000}{99,500} - 1 = 1.01$$

Before analysis of the main strut of the gear can be completed, the forces, moments, and torques, perpendicular to, and along the upper tube must be computed. The following torque and moment will be resolved. All others can be used as they are.





$T' = T \cos 40^\circ = (28{,}100)(0.766) = 21{,}600$ in. lbs.

$T'' = T \sin 40^\circ = (28{,}100)(0.643) = 18{,}100$ in. lbs.

$M' = M \cos 50^\circ = (137{,}000)(0.643) = 88{,}200$ in. lbs.

$M'' = M \sin 50^\circ = (137{,}000)(0.766) = 105{,}000$ in. lbs.

T_r along strut $= T' + M' = 21{,}600 + 88{,}200 = 109{,}800$ in. lbs.

M_r perpendicular to strut $= M'' - T'' = 105{,}000 - 18{,}100$ $= 86{,}900$ in. lbs.

Combining the above moment with the constant moment of 45,700 in. lbs.

$$M_r = \sqrt{(45{,}700)^2 + (86{,}900)^2} = 98{,}000 \text{ in. lbs.}$$

The proceedure for selection of a tube size is similar to what was used previously.

$$t = 0.156$$

$$\frac{D}{t} = \frac{3}{0.156} = 19.2$$

$$\frac{L'}{\rho} = \frac{15.55}{1.007} \cong 15.50$$

$$\frac{I}{y} = 0.9436 \text{ in.}^3$$

$$A = 1.3959 \text{ in.}^2$$

$$F_{tu} = 180 \text{ ksi}$$

$$F_{cc} = 144 \text{ ksi}$$

$$F_b = (1.21)(180{,}000) = 218{,}000 \text{ psi}$$

$$F_{st} = (0.585)(180{,}000) = 111{,}000 \text{ psi}$$

$$f_c = \frac{8910}{1.3959} = 6400 \text{ psi}$$

$$f_b = -\frac{M}{I/y} = \frac{98{,}000}{0.9436} = 104{,}000 \text{ psi}$$

$$f_s = \frac{T}{2(I/y)} = \frac{109{,}800}{2(0.9436)} = 58{,}200 \text{ psi}$$

$$R_c = \frac{6400}{144{,}000} = 0.045$$

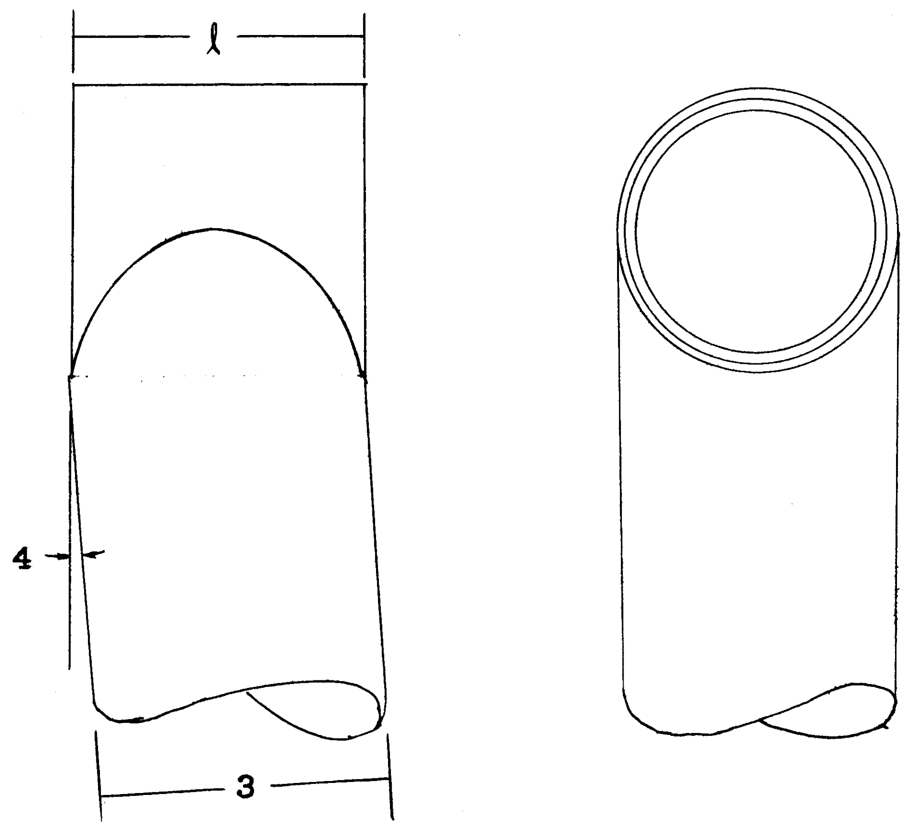
$$R_b = \frac{104{,}000}{218{,}000} \approx 0.478$$

$$R_s = \frac{58{,}200}{111{,}000} = 0.525$$

$$M.S. = \frac{1}{R_c + \sqrt{R_b^2 + R_s^2}} - 1 = \frac{1}{0.045 + \sqrt{(0.478)^2 + (0.525)^2}} - 1$$

M.S. = 0.33

Since rotation of the main strut takes place at the upper end, a bearing surface must be provided.



$$l = \frac{3}{\cos 4^{\circ}} = \frac{3}{0.998} \cong 3 \text{ in.}$$

The bushing material is aluminim bronze and is made 0.187 in. thick, so that it will be a push fit on a 2.625 inch shaft. The bushing will be subject to a bearing force only. Therefore it is necessary to check the margin of safety in bearing.

$$f_{br} = \frac{P}{dt} = \frac{8910}{(2.625)(3)} = 1130 \text{ psi}$$

F_{bru}(Al bronze) = 80,000 psi

$$M.S. = \frac{80,000}{1130} - 1 = 69.8$$

The drag tube is designed to take compression only, since all torques are taken by the main strut.

F_D = Force along drag strut

$$F_D (\sin 17^\circ)(15.55 \sin 50^\circ) = 6240(21.9 + 15.55 \sin 50^\circ)$$

F_D = 60,700 lbs.

t = 0.120 in.

