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**DESIGN OF AN
ARTILLERY TOWING LIGHTWEIGHT
AUXILIARY SYSTEM (ATLAS)**

H.G. Kirchner

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MAY 1974

Final Report



PREPARED BY

PACIFIC CAR AND FOUNDRY COMPANY
RENTON, WASHINGTON

CONTRACT NO. DAAFO3-C-0138

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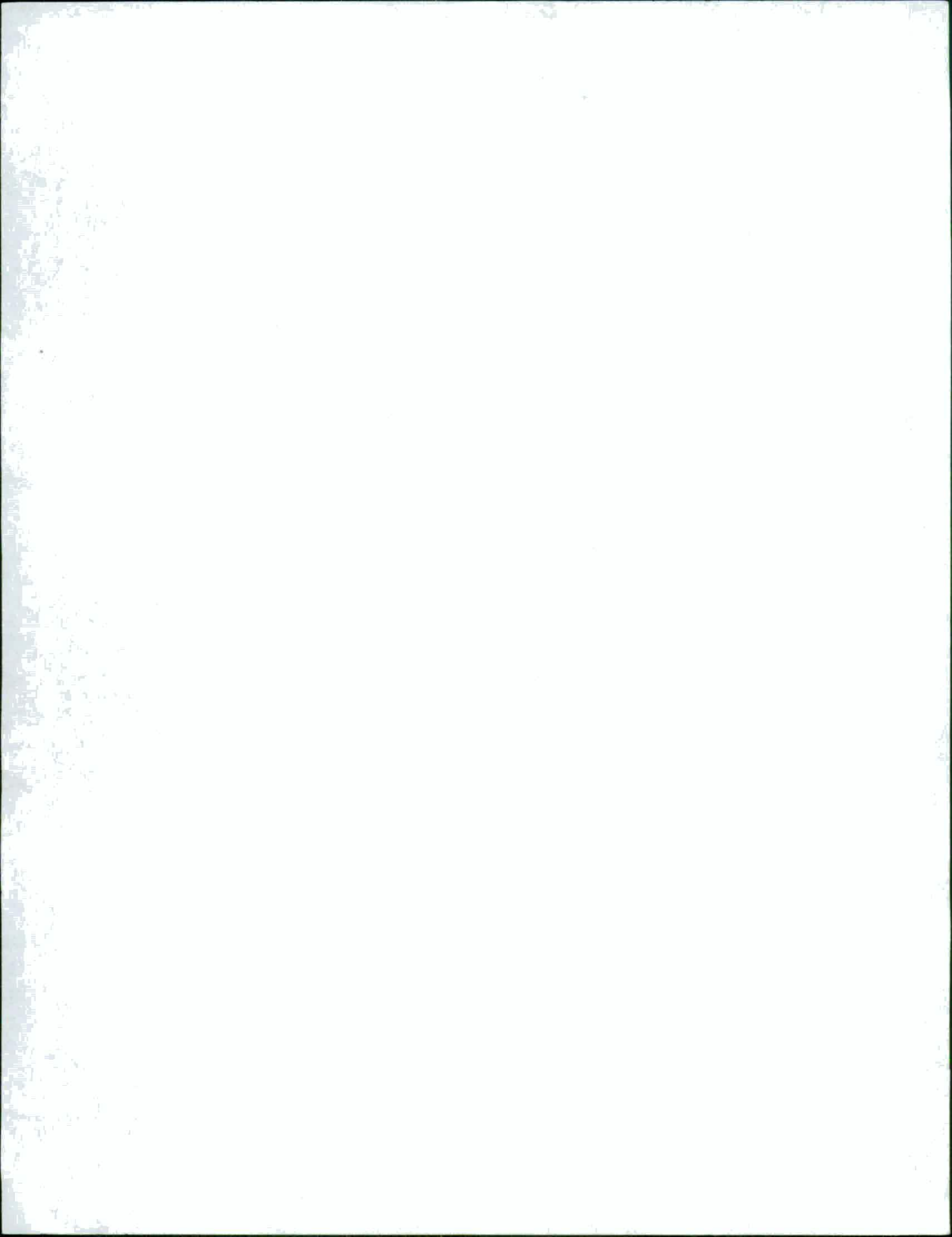
**Prepared for
ARTILLERY & ARMORED WEAPONS SYSTEMS DIRECTORATE
GENERAL THOMAS J. RODMAN LABORATORY
ROCK ISLAND ARSENAL
ROCK ISLAND, ILLINOIS 61201**

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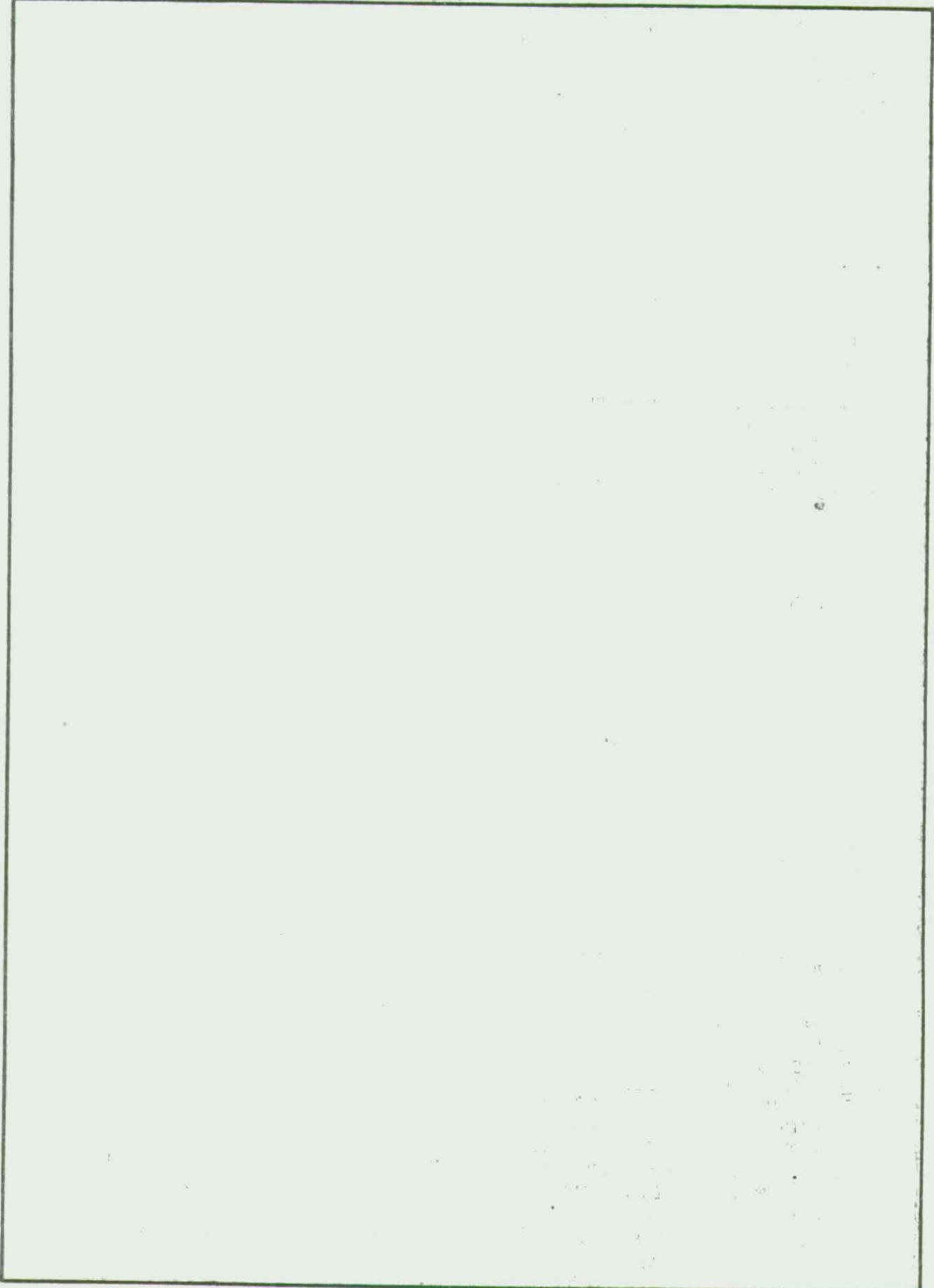
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) THIS TECHNICAL REPORT SUMMARIZES THE CONTRACTOR'S DESIGN AND ANALYSIS EFFORT ON A CONCEPT TITLED ARTILLERY TOWING LIGHTWEIGHT AUXILIARY SYSTEM (ATLAS). THE ATLAS CONCEPT WILL PROVIDE IMPROVED MOBILITY AND AUXILIARY POWER FOR TOWED ARTILLERY WEAPONS IN REMOTE BATTLEFIELD AREAS. SPECIFICATIONS AND DETAILED CALCULATIONS ARE PRESENTED FOR ALL MAJOR COMPONENTS SUCH AS THE POWER TRAIN, THE SUSPENSION, THE AUXILIARY POWER SYSTEMS, ETC. THIS HELICOPTER TRANSPORTABLE PROPULSION DEVICE WILL SIGNIFICANTLY ENHANCE THE EFFECTIVENESS OF TOWED ARTILLERY WEAPONS.		

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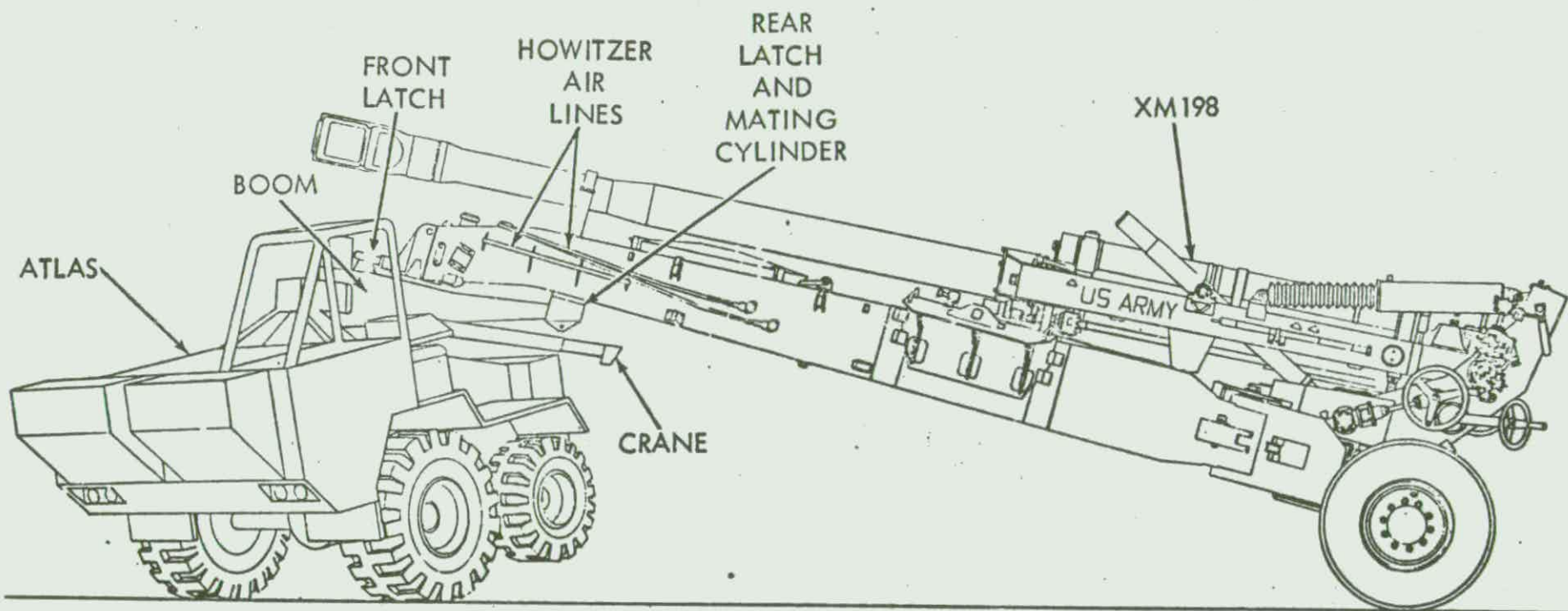
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AT



ATLAS

Technical Report

Introduction

ATLAS, an Artillery Towing Lightweight Auxilliary System is primarily a highly maneuverable tractor designed for transport and support of the XM198 155mm Howitzer. It is lightweight, helicopter transportable, and in addition is capable of supplying auxilliary power, electrical, compressed air and hydraulic to support associated equipment. In support of the howitzer it can carry 30 rounds each of projectiles, mixed zone powder charges and up to 64 varied types of fuzes. It is a front wheel drive, rear wheel steer tractor and when hooked to the howitzer can pivot steer about the center of its drive axle, enabling it to traverse the howitzer while it sits on its own base plate. It is also capable of cross-country operation and has fording capability, and is powered by a dependable, derated diesel engine using primarily dependable commerical powertrain components.

Specifications

Powertrain

Engine - Detroit Diesel Allison 4-53N40, water cooled

116 horsepower @ 2,800 rpm

252 lb-ft torque @ 1,500 rpm

212 in³ displacement

Clutch - 12" RT, Rockford clutch

Transmission - Clark Equipment Company Model 280V

5 speed manual - remote shift

Transfer Case - Pacific Car and Foundry Company Special Design

2 speed, 1.08: 1 and 1.985: 1, with/disc

Parking brake

Drive Axle - Rockwell Standard, H-140, modified with/offset bowl,

limited slip differential, 8.2: 1 ratio, disc brakes

Suspension

Drive Axle - trailing arm, torsion bar, tubular shock absorbers,

6" jounce, 6" rebound

Steering Axle - Trailing arm, coil spring, tubular shock absorbers,

4.25" jounce, 4.75" rebound

Tires - Drive axle, 16.5 - 19.5 wide profile Load range H (16 ply rating)

inflation 60 psi

Steer axle, 12-16.5 wide profile, load

Range D (8 ply rating) inflation 30 psi

Wheel base - 45.5" (solo), 252" (with/howitzer)

Steering (Power)

Solo operation, hydraulic cylinders on axle

Towing howitzer, hydraulic rotary actuator acting through King pin on boom

Minimum turning radius, solo, 146" wall-wall

Performance

Maximum drawbar pull - 13,500 pounds

Maximum speed - 38 mph with/howitzer, 20 mph solo

Gradeability - 50% ascending, 60% descending (with/howitzer)

Side slope - 40% (with/howitzer)

Fording depth - 48 inches

Cross-country speed - in excess of 5 mph

Ammunition stowage

30 rounds projectiles, 155mm

30 rounds powder charges, mixed zones

64 fuzes, M557 and/or M514

Auxiliary Systems

Electrical - 24 VDC, 200 amp alternator

2 type 6TN batteries

Compressed Air - 100 psi maximum, 2240 in³ reservoir capacity,

12 CFM compressor

Hydraulic - 2,000 psi maximum, 23 GPM, open center

Crane - Hydraulic - Electric, 2,000 pound maximum lift

9 foot maximum reach (manual telescoping)

Winch - 10,000 pound planetary, 2 speed

Overall Dimensions

Length 110 Inch (less front and rear ammo racks)

Width 96 Inch

Height 109 Inch (reducible to 94 inch)

Weight 8,000 pound (less ammo)

DESIGN DISCUSSION

1. Frame

Since the completion of the final weight analysis is apparent that additional consideration needs be given to whether the frame is indeed over designed and if its weight could be reduced. In the case of the boom, cross beam, and side frames this is likely and also the frame itself. If new analysis is undertaken use should be made of one of the more simple commercially available computer analysis programs such as ANSYS, Engineering Analysis System from Swanson Analysis Systems Inc. Some additional shop drawings will need to be made when the test rig is built. The engine covers were never layed out but a concept sketch is included at the end of this section. Lifting eyes and towing eyes have not been incorporated. However it is apparent that the strong front and rear corners of the sub-frame provide suitable points for attachment of these eyes.

2. Powertrain

Detroit Diesel Allison should be contacted and consulted on problems to be encountered in fording and the engine oil pan, oil pump strainer bracket and pickup tube need to be modified to accommodate the 9° nose down or up engine attitude (depending on whether operating solo or with/howitzer) on extreme slopes. No detail drawings were made for the idler (pulley, arm, etc.) the alternator pulley, mounting bracket and brace or for the air compressor spacer. The exhaust piping is well defined but needs shop drawings.

The oil seal between the transmission and transfer case may not be effective in separating the oil in the two cases. It is suggested that a double lip tandem seal be installed in place of the single lip seal shown as in the sketch at the end of this section.

A conflict occurred in regard to the driveshaft in that the Mechanics Universal CV (constant velocity) universal joint doesn't fit the standard yokes. This wasn't discovered until too late, however, when Mechanics Universal was contacted by phone they assured us that they had the necessary yokes in production to fit the CV joints. Any information on this will be passed on.

3. Suspension

The H-140 axle may need to be reinforced, mainly because of the additional deflection of the axle housing from offsetting the differential. This deflection will create an end moment on the splined end of the axle shaft and will cause additional wear on the splines and possibly lead to fatigue failure of the axle shaft. Clarification is forthcoming from Rockwell Standard on this. If there is no definite danger of fatigue failure in the axle shaft, it is recommended that the reinforcing be omitted for the test rig in the interest of saving weight and cost. (See Axle Housing Stress Analysis).

The shock absorbers are a special design, depending on the length of stroke and this wasn't a long lead item so it was left to be completed concurrently with the test rig fabrication.

The suspension arms on the steer axle are somewhat unconventionally designed in that they must deflect sideways and twist as well as support a vertical bending moment when the axle tips. There is also an axial load imposed if the vehicle is braked. When the vehicle is traveling forward the axial load is a tensile load and as such doesn't cause great concern. The crucial loading would come from braking when backing up or backing into something especially if the axle is at full tip as well.

This distortion of the suspension arm is what provides roll resistance to the entire suspension. If the ends of the arms are not held fixed then an additional roll stabilizer bar must be added. (One may be needed even with the present suspension arms. The roll stiffness desired is not a hard and fast number).

As noted on the suspension installation, and in the calculations, a better procedure would be to mount the arm or rubber connections so that approximately 75% of the horizontal deflection and twist are taken up in the rubber. The suspension arm has been redesigned for this and thickened to .50 thick.

If the above approach is still suspected or if a roll stabilizer bar is still needed, then the suspension must be modeled after the drive axle suspension, although only a single arm need be used. It will require a heavy roll stabilizer and will increase the vehicle weight.

4. Hydraulic and Air Systems

No piping layouts were attempted because of lack of overall vehicle definition at the time they would have had to be initiated. At any rate, the usual test rig practice is to create the piping drawings after the piping is installed in the test rig. All the large hydraulic and air components were located in the vehicle however (see 8019-502). Note should be taken of the Safety Evaluation comments on additional safety that could be built into the steering circuits.

5. Electrical System

For the same reasons as for the hydraulic and air systems, only an electrical schematic was created. One oversight was the provision for mounting the batteries. Space was reserved for them above the transfer case and transmission however, and the rear frame number and upper cross member provide handy mounting supports. The batteries should be mounted in a tray, parallel to the engine crankshaft centerline as shown in the sketch at the end of this section.

6. Controls and Driver's Compartment

Several shop drawings will be needed for such things as the instrument panel, clutch pedal, light brackets, etc. The control cable lengths will have to be determined by trial installation on the test rig (this can be done with stiff 3 conductor electrical wire). Shop drawings will be needed for the roll cage also.

7. Crane and Winch

The design of the crane is incomplete in the area of the base and hinge for the boom. The informational drawings on details of the boom base, on the mounting of the overload limiter and details of standard swing brakes for the BA-2 Workhorse were not received from the Autocrane Company. The information will, however, be passed on when and if it is received.

The submersion capability for the two speed winch requires only the addition of a double lip seal to the shifting spool. If the two speed capability is forgone, a single speed locked in low gear winch has been produced for the Engineer Corps, Ribbon Bridge (USAMERDC) and the winch can be reversed and held in neutral hydraulically without any change in the hydraulic system.

8. Stowage

No racks for the 9 powder charges on the front of the vehicle or for the 16 projectiles on the rear of the vehicle have been designed. (See sketch at end of this section). Also details of the straps and ratchet tighteners were not worked out. The scheme was to strap down the powder charges and projectiles in groups of 3 or 4 using nylon straps anchored with footman's loops and tensioned with ratchet fasteners.

Additional stowage needs to be considered, such as 5LB CO₂ fire extinguisher, operators rifle, tools, tire inflater gage and hose, etc.

9. General

Detailing of minor (and of most) brackets was ignored in order to concentrate on the other major areas. These are most often more ingeniously designed as the vehicle is being built.

VENDOR INFORMATION

AXLE, DRIVE

Rockwell Standard Division
Rockwell International
1055 West Maple Road
Clawson, Michigan 48017
Mr. Fred M. Cole Jr.
Military Products Engineer

AXLE, STEER

Hadco Engineering
2000 Canfield Avenue
Los Angeles, California 90040
Mr. Mike Guarino

BRAKES, DRIVE AXLE AND PARKING

B. F. Goodrich Company
Troy, Ohio 45373
Mr. Joe Moore
PCF contact was through Zink Enterprises, P.O. Box 771
Bellevue, Washington 98004
Mr. Bill Zink

BRAKES, STEER AXLE

Tol-O-Matic
246 Tenth Avenue South
Minneapolis, Minnesota 55415
Mr. William C. Branham
Marketing Manager

ENGINE

Detroit Diesel Allison
Division of General Motors
13400 West Outer Drive
Detroit, Michigan 48228
Mr. Ron Lund
Engineering Department

TRANSMISSION

Clark Equipment Company
Western Sales Office
1902 SeMorrison Street
Portland, Oregon 97214
Mr. W. A. VanLannen, Manager

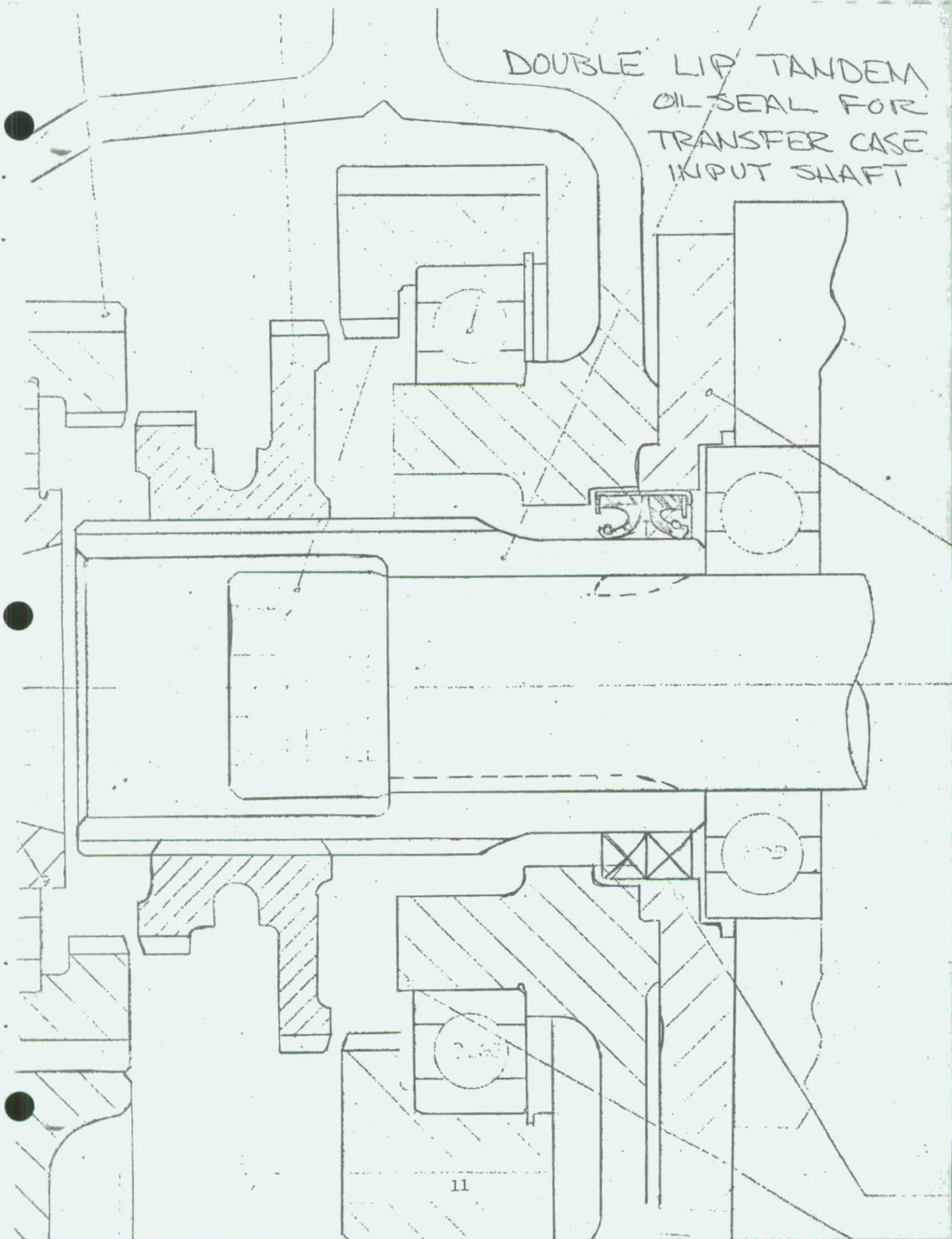
SHOCK ABSORBERS

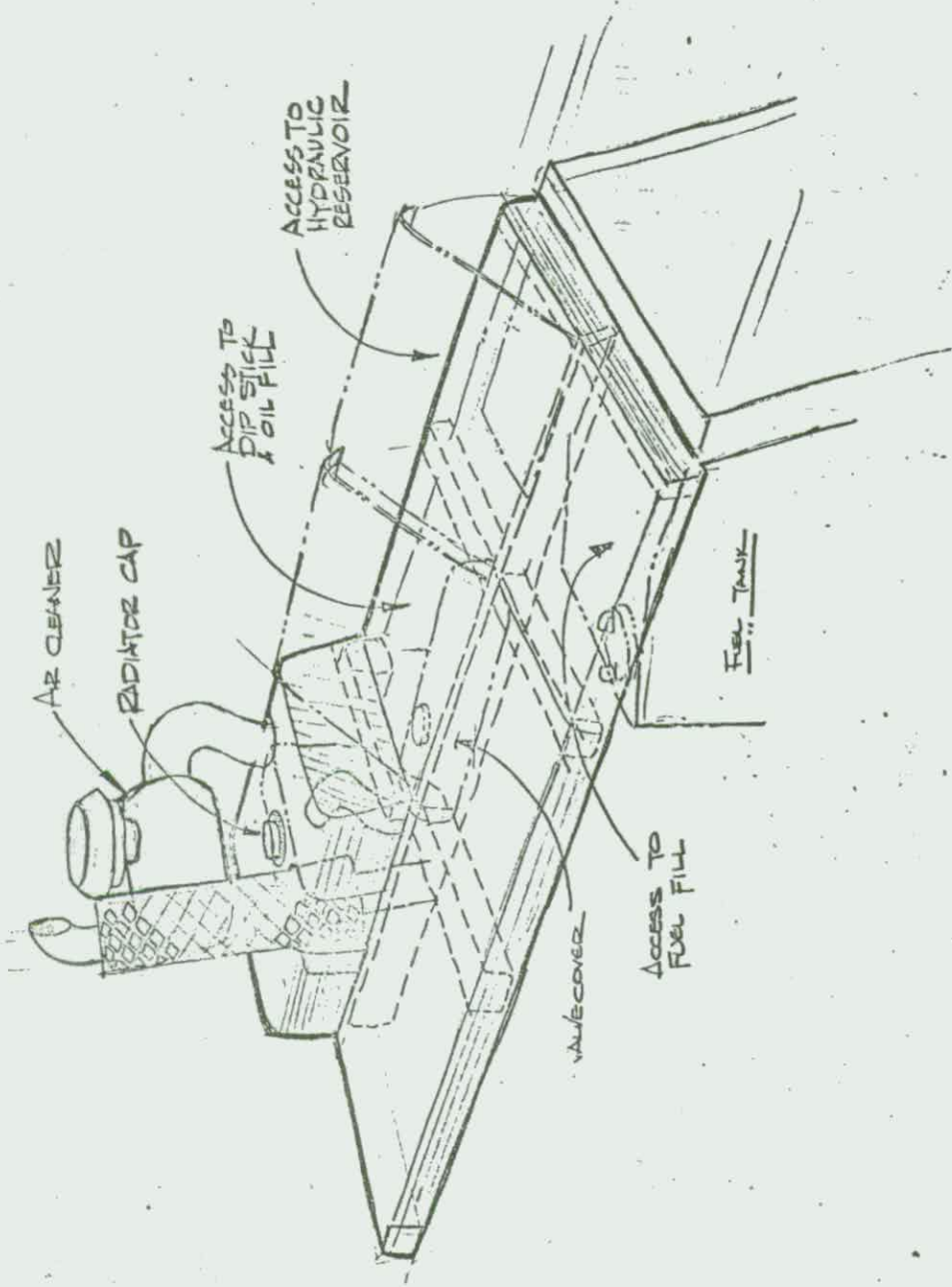
Gabriel of Canada
22371 Newman Avenue
Dearborn, Michigan 48124
Mr. Ral F. Homovic, General Sales Manager

CRANE

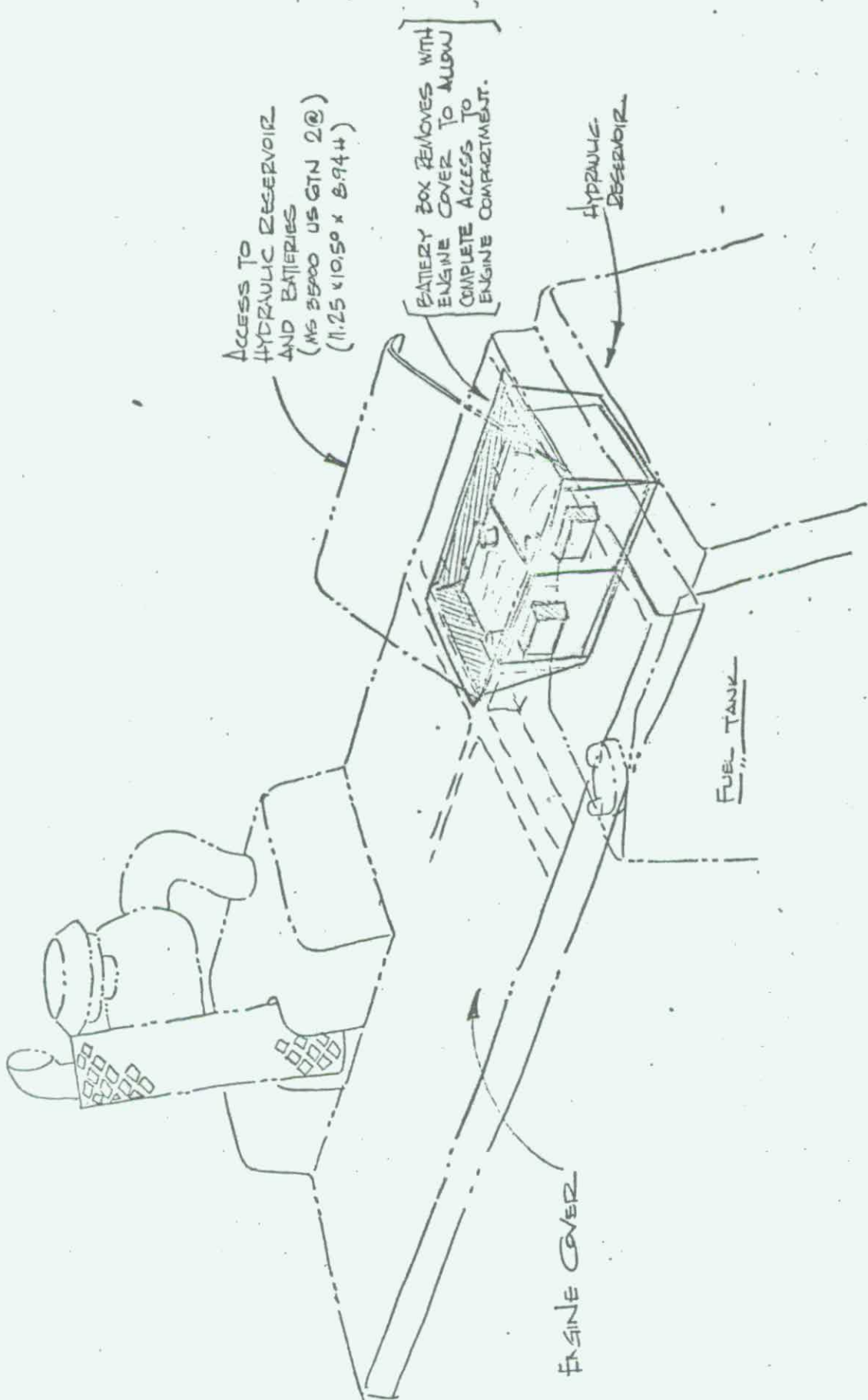
Auto Crane Company
9260 Broken Arrow Expressway
Tulsa, Oklahoma 74145
Mr. Jack Hamilton

DOUBLE LIP TANDEM
OIL SEAL FOR
TRANSFER CASE
INPUT SHAFT





ENGINE COVER



ACCESS TO
HYDRAULIC RESERVOIR
AND BATTERIES
(MS 35000 US GTN 20)
(11.25 x 10.50 x 8.944)

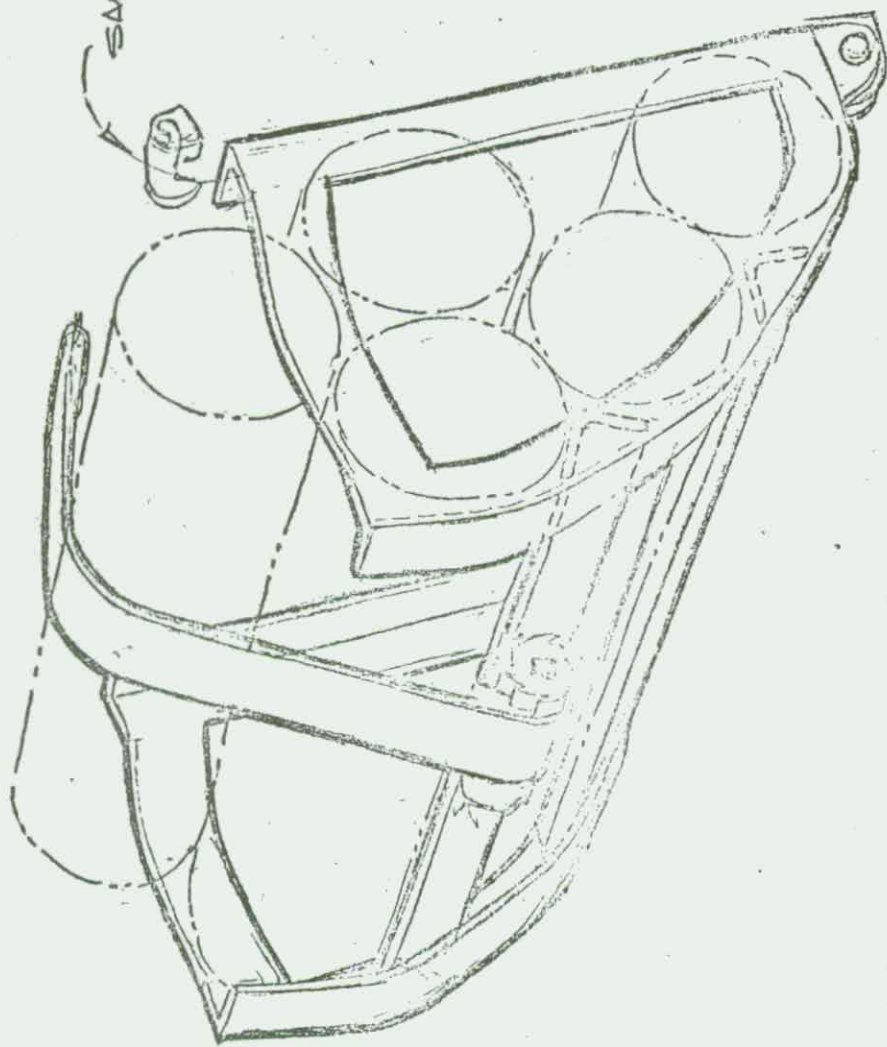
BATTERY BOX REMOVES WITH
ENGINE COVER TO ALLOW
COMPLETE ACCESS TO
ENGINE COMPARTMENT.

HYDRAULIC
RESERVOIR

FUEL TANK

ENGINE COVER

BATTERY BOX

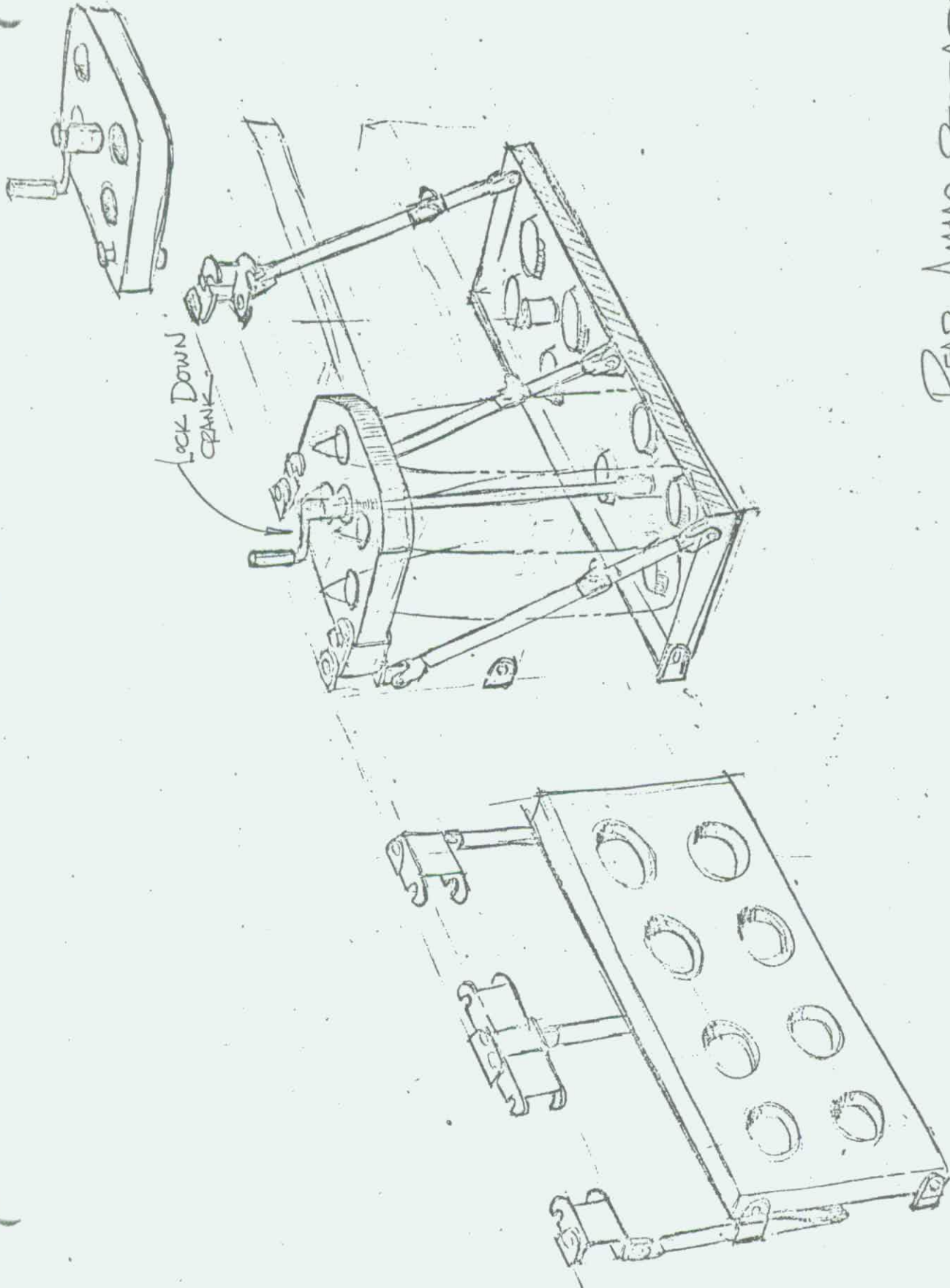


SAFETY LATCH

FRONT POWDER CHARGE

RACK

Z-REQD 81 JAN 74 RSH



REAR AMMO STORAGE
E1 JAN 74 ~~REU~~

TECHNICAL DATA AND CALCULATIONS

1. Frame
2. Powertrain
3. Suspension
4. Hydraulic
5. Air
6. Brakes
7. Miscellaneous

Electrical Schematic

Final Weight C.G. Analysis

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY S. BLACK

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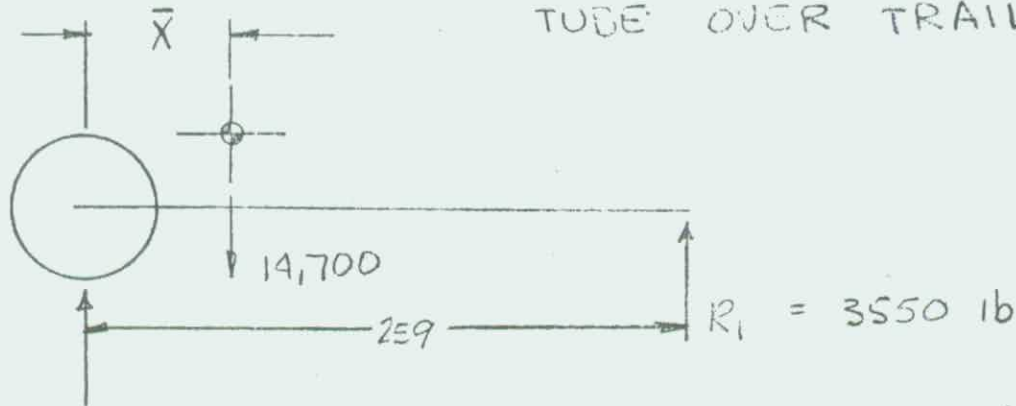
PAGE 1 OF _____

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HOWITZER HORIZONTAL C.G. LOCATION WITH

TUBE OVER TRAILS:



$$R_2 = 14,700 - 3550 = 11,150 \text{ lb}$$

$$\bar{X} = \frac{259 (3550)}{14,700} = \underline{\underline{62.5 \text{ INCHES}}}$$

WHEN THE LUNETTE IS 84.0 INCHES ABOVE
THE GROUND LINE, THE DISTANCE TO
THE C.G. CHANGES TO 55 INCHES.

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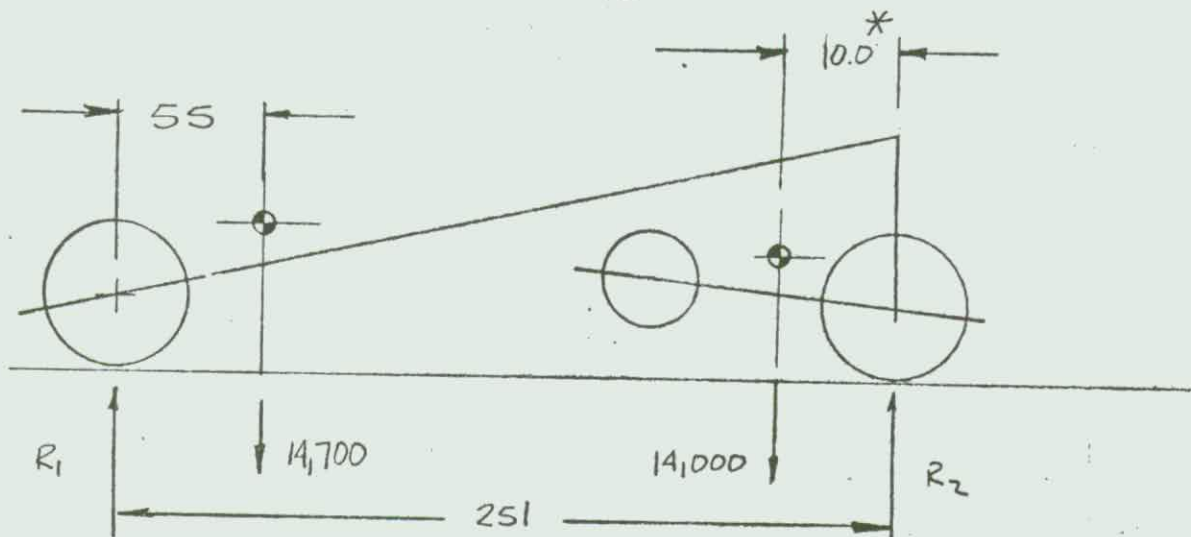
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$$R_2 = \frac{14,700(55) + 14,000(251)}{251} = 16,663 \text{ lb}$$

$$R_1 = 12,037 \text{ lb}$$

* NOTE: THE C.G. OF THE ATLAS CHANGES FROM 16.0 FROM THE AXLE TO 10.0 FROM THE AXLE WHEN IT IS IN THE RAISED POSITION (TILTED ~15°).

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ENGINEERING DEPARTMENT

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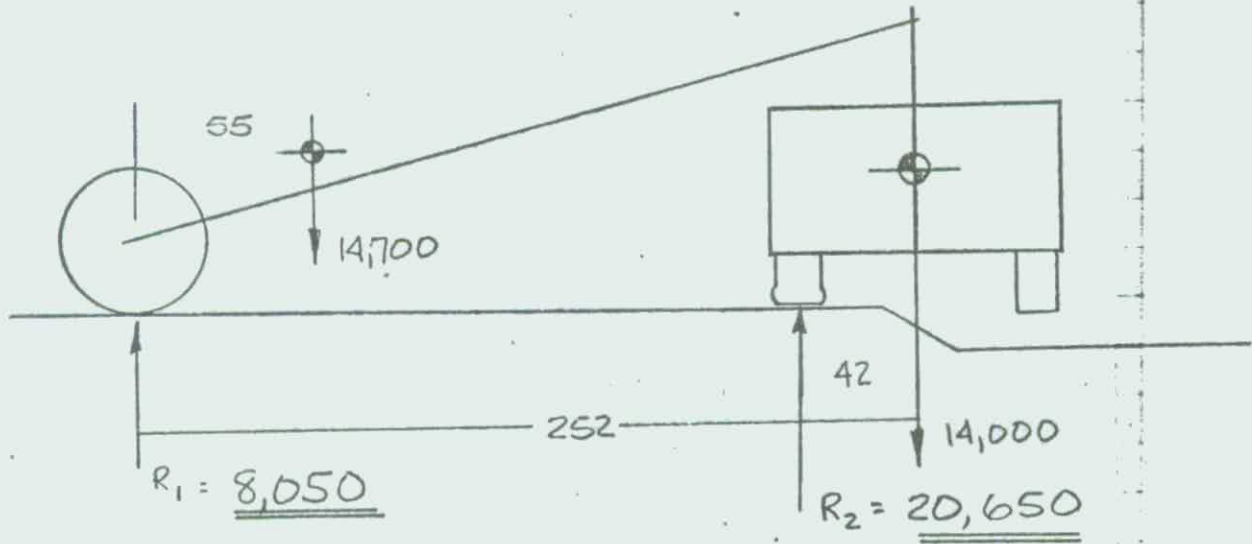
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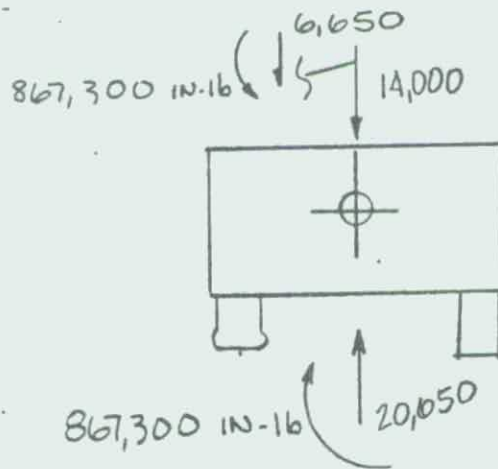
$$\sum M_{R_1} = 0$$

$$\therefore (55)(14,700) + (252)(14,000) - .210(R_2) = 0$$

$$R_2 = 20,650$$

$$R_2 = 8,050$$

VEHICLE
FREE BODY:
(AT PIN)



PACIFIC CAR AND FOUNDRY COMPANY

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PREPARED BY _____

ORDER NO. ATLAS

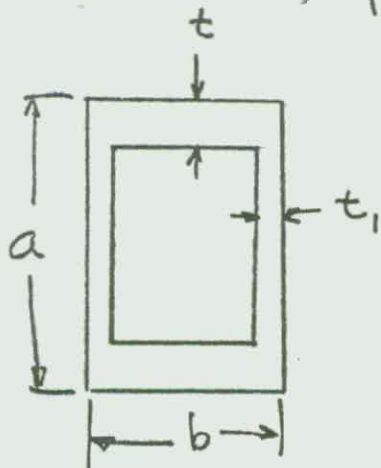
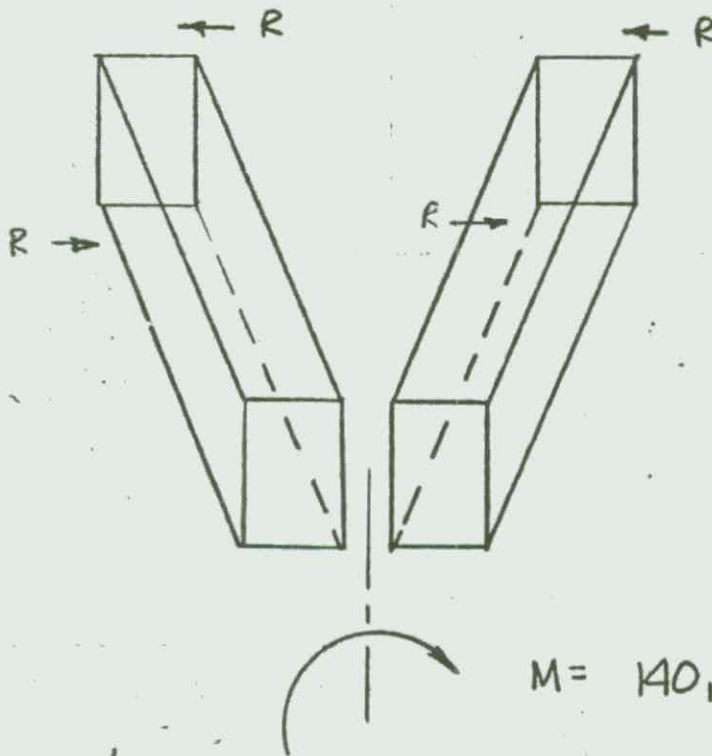
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FIND THE ANGULAR DEFLECTION OF THE HOWITZER TRAILS WHEN THE ATLAS IS 90° TO THE TRAILS



$$\theta = \frac{TL}{KG}$$

$$K = \frac{2tt_1(a-t)^2(b-t_1)^2}{at + bt_1 - t^2 - t_1^2}$$

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FOR THE HOWITZER TRAILS ASSUME:

$$a = 16.0 \text{ (ave) } (12.75 \text{ to } 22)$$

$$b = 10.0$$

$$t = .50$$

$$t_1 = .25$$

$$T = 70,000 \text{ IN-16}$$

$$L = 175 \text{ IN}$$

$$G = 3.8 \times 10^6$$

$$K = \frac{(2)(.50)(.25)(16.0 - .5)^2 (10.0 - .25)^2}{(16)(.5) + 10(.25) - .5^2 - .25^2}$$

$$K = \frac{.25(15.5)^2 (9.75)^2}{8 + 2.5 - .25 - .06}$$

10.19

$$K = 560$$

$$\theta = \frac{(70,000)(175)}{560(3,800,000)} = .0057 \text{ RADIANS}$$

$$.0057 (57.29) = .32^\circ$$

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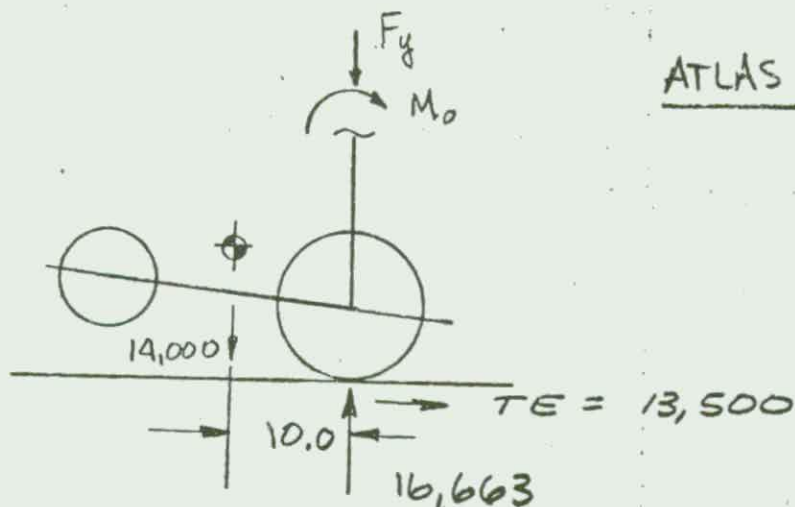
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ATLAS FREEBODY



$$M_o = 14,000 (10) = 140,000 \text{ lb-in}$$

$$F_y = 16,663 - 14,000 = 2,663 \text{ lb}$$

TO DETERMINE THE STRESS IN THE FRAME,
 HINGE, AND HOWITZER TRAILS, IT WILL
 BE ASSUMED THAT THE TRACTIVE
 EFFORT IS 13,500.