A = 0.5202 in.^{2}

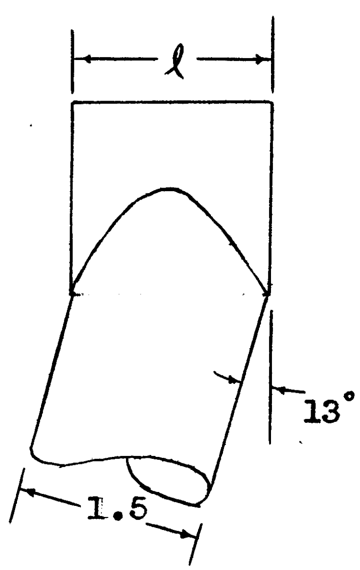
$$\frac{l'}{\rho} = \frac{15.95}{0.4898} = 32.6$$

F_{cc} = 140,000 psi

$$f_c = \frac{60,700}{0.5202} = 117,000 \text{ psi}$$

$$M.S. = \frac{140,000}{117,000} - 1 = 0.195$$

Rotation takes place at the upper end of the drag strut so an aluminum bronze bushing will be provided.



$$\ell = \frac{1.5}{\cos 13^\circ} = 1.54 \text{ in.}$$

To provide rotation on a one inch shaft, the bush ing must be 0.130 in. thick.

$$f_{br} = \frac{P}{dt} = \frac{(60,700)(\cos 13^\circ)}{(1)(1.54)} = \frac{(60,700)(0.974)}{1.54}$$

$$= 38,400 \text{ psi}$$

$$F_{bru} = 80,000 \text{ psi}$$

$$M.S. = \frac{80,000}{38,400} - 1 = 1.08$$

For retraction, an electrically operated unit manufactured by Airbourne Accessories Corp., Hillside, New Jersey, will be used. This linear actuator is known by the manufacturer as a "lineator", Model R - 550. (Figure 5) This model will be used due to its ability to resist the ultimate of 6630 pounds. It was designed to withstand a static load of 7500 pounds. In order that the maximum operating load of 2500 pounds is not exceeded, a proper geometry for the location of the unit must be observed. To determine this geometry, the center of gravity of the gear must be calculated.

Axle

$$V \cong \pi r^2 \ell = \pi \left(\frac{1.375}{2}\right)^2 \times 6 = 8.8 \text{ in.}^3$$

$$W = (8.9)(0.283) = 2.52 \text{ lbs.}$$

Oleo piston tube

$$W = (A)(\ell)(\text{unit wt.}) = (0.9050)(11)(0.283)$$

$W = 2.82$ lbs.

Oleo cylinder tube

$W = (1.3959)(12)(0.238) = 4.75$ lbs.

Elbow

$W = (1.180)(7)(0.283) = 2.34$ lbs.

Upper tube

$W = (1.3959)(15.55)(0.283) = 6.14$ lbs.

Drag tube

$W = (0.5202)(15.95)(0.283) = 2.35$ lbs.

Volume of oil in shock strut

= internal volume of cylinder - volume of piston

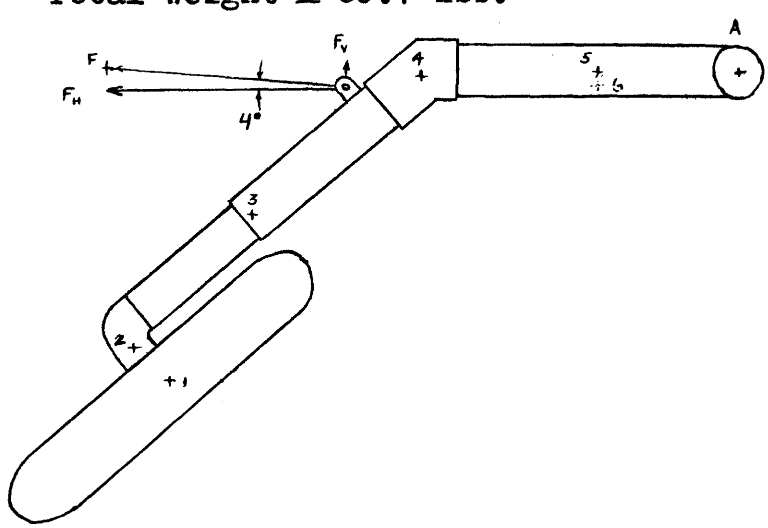
$$V = \pi\left(\frac{2.688}{2}\right)^2 (12) - 9.95 = 58.25 \text{ in.}^3$$

W (of oil) $= (58.25)(0.0325) = 1.9$ lbs.

Total for calculated parts = 22.82 lbs.

Add 10% for welds, 10% for small fittings, 10% for internal parts of the shock strut, and 30 pounds for the wheel and brake.

Total weight ≅ 59.7 lbs.



$$\sum M_A = W_1 l_1 + W_2 l_2 + W_3 l_3 + W_4 l_4 + W_5 l_5 + W_6 l_6 + 20\%W\ x = W_t\ x$$

l = perpendicular distance from the center of gravity of the component to a vertical line through A.

$$\sum M_A = 30(31.5) + 2.52(32.2) + 11.75(25.1) + 2.34(16.1)$$
$$+ 6.14(8.1) + 2.35(8.1) + 2(2.28)(x) = 59.7\ x$$

Therefor:

$$x = \frac{1417.5}{55.14} = 26.7 \text{ in.}$$

The distance from F_v to A $= 16.1 + 4 \cos 40^\circ = 18.7$ in.

$$F_v (18.7) = 59.7(26.7) = 1595 \text{ in. lbs.}$$

$$F_v = F \sin 4^\circ = \frac{1595}{18.7} = 85 \text{ lbs.}$$

$$F = \frac{F_v}{\sin 4^\circ} = \frac{85}{0.070} = 1220 \text{ lbs.}$$

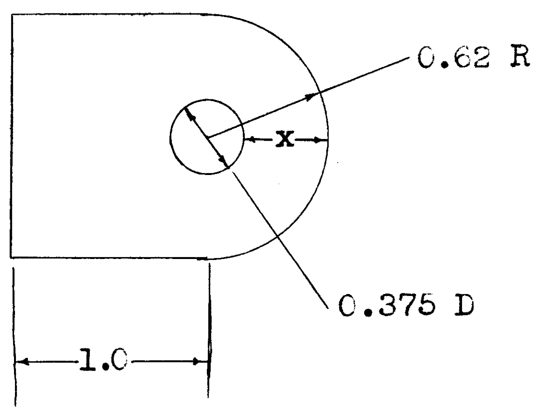
The maximum operating load for the strut is 2500 lbs. which leaves an operating margin of safety of:

$$\frac{2500}{1220} - 1 = 1.25$$

CHAPTER IV

Various small fittings, such as for the attachment of the Lineator to the gear, must now be designed.

The end fitting on the Lineator is 0.625 in. thick. Therefor to make the attachment fittings seem properly proportioned, they have the following design.