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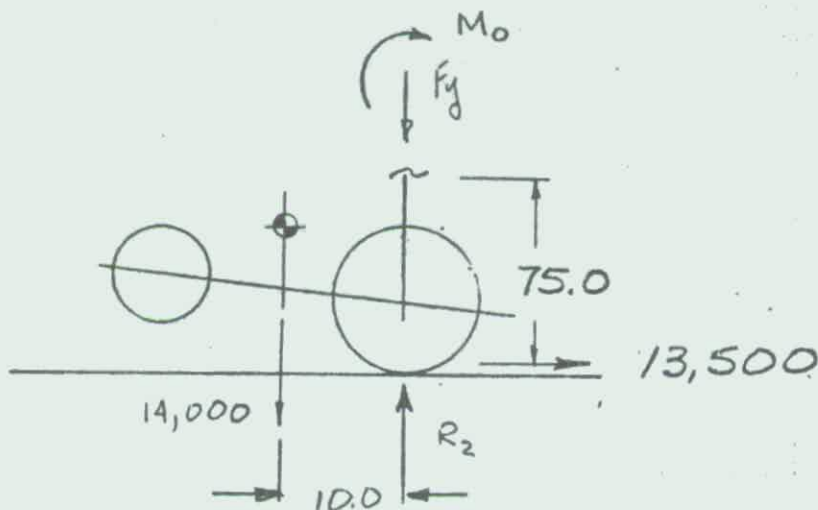
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FOR THE WORST CASE ASSUME THAT
THE ATLAS DEVELOPS MAXIMUM TRACTIVE
EFFORT TO MOVE THE HOWITZER:



$$F_y = 2,663 \text{ lb}$$

$$M_o = 14,000 (10.0) + (13,500)(75) = \underline{\underline{1,152,500 \text{ in-lb}}}$$

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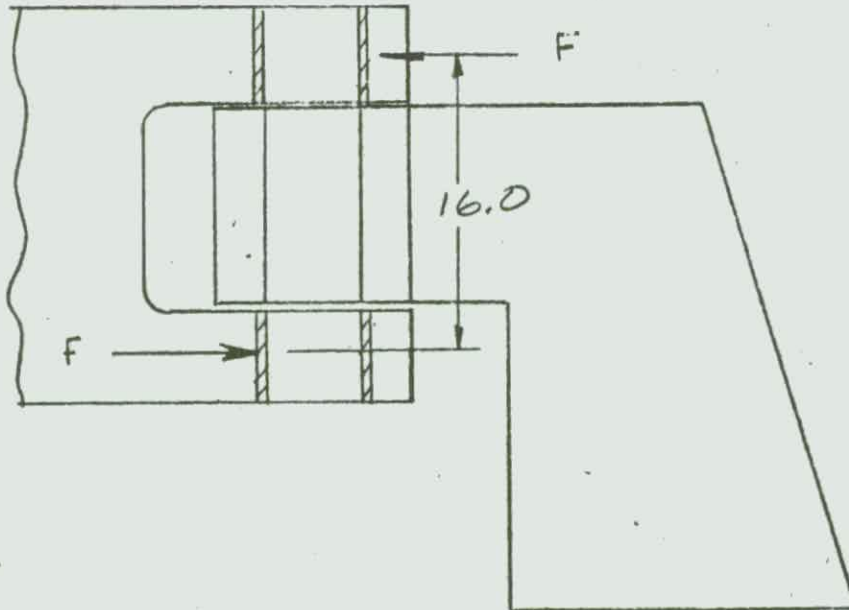
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"DU" BUSHING LOAD = $M_0/16$

LOAD = 92,200 lb

FOR (6ADUGD) 4.0 ID X 3.75 LONG:

PROJECTED AREA = 15.0 IN²

STRESS = $92,200 / 15 = \underline{\underline{6,150 \text{ lb}/\text{IN}^2}}$

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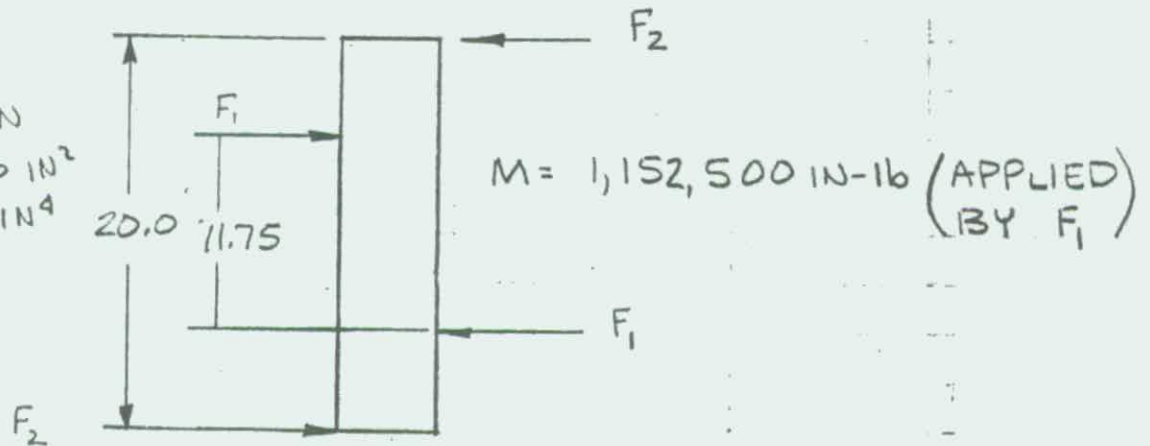
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CONSIDERING NO LOAD DISTRIBUTION TO FIND
PIN STRESS:

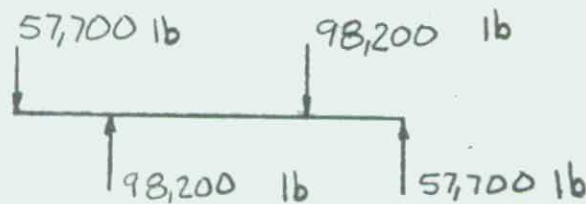
DIA = 4.0 IN
AREA = 12.56 IN²
"I" = 12.6 IN⁴



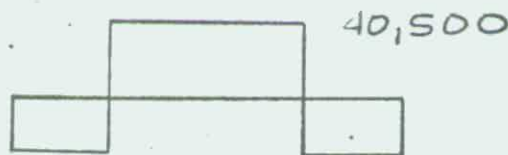
$$F_1 = 1,152,500 / 11.75 = 98,200 \text{ lb}$$

$$F_2 = 1,152,500 / 20 = 57,700 \text{ lb}$$

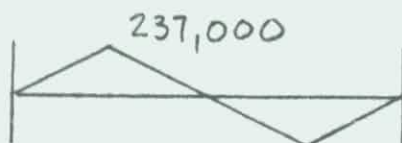
LOAD



SHEAR



MOMENT



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BENDING STRESSES:

$$\sigma = \frac{M y}{I} = \frac{(237,000)(2)}{12.6} = 37,600 \text{ lb/in}^2$$

SHEAR STRESS:

$$\tau = F/A = 40,500/12.56$$

$$\tau = 3,230 \text{ lb/in}^2$$

PRINCIPAL STRESS:

$$\sigma_1 = \frac{37,600}{2} + \left[\left(\frac{37,600}{2} \right)^2 + (3,230)^2 \right]^{\frac{1}{2}}$$

$$\sigma_1 = 37,875 \text{ lb/in}^2 \leftarrow \text{MAXIMUM PRINCIPAL}$$

$$\tau_{\text{MAX}} = \pm \left[\left(\frac{37,600}{2} \right)^2 + (3,230)^2 \right]^{\frac{1}{2}}$$

$$\tau_{\text{MAX}} = 19,075 \text{ lb/in}^2 \leftarrow \text{MAXIMUM SHEAR}$$

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PREPARED BY R. FLOTH

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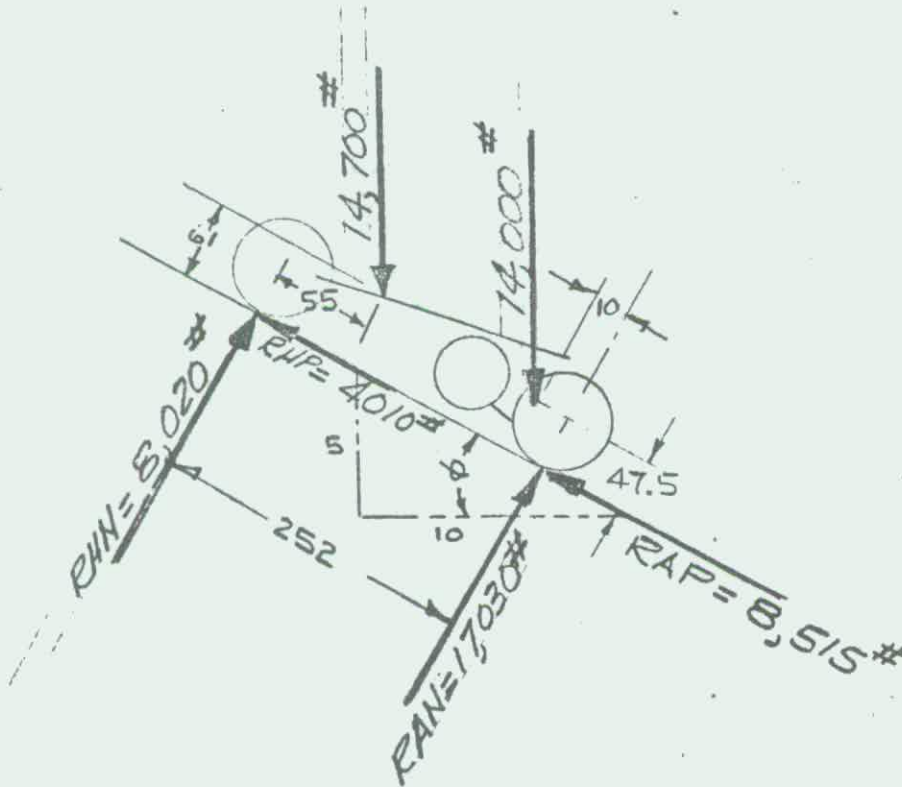
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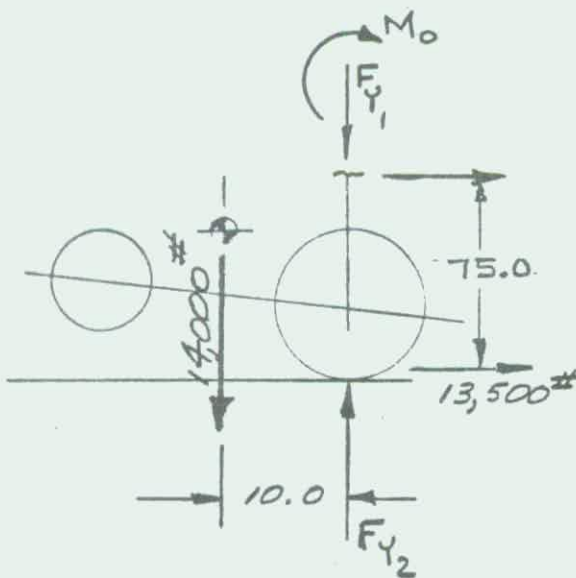
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EVALUATION FOR "KING PIN" WELDED TUBE STRUCTURE



REQUIRED DROP BAR PULL = 13,500 #



$$M_o = (14,000\#)(10\text{in}) + (13,500\#)(75\text{in})$$

$$M_o = 140,000\text{ in-lb} + 1,012,500\text{ in-lb}$$

$$M_o = 1,152,500\text{ in-lb}$$

PACIFIC CAR AND FOUNDRY COMPANY

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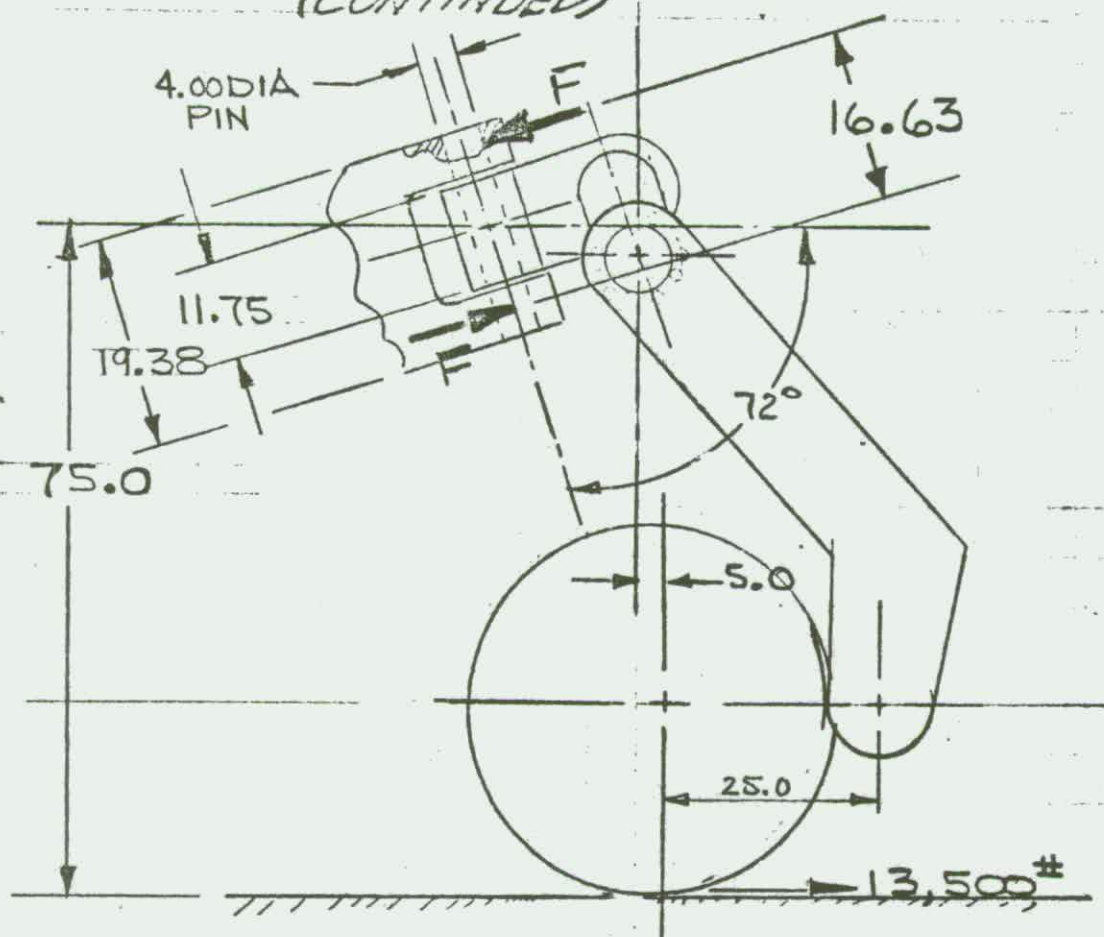
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EVALUATION FOR "KING PIN" WELDED TUBE STRUCTURE (CONTINUED)



$$F = \frac{1,152,500 \text{ lb} - \text{Lb}}{16.63 \text{ in}}$$

$$F = 69,303 \text{ Lb}$$

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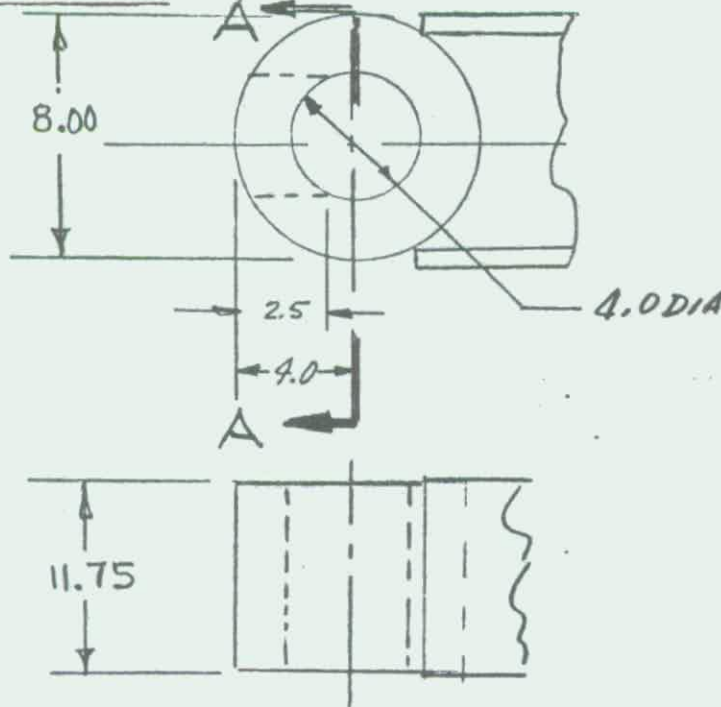
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EVALUATION FOR "KING PIN" WELDED TUBE STRUCTURE (CONTINUED)

BAR EVALUATION (ITEM #1)



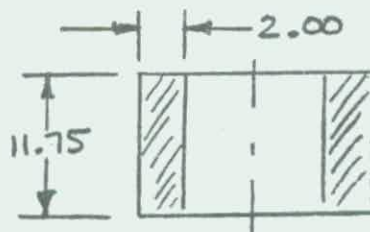
SHEAR TEAROUT

$$S_s = \frac{F}{A}$$

$$S_s = \frac{69,303 \text{ Lb}}{(2)(2.5)(8.00)} = \frac{69,303 \text{ Lb}}{40.0 \text{ in}^2}$$

$$S_s = 1,733 \text{ PSI}$$

TENSION



SECTION A-A

$$\text{AREA} = (2.00)(11.75) = 23.50 \text{ in}^2$$

$$\text{LOAD} = \frac{69,303 \text{ Lb}}{2} = 34,651 \text{ Lb}$$

$$S_t = \frac{34,651 \text{ Lb}}{23.50 \text{ in}^2}$$

$$S_t = 1475 \text{ Lb/in}^2$$

BEARING

$$\text{AREA} = (4.0)(11.75) = 47.0 \text{ in}^2$$

$$S_B = \frac{69,303 \text{ Lb}}{47.0 \text{ in}^2} ; S_B = 1474 \text{ Lb/in}^2$$

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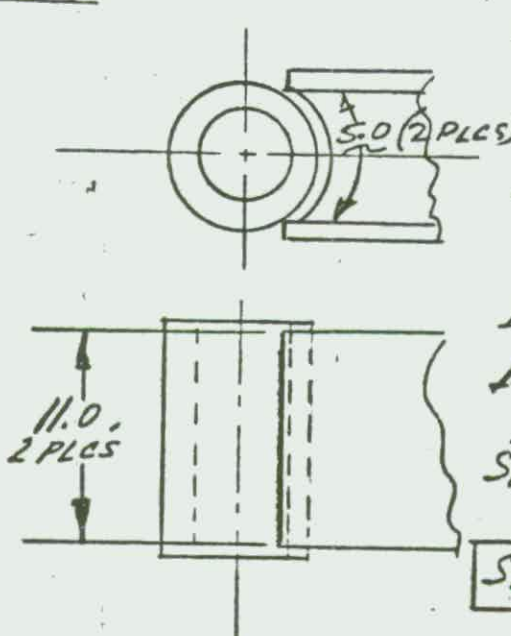
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EVALUATION FOR "KING PIN" WELDED TUBE STRUCTURE (CONTINUED)

WELDED SUPPORT



$$\text{LENGTH OF WELD} = (2)(11.0) + (2)(5.0)$$

$$\text{LENGTH OF WELD} = 22.0 + 10.0$$

$$\text{LENGTH OF WELD} = 32.0 \text{ in}$$

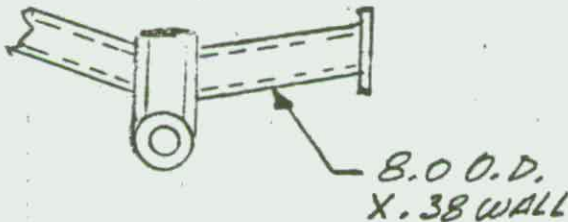
$$\text{AREA} = (.707)(.50)(32.0)$$

$$\text{AREA} = 11.31 \text{ in}^2$$

$$S_{\text{WELD}} = \frac{69,303 \text{ Lb}}{11.31 \text{ in}^2}$$

$$S_{\text{WELD}} = 6,127 \text{ Lb/in}^2$$

TORSIONAL STRESS IN TUBE (ASSUME WORST CONDITION) (MOMENT TAKEN OUT ONE SIDE OF SUPPORT ONLY)



$$r_o = 4.00$$

$$r_i = 3.62$$

$$S_{\text{MAX}} = \frac{2 T r_o}{3.14(r_o^4 - r_i^4)} \quad (\text{ROARK})$$

$$S_{\text{MAX}} = \frac{(2)(1,152,500)(4.00)}{3.14[(4.0)^4 - (3.62)^4]} = \frac{9,220,000}{3.14[256.0 - 172]}$$

$$S_{\text{MAX}} = \frac{9,220,000}{263.7} = 34,963$$

$$S_{\text{MAX}} = 34,963 \text{ PSC}$$

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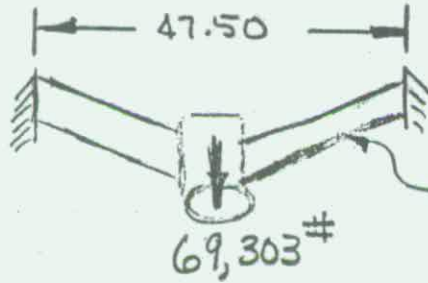
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EVALUATION FOR "KING PIN" WELDED TUBE STRUCTURE (CONTINUED)

BENDING STRESS IN TUBE



$$I = \frac{\pi [(4.0)^4 - (3.62)^4]}{4}$$

$$I = \frac{(3.14) [256.0 - 172]}{4}$$

$$I = 65.9$$

$$M = \frac{(69,303\#)(47.50)}{8} = 411,486 \text{ in-Lb}$$

$$S_B = \frac{M Y}{I} = \frac{(411,486)(4.0)}{65.9}$$

$$S_B = 24,976 \text{ Lb/in}^2$$

TUBE WELDED



$$\text{CIRCUMFERENCE} = 11.38$$

$$\text{LENGTH OF WELD} = (11.38)(2) = 22.76 \text{ in}$$

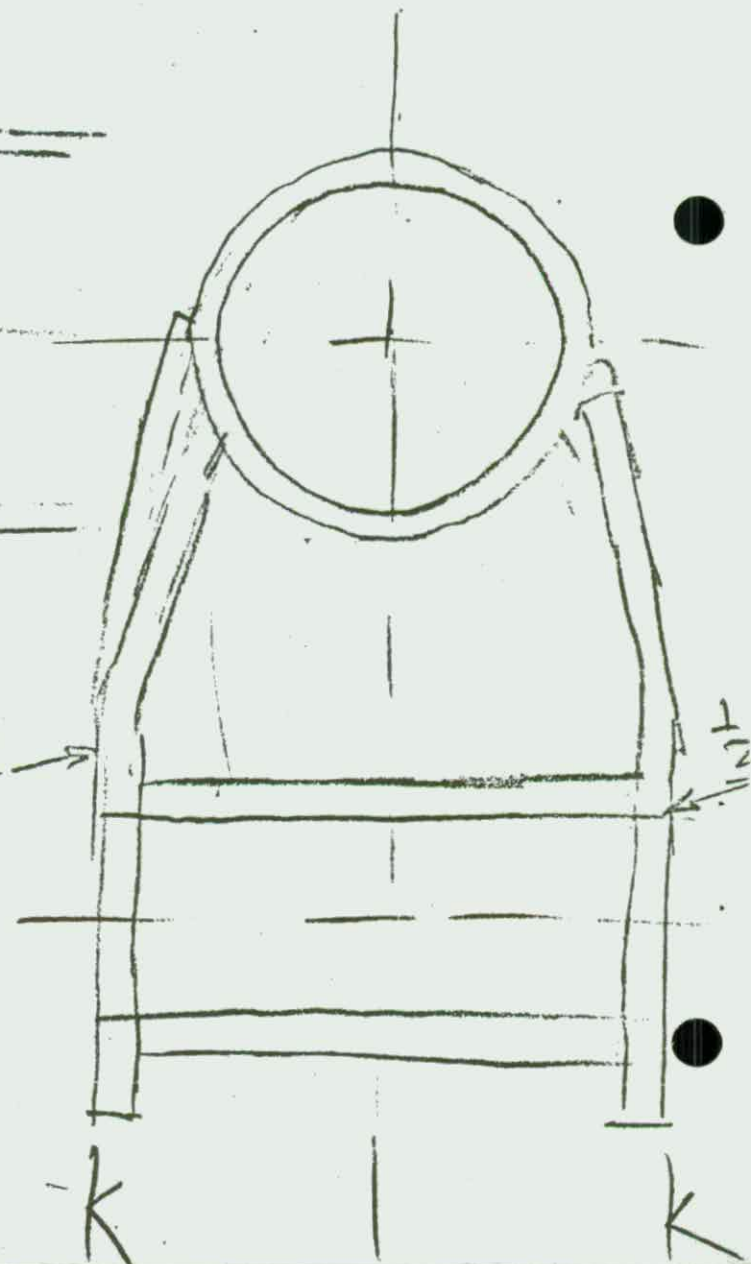
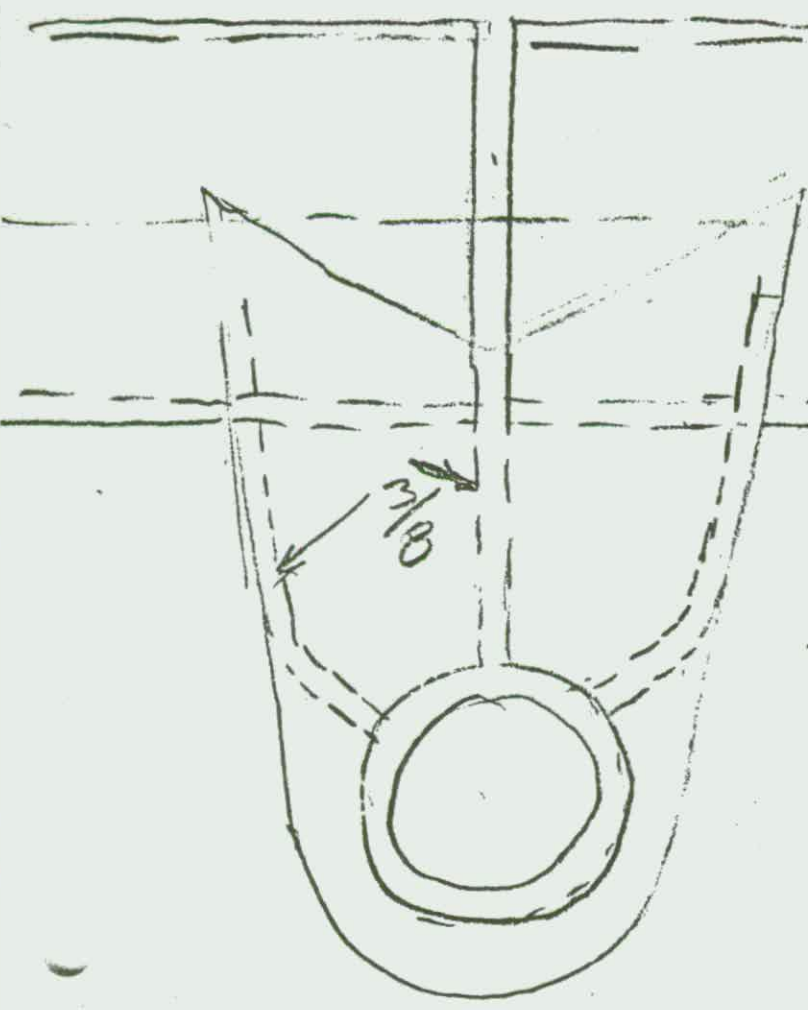
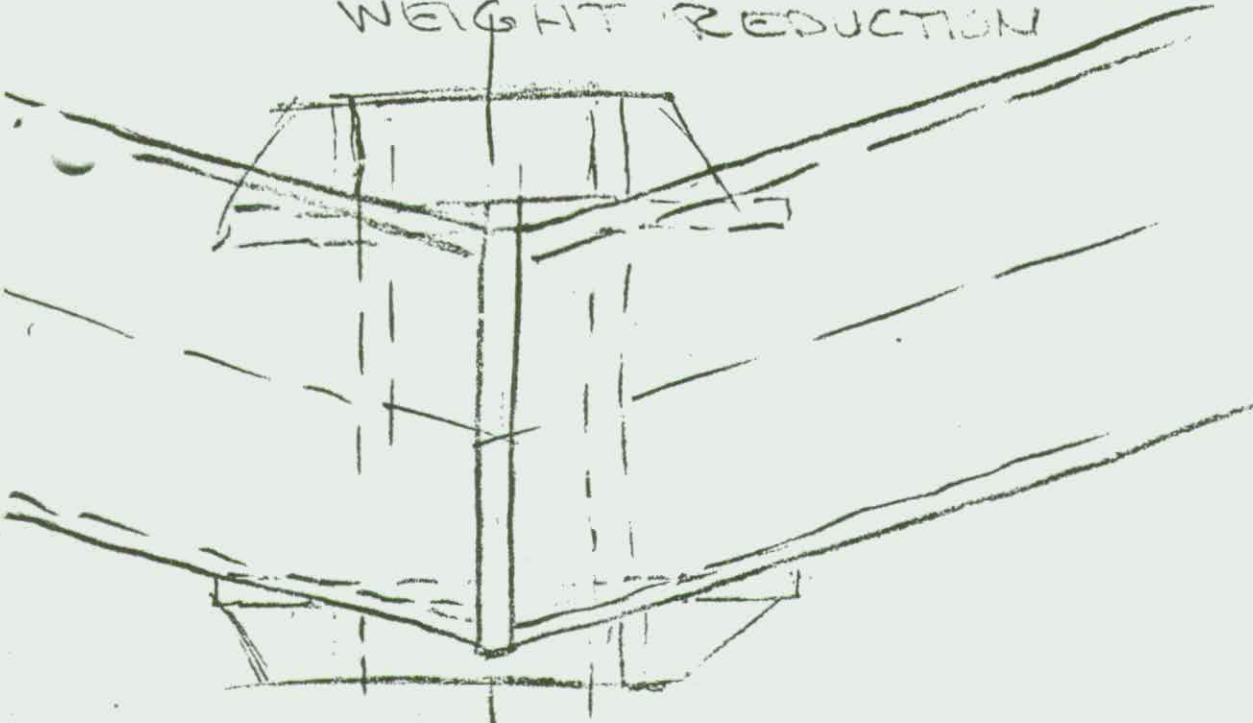
$$\text{AREA} = (.707)(.38)(22.76)$$

$$\text{AREA} = 6.1 \text{ in}^2$$

$$S_{\text{WELD}} = \frac{69,303 \text{ Lb}}{6.1 \text{ in}^2}$$

$$S_{\text{WELD}} = 11,361 \text{ PSI}$$

POSSIBLE REDESIGN FOR WEIGHT REDUCTION



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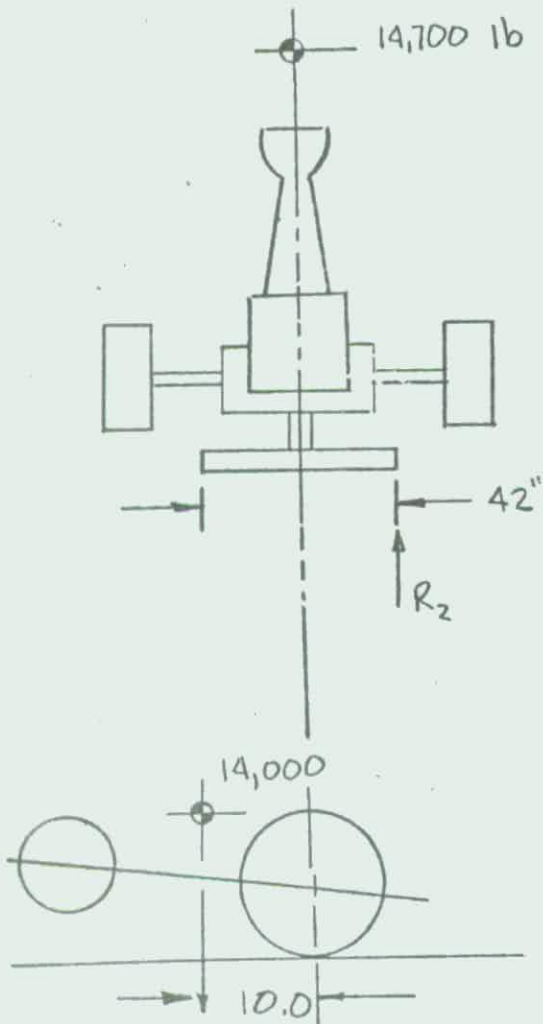
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TIPPING RESISTANCE

WITH TUBE IN-LINE:

$$\text{RESISTANCE} = 14,700 \times 21 = 309,000 \text{ lb-in}$$



$$\begin{aligned} \text{TIPPING MOMENT} &= 14,000 \times 10 \\ &= 140,000 \text{ lb-in} \end{aligned}$$

POSITION OF TUBE AND TERRAIN MAY
REDUCE STABILIZING MOMENT. A LARGER
BASE PLATE MAY BE REQUIRED.

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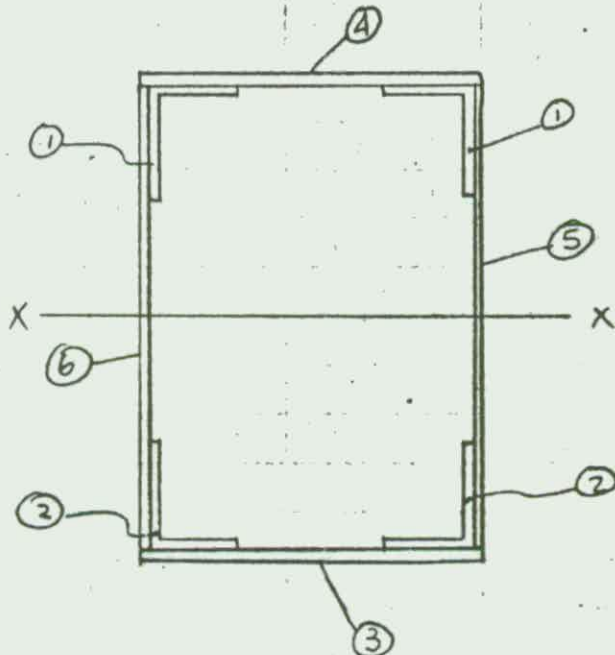
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FIND STRESS IN HOWITZER TRAILS DUE
TO MOMENT DEVELOPED BY ATLAS.

SECTION F-F OF TRAIL ASSY RIGHT (72F515) :

1	72B909	TOP ANGLE	3 X 2 X 3/16
2	72B813	BOTTOM ANGLE	3 X 2 X 3/16
3	72D828	BOTTOM	10.0 X .50
4	72D810	TOP	10.0 X .50
5	72F802	SIDE	12.75 X .25
6	72F825	SIDE	12.75 X .25



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<u>SECTION</u>	<u>I_{x-x}</u>	
①	$2 [.82 \text{ IN}^4 + .91 (5.43)^2] =$	55 IN^4
②	$2 [.82 \text{ IN}^4 + .91 (5.43)^2] =$	55 IN^4
③ & ④	$2 [.104 \text{ IN}^4 + 5.0 (6.62)^2] =$	438 IN^4
⑤ & ⑥	$(43) 2 =$	<u>86 IN^4</u>
	TOTAL	634 IN^4

FOR A MOMENT OF 1,152,500 IN-LB

AT THE SUPPORT THE MOMENT IS

$$\left(\frac{192}{252}\right)(1,152,500) = 878,100 \text{ lb-IN } 60$$

INCHES FROM THE HINGE WHERE THE TRAILS ALONE MUST CARRY THE LOAD.

FOR TWO (2) TRAILS THE TOTAL "I"

$$\text{IS: } 1268 \text{ IN}^4$$

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$$\sigma = \frac{M_y}{I''} = (878,100)(6.88) / 1268$$

$$\sigma = \underline{\underline{4,760 \text{ lb/in}^2}} \quad (\text{BENDING STRESS ONLY})$$

FIND THE FORCE REQUIRED IN THE
HYDRAULIC CYLINDER TO HOLD THE
MOMENT DEVELOPED AT THE HINGE:

$$F = 1,152,500 / 56 / \cos 15^\circ = \underline{\underline{21,300 \text{ lbs}}}$$

SINCE A HYDRAULIC CYLINDER WOULD
"PRE-STRESS" THE STRUCTURE UNDULY IT
IS RECOMMENDED THAT A CLAMPING
MECHANISM BE USED WHICH WILL
HOLD THE 21,300 lb LOAD.

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FIND THE TORSIONAL STRESS IN THE
TUBE HOLDING THE "KING PIN"

$$T = 1,152,500 \text{ lb-in} / 2 = 576,250 \text{ lb-in}$$

TUBE SIZE 8.00 OD X .375 WALL

$$S_{MAX} = 2T r_i / 3.14 (\eta^4 - r_o^4)$$

$$\eta = 4.00$$

$$r_o = 3.625$$

$$S_{MAX} = (2)(576,250)(4) / 3.14 (4^4 - 3.625^4)$$

$$S_{MAX} = 17,610 \text{ lb/in}^2$$

PACIFIC CAR AND FOUNDRY COMPANY

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REPORT NO. ATLAS

FIND DRUM DIA OF P-10 WINCH FOR 100 FT
OF 1/2 INCH CABLE.

N = NUMBER OF LAYERS

$$W = 10.0$$

$$d = .50$$

$$D = 6.50$$

$$L = 100$$

$$L = \frac{\pi N}{12} (W-d) \left(\frac{D}{d} + N \right)$$

$$100 = \frac{(3.14)(N)}{12} (9.5) (13 + N)$$

$$100 = 2.45 N (13 + N)$$

$$40.82 = 13N + N^2$$

$$N^2 + 13N - 40.82 = 0$$

$$N = \frac{-13 \pm (13)^2 + [(4)(1)(40.82)]^{\frac{1}{2}}}{2}$$

$$N = -6.5 + 9.11 = 2.61$$

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REPORT NO. ATLAS

$$D = 6.50$$

$$X = .25 / \tan 30^\circ = .257 / .577 = .433$$

$$\begin{aligned} \text{OVER WIRE ROPE} &= 6.50 + 2(.50 + .866) \\ &= 6.50 + 2.73 \\ &= 9.23 \end{aligned}$$

$$\text{WINCH DRUM O.D.} = 9.23 + 2.00 = 11.23$$

DESIGN DIMENSION =
FOR DRUM DIA

11.25 ± .03 ←

SPECIFICATIONS

CARCO MODEL P-10 PLANETARY WINCH

BASIC WINCH DATA

Rated Line Pull	10,000 Lbs.
Total Reduction	
Low Speed	140:1
High Speed	26:1
Overall Gear Efficiency	85%
Brake Release Pressure	80 PSI
Weight (Less Wire Rope & Motor)	150 Lbs.
Oil Capacity	1-1/2 Qts.
Drum Size:	
Barrel Diameter	6-1/2"
Flange Diameter	13-1/2"
Barrel Length	10-0"
Drum Capacity	366' of 1/2" Wire Rope

EQUATIONS

Line Pull (Lbs.)	=	$\frac{(\text{Motor Torque, In-Lb}) (\text{Gear Ratio}) (\text{Efficiency \%})}{(\text{Radius to Center of Wire, In}) (100)}$
Line Speed (FPM)	=	$\frac{(.5236) (\text{Motor RPM}) (\text{Radius to Center of Wire, In})}{(\text{Gear Ratio})}$

* WINCH PERFORMANCE DATA

Motor Input

Torque (In-lb) 300	Speed (RPM) 1800
------------------------------	----------------------------

Line Pull

Low Speed:

Bare Drum . . . 10,000 Lbs.
Full Drum . . . 5,380 Lbs.

High Speed:

Bare Drum 1,860 Lbs.
Full Drum 1,000 Lbs.

Line Speed

Low Speed:

Bare Drum 24 FPM
Full Drum 44 FPM

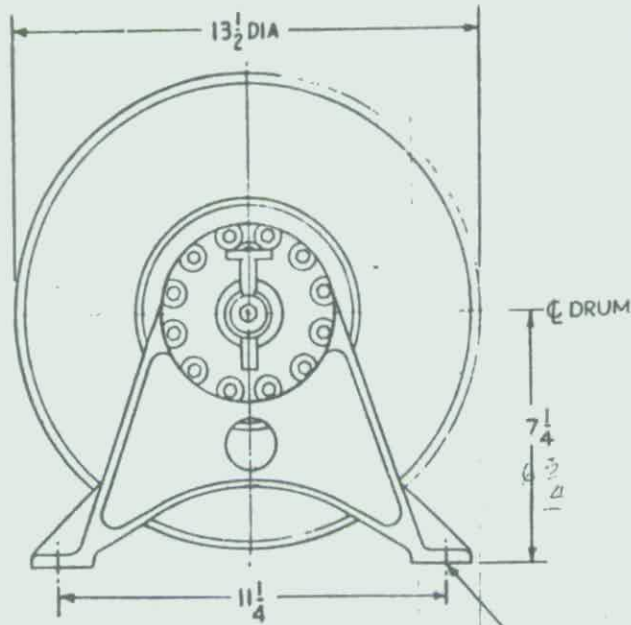
High Speed:

Bare Drum 127 FPM
Full Drum 236 FPM

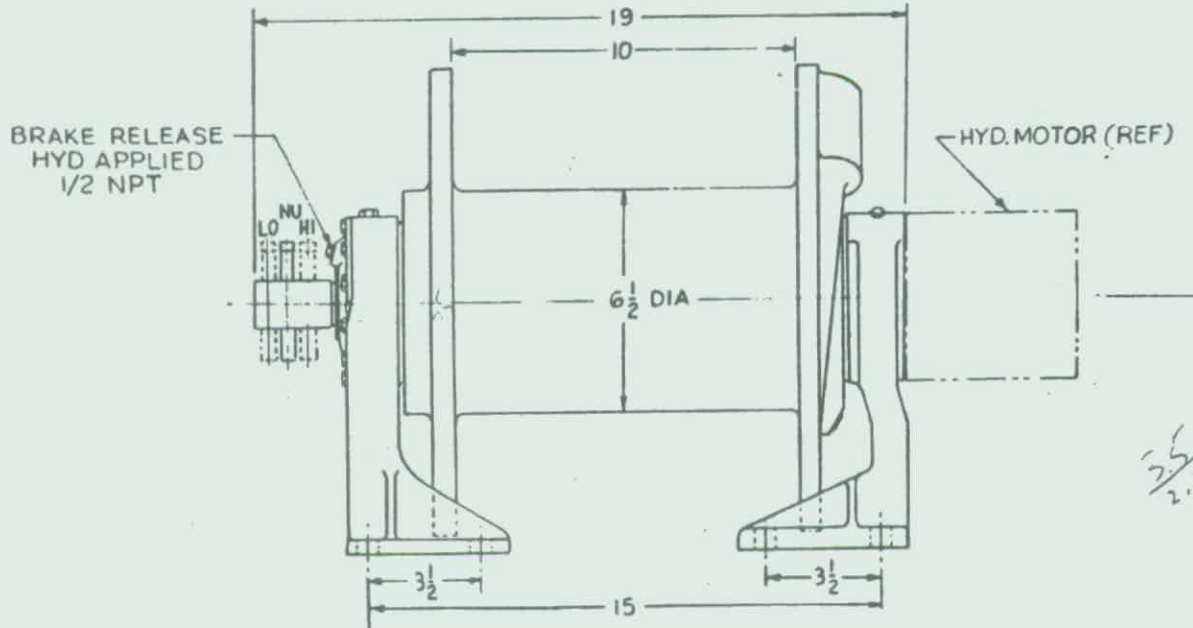
* The above are representative performance figures - based on 1800 RPM input speed.

13.5
7.6

6.5
4.6

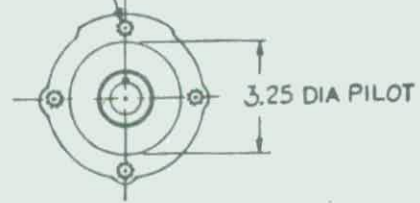


8 HOLES FOR 5/8 DIA BOLT



25
2.3

HYD MOTOR MOUNT
3/8-24UNF-2B 4-PLACES
ON A 4-3/16 DIA B.C.
SAE, SIZE A
2 BOLT FLANGE
(OPT ORIENTATION)



For specifications see reverse.

Form No. L-250F 7-68

	PACIFIC CAR AND FOUNDRY COMPANY <small>SENYON, WASHINGTON</small>		
	OUTLINE DRAWING P-10 WINCH	<table border="1"> <tr> <td>D</td> <td>43201</td> </tr> </table>	D
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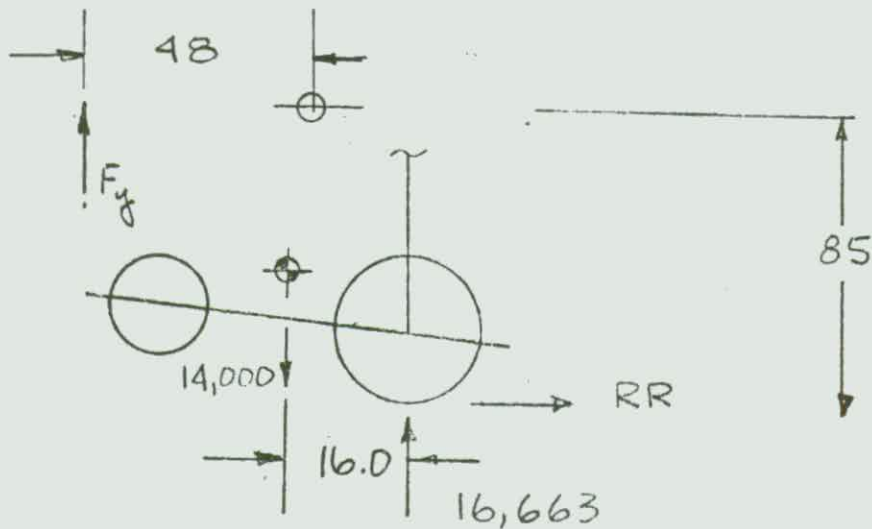
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$$RR = GVW \times R / 1000$$

$$RR = 16,663 \times 150 / 1000 = 2500 \text{ lb}$$

$$F_y(48) = 14000(16) + 2500(85)$$

$$F_y = 9100 \text{ lb}$$

F_y = FORCE REQUIRED TO RAISE VEHICLE
TO TOW POSITION.

ADDING 10% FOR FRICTION LOSSES

$$F_y = 10,000 \#$$

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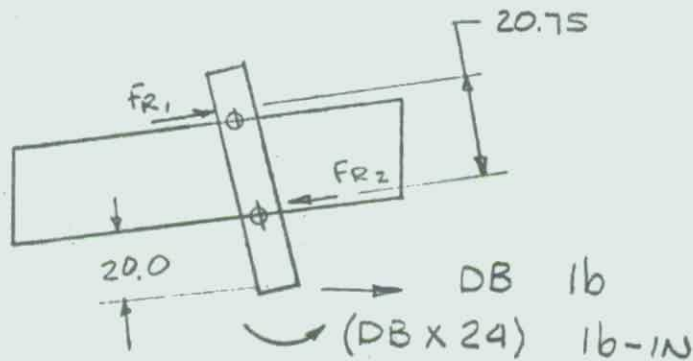
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MAXIMUM DRAWBAR PULL WITHOUT HOWITZER

DB =

$$F_{R2} = \frac{DB(40.75) + DB \times 24}{20.75} = \frac{DB(64.75)}{20.75} = 3.12 \text{ DB}$$

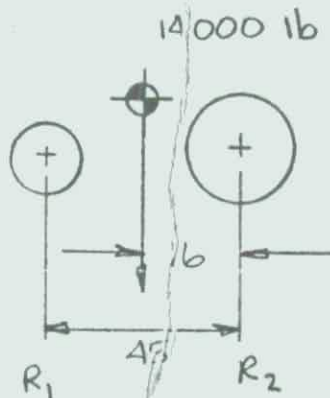
$$F_{R1} = \frac{20 \text{ DB} + 24 \text{ DB}}{20.75} = 2.12 \text{ DB}$$

FOR DB = WHEEL LOAD: ALL WEIGHT AND FULL TORQUE TO ONE (1) WHEEL

$$\sum M_{R1} = 0$$

$$\therefore R_2 = \frac{(29)(14,000)}{45}$$

$$R_2 = 9,030 \text{ lb}$$



$$F_{R1} = 19,100 \text{ lb}$$

$$F_{R2} = 28,200 \text{ lb}$$

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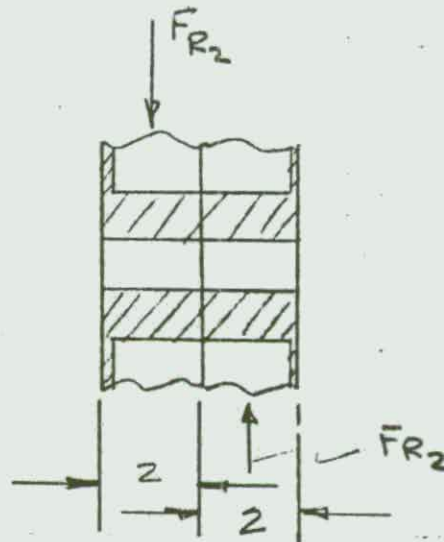
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FOR A 1.50 INCH DIA PIN IN SINGLE SHEAR

(AREA = 1.76 IN²)

$$Z = F/A = 28,200 / 1.76 = 16,000 \text{ PSI}$$



$$F_{R2} = 16,000 \text{ lb}$$

$$16,000 \times 6 = 96,000 \text{ PSI}$$

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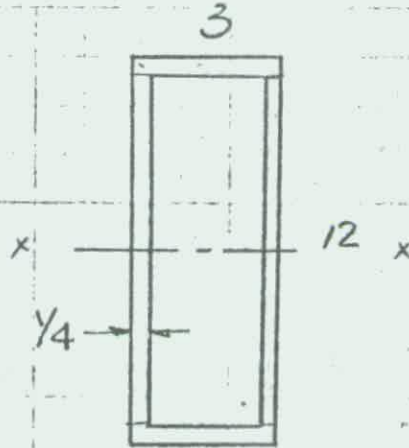
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FIND THE STRESS IN THE TUBE SUPPORT BEAM.

$$T = 1,120,000 \text{ lb-in} / 2 = 560,000 \text{ lb-in}$$

BEAM SECTION:



$$I_{xx} = 2(31.69) + 2(.004 + .750(5.87)^2)$$

$$I_{xx} = 64 + 52 = 116 \text{ IN}^4$$

$$\sigma = \frac{My}{I} = \frac{(560,000)(6)}{116} = 28,965 \frac{\text{lb}}{\text{IN}^2}$$

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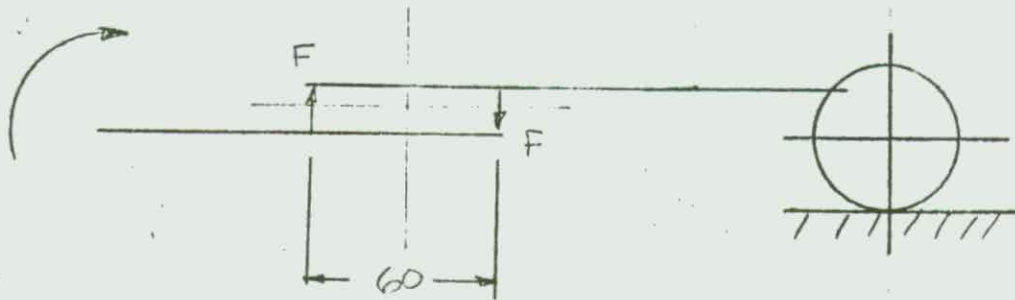
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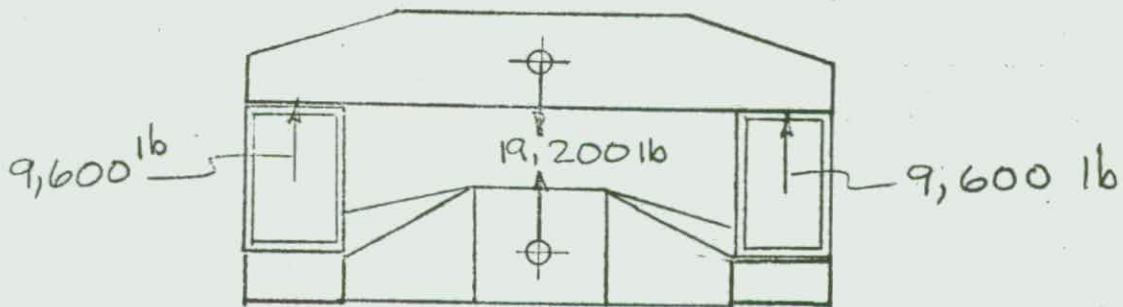
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FIND FORCE IN BEAM HOLDING TRAILS :

$$M = 1,157,500 \text{ lb-in}$$



$$F = 1,152,500 / 60 = 19,200 \text{ lb}$$



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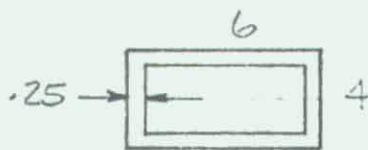
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BEAM SECTION: 6 X 4 X .25



$$I = (.008 + 1.5 \left(\frac{3.75}{2} \right)^2 + .89) 2$$

$$I = 12 \text{ IN}^4$$

$$M = 9,600(13) = 124,800$$

$$S = \frac{My}{I} = \frac{(124,800)(2)}{12} = 20,800 \frac{\text{lb}}{\text{IN}^2}$$

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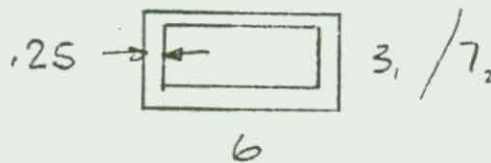
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$$F = 9,600$$

$$M = 36,825 \text{ lb-in}$$

$$I = (3 \times 6 \times .25)$$



$$I = 2 \left(.008 + 1.5 \left(\frac{2.75}{2} \right)^2 + .326 \right)$$

$$I = 5.8 \text{ in}^4$$

$$\sigma = \frac{M_y}{I} = \frac{36,825 (1.5)}{5.8} = 9523 \frac{\text{lb}}{\text{in}^2}$$

$$\text{FOR } M = 12,275 (12) = 147,000 \text{ in-lb}$$

$$I = 2 \left(.008 + 1.5 \left(\frac{6.75}{2} \right)^2 + 5.72 \right)$$

$$I = 23 \text{ in}^4$$

$$\sigma = \frac{M_u}{I} = \frac{(147,000)(3.5)}{23} = 22,370 \frac{\text{lb}}{\text{in}^2}$$

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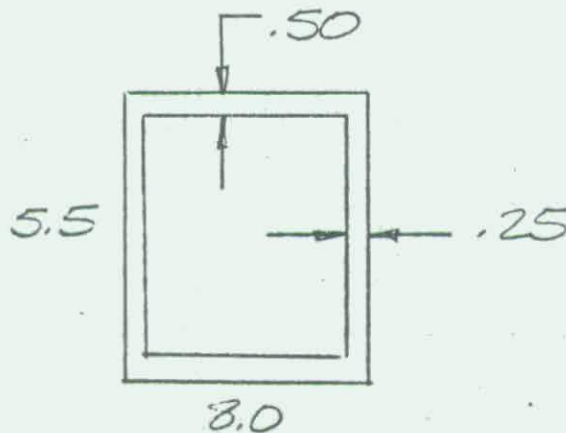
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$$T = 140,000 \text{ IN-LB}$$



$$S_1 = T / 2(.25)(8 - .25)(5.5 - .50)$$

$$S_2 = T / 2(.5)(8 - .25)(5.5 - .5)$$

$$S_1 = 140,000 / .5(7.75)(5) = 7,225 \text{ LB/IN}^2$$

$$S_2 = 140,000 / (.775)(5) = 3,612 \text{ LB/IN}^2$$

ASSUMING ALL TORQUE IS TRANSMITTED
THRU BOOM.