$t = 0.25$ in.

$$f_{br} = \frac{P}{dt} = \frac{6630/2}{0.375(0.25)} = 35{,}400 \text{ psi}$$

$$F_{br} = 200{,}000 \text{ psi}$$

$$M.S. = \frac{200{,}000}{35{,}400} - 1 = 4.65$$

Check the fitting for tear-out

$$f_s = \frac{P}{2tx}$$

x = distance from edge of hole to end of fitting

P = side load due to ground handling.

$$f_s = \frac{3990/2}{2(0.25)(0.432)} = 9{,}220 \text{ psi}$$

$$F_{su} = 105{,}000 \text{ psi}$$

$$\text{M.S.} = \frac{105{,}000}{9{,}220} - 1 = 10.4$$

Checking the fitting for compressive and tensile stress due to bending:

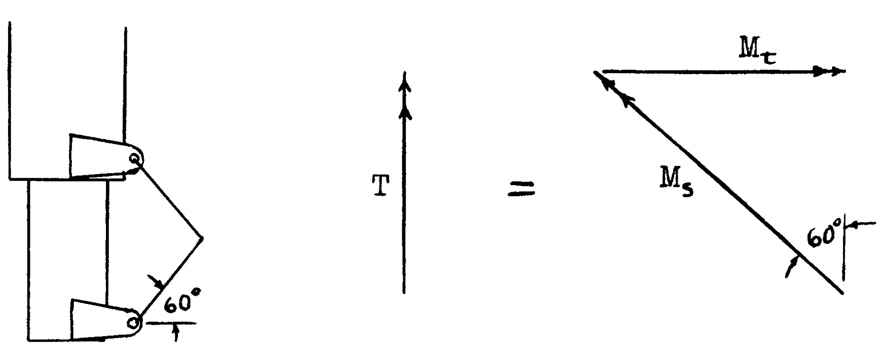
$$f_b = \frac{My}{I} = \frac{\frac{1}{2}(F_N)(\cos 40^\circ)(0.62)}{0.041}$$

$$= \frac{\frac{1}{2}(F \cos 4^\circ)(\cos 40^\circ)(0.62)}{0.041}$$

$$= \frac{(610)(0.998)(0.766)(0.62)}{0.041} = 17{,}050 \text{ psi}$$

$$\text{M.S.} = \frac{180{,}000}{7{,}050} - 1 = 24.5$$

In order to transmit the torque due to the drag load from the axle to the oleo cylinder tube, a scissors arrangement is provided. The scissors arrangement is designed to transmit a bending moment only. Therefore the torque vector is resolved into a moment taken out by the scissors and onother moment resisted by the stiffness of the oleo piston tube.



$M_s =$ moment taken by scissors $= \frac{T}{\cos 60^\circ}$

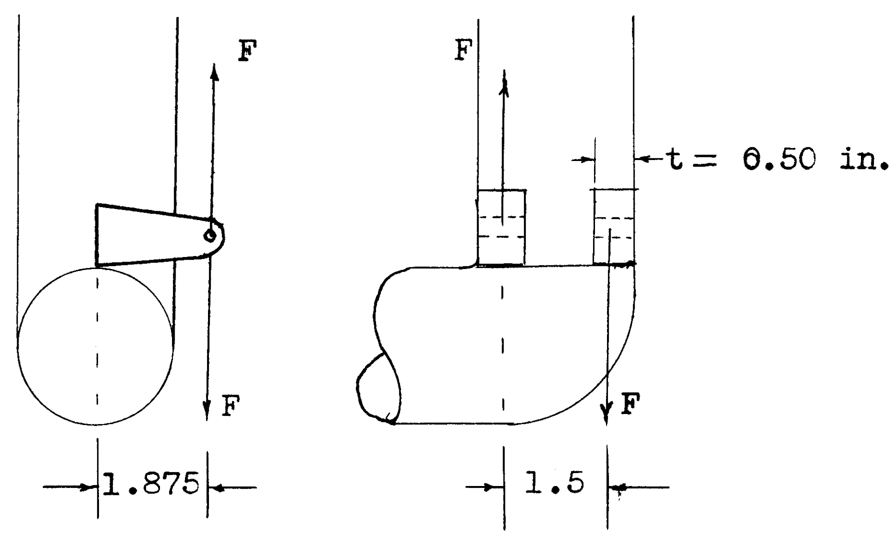
$= \frac{28,100}{0.5} = 56,200$ in. lbs.

$M_t =$ moment taken by oleo piston tube $= T \tan 60^\circ$

$= 28,100(1.732) = 48,700$ in.lbs.

The above moment of 48,700 inch pounds is in such direction as to oppose the moment caused by the normal force acting at the tire center line, but because the landing gear may meet unusual landing conditions (such as the airplane stalling a short distance above the ground), this relieving moment is ignored.

The fittings that hold the scissors arrangement must be strong enough to resist the maximum moment acting on them.

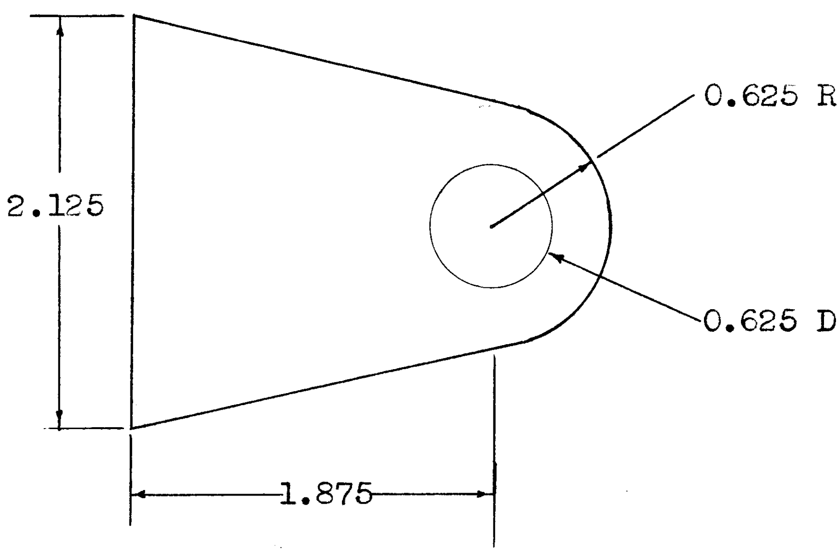


Side view Rear view

The moment of 48,700 in. lbs. is resisted by shear forces which are assumed to be acting at the center line of each fitting.

$$F = \frac{48,700}{1.5} = 32,500$$

This shear force leads to a compressive stress on one side and a tensile stress on the opposite side of each fitting. These stresses are a maximum at the point of attachment of the fittings to the strut. To resist these increasing stresses the fitting is tapered as viewed from the side. The dimensions are as follows:



$$M = F\,(1.875) = 32,500(1.875) = 61,000 \text{ in. lbs.}$$

$$f_c = -\frac{My}{I} = -\frac{61,000(2.125/2)}{0.400} = -162,000 \text{ psi}$$

$$F_{cc} = -\ 180,000 \text{ psi}$$

$$M.S. = \frac{180,000}{162,000} - 1 = 0.11$$

The shear that is resisted by the bolt attaching the scissors to the fitting is:

$$S = \frac{56,000}{2} = 28,000 \text{ lbs. (single shear)}$$

To resist this shear requires two NAS 150 bolts (0.625 inches in diameter).