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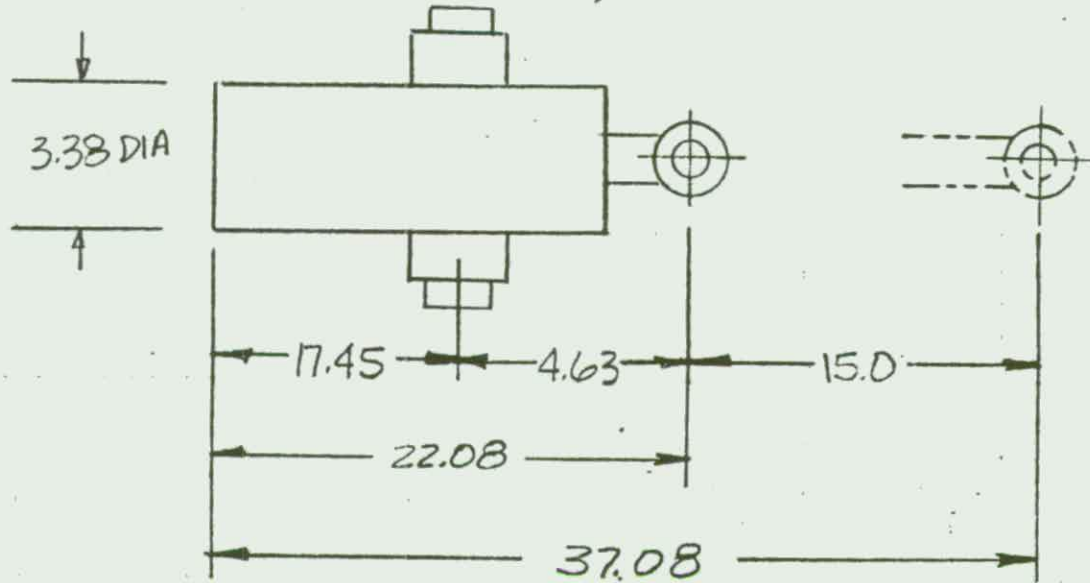
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BRUNING MALE TRUNNION TYPE 16
3.0" BORE, 1.00" ROD, RE3 TUBE EYE



$$6.283 \frac{\text{lb}_f}{\text{psi}}$$

$$F = 12,566 \text{ lbs @ } 2000 \text{ psi SYS HYD PRESS}$$

$$\frac{3.38}{2} = 1.69$$

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TAPERED SHEAR PIN :

$$T = 100,000 \text{ lb-IN}$$

$$\text{DIA PIN} = 1.000 \text{ AVE}$$

$$\text{AVE AREA} = .78 \text{ IN}^2$$

$$F = 100,000/4 = 25,000 \text{ lb}$$

FOR A ROUND SHAFT IN TORSION WITH TRANSVERSE HOLE :

$$\tau_0 = \frac{Tc}{J} \quad T = 100,000 \text{ lb-IN}$$

$$\frac{J}{c} = \frac{\pi D^3}{16} - \frac{dD^2}{6} \quad d = 1.000$$

$$\frac{J}{c} = \frac{(3.14)(4)^3}{16} - \frac{(1.0)(4)^2}{6}$$

$$\frac{J}{c} = 12.56 - 2.66 = 9.90$$

$$\tau_0 = 100,000 / 9.90 = 10,100 \text{ lb/IN}^2$$

$$K_{ts} = 2.9$$

$$\tau = K_{ts} \tau_0 = 2.9 (10,100) = 29,300 \text{ lb/IN}^2$$

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FOR A PIN LENGTH OF 8.00, THE
SMALL END IS: $[1.000 - 2(8)(.02083)]$

$$\begin{array}{r} 1.00000 \\ .33333 \\ \hline .66666 \end{array}$$

FOR A TAPER OF
 $\frac{1}{4}$ " PER FOOT.

DIA OF SMALL END = .667

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY _____

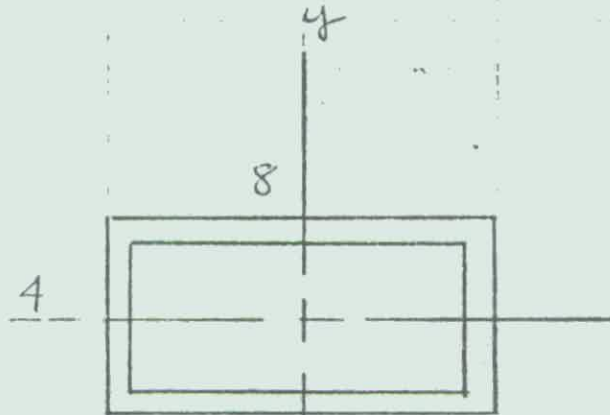
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BOOM SECTION
AT SUPPORT

STRESS DUE TO FULLY STEERING TORQUE:

$$I_{yy} = 2(21.33 + .03 + (1.50)(3.75)^2)$$

$$I_{yy} = 85 \text{ IN}^4$$

$$M = 100,000$$

$$y = 4.0$$

$$\sigma = \frac{(100,000)(4.0)}{85} = \underline{\underline{4700 \text{ lb/IN}^2}}$$

+ STRESS DUE TO STATIC
OR DYNAMIC LOAD.

SECTION 3

ENGINE MOUNTING

The mounting of an engine must not only provide adequate support for the engine in the desired location, but should do so in a manner that the engine is not subjected to excessive stresses imposed by power take-offs, shock load, or deflection of supporting members. The following section includes recommendations and cautions applicable to the mounting of the engine.

A. Bending Moment Restrictions

In the design of an engine mounting arrangement or power transmitting drive, care must be taken to ensure that the bending moment imposed on the engine is not excessive. A zero bending moment at the point the flywheel housing mounts to the engine is preferred for all engine installations. This is with the unit operational, i. e., all components installed and fluids included.

Where it is impossible to locate the engine mounts so as to achieve a zero bending moment, the allowable bending moment is limited to 1000 ft-lb. A sketch and formulae to determine the bending moment of an existing installation are contained on Chart 3-1.

1. In an automotive installation, the bending moment is based on the static load of the engine installed with all components and accessories and complete with water, oil, etc.
2. For industrial installations, the bending moment must not only include the static loading as for an automotive application, but must also include any dynamic loading imposed by the driven component such as a side pull through a chain drive.

Although the engine weight and center of gravity will vary depending upon the optional components and the mounting of accessories, the weight and center of gravity of the automotive version of all engines are included in Chart D of the Appendix for reference in making the necessary calculations.

11-3-73

306. BERRIC - LORD MFG. CALLED
(215-788-3321)

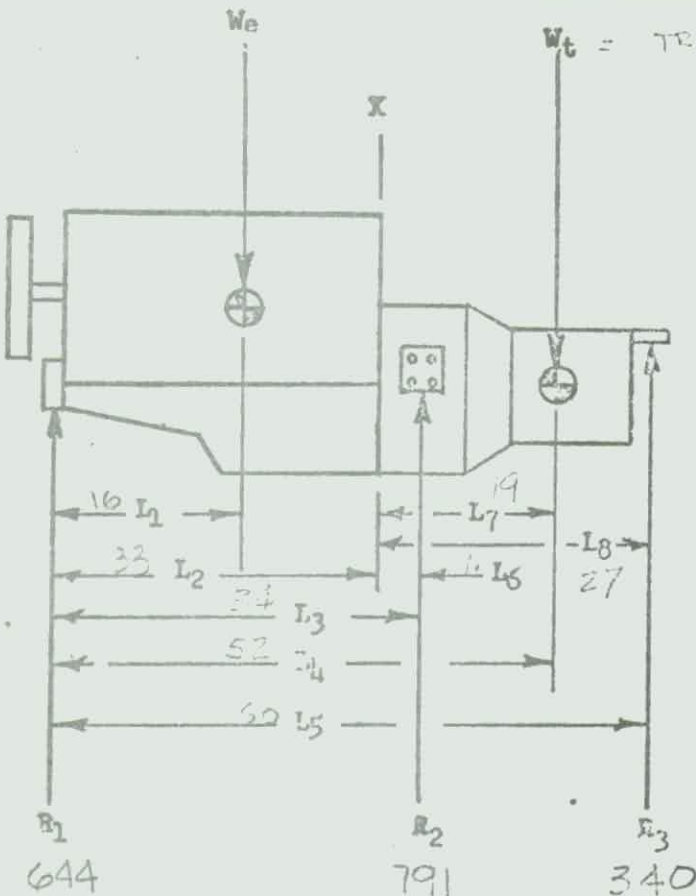
1. FIRING IMBALANCE OF ENGINE ARE MAJOR
HINDR TO ISOLATE (NOT SHAFT SPEED)
(2 CYCL - 4cyl is 4 TIMES SHAFT SPEED)
2. 0.25" IS TYPICAL STATIC DEFLECTION AT
RATED LOAD
3. SOFT MOUNT ON GEAR BOX. 3-400 lbs/in
CHANGE BEHAVIOR ON MILITARY APPLICATION
4. WILL SEND ADDITIONAL INFO - WILL BE
IN OWN WEEK OF 11-12-73

11-7-73

DOIT (243-3123) - YOUNG RAD (243-3123)

1. 4-6 ROW CORE DEPENDING ON FAN
(6200 ACFM @ 175°) < 1" STATIC PRESS.
2. NEEDS SPECIFIC FAN INFORMATION.
(SKIP WILL CALL D.D. ON THURS IF NOT
RECEIVED.)

DETERMINATION OF BENDING MOMENT FOR AN EXISTING INSTALLATION



$W_e =$ _____ lbs. eng. dry 1100
 oil 20
 water 15
 Total (W_e) 1135
 1250 *

$W_t =$ _____ lbs. trans. dry 255
 oil 7
 clutch 2
 Total (W_t) 264

GEARBOX 275
525 *

The engine mount reactions, R_1 and R_2 , must first be determined. To do this the tail support reaction, R_3 , must be assumed to be zero or a predetermined value which may be built into the unit.

$$R_2 = \frac{W_e L_1 + W_t L_4 - R_3 L_5}{L_3} = \text{_____ lbs.}$$

$$R_1 = W_e + W_t - R_2 - R_3 = \text{_____ lbs.}$$

M_x (Bending Moment Between Flywheel Housing & Block Face) =

$$M_x = R_2 L_6 + R_3 L_8 - W_t L_7 = \text{_____ in. lbs.}$$

$$\text{Check: } M_x = R_1 L_2 - W_e (L_2 - L_1) = \text{_____ in. lbs.}$$

Determination of transmission support preload to give zero M_x when locations of R_1 and R_2 are fixed.

$$R_3 = \frac{W_e L_1 + W_t L_4 - (\frac{W_t L_7 L_3}{L_6})}{L_5 - (\frac{L_8 L_3}{L_6})} = \frac{(1250)(16) + 525(52) - (525)(11)(34)}{60 - (27)(34)} = \frac{47300 - 29850}{858} = \underline{340} \text{ lbs.}$$

$$R_2 = \frac{W_e L_1 + W_t L_4 - R_3 L_5}{L_3} = \frac{(1250)(16) + (525)(52) - (340)(60)}{34} = \underline{791} \text{ lbs.}$$

$$R_1 = W_e + W_t - R_3 - R_2 = \underline{644} \text{ lbs.}$$

$$\text{Check: } M_x = 0 = R_1 L_2 - W_e (L_2 - L_1) = \underline{2.0} \text{ in. lbs.}$$

$$644(33) - 1250(17) =$$

PACIFIC CAR AND FOUNDRY COMPANY

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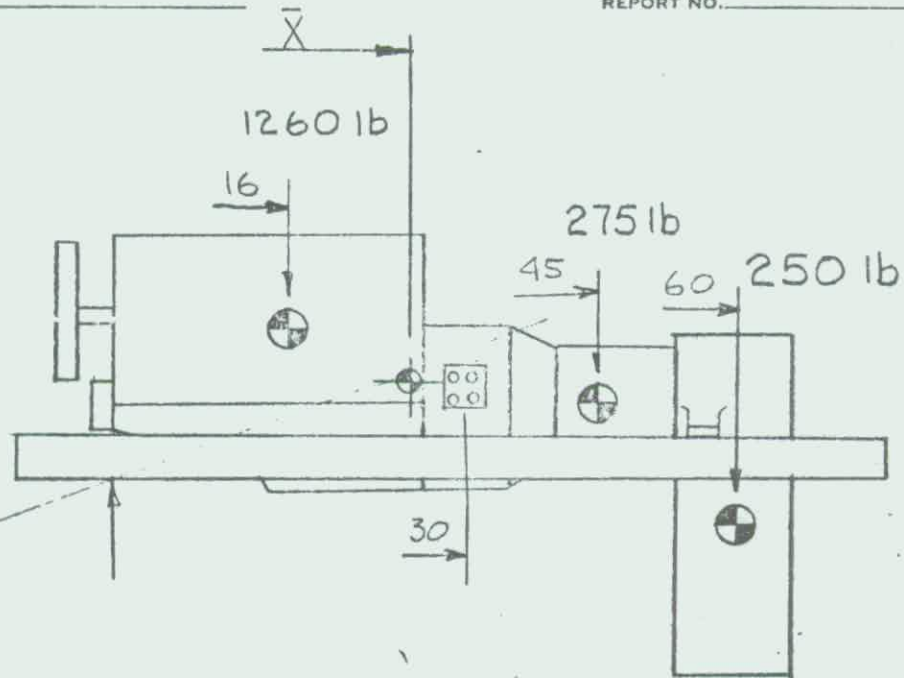
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$$\bar{X} = \frac{16(1260) + 45(275) + 60(250)}{1785}$$

$$\bar{X} = 27$$

ASSUMING THE TRANSMISSION AND GEAR BOX TO BE ONE UNIT, FIND THE COMBINED C.G. :

$$x = \frac{45(275) + 250(60)}{525} = 52$$

$$R_1 = 644 \text{ lb}$$

$$R_2 = 791 \text{ lb}$$

$$R_3 = 340 \text{ lb}$$

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MOUNT #1 (2) LOAD = 644 lb

MOUNT #2 (2) LOAD = $791/2 = 364$ lb

MOUNT #3 (1) LOAD = $340/2 = 170$ lb

DISTURBING FREQUENCY RANGE :

$$\frac{600(4)}{60} = 40$$

$$\frac{2800(4)}{60} = 254$$

PART NO.	$\frac{\text{ACTUAL LOAD}}{\text{RATED LOAD}}$	x	$\frac{\text{RATED DEF.}}{\text{ACTUAL DEF.}}$	=	ACTUAL DEF.
----------	--	---	--	---	-------------

J-6210-1	$644/765$	x	.078	=	.0655 IN
----------	-----------	---	------	---	----------

J-8006-1	$364/500$	x	.075	=	.0546 IN
----------	-----------	---	------	---	----------

WHAT ABOUT 10G SHOCK LOAD ?? (OK.)

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

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TRANSFER CASE MOUNT

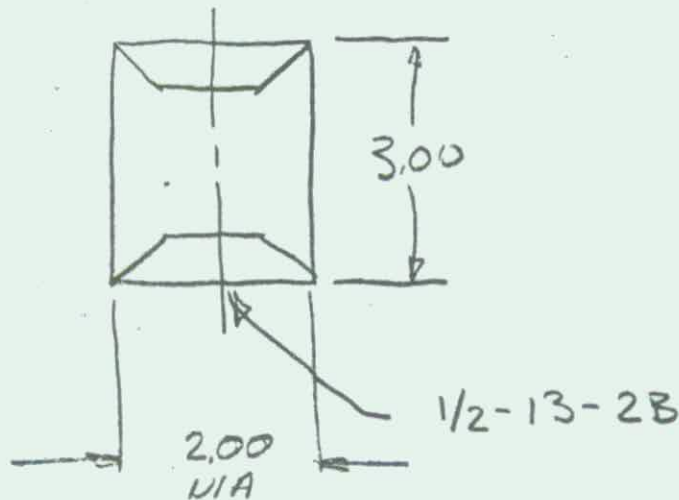
REQUIRED LOAD = 340 lb -

USING TWO MOUNTS $340/2 = 170$ lbs

J-3424-49 $K_c = 450$

$$\Delta = \frac{170}{450} = .378$$

} VERTICAL DEFLECTION



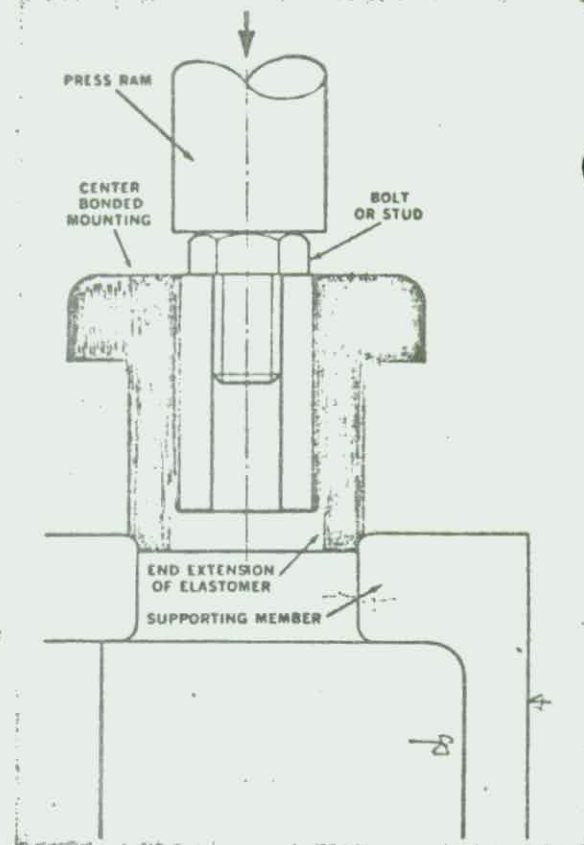
Center Bonded Mountings

1. ISOLATE FIRING IMPULSES
2. AVE STATIC DEF (.080") AT RATED LOAD
3. SOFT MOUNT ON GEAR BOX 4-800 LBS./INCH

The Center Bonded Mountings have been designed to provide multi-directional isolation of engines, pumps, compressors, etc. The mount provides a simple one-piece design. Because of the complexity of vibrating masses and the interaction between them, it is difficult to select a mounting based on load rating only. Further, since a vibration source may excite the total mass in two or more directions, commonly called coupling, one may have to try several stiffnesses in order to obtain an optimum solution. Generally speaking, one should begin with isolating the primary excitation, usually the operating speed of the unit.

The load ratings indicated are for static gravity loads. The mountings are capable of handling dynamic torque loads of 2.5 to three times the rated loads.

The spring rate can be determined by dividing the rated load by the rated deflection. The rated load is in the axial direction. Radial loads are acceptable; but static radial loads are not recommended.

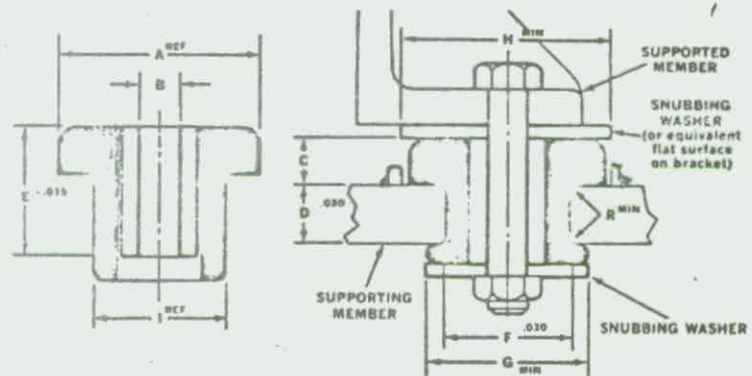


Installation: easy economical, uniform

Mountings may be installed in supporting member with rebound shoulder down, or in supported member with rebound shoulder up.

Installation is a simple, four-step procedure:

1. Lubricate the mounting and socket lightly with rubber lubricant or water. Lubricant available from International Products Corporation, Trenton, N. J.
2. Insert assembly fixtures or driving bolt through inner member. Take care that driving members do not overhang spine outside diameter or damage may result to the elastomer.
3. Apply sufficient pressure to seat the mounting socket.
4. Tighten the nut until the supported member and snubbing washer are snug against the inner member. The rebound shoulder is formed automatically. The resultant precompression will deliver published design ratings and assure optimum performance.



specifications

Part No.	Rated Load (Lbs.)	Rated Deflection (In.)	A	B	C ³	D	E	F	G	H	I	R
J-7153-1	125	.018	1.09	.40	.19	.31	.69	.75	1.12	1.25	.81	.06
J-6642-1	225	.035	1.75	.47	.38	.38	1.00	1.12	1.50	2.00	1.24	.06
J-6642-7	225	.035	1.75	.52	.38	.38	1.00	1.12	1.50	2.00	1.24	.06
J-6642-9	225	.028	1.75	.52	.38	.38	1.00	1.12	1.50	2.00	1.24	.06
J-6256-1	220	.037	2.00	.53	.45	.62	1.38	1.25	1.75	2.25	1.35	.06
J-6256-3	320	.029	2.00	.53	.45	.62	1.38	1.25	1.75	2.25	1.35	.06
J-6256-9	250	.028	2.00	.65	.45	.25	1.00	1.25	1.75	2.25	1.35	.03
J-6256-10	200	.040	2.00	.53	.45	.25	1.00	1.25	1.75	2.25	1.35	.03
J-6256-12	320	.033	2.00	.64	.45	.62	1.38	1.25	1.75	2.25	1.35	.06
J-6256-22	220	.037	2.00	.64	.45	.62	1.38	1.25	1.75	2.25	1.35	.06
J-8006-1	500	.075	2.50	.64	.56	.75	1.75	1.50	2.25	2.88	1.62	.06
J-8006-6	650	.042	2.50	.64	.56	.75	1.75	1.50	2.25	2.88	1.62	.06
J-6210-1	765	.078	2.98	.64	.71	.93	2.00	1.81	2.50	3.50	1.98	.12
J-6210-4	950	.068	2.98	.64	.71	.93	2.00	1.81	2.50	3.50	1.98	.12
J-6210-36	765	.078	2.98	.64	.71	.50	1.50	1.81	2.50	3.50	1.98	.12
J-6198-1	900	.10	3.75	.77	.94	.75	2.12	2.00	2.75	4.25	2.25	.12
J-6198-2	900	.07	3.75	.77	.94	.75	2.12	2.00	2.75	4.25	2.25	.12
J-6198-3	900	.05	3.75	.77	.94	.75	2.12	2.00	2.75	4.25	2.23	.12
CB-1009-1	1200	.12	3.75	.77	.94	.75	2.00	2.00	3.00	4.25	2.23	.12
CB-1009-9	1600	.08	3.75	.77	.94	.75	2.00	2.00	3.00	4.25	2.23	.12
J-8635-1	1800	.145	4.50	1.02	1.12	1.75	3.50	2.75	4.00	5.00	2.98	.12

(2) ENG FLWHL HSG → J-8006-1
 (1) ENG. FRONT → J-6210-1

NOTES:
³ Special load ratings available on request.
² Unless otherwise specified all dimensions are nominal. See drawings for tolerances.
¹ Head thickness after assembly with no external load.