$$\text{Bolt M.S.} = \frac{58,300}{2(28,100)} - 1 = 0.03$$

Checking the fitting for sufficient safety in bearing:

$$f_{br} = \frac{P}{dt} = \frac{28,100}{(0.625)(0.50)} = 90,000 \text{ psi}$$

$$F_{br-u} = 200,000 \text{ psi}$$

$$\text{M.S.} = \frac{200,000}{90,000} - 1 = 1.22$$

The distance from the edge of the hole to the edge of the fitting is determined by the tear-out condition. Assume the edge distance (x) is 0.3125 (5/16 in.).

$$f_s = \frac{P}{2tx} = \frac{28,100}{2(0.5)(0.3125)} = 90,000 \text{ psi}$$

$$F_{su} = 105,000 \text{ psi}$$

$$\text{M.S.} = \frac{105,000}{90,000} - 1 = 0.17$$

An additional condition to check is the tensile stress of the material on either side of the hole due to direct outward pull on the bolt.

$$f_t = \frac{F}{A} = \frac{(28,100)(\cos 60\ )}{2\ (0.3125)(0.5)} = 45,000 \text{ psi}$$

$$F_{tu} = 180,000 \text{ psi}$$

$$\text{M.S.} = \frac{180,000}{45,000} - 1 = 3.0$$

The fittings on the oleo cylinder tube resist the same moment of 48,700 inch pounds but the shear force in them is less, due to the fact that they are 2.5 inches apart.

$$F = \frac{48,700}{2.5} = 19,500$$

Therefore the maximum stress at the outer fibre of the fitting is (t = 0.375 in.):

$$f_c = -\frac{My}{I} = \frac{-(19,500)(1.875)(2.125/2)}{0.300} = -130,000 \text{ psi}$$

$$M.S. = \frac{180,000}{130,000} - 1 = 0.38$$

Since the same bolt is being used here as was used in the lower fitting:

$$f_{br} = \frac{P}{dt} = \frac{19,500}{(0.625)(0.375)} = 83,300 \text{ psi}$$

$$M.S. = \frac{200,000}{83,300} - 1 = 1.4$$

Checking for tear-out:

$$f_s = \frac{P}{2tx} = \frac{19,500}{2(0.375)(0.3125)} = 83,300 \text{ psi}$$

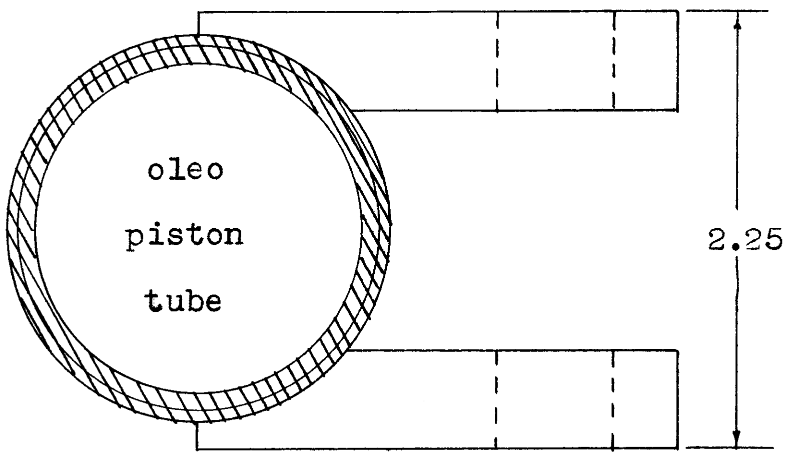
$$M.S. = \frac{105,000}{83,300} - 1 = 0.26$$

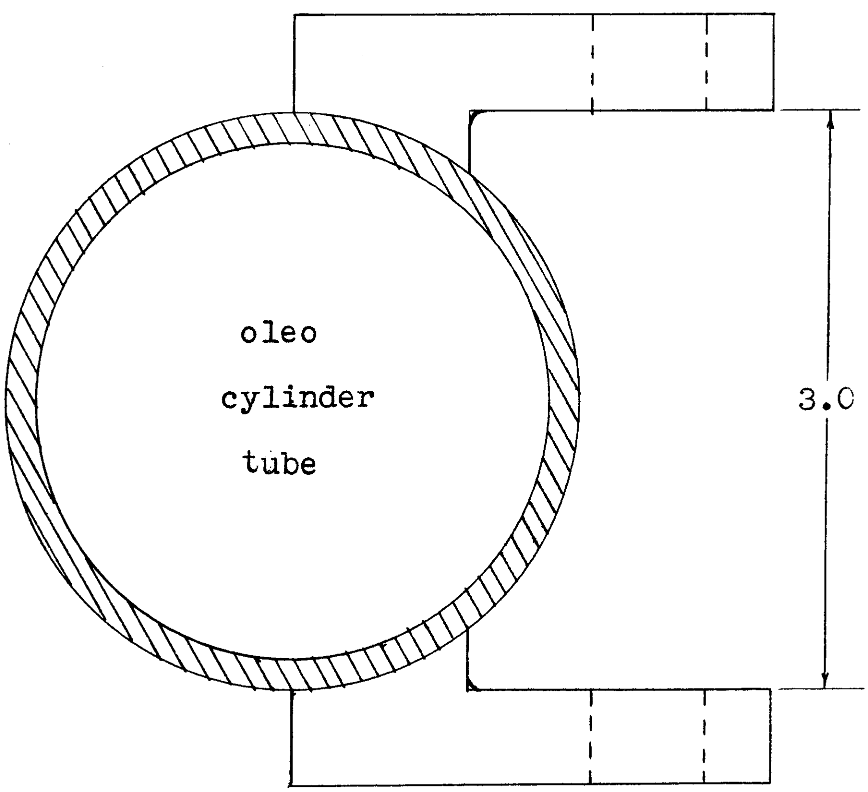
Checking tensile stress:

$$f_t = \frac{F}{A} = \frac{(19,500)(\cos 60)}{2(0.3125)(0.375)} = 41,600 \text{ psi}$$

$$M.S. \quad \frac{180,000}{41,600} - 1 \quad 3.33$$

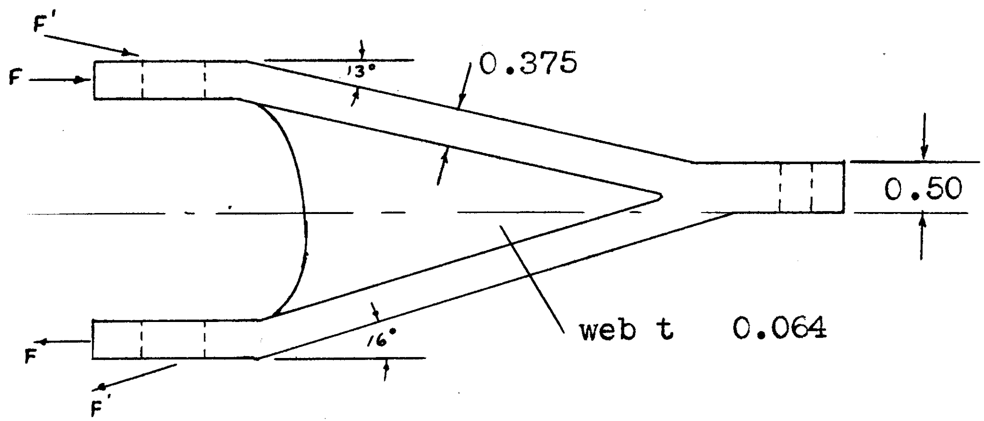
The arrangement of these fittings on their appropriate tubes are as follows:

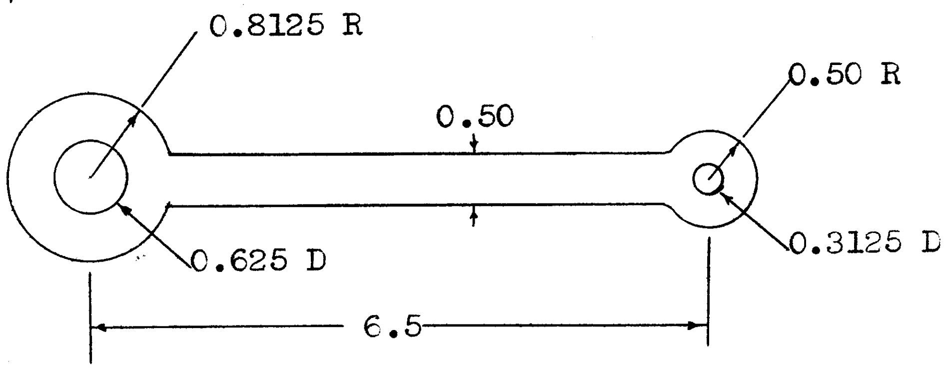




This arrangement allows for the two halves of the scissors to be made exactly alike, the upper section fitting inside the cylinder tube fittings, and the lower section fitting on the outside of the piston tube fittings.

Each scissors section has the following design:





$$F = \frac{M_s}{2.625} = \frac{56,200}{2.625} = 21,300 \text{ lbs.}$$

$$F'_{max} = \frac{F}{\cos 16^\circ} = \frac{21,300}{0.961} = 22,200 \text{ lbs.}$$

Assume the area resisting the force F is 0.1875 square inches.

$$f_c = \frac{P}{A} = \frac{22,200}{0.1875} = 119,000 \text{ psi}$$

The edge distance around the 0.625 inch hole is 0.50 in. Investigation is made to find out if this is enough to resist the applied loads.

Checking bearing stress

$$f_{br} = \frac{P}{dt} = \frac{21,300}{0.625(0.375)} = 91,000 \text{ psi}$$

$$M.S. = \frac{200,000}{91,000} - 1 = 1.20$$

Checking tear-out

$$f_s = \frac{P}{2tx} = \frac{21,300}{2(0.375)(0.50)} = 57,000 \text{ psi}$$

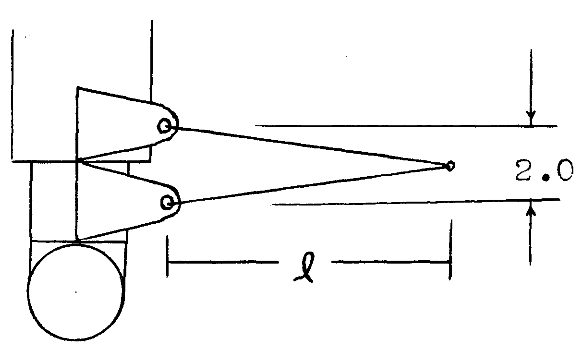
$$M.S. = \frac{105,000}{57,000} - 1 = 0.84$$

Checking tensile stress

$$f_t = \frac{F}{A} = \frac{21,300}{2(0.375)(0.5)} = 57,000 \text{ psi}$$

$$M.S. = \frac{180,000}{57,000} - 1 = 2.16$$

The maximum force on the bolt holding the two halves of the scissors together at the small end occurs when the oleo is as compressed as possible.



$\ell = 6.5 \cos(\sin 1/6.5) = 6.5\ (0.988) = 6.42$

Tensile stress in bolt $= \frac{T/\cos(\sin 1/6.5)}{\ell}$

$= \frac{28,100}{(6.42)(0.988)} = 4430$ lbs.

Use a NAS 145 bolt (5/16 in. bolt)

$$M.S. = \frac{8200}{4430} - 1 = 0.85$$

The application of tension on the bolt leads to bending and shear at this end of the scissors which are maximums where the area is a minimum. This minimum area occurs at the point where the outside dimension begins to increase due to the 0.50 inch radius, about 0.50 inches inboard from the bolt center.

$$f_s = \frac{S}{A} = \frac{4430}{0.1875} = 23,700 \text{ psi}$$

$$M.S. = \frac{105,000}{23,700} - 1 = 3.43$$

Checking stress due to bending:

$$f_c = - \frac{My}{I}$$

$$M = (4430)(0.50) = 2215 \text{ in.lbs.}$$

$$y = 0.50/2 = 0.25 \text{ in.}$$

$$I = \frac{1}{12}(0.5)(0.5)^3 = 0.0052 \text{ in.}^4$$

$$f_c = - \frac{(2215)(0.25)}{0.0052} = 106,500 \text{ psi}$$

$$M.S. = \frac{180,000}{106,500} - 1 = 0.69$$

CHAPTER V

The design of the landing gear consisted of stress analysis of the various components. The writer is aware that lubrication must be provided for at the joints where relative motion exists. It would be a simple matter to just go ahead and provide grease fittings at each joint, but would that be the best way? The writer feels that some knowledge of the lubrication requirements for different types of rubbing surfaces must be acquired before the grease fittings are provided. In industry, there are personnel in each company who can provide this knowledge and therefor, with their advice, the last details can be included in the specifications and design.

REFERENCES

1) ANC-5 Strength of Metal Aircraft Elements, U.S. Government Printing Office, Washington 25, D.C.