Shear Sandwich Mountings

WICH
MOUNTS

Part No.	Elastomer	SHEAR		COMPRESSION		T (inch)	SIDE ONE		SIDE TWO	
		Load (lbs.)	K-	Load (lbs.)	Kc		Stud Length or Minimum Thread Depth (inch)	Thread	Stud Length or Minimum Thread Depth (inch)	Thread
J-3424-49	NR	24	60	190	450	3.00	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-9	NR	25	75	210	638	2.62	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-8	NR	33	100	280	850	2.62	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-7	NR	34	150	310	1,350	2.12	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-30	NR	39	250	385	2,500	1.75	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-6	NR	46	200	415	1,800	2.12	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-16	NR	57	375	575	3,750	1.75	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-80	NR	57.5	250	515	2,250	2.12	.50	1/2-13-2B	.40	1/2-13-2B
J-3424-70	NR	103	450	931	4,050	2.12	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-5	N	34	150	310	1,350	2.12	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-2	N	46	200	415	1,800	2.12	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-3	N	80	350	725	3,150	2.12	.40	1/2-13-2B	.40	1/2-13-2B
J-3424-21	N	92	400	828	3,600	2.12	.40	1/2-13-2B	.40	1/2-13-2B
J-4624-165	NR	4	95	23	560	.75	.62	1/4-20-2A	.25	1/4-20-2B
J-4624-150	NR	5	125	40	750	.75	.38	1/4-20-2A	.25	1/4-20-2B
J-4624-19	NR	6	55	36	320	.75	.38	1/4-20-2A	.38	1/4-20-2A
J-4624-17	NR	6.5	60	40	360	.75	.38	1/4-20-2A	.38	1/4-20-2A
J-4624-10	NR	40	750	168	4,200	.75	.38	1/4-20-2A	.25	1/4-20-2B
J-4624-109	N	3	28	22	195	.75	.38	1/4-20-2A	.38	1/4-20-2A
J-4624-176	N	6	55	32	290	.75	.38	1/4-20-2A	.38	1/4-20-2A
J-4624-53	N	8	170	44	1,100	.75	.50	1/4-20-2A	.25	1/4-20-2B
J-4624-225	N	9.5	190	56	1,121	.75	.75	1/4-20-2A	.25	1/4-20-2B
J-4624-69	N	11	100	70	620	.75	.38	6-32-2A	.38	6-32-2A
J-4624-4	N	11	100	70	620	.75	.38	1/4-20-2A	.38	6-32-2A
J-4624-1	N	12	105	64	620	.75	.38	1/4-20-2A	.38	1/4-20-2A
J-4624-14	N	12	105	64	620	.75	.50	1/4-20-2A	.50	1/4-20-2A
J-4624-27	N	12	105	64	620	.75	.75	1/4-20-2A	.75	1/4-20-2A
J-4624-23	N	17	150	110	1,000	.75	.62	1/4-20-2A	.62	1/4-20-2A
J-4624-3	N	27	200	135	1,200	.75	.88	5/16-18-2A	.50	5/16-18-2A
J-4624-16	N	22	200	135	1,200	.75	.88	5/16-18-2A	.50	5/16-18-2A
J-4624-45	N	22	200	135	1,200	.75	.75	5/16-18-2A	.75	5/16-18-2A
J-4624-85	N	22	200	135	1,200	.75	1.38	5/16-18-2A	1.12	5/16-18-2A
J-4624-32	N	29	260	200	1,800	.75	.50	1/4-20-2A	.50	1/4-20-2A
J-5130-55	NR	450	1,400	3,400	10,500	2.12	.53	1/2-20-2B	.53	1/2-20-2B
J-5130-1	N	550	1,700	4,130	12,750	2.12	.53	1/2-20-2B	.53	1/2-20-2B
J-5294-21	N	130	350	1,050	2,800	2.25	.81	5/16-18-2A	.38/19**	Flange
J-5294-2	N	190	500	1,500	4,000	2.25	.81	5/16-18-2A	.38/19**	Flange
J-5425-15	NR	92	175	596	1,140	3.00	1.25	1/2-13-2A	1.25	1/2-13-2A
J-5425-30	B	138	250	900	1,625	3.00	1.31	1/2-13-2A	1.31	1/2-13-2A
J-5425-1	NR	157	300	1,021	1,950	3.00	1.25	1/2-13-2A	1.25	1/2-13-2A
J-5425-16	B	221	550	1,670	4,125	2.25	1.50	1/2-13-2A	1.31	1/2-13-2A
J-5425-6	NR	300	600	2,040	3,900	3.00	1.25	1/2-13-2A	1.25	1/2-13-2A

N - Neoprene; NR - Natural Rubber; B - Buna; BTR - Lord Broad Temperature Range Elastomer

DETROIT DIESEL

TRUCK MODELS

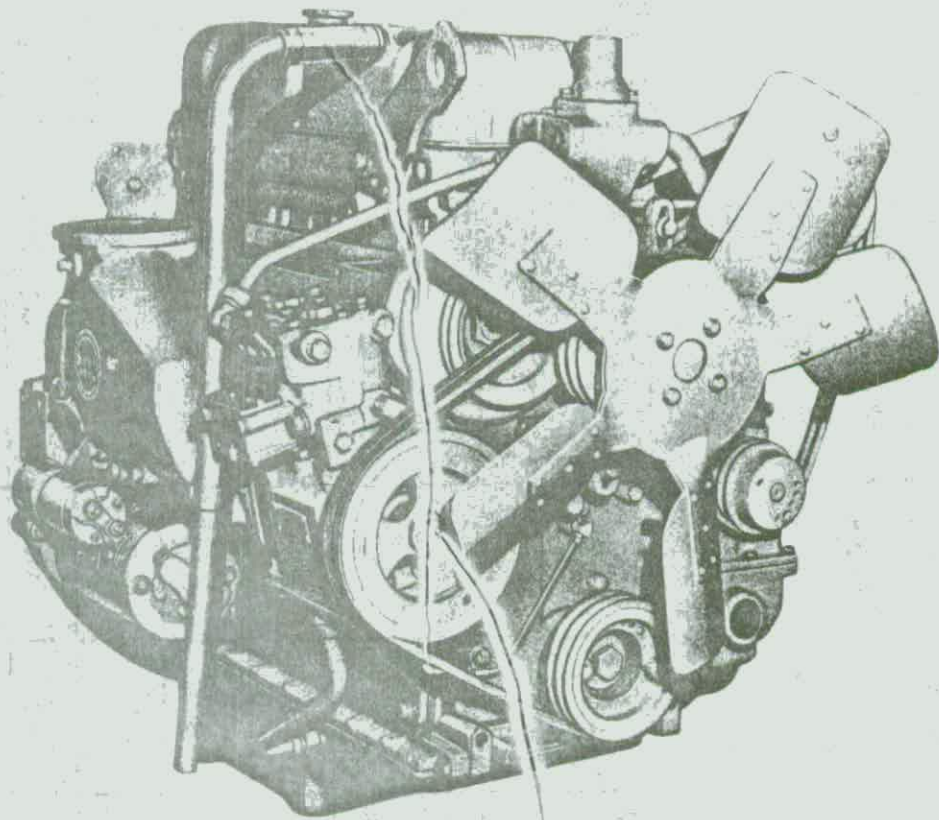
4-53N

116 HP | 130 HP | 140 HP

MODELS

4-53N

5047-5040
5047-7040
5047-7041



Model 5047-7040



SPECIFICATIONS

	4-53 (N40)	4-53 (N45)	4-53 (N50)
Model	Hi-Economy	Balanced Match	Hi-Performance
Engine Type	Two Cycle	Two Cycle	Two Cycle
No. of Cylinders	4	4	4
Bore and Stroke	3 3/8 in. x 4 1/2 in.	3 3/8 in. x 4 1/2 in.	3 3/8 in. x 4 1/2 in.
Two Cycle Displacement (Every Downstroke a Powerstroke)	212 cu. in.	212 cu. in.	212 cu. in.
Rated Brake Horsepower—2800 RPM	116	130	140
Torque	259 lb. ft. @ 1500 RPM	278 lb. ft. @ 1800 RPM	286 lb. ft. @ 1800 RPM
Compression Ratio	21 to 1	21 to 1	21 to 1
Net Weight (Dry) with Standard Equipment	1190 lbs.	1190 lbs.	1190 lbs.

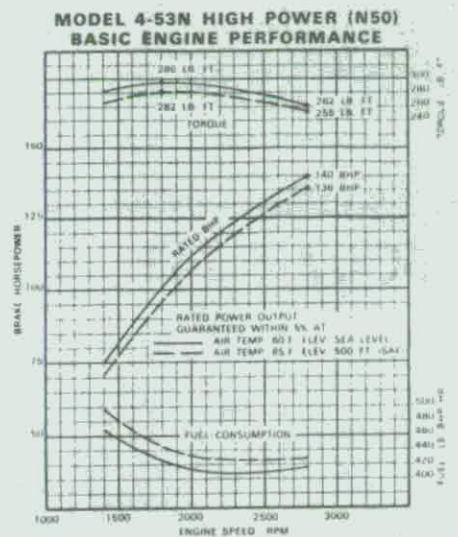
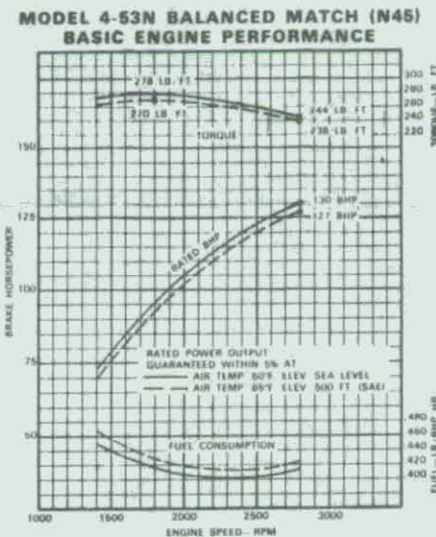
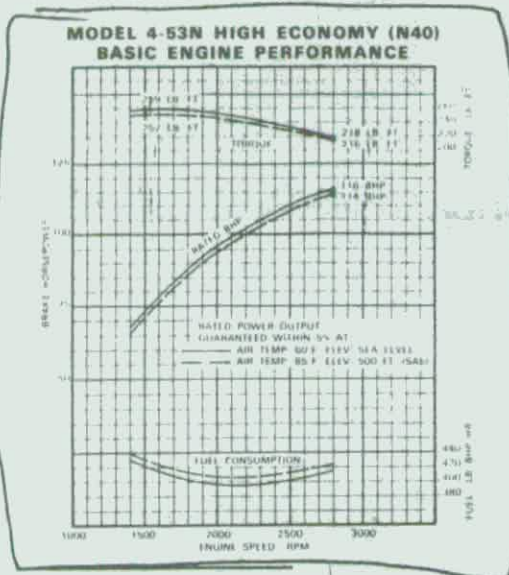
STANDARD EQUIPMENT

- Air Compressor—7 1/4 CF
- Air Inlet Housing—Manual shutdown with 50" cable
- Exhaust Manifold—With center horizontal outlet and flange
- Fan—22"—5 blade, suction
- Flywheel—SAE #2 for 14" clutch
- Flywheel Housing—SAE #2

- Generator—12 volt—25 amp.
- Governor—Limiting speed
- Injectors—Cam-operated, Unit type
- Lube Oil Filter—Full flow filter
- Oil Pan and Distribution System—10 degree inclination
- Starting Equipment—12 volt—Sprag clutch

OPTIONAL AND EXTRA EQUIPMENT AVAILABLE

PERFORMANCE



Notes, Explanation

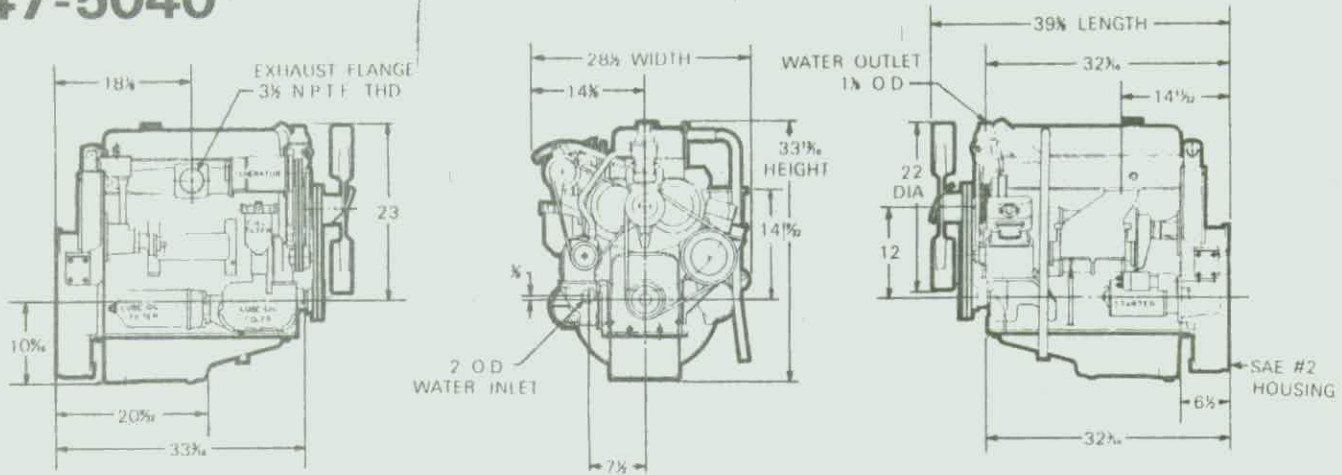
1. The graph on the power rating for various applications is based on the net weight of the engine and required accessories.

2. The Fuel Consumption Curve shows fuel used in pounds per Brake horsepower hour.

3. The BALANCED MATCH engine meets power requirements for accessories, if standard equipment.

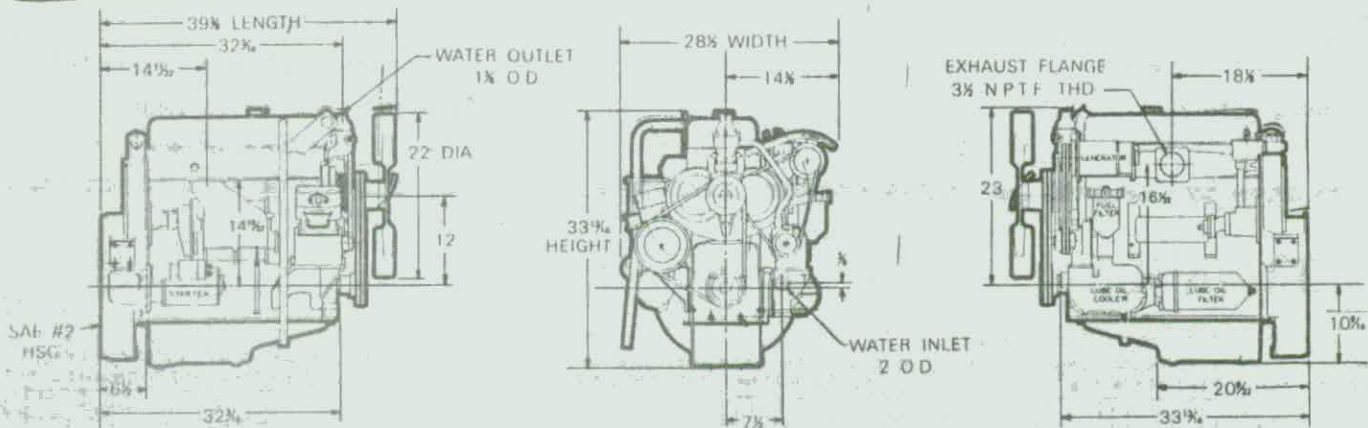
PRINCIPAL DIMENSIONS

5047-5040



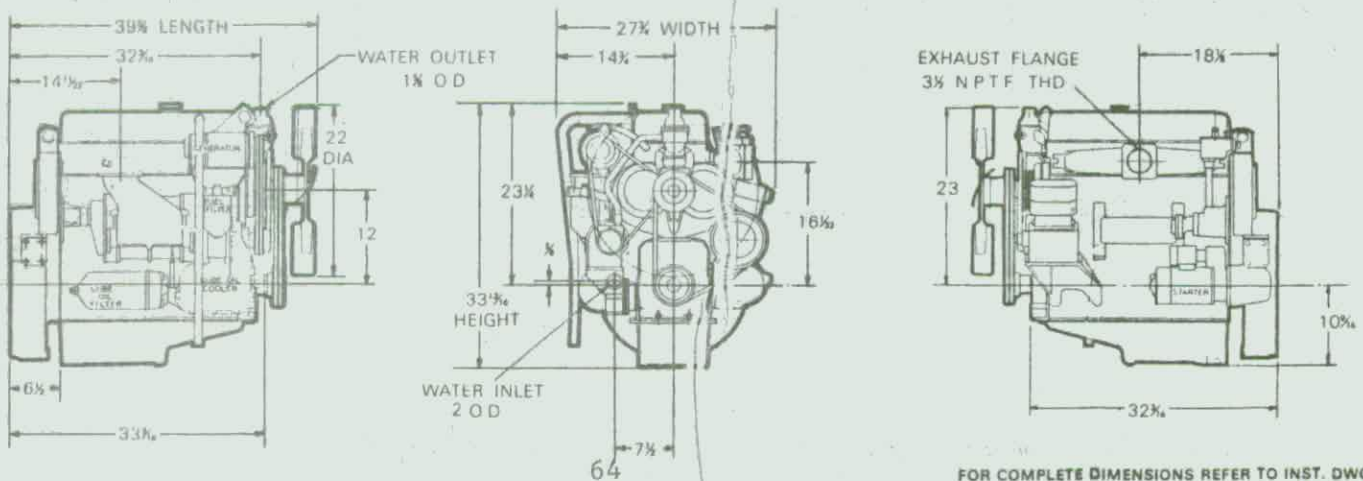
FOR COMPLETE DIMENSIONS REFER TO INST. DWG. 2SA297

5047-7040

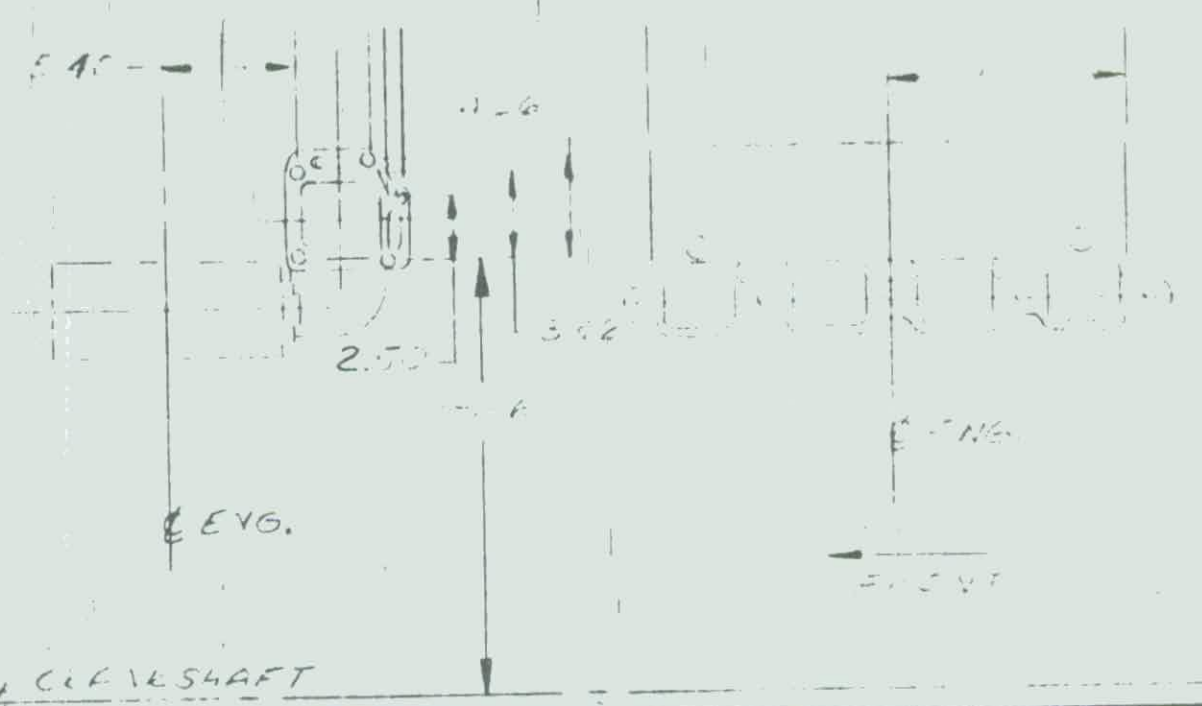


FOR COMPLETE DIMENSIONS REFER TO INST. DWG. 2SA270

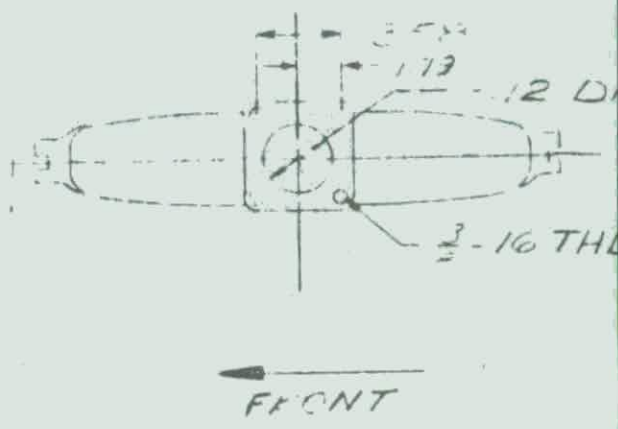
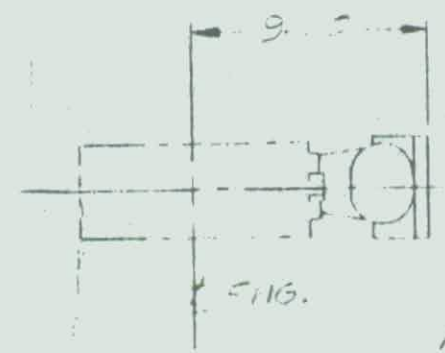
5047-7041



FOR COMPLETE DIMENSIONS REFER TO INST. DWG. 2SA314



5100146
 5100146
 2



CRANKSHAFT

4-53

PARTS LIST

DETROIT DIESEL ENGINE DIVISION
GENERAL MOTORS CORPORATION

05T00 1364 01
NO.

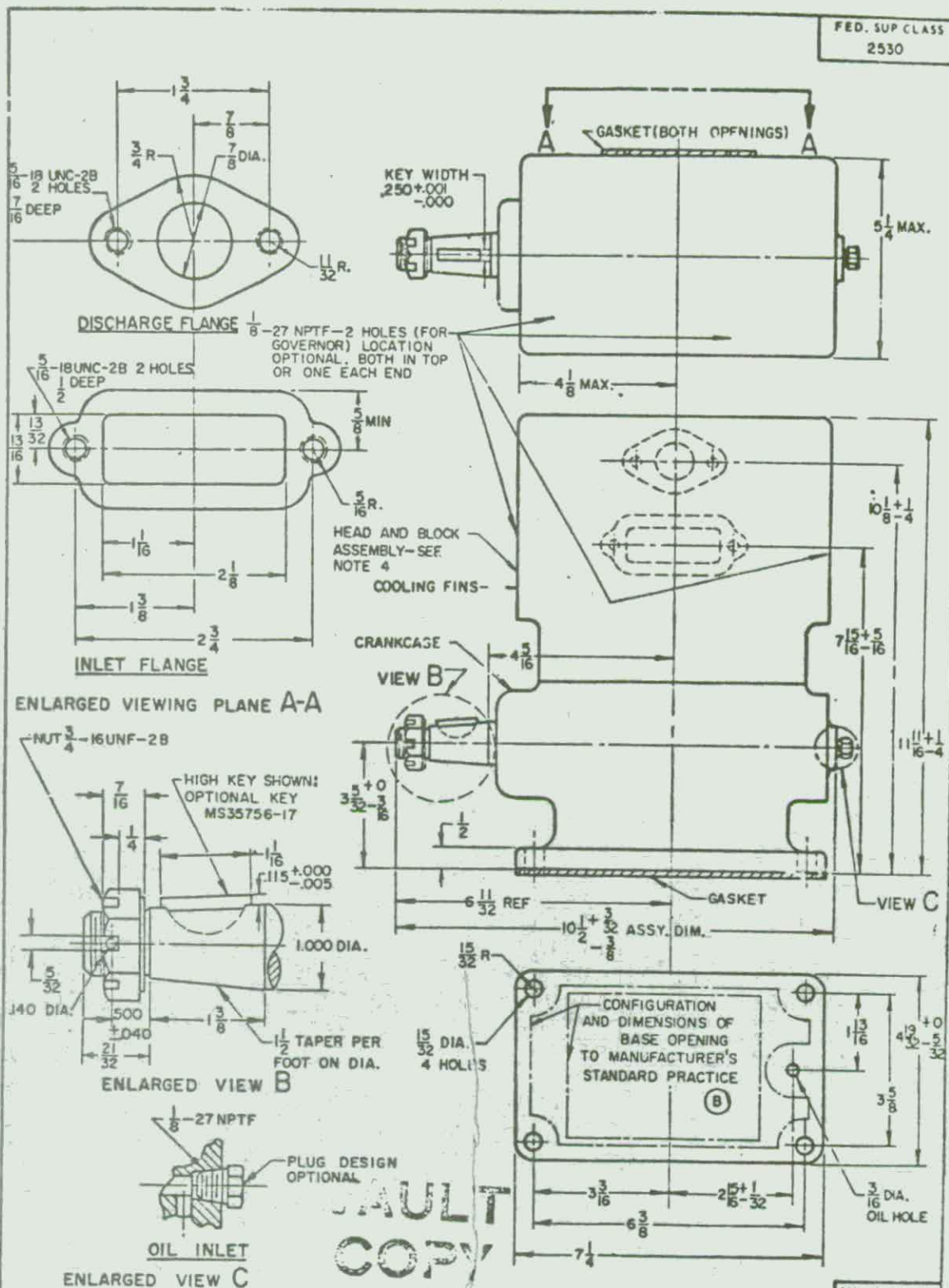
LINE	PART NO.	QTY.	CODE	PART NAME	REMARKS
1	5145617	1		COMPRESSOR AIR 12CFM □BW 283390	PA
	5145014	6		PLUG PIPE 3/8-18PTF SHTHEXSOC TEF	HOLES □WAT □/
3	5145009	1		PLUG PIPE 1/8-27NPTFHEXSOC TEF	O/HOLE WHR REQ
4	5130964	1		PULLEY COMP 6.25 DIA 2 GROOVE	
5	103385	1		PIN COT 1/8X1	
6	5145594	1		BRACKET AIR COMP MTG	□#
7	106337	6		BOLT 7/16-14X1 3/8 □COMP TJ	
8	103322	6		LW 7/16 □BRKT	
9	186622	4		BOLT 3/8-16X1 1/4 □BRKT	
10	5157244	1		BOLT 3/8-16X1 3/4 SPL □TJ	
11	103321	5		LW 3/8 □CYL BLK	
12	5131939	1		GASKET AIR COMP □TO COMP BASE	
13	5110411	1		STRAINER ASM AIR COMP	PA
14	5110410	1	R	GASKET AIR COMP	FURN WITH COMP
15	179827	2		BOLT 5/16-18X2 1/4	
16	103320	2		LW 5/16	
17	DED 5150023	1	D	COVER CYL BLK WAT HOLE PLN	IN 6A1
18	5115097	1		COVER CYL BLK WAT HOLE 3/8NPT	
19	DED 179816	2	D	BOLT 5/16-18X3/4 □CVR TO BLK	IN 6A1
20	186625	2		BOLT 5/16-18X7/8 □CVR TO BLK	
21	5109116	1		HOSE 1/2 X 10	
22	5120020	2		FITTING FEM SWIV 1/2 HOSE 3/4-16	
23	118757	2		EL 90 DEG 1/2 TUB 3/8 NPT □COMP FRT	
24	DED 5145014	1	D	PLUG PIPE 3/8-18PTF SHTHEXSOC TEF	IN 6K3d
25	5121810	1		HOSE 1/2 X 12 1/2	
26	5120020	1		FITTING FEM SWIV 1/2 HOSE 3/4-16 □IN	PA
27	118752	1		CONN 1/2 TUB 3/8 NPT □IN THERM HSG	PA
28	118757	1		EL 90 DEG 1/2 TUB 3/8 NPT	□COMP □
29	DED 5145009	1	D	PLUG PIPE 1/8-27NPTFHEXSOC TEF	IN 6J
30	AAAL0105	1		HOSE ASM NO6 13.8IN A&A	
31	5100136	1		ADAPTOR 1/8 PIPE	COMP BRKT
32	187322	2		EL 90 DEG 3/8 TUB 1/8 NPT	
33	2488022	1		CLIP 3/8 NEOPRENE □LWR FRT	CVR 3OLT
34	5163529	1		CLIP 5/16 DIA □TO LWR FRT CVR BOLT	
35	5118398	1		HOSE 5/8X10 □COMP BRKT TO OIL PAN	□□
36	5113215	1		FITTING MALE 5/8 HOSE 1/2-14 □BRKT END	
37	DED 5145012	1	D	PLUG PIPE 1/2-14HEXSOC TEF □OIL PAN	IN 6E
38	9402801	1		ELBOW 45 DEG 5/8 TUB 7/8-14 EXT □IN	OIL PAN
39	5113192	1		FITTING FEM SWIV 5/8 HOSE 7/8-14	TO 9402801
40	SK 3253	1	R	SKETCH INSTAL	
41					
42				SEE 6K1B FOR DRIV BELTS	L-08-234
43				/IN COMP WHERE REQ	
44				HOSE 5134580 HOSE 5/8X16 WHEN	L-27-274(5)
45				6E-175 IS SPEC W/SUMP TO FRT	
46				#MOVE VOLT REG TO RR-MT TO SI OF BLK	
47				IMMEDIATELY AHEAD OF STARTER	
48					
49					
50					
51					
52					
53					
54					
55	REVISIONS:			REINSTATED	
56					
57	DISTRIBUTION			53-0	REF 6T-550
58	DESCRIPTION			COMP 12 CR/SHF TRI DR RHSI STMP PAN 53	
59					
60				DATE ISSUED 05-08-73 DATE SUPERSEDED 07-05-72 SHEET 01 OF 0001 GROUP NO. 06T00 136	

DE 2400 EDP 235

1

FED. SUP CLASS
2530

REVIEW ACTIVITY: AF-85 *
USER ACTIVITIES: ARMY-ME; NAVY-MC, YD



This military standard is mandatory for use by all Departments and Agencies of the Department of Defense. Selection for all new engineering and design applications and for repetitive use shall be made from this document.

APPROVED 12 SEPT 1962 REVISED (A) 5 APRIL 1967 (B) 25 MARCH 1968

P.A. ARMY-AT		TITLE		MILITARY STANDARD	
Other Cust		COMPRESSOR, RECIPROCATING, POWER DRIVEN: AIR COOLED, 12 C.F.M. W/ UNLOADER		MS 53032	
AF-84		SUPERSEDES		MS PART NO. MS 53032-1	
PROCUREMENT SPECIFICATION MIL-C-45334				SHEET 1 OF 2	

DD FORM 672-1 (Continued)

PREVIOUS EDITIONS OF THIS FORM ARE OBSOLETE.

(B)

NOTES:

1. THIS COMPRESSOR SHALL CONFORM TO MIL-C-45334, TYPE II.
2. RECOMMENDED MAXIMUM OPERATING SPEED 2400 RPM.
3. MAXIMUM BUILD-UP TIME TO PUMP A 1000 CU. IN. RESERVOIR TO 100 PSIG:
21 SECONDS - AT COMPRESSOR SPEED OF 2400 RPM.
33 SECONDS - AT COMPRESSOR SPEED OF 1200 RPM.
4. COMPRESSOR SHALL BE CONSTRUCTED TO ALLOW 180° ROTATION OF THE HEAD AND BLOCK ASSEMBLY TO PROVIDE FOR RIGHT OR LEFT LOCATION OF THE INLET AND DISCHARGE FLANGES. MANUFACTURER SHALL FURNISH AN INSTRUCTION SHEET, TOGETHER WITH A SPARE GASKET FOR EACH COMPRESSOR.
5. MAXIMUM WEIGHT: 37 POUNDS.
6. DIMENSIONS ARE IN INCHES. UNLESS OTHERWISE SPECIFIED, TOLERANCES ARE $\pm 1/64$ ON FRACTIONS, $\pm .005$ ON DECIMALS, $\pm 1^\circ$ ON ANGLES, $\pm 1/16$ ON CASTINGS.
7. THREADS SHALL BE IN ACCORDANCE WITH SCREW THREAD STANDARDS FOR FEDERAL SERVICES HANDBOOK H28.
8. THIS STANDARD IS NOT INTENDED TO LIMIT CONSTRUCTION TO FEATURES OTHER THAN AS SHOWN HEREON, BY DIMENSIONS, NOTATIONS, OR REFERENCED DOCUMENTS.
9. REFERENCED DOCUMENTS SHALL BE OF THE ISSUE IN EFFECT ON DATE OF INVITATIONS FOR BID.
10. FOR DESIGN FEATURE PURPOSES, THIS STANDARD TAKES PRECEDENCE OVER PROCUREMENT DOCUMENTS REFERENCED HEREIN.
11. MARKING SHALL CONSIST OF THE MS PART NUMBER, MANUFACTURER'S IDENTIFICATION AND SERIAL NUMBER IN ACCORDANCE WITH MIL-STD-130.

This military standard is mandatory for use by all Departments and Agencies of the Department of Defense. Selection for all new engineering and design applications and for repetitive use shall be made from this document.

**VAULT
COPY**

APPROVED 12 SEPT 1962 REVISED (A) FOR CHANGES SEE SHEETS 1 AND 2 (B) FOR CHANGES SEE SHEETS 1 AND 2

P.A. ARMY-AT Other Cust	TITLE COMPRESSOR, RECIPROCATING, POWER DRIVEN: AIR COOLED, 12 C.F.M. W/ UNLOADER	MILITARY STANDARD MS53032
AF-84	SUPERSEDES	SHEET 2
PROCUREMENT SPECIFICATION MIL-C-45334		

DD FORM 672-1 COORDINATED

PREVIOUS EDITIONS OF THIS FORM ARE OBSOLETE

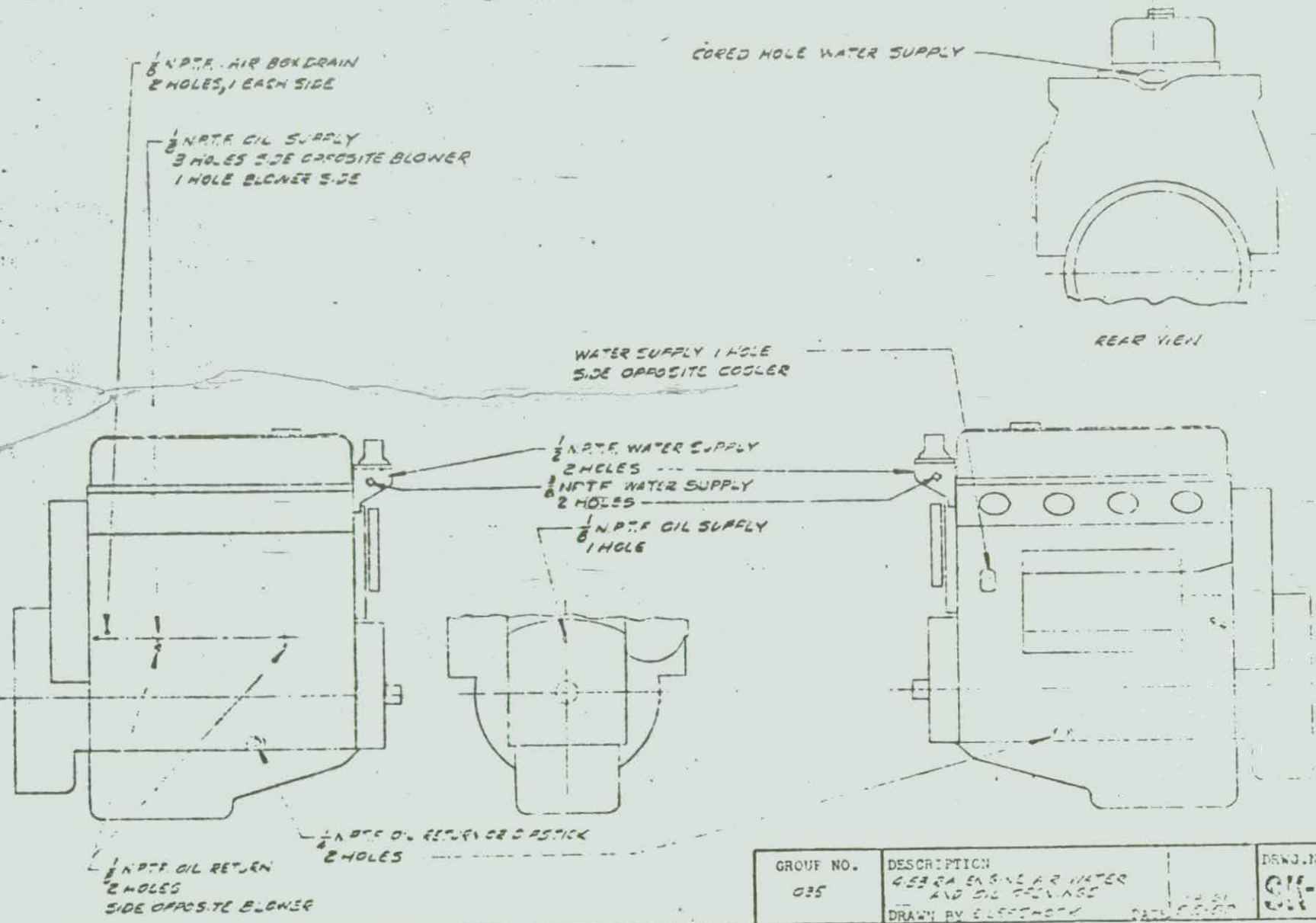
★ U. S. GOVERNMENT PRINTING OFFICE: 1960-301-519/5477

— OPTION INSTALLATION —

GENERAL MOTORS CORPORATION, DETROIT 26, MICH.

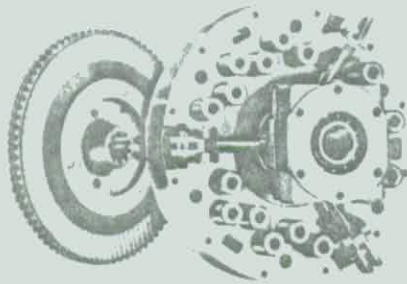
70

DETROIT DIESEL ENGINE DIVISION



GROUP NO. 035	DESCRIPTION GEE RA ENGINE AIR WATER AND OIL FEEDLINE DRAWN BY ELECTROTEK	DWG. NO. 51-3000
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NEW! HEAVY DUTY PULL-TYPE, FULL OIL-FLOW CLUTCHES



These full oil flow wet clutches deliver top performance in motor trucks and as part of the drive lines for other heavy duty automotive equipment. Handle range of engine sizes from 300 to 1100 lbs.-ft. Long life and exceptional dependability are due to full oil flow continuously pumped by internal-external gear pump to all friction surfaces of the plates. Included are integral oil pump, output shaft brake and 2, 3, or 4 plates depending on torque outputs. Compact clutch, brake, pump and sump are in one housing. Remote sump available when required.

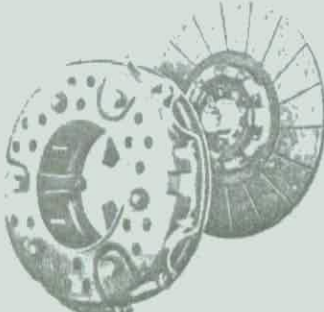
NEW OIL spring loaded clutch



NEW OIL over center clutch



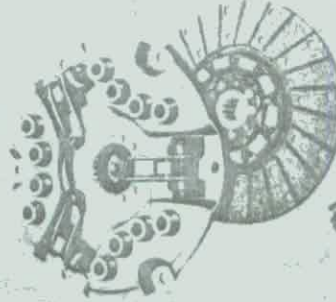
MOST ROCKFORD SPRING LOADED CLUTCHES MAY BE HAD WITH OUR VIBRATION DAMPENER FEATURE



Model RT Single plate, dry, for flat flywheel, Sizes 11, 12, 13, 14 in.; for counterbored flywheel in 14 and 15 in. sizes. Multiple plate available.



Model RM 6 1/2, 8, 8 1/2, 9, 10, 11 in. dia. Available with flat cover plate for use with counterbored flywheel; some sizes available with cupped cover plate for use with flat flywheel.



Model FA Wet or dry operation. With or without dual drive design for constant-running auxiliary drive, with PTO splined hub.



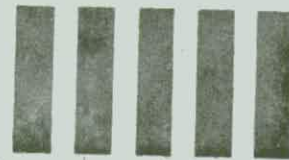
Model TS Single plate, dry; 5 1/2, 6, 6 1/2 in.; 50 to 75 ft. lb. max. capacity. Used in midget cars and trucks, small garden tractors, and auxiliary drives in industrial trucks.

These partial specifications represent only a few of the available sizes

CLUTCH		TORQUE CAPACITY	BELL HOUSING MIN. SIZE S.A.E. NO.	CLUTCH WEIGHT	OUTSIDE DIA. OF CLUTCH	CLUTCH MOUNTING BOLT CIRCLE
PULL TYPE FULL OIL FLOW	2 PLATE	400-800 FT. LBS.	2	74.7 LBS.	14.997	14.250
	3 PLATE	600-1200 FT. LBS.	2	84.0 LBS.	14.997	14.250
	4 PLATE	1000-1600 FT. LBS.	2	94.6 LBS.	14.997	14.250
11" RT		320 FT. LBS.	4	26 LBS.	13.000	12.377 12.373
12" RT		430 FT. LBS.	3	31 LBS.	14.690	13.502 13.498
13" RT		520 FT. LBS.	3	48 LBS.	15.380	14.630 14.620
14" RT		600 FT. LBS.	2	66 LBS.	16.250	15.505 15.494
15" RT		950 FT. LBS.	2	76 LBS.	16.628 16.625	15.88
11" FA		380-530 FT. LBS.	4	28 LBS.	12.436 12.433	11.750
12" FA		480-770 FT. LBS.	4	38 LBS.	13.999 13.996	13.000
14" FA		600-960 FT. LBS.	3	54 LBS.	15.502 15.499	14.750

Division of Borg-Warner Corporation

MODEL 280V



Features

- 5 speeds forward, 1 reverse
- Clark-designed split pin synchronizers
- positive spline locks
- high speed P.T.O.
- six available ratios
- extra capacity synchronizers
- constant mesh gears in 4 top speeds
- wide gear faces

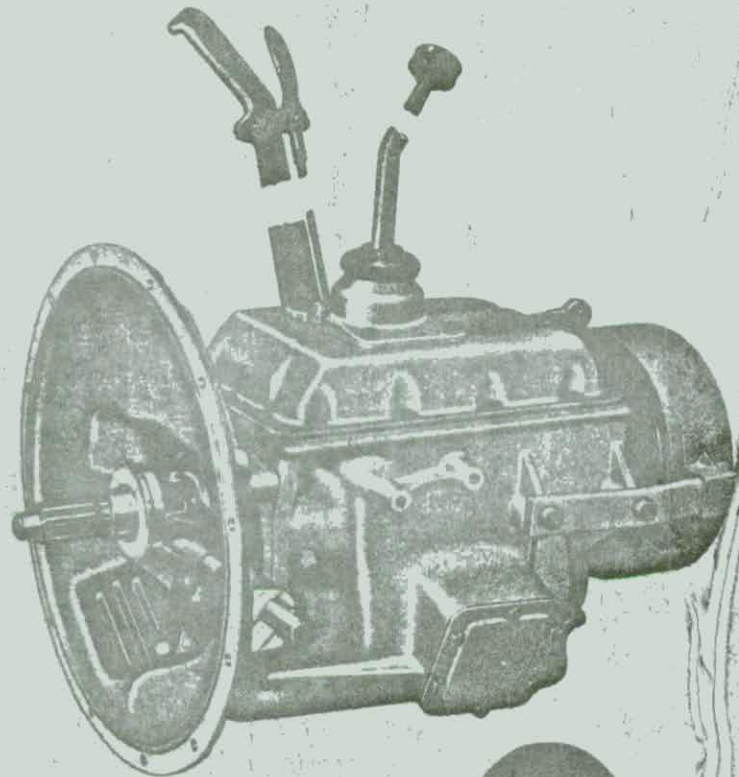
Options

- brake equipment
- shift lever to suit requirements
- mechanical remote control
- bell housing—S.A.E. No. 2 or No. 3

A new modern design transmission with greater torque output capability to handle bigger engines and increased GCW loads, the Clark 280V series of synchronized transmissions incorporates the latest innovations in engineering design, materials specification and manufacturing techniques. These include: Clark designed split-pin synchronizers; positive spline locks to prevent gear popout; and wide gear faces to provide conservative gear loads for increased transmission life.

Tested and proved in dump trucks, transit mixers and car and freight haulers, the GTO-280 is designed for use with engines in the 280-350 lb. ft. torque range.

See reverse side for specifications. Phone, write or wire for additional information pertinent to your application.



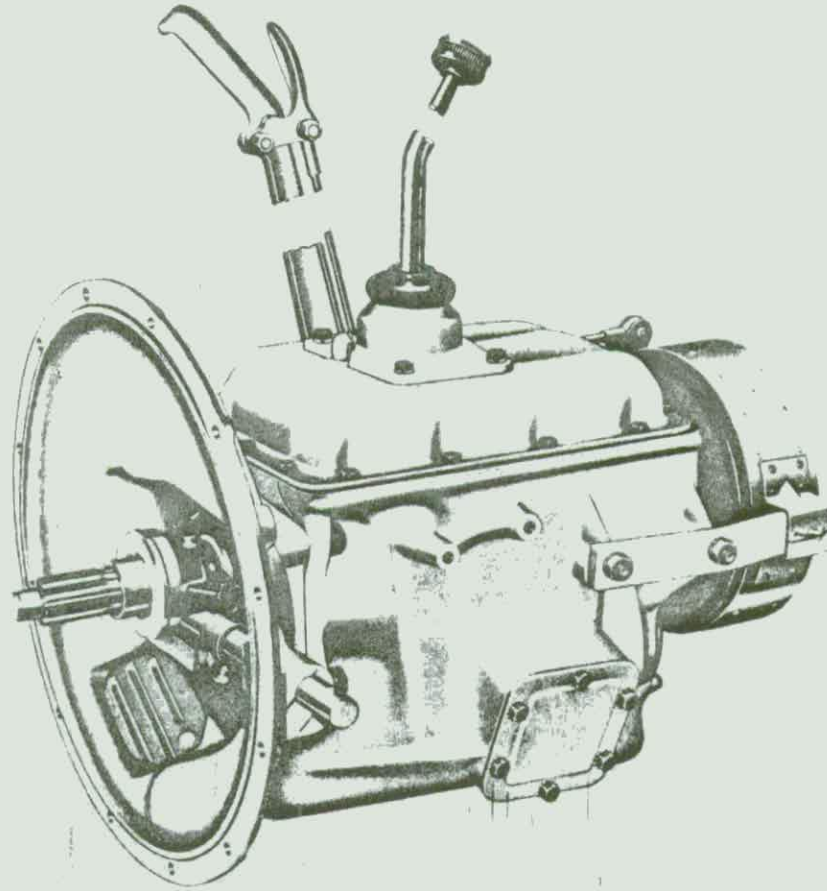
THE CLARK SPLIT PIN SYNCHRONIZER...
It synchronizes transmission gears to the speed of the vehicle... smooths up-or-down shifting... prevents gear clash... makes shifts much easier. Results include longer life for heavy duty transmissions, less driver fatigue, reduced driving time.

CLARK EQUIPMENT

CLARK EQUIPMENT COMPANY
AUTOMOTIVE DIVISION
Jackson, Michigan (517) 764-6000

MODEL 280V

FIVE SPEED FORWARD
AND ONE REVERSE



GENERAL SPECIFICATIONS

- GEARS**—Quiet constant mesh gears for the four top speeds.
- SYNCHRONIZED**—In 2nd, 3rd, 4th, and 5th Speeds.
- NOMINAL TORQUE RANGE**—280-350 LBS. FT.
- WEIGHT**—275 lbs. with brake as shown.
- CLUTCH HOUSING**—S.A.E. No. 2 and No. 3.

SPECIFICATIONS AND/OR DESIGNS ARE SUBJECT
TO CHANGE WITHOUT NOTICE OR OBLIGATION

FOR ATLAS USE REMOTE SHIFT
LEVER 230729
REMOTE CONTROL 21613
CLARK EQUIPMENT COMPANY

JACKSON - - MICHIGAN, U.S.A.



PTO DATA - RIGHT SIDE

MODEL	DRIVING GEAR 7/9 PITCH 19-51-46 P.A. 27-12-57 HARKH	SPEED AT 1000 RPM OF ENGINE	SPACER "A"
280V	40T	456 RPM	131
280VO	46T	571 RPM	560
282V	42T	489 RPM	332
285V	39T	489 RPM	060
288V	37T	571 RPM	000
289V	40T	571 RPM	131

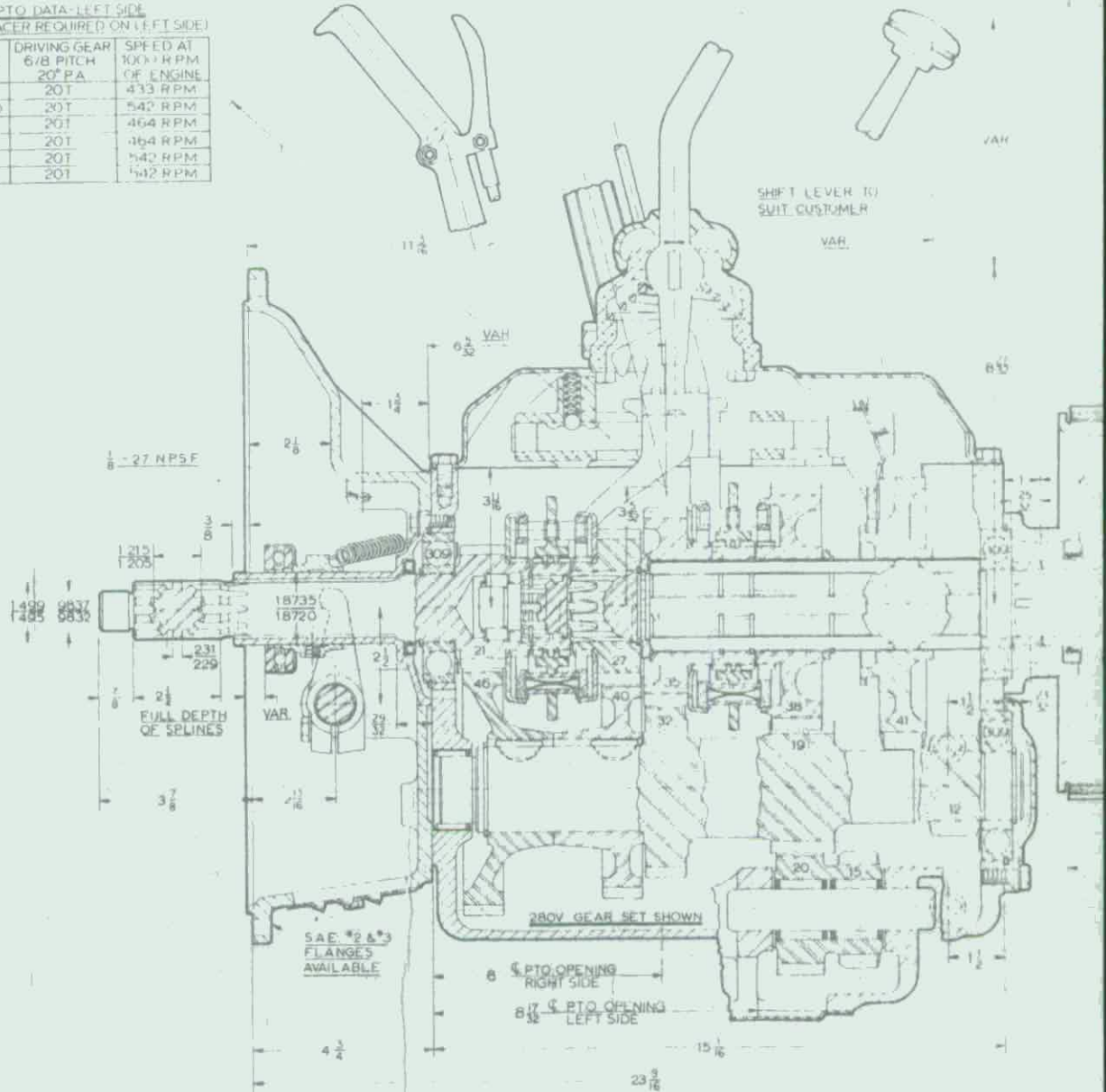
*PLANE OF ROTATION

PTO DATA - LEFT SIDE
(NO SPACER REQUIRED ON LEFT SIDE)

MODEL	DRIVING GEAR 6/8 PITCH 20" P.A.	SPEED AT 1000 RPM OF ENGINE
280V	20T	433 RPM
280VO	20T	542 RPM
282V	20T	464 RPM
285V	20T	464 RPM
288V	20T	542 RPM
289V	20T	542 RPM

GEAR RATIOS

SPEED	MODEL 5					
	280V	280VO	282V	285V	288V	289V
FIFTH	DIRECT	0.831	DIRECT	DIRECT	DIRECT	DIRECT
FOURTH	1.481	DIRECT	1.171	1.471	1.371	1.181
THIRD	2.401	1.861	2.171	2.241	1.911	1.911
SECOND	4.381	3.501	4.091	4.091	3.501	3.501
FIRST	7.481	5.981	6.991	6.991	5.981	5.981
REVERSE	6.301	5.041	5.891	5.891	5.041	5.041



CLARK
EQUIPMENT

280-350 LBS. FT. NOMINAL TORQUE RANGE—When desired, our engineers will give the definite torque rating for these units upon receipt of complete data as requested on our standard specification sheets. We cannot assume responsibility for installations which have not been approved by our Engineering Department.

BELL HOUSING—S.A.E. No. 2 and No. 3

CLUTCH SHAFT END—1½" Dia. Std. 10 Spline.

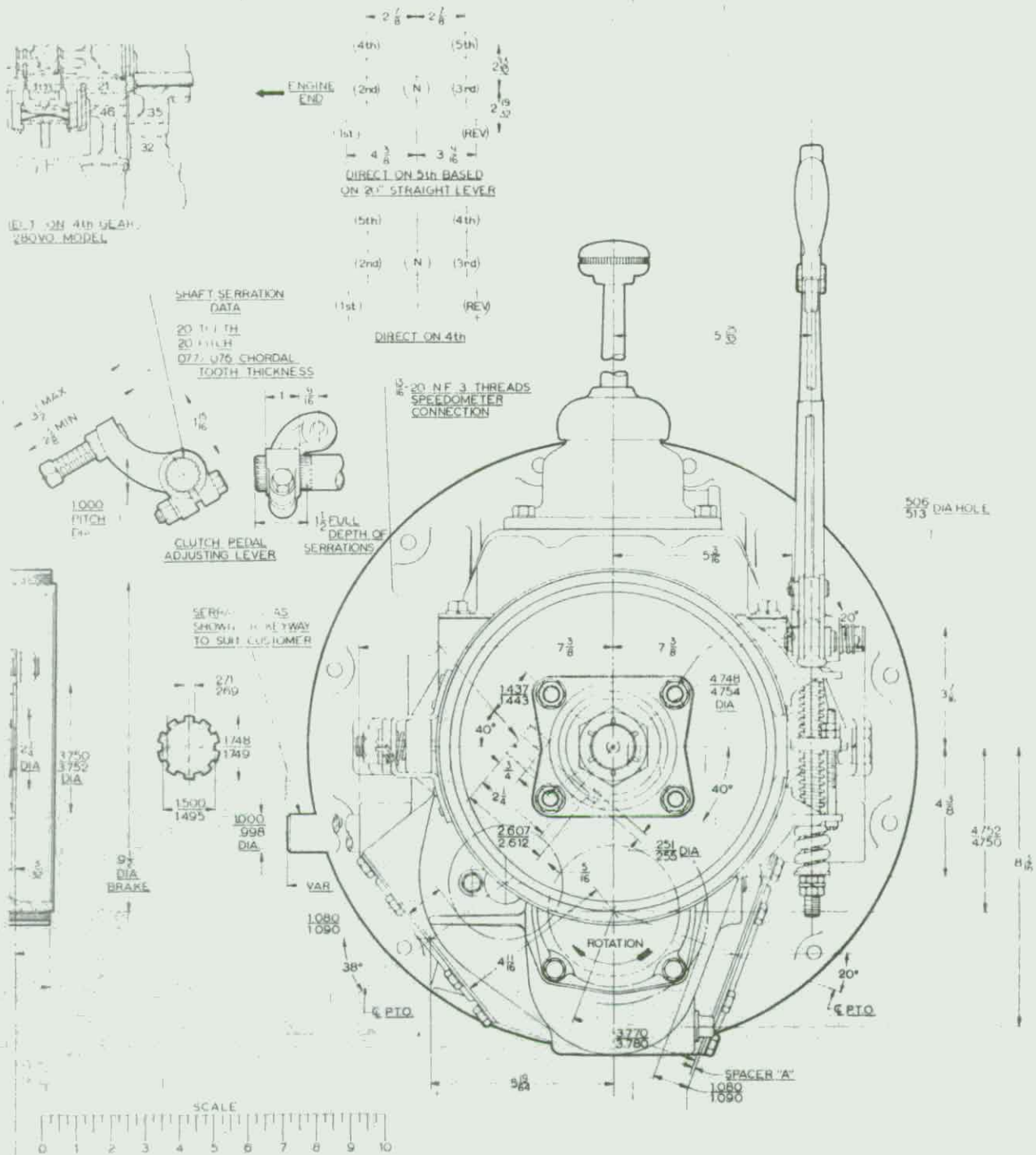
CLUTCH INSTALLATION—Arranged to suit Standard makes.