2) Niles, A.S. and Newell, J.S., Airplane Structures Vol. 1, John Wiley and Sons, Inc., New York

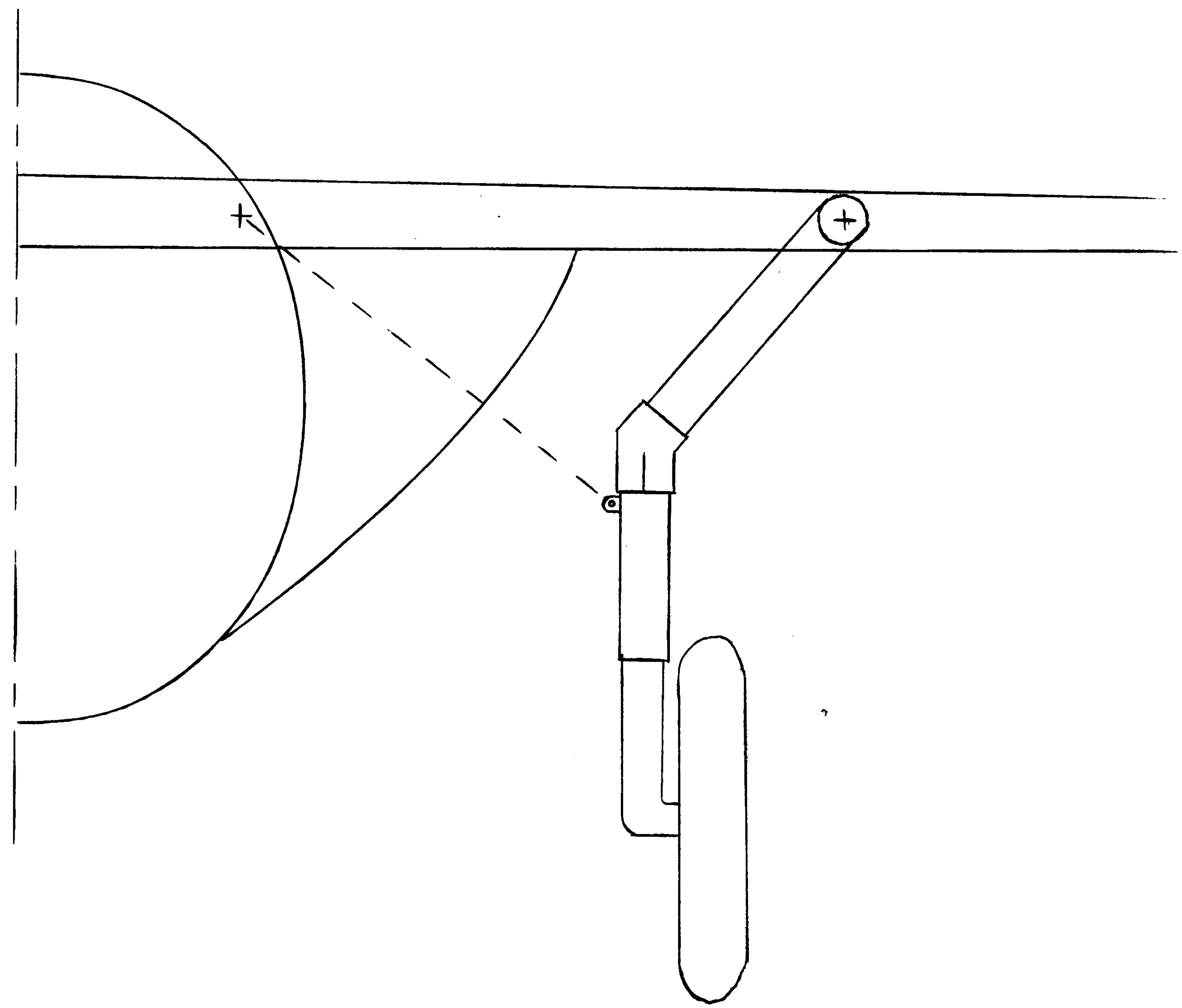


Figure 1

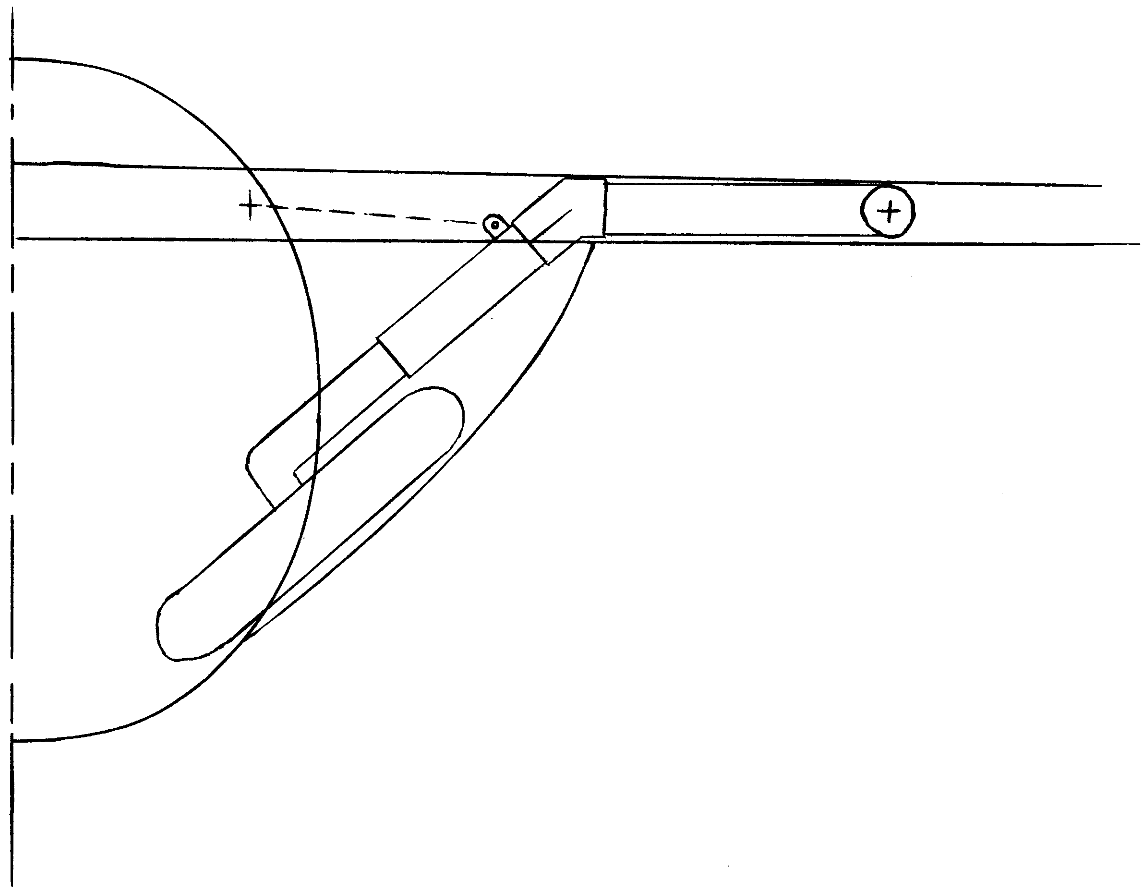


Figure 2

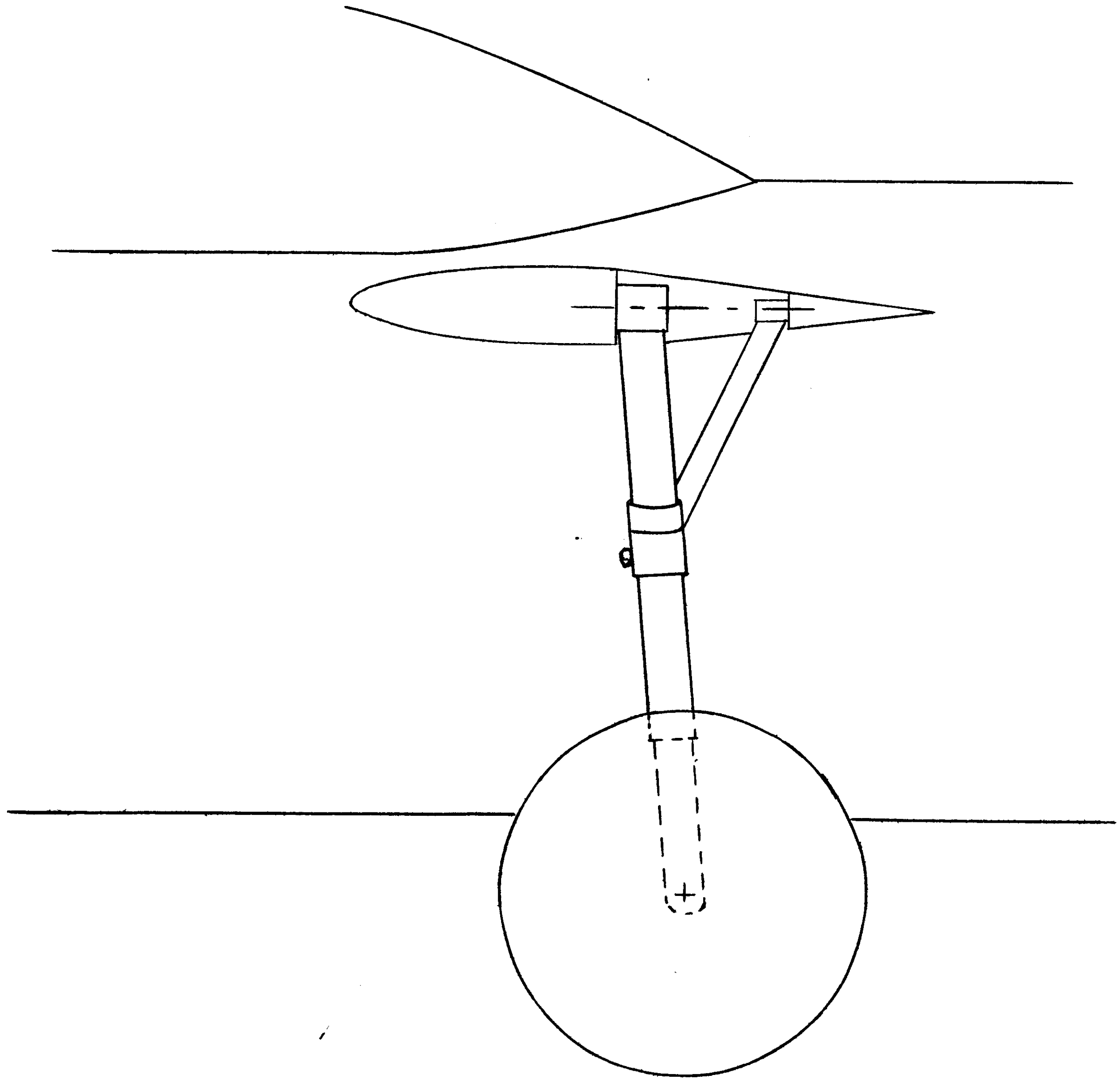


Figure 3

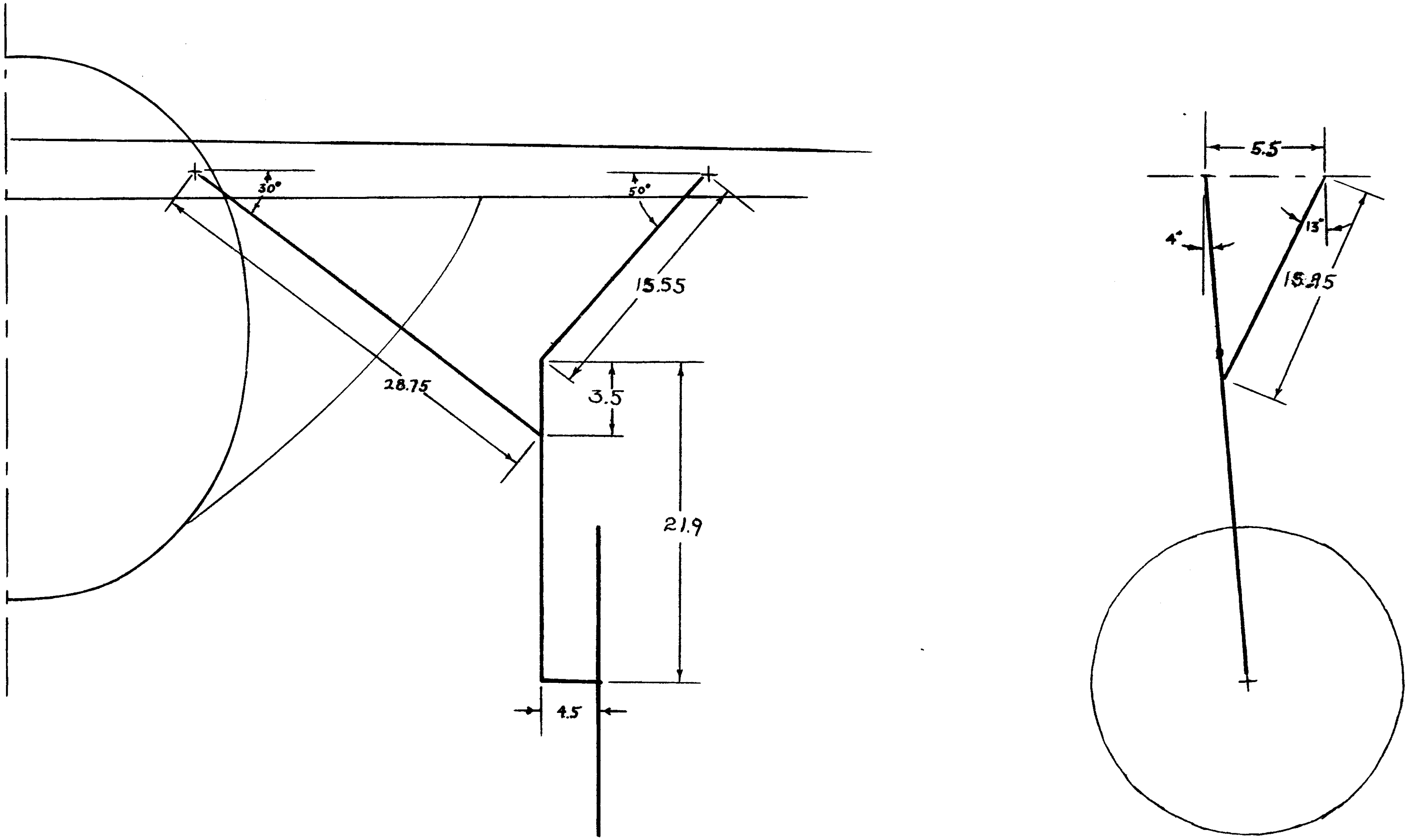


Figure 4

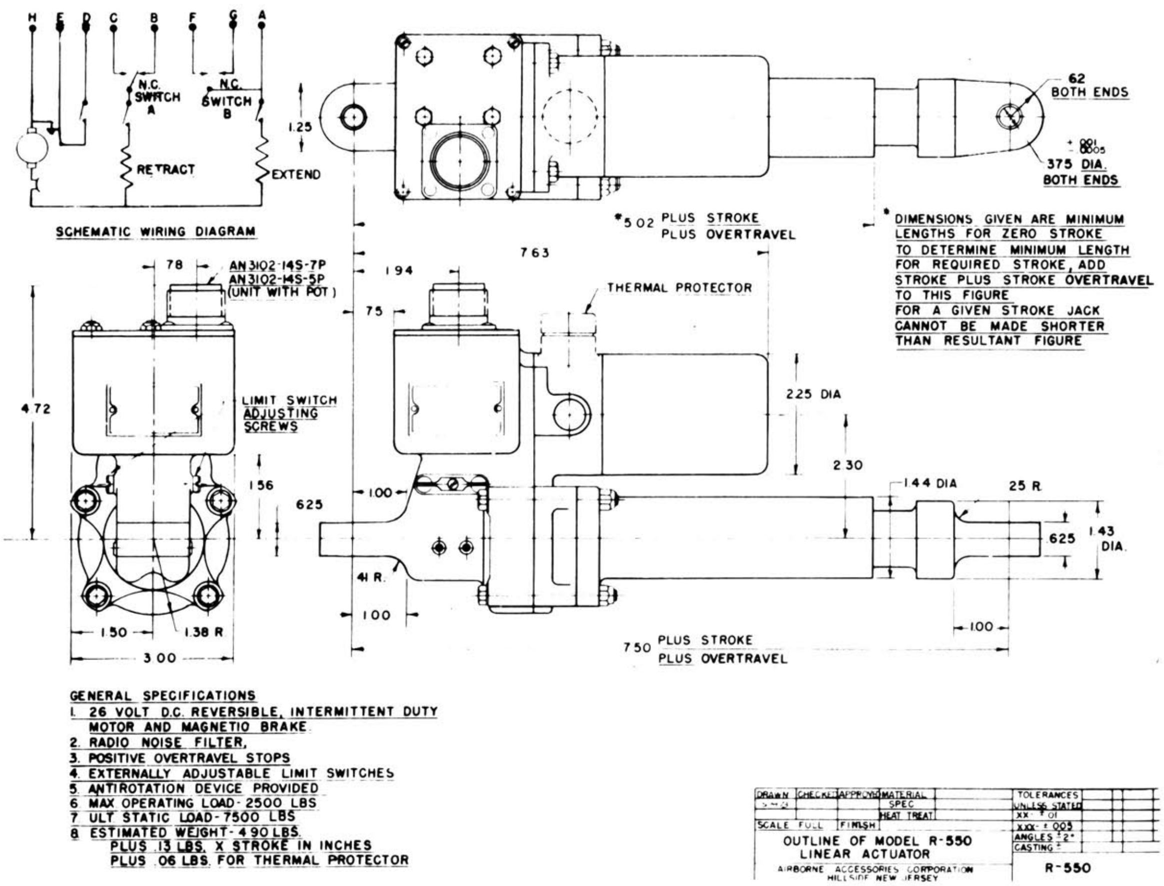

H E D C B F G A
N.C. SWITCH A
N.C. SWITCH B
1.25
RETRACT
EXTEND
SCHEMATIC WIRING DIAGRAM
62 BOTH ENDS
375 DIA. BOTH ENDS
*5.02 PLUS STROKE PLUS OVERTRAVEL
* DIMENSIONS GIVEN ARE MINIMUM LENGTHS FOR ZERO STROKE TO DETERMINE MINIMUM LENGTH FOR REQUIRED STROKE, ADD STROKE PLUS STROKE OVERTRAVEL TO THIS FIGURE FOR A GIVEN STROKE JACK CANNOT BE MADE SHORTER THAN RESULTANT FIGURE
.78
AN3102-14S-7P
AN3102-14S-5P (UNIT WITH POT)
7.63
1.94
THERMAL PROTECTOR
.75
4.72
LIMIT SWITCH ADJUSTING SCREWS
2.25 DIA
2.30
1.56
.625
1.00
1.44 DIA
.25 R.
1.43 DIA.
.625
.41 R.
1.00
1.50
1.38 R
3.00
1.00
7.50 PLUS STROKE PLUS OVERTRAVEL
GENERAL SPECIFICATIONS
1. 26 VOLT D.C. REVERSIBLE, INTERMITTENT DUTY MOTOR AND MAGNETIO BRAKE.
2. RADIO NOISE FILTER,
3. POSITIVE OVERTRAVEL STOPS
4. EXTERNALLY ADJUSTABLE LIMIT SWITCHES
5. ANTIROTATION DEVICE PROVIDED
6. MAX OPERATING LOAD - 2500 LBS
7. ULT STATIC LOAD - 7500 LBS
8. ESTIMATED WEIGHT - 4.90 LBS.
PLUS .13 LBS. X STROKE IN INCHES
PLUS .06 LBS. FOR THERMAL PROTECTOR
DRAWN
CHECKED
APPROVED
MATERIAL
SPEC
HEAT TREAT
SCALE FULL
FINISH
TOLERANCES UNLESS STATED
XX - ± .01
XXX - ± .005
ANGLES ± 2°
CASTING ±
OUTLINE OF MODEL R-550 LINEAR ACTUATOR
AIRBORNE ACCESSORIES CORPORATION
HILLSIDE NEW JERSEY
R-550

Figure 5