MAINSHAFT END—1¾" Dia. S.A.E. 10 Spline.

MATERIAL—Gears and mainshaft made of Alloy steel carburized. Made from fine grain, full upset forgings heat treated to obtain the maximum properties of the steel for clash, wear, distortion and strength.

CONTROLS—Furnished with Center Control Only.

SPEEDOMETER DRIVE—Rear Bearing Cover is arranged for Stewart-Warner type speedometer gears.



MODEL 280V

RAKE EQUIPMENT AVAILABLE—Band Brake 9 1/2" Dia. x 3" wide, with 3/16" lining. Hydraulic pressed and ground heavy duty lining with counterbores for all rivet heads used. Brake is cam operated with equalizing shoes.

OIL CAPACITY—4 Quarts.

WEIGHT—238 lbs. with all equipment as shown (except brake). 37 lbs. 3" x 9 1/2" brake, cast drum and flange.

FOR RECOMMENDATIONS AND APPROVAL PLEASE FILL OUT SPECIFICATION SHEET PROVIDED.

NUMERICAL LISTING SPECIFICATION

CHROME PLATE COMPONENTS

EXHAUST SYSTEM COMPONENTS

ENGINE MAKE AND MODEL

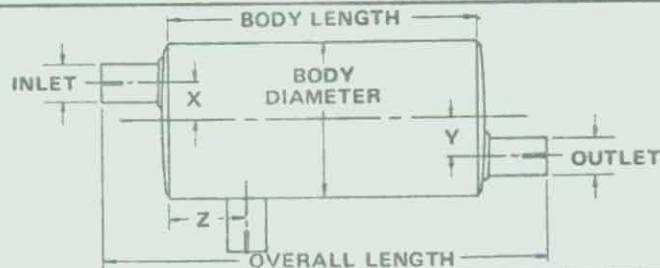
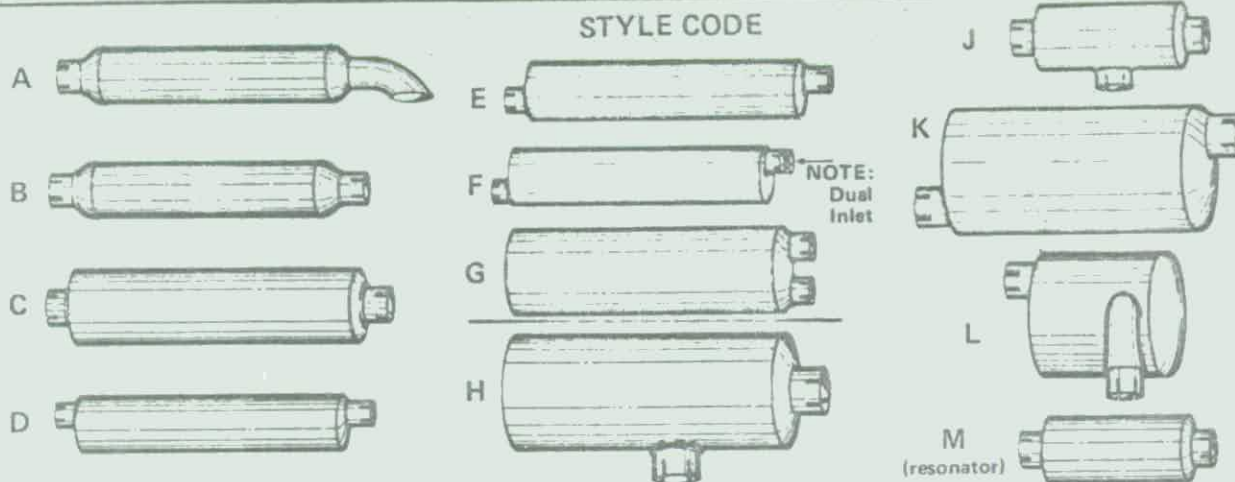
TRUCK MAKE AND MODEL

SPECIFICATIONS BY INLET SIZE

OEM CROSS REFERENCE

NOISE REDUCTION TIPS

MUFFLERS TO MEET NOISE LEVELS



- DIMENSION CODE -
(All Dimensions in Inches)

MUFFLER NO.	STYLE	INLET ID Notched Except Noted	BODY DIA.	BODY LENGTH	OUTLET ID Notched Except Noted	OVERALL LENGTH	X Inlet Distance Off-Center	Y Outlet Distance Off-center	Z Distance From Body Edge
201	A	2	6	14 $\frac{1}{2}$	2 Spout	30 $\frac{1}{2}$			
202	A	2 $\frac{1}{2}$	6	14 $\frac{1}{2}$	2 Spout	30 $\frac{1}{2}$			
204	A	2 $\frac{1}{2}$	6	22 $\frac{1}{2}$	2 $\frac{1}{2}$ Spout	39 $\frac{1}{2}$			
205	A	2 $\frac{1}{2}$	6	22 $\frac{1}{2}$	2 $\frac{1}{2}$ Spout	39 $\frac{1}{2}$			
206	A	3	6	22 $\frac{1}{2}$	3 Spout	40 $\frac{1}{8}$			
207	A	3 $\frac{1}{2}$	6	22 $\frac{1}{2}$	3 Spout	39 $\frac{7}{8}$			
209	A	3 $\frac{1}{2}$	6	22 $\frac{1}{2}$	3 Spout	40			
212	A	2 $\frac{1}{2}$	6	28 $\frac{1}{2}$	2 $\frac{1}{2}$ Spout	45 $\frac{1}{2}$			
401	B	2	6	14 $\frac{1}{2}$	2	28 $\frac{1}{2}$			
404	B	2 $\frac{1}{2}$	6	22 $\frac{1}{2}$	2 $\frac{1}{2}$	36 $\frac{1}{2}$			
406	B	3	6	22 $\frac{1}{2}$	3	36 $\frac{1}{2}$			
407	B	3 $\frac{1}{2}$	6	32 $\frac{1}{2}$	3	46 $\frac{1}{2}$			
408	B	4	6	32 $\frac{1}{2}$	4	45 $\frac{1}{2}$			
409	B	3 $\frac{1}{2}$	6	22 $\frac{1}{2}$	3	36 $\frac{1}{8}$			
9004	C	2 $\frac{1}{2}$	6	24	2 $\frac{1}{2}$	30			
9006	E	2	6	24	2	30	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9007	E	2 $\frac{1}{2}$	6	28 $\frac{1}{2}$	2 $\frac{1}{2}$	34 $\frac{1}{2}$	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9008	E	2 $\frac{1}{2}$	6	28 $\frac{1}{2}$	2	36 $\frac{1}{2}$	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9012	E	2 $\frac{1}{2}$	6	22 $\frac{1}{2}$	2	28 $\frac{1}{2}$	0	1 $\frac{1}{2}$	
9013	C	3	6	24	3	30			
9016	C	2	6	19 $\frac{1}{2}$	1 $\frac{1}{2}$	23 $\frac{1}{2}$			
9017	E	2	6	19 $\frac{1}{2}$	2	26	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9018	E	2 $\frac{1}{2}$	6	24	2 $\frac{1}{2}$	29	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9022	E	2	6	28 $\frac{1}{2}$	2	33	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9027	C	2	5	22 $\frac{1}{2}$	2	28			
9051	C	2 $\frac{1}{2}$	5	34	2	40			
9100	C	2 $\frac{1}{2}$	7	30	2 $\frac{1}{2}$	36			
9103	C	3	7	32	3	39			
9105	C	3 $\frac{1}{2}$	7	32 $\frac{1}{2}$	3 $\frac{1}{2}$	38			
9111	E	2 $\frac{1}{2}$	7	30	2 $\frac{1}{2}$	36	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9113	E	3 $\frac{1}{2}$	7	23 $\frac{1}{2}$	3	29	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9121	M	3	7	12	3	20			
9122	M	3 $\frac{1}{2}$	7	12	3 $\frac{1}{2}$	20			
9123	M	4	7	12	4	20			
9125	C	3 $\frac{1}{2}$	7	27	3	32 $\frac{1}{2}$			
9127	E	3 $\frac{1}{2}$	7	26	2 $\frac{1}{2}$	32	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9129	C	4	7	32 $\frac{1}{2}$	4	39			
9131	E	2 $\frac{1}{2}$	7	28 $\frac{1}{2}$	2 $\frac{1}{2}$	34 $\frac{1}{2}$	0	1 $\frac{1}{2}$	
9134	J	2-3	7	15	4	20			7 $\frac{1}{2}$
9135	C	3 S.P.T.	7	14	3 S.P.T.	19 $\frac{1}{2}$			
9136	F	2-2 $\frac{1}{2}$	7	28 $\frac{1}{2}$	2 $\frac{1}{2}$	34 $\frac{1}{2}$	1 $\frac{1}{2}$	1 $\frac{1}{2}$	
9146	C	3 $\frac{1}{2}$ S.P.T.	7	19	3 $\frac{1}{2}$ S.P.T.	25 $\frac{1}{2}$			
9147	C	4	7	38 $\frac{1}{2}$	4	48			
9201	C	4	8 $\frac{1}{2}$	48	4	54 $\frac{1}{2}$			
9202	C	4	8 $\frac{1}{2}$	33	3.84 OD	39			
9203	E	4	8 $\frac{1}{2}$	42	3 $\frac{1}{2}$ OD	47 $\frac{1}{2}$	0	1 $\frac{1}{2}$	
9204	E	4	8 $\frac{1}{2}$	33	3.84 OD	41	0	1 $\frac{7}{8}$	
9205	E	2 $\frac{1}{2}$ Flared	8 $\frac{1}{2}$	36	2 $\frac{1}{2}$	45	2	2 $\frac{1}{2}$	
9206	E	3	8 $\frac{1}{2}$	20 $\frac{1}{2}$	2 $\frac{1}{2}$	28	2	2 $\frac{1}{2}$	

NUMERICAL LISTING SPECIFICATION

MUFFLER NO.	STYLE	INLET ID Notched Except Noted	BODY DIA.	BODY LENGTH	OUTLET ID Notched Except Noted	OVERALL LENGTH	X Inlet Distance Off-Center	Y Outlet Distance Off-center	Z Distance From Body Edge
9207	E	3 3/4	8 1/2	36	3	43	1 1/4	1 1/4	
9208	E	3	8 1/2	30	3	36 1/2	2	2	
9210	E	3	8 1/2	28	2.84 OD	34	2	2 1/8	
9213	E	3 3/4	8 1/2	33	3.84 OD	40	1 1/4	1 7/8	
9217	C	4	8 1/2	42	3 5/8 OD Hump	48 1/2			
9218	G	3	8 1/2	28	3	32	2	2	
9225	E	4	8 1/2	44	3.84 OD	52	-0-	1 7/8	
9229	C	4 Flared	8 1/2	42	3 5/8 OD Hump	48 1/2			
9232	H	4	8 1/2	48	4	51 1/2			
9300	C	4	9	44	4	51			33
9302	E	4	9	15	4	22	2	2	
9303	M	4	9	15	4	22			
9305	C	4	9	28	4	34 1/2			
9306	C	4	9	40	4	48			
9310	M	5	9	12	5	20			
9315	C	3 1/2	9	25	3 1/2	31			
9321	E	5	9	40	5	48	1 3/8	-0-	
9327	C	5	9	44	5	51			
9330	C	3 1/2	9	48	5 OD	92			
9333	C	4 S.P.T.	9	20	4 S.P.T.	26			
9336	C	4	9	44	4	51			
9337	C	5	9	44	5	51			
9338	C	4	9	44	4	51			
9340	C	3 1/2	9	44	3 1/2	51			
9341	C	3 1/2	9	44	3 1/2	51			
9344	E	4	9	44	4	51	2	2	
9345	C	3 1/2	9	36	3 1/2	43			
9348	C	4	9	44	4	51			
9349	C	4	9	44	4	51			
9350	C	4	9	44	4	51			
9400	K	4	10x15	26	4	32 1/2	4 1/2	4 1/2	
9401	K	5	10x15	26	5	32 1/2	3 1/8	3 1/8	
9408	H	5	10x15	16	5	20 1/2		1 1/2	4
9412	H	4	10x15	16	4	20 1/2	4 1/2		3 1/2
9413	H	4	10x15	21 1/2	4	24 1/2	4 1/2		14 1/2
9414	H	5	10x15	21 1/2	5	24 1/2	3 1/2		14 1/2
9416	G	4	10x15	39	4	42 1/2	3	3	
9500	K	3 1/2	8 1/2 x 11 1/2	32	3.34 OD	39	2 1/2	2 1/2	
9505	K	4	8 1/2 x 11 1/2	28 1/2	4	36 1/2	2 1/2	2 1/2	
9509	G	3 1/2	8 1/2 x 11 1/2	20	3 1/2	23 1/2	3	3	
9510	G	4	8 1/2 x 11 1/2	39	4	42	2 5/8	2 5/8	
9512	K	3 Flared	8 1/2 x 11 1/2	18	3 OD	22 1/2	3 3/8	2 1/2	
9513	G	3 Flared	8 1/2 x 11 1/2	20	2 1/2 OD	22 1/2	2 1/2	2 1/2	
9514	D	4	8 1/2 x 11 1/2	40	4	46	2 5/8	2 5/8	
9515	H	4	8 1/2 x 11 1/2	16	4	21 1/2	2 5/8	2 1/2	3 1/2
9523	K	4	8 1/2 x 11 1/2	30	3.84 OD	40	2 5/8	2 5/8	
9524	H	3 1/2	8 1/2 x 11 1/2	18	3 1/2	21		2 1/2	2 1/2
9525	H	2 1/2	8 1/2 x 11 1/2	20	2 1/2	23 1/2		2 7/8	2 3/8
9526	H	3 1/2	8 1/2 x 11 1/2	30 1/2	3 1/2 Spout	38 1/2		1 1/2	15 3/8
9601	F	2-2 1/2 Flared	8 1/2	36	3 1/8 OD Hump	43 1/2	1 1/2	2	
9604	F	2-2 1/2	8	30	3 1/2 OD	37 1/2	1 1/2	1 1/2	
9605	F	2-2 1/2 Flange	8	30	3 1/2 OD	37 1/2	1 1/2	1 1/2	
9606	F	2-2 1/2	8 1/2	38	3 1/8 OD	44 1/2	1 1/2	2	
9607	F	2-2 1/2 Flared	8 1/2	40	4	48 1/2	1 1/2	1 1/2	
9608	F	2-2 1/2	8 1/2	38	3	43 1/2	1 1/2	2	
9700	E	2 1/2	6 1/2	27 5/8	2 1/2 OD	35 1/2	1 1/2	1 1/2	
9701	D	2 1/2	6 1/2	24	2 1/2	29	1 1/2	1 1/2	
9801	C	4	8	30	4	37			
9803	M	4	8	12	4	20			
9807	C	3 1/2	8	30	3 1/2	37			
9808	M	4	8	20	4	25 1/2			
9809	C	4	8	45	4	52			
9817	E	4	8	30	4	36 1/2	1 5/8	1 5/8	
9851	H	4	10	36	4	38 1/2	-0-		3 1/2
9852	E	4	10	28	4	33 1/2	2 1/8	2 1/8	
9854	G	4	10	40 1/2	4	43 1/2	2 1/2	2 1/2	
9855	G	4	10	28	4	32	2 1/2	2 1/2	
9860	H	4	10	28	4	31 1/2	2 1/8		20 1/2
9864	E	4	10	33	4	38 1/2	2 1/2	2 1/2	
9866	E	4	10	44	4	51	2 5/8	2 5/8	
9876	G	4	12	26 1/2	4	29 1/2	2 1/2	2 1/2	
9877	G	6 1/2 OD Flared	12	22	3 OD	25 1/2	2 1/2	3 1/2	
9901	L	4	10x15	12	4	16 1/2		4 3/8	9
9905	L Special	4 Flared	10x15	26	3 1/2 OD Hump	26	N/A	N/A	N/A
9906	L Special	4 Flared	10x15	23	3 1/8 OD Hump	24		Special	N/A
9907	L Special	4 Flared	10x15	23	3 1/8 OD Hump	23	N/A	N/A	N/A
9909	L Special	4 Flared	10x15	26	3 1/2 OD Hump	26	N/A	N/A	N/A
9913	L	4	10x15	12	4	16 1/2		4 3/8	9
9915	L	4	10x15	12	4	16 1/2		4 3/8	3
9917	L	4	10x15	12	4	16 1/2		4 3/8	9
9918	L	4	10x15	16	3.84 OD	19 1/2		4 3/8	13
9926	L	4	10x15	16	3.84 OD	19 1/2		4 3/8	3
9930	L	5	10x15	16	4.84 OD	19 1/2		4	3 1/2
9938	Special	4	10x15	12	4	12	N/A	N/A	3

CHROME PLATED COMPONENTS

EXHAUST SYSTEM COMPONENTS

ENGINE MAKE AND MODEL

TRUCK MAKE AND MODEL

MUFFLER SPECIFICATIONS BY INLET SIZE

OEM CROSS REFERENCE

NOISE REDUCTION TIPS

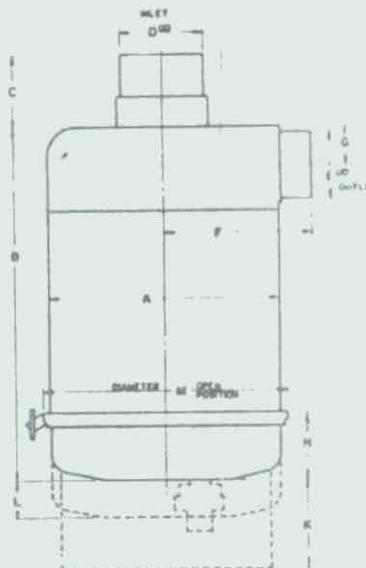
MUFFLERS TO MEET NOISE LEVELS

DONALDSON

SPECIFICATIONS

AIR CLEANERS

FWA and FWG cleaners can be mounted either horizontally or vertically.



Standard Air Cleaner Model	Vacuator Air Cleaner Model**	*Air Flow Rating		A	B	C	D	E	F	G	H	K	L	M	Approx. Wt. Lbs.
		At 8" H ₂ O Gas	At 12" H ₂ O Diesel												
FWA05-2526	FWA05-2527	80	95	5 1/4	14 3/16	1 3/8	2	2	4	1 3/8	3 3/8	8 1/2	1 3/8	6 3/8	6
FWA06-5007	FWA06-5015	110	135	6 1/2	17 3/16	2 1/8	2 1/2	2 1/4	4 3/8	1 7/16	3 3/8	8 3/8	1 3/8	7 3/8	8
FWA08-0022	FWA08-0031	190	235	8	18 3/16	2 3/8	3	3	6 1/8	2 3/8	3 3/8	9 3/8	1 3/8	8 3/8	10 1/2
FWA10-0017	FWA10-0019	290	360	10 3/8	18 1/2	4 3/8	3 3/4	4	7 1/8	2 3/4	4	7 3/16	1 3/4	11 3/8	20
FWA12-0003	FWA12-0036	350	440	11 3/8	18 3/16	3 7/16	4 1/2	4	7 3/8	2 1/2	4	7 3/16	1 1/2	13 3/8	24
FWA14-0002	FWA14-0036	460	530	14	21 7/8	3 7/16	5	4	9	2 1/2	4	9 3/16	1 1/2	15 3/8	33
FWA14-0003	FWA14-0033	560	700	14	21 7/8	3 7/16	5	5 1/2	9	3 3/16	4	9 3/16	1 1/2	15 3/8	32
FWA16-0001	FWA16-0013	760	945	16	24 1/2	4 3/8	6	6	10 1/2	3 3/8	4	11 3/16	1 1/2	17 3/8	52

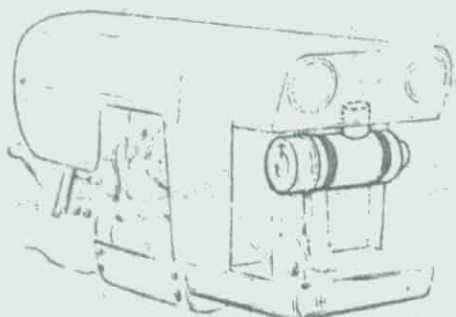
*Ratings are ± 1" H₂O

**Vacuator cup dimensions will vary slightly from Standard

Standard Air Cleaner Model	Vacuator Air Cleaner Model**	*Air Flow Rating		A	B	C	D	E	F	G	H	K	L	M	Approx. Wt. Lbs.
		At 8" H ₂ O Gas	At 12" H ₂ O Diesel												
FWG04-2503	None	70	85	4 3/8	9 1/8	1	1 3/4	1 3/4	3 3/8	1 1/2	3	5	1	5 1/4	3 1/4
FWG05-2510	FWG05-2512	95	115	5 1/4	12 3/16	1 3/8	2	2	4	1 1/8	3 3/16	8 3/16	1 3/8	6 3/8	4 3/4
FWG06-5008	FWG06-5012	140	170	6 1/2	13 3/16	1 3/8	2 1/2	2 1/4	4 3/8	1 3/8	3 3/8	8 3/8	1 3/8	7 3/8	6 1/2
FWG08-0023	FWG08-0026	225	280	8	14 3/16	1 3/8	3	3	6 1/8	1 3/8	3 3/8	9 1/8	1 3/8	8 3/8	9 1/2
FWG10-0003	FWG10-0004	330	405	10 3/8	16 3/16	1 3/8	4	4	7 1/2	2 1/8	4	7 3/16	1 3/4	11 3/8	17
FWG12-005	FWG12-0063	415	515	11 3/8	16 3/16	2 1/8	4 1/2	4	7 7/8	3 1/4	4	7 3/16	1 1/2	13 3/8	23
FWG14-0077	FWG14-0083	590	720	14	19 3/16	2 1/8	5	5 1/2	9	3 3/16	4	9 3/16	1 1/2	15 3/8	32
FWG16-0104	FWG16-0107	870	1080	16	21 3/16	2 3/16	6	6	12	4	4	11 3/16	1 1/2	17 3/8	45

*Ratings are ± 1" H₂O

**Vacuator cup dimensions will vary slightly from Standard



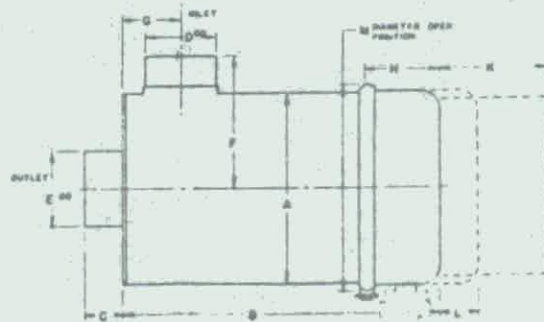
FWG CYCLOPAC installed horizontally on farm tractor.



D-1400 FILTER CLEANER
Detergent with carbon dissolving additive. Mix with water. Cleans any washable paper filter.

RESTRICTION GAUGE

Signal locks in view when filter element requires servicing. Mount on dash of cleaner.



NOTE: Dimensions not certified. Request prints for specific design applications.

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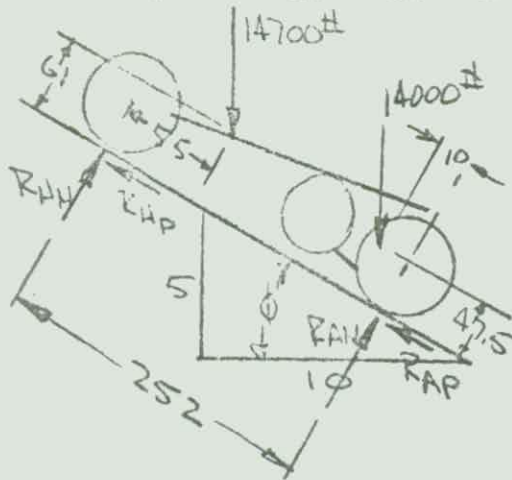
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DRAWBAR PULL FOR CLIMBING
50% SLOPE
(BACKING UP)



$$R_N = \frac{14700(53.5 \cos 23.7^\circ + 61 \sin 23.7^\circ) + 14000(242 \cos 23.7^\circ + 47.5 \sin 23.7^\circ)}{252} = 17030 \text{ lb}$$

$$R_H = \frac{14700(197 \cos \phi - 61 \sin \phi) + 14000(10 \cos \phi - 47.5 \sin \phi)}{252} = 8020 \text{ lb}$$

$$R_{AP} = R_N \tan \phi = 17030 (.5) = 8515 \text{ lb}$$

$$R_{HP} = R_H \tan \phi = 8020 (.5) = 4010 \text{ lb}$$

$$\text{REQD DBP} = R_{AP} + R_{HP} + \frac{24700(17)}{1000} = 13015 \text{ lb}$$

$$\mu \text{ (BETWEEN TIRES \& SLOPE)} = \frac{13015}{17030} = .764$$

THIS IS POSSIBLE ON A CLEAN SLOPE

DESIGN POWERTRAIN FOR 13500^{lb} DBP



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POWERTRAIN GEARING

1. 13500# DBP REQ'D FOR BACKING UP 50% SLOPE

2. TOP SPEED 35-40 MPH

1. TORQUE REQ'D = 13,500# (19.3 IN)
= 260,600 LB-IN

MAX TRANS RATIO = ^{IN REV} 6.30:1
MAX ENG TORQUE = .85 x 252 LB-FT

AXLE-DROP BOX RATIO REQ'D
= $\frac{260,600}{6.30(.85)(252)(12)} = 16.10$

2. AXLE-DROP BOX RATIO FOR 35 MPH

= $\frac{RPM_{GOV}}{RPM_{WHEEL}} = \frac{RPM_{GOV}}{\frac{REV/MIX \cdot MPH}{60}}$
= $\frac{2800 \times 60}{496 \times 35} = 9.68$

AXLE RATIOS AVAILABLE ARE

8.2 & 7.8

SO TWO SPEED DROP BOX WILL BE NEEDED

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WITH 8.2 AXLE

$$\text{MAX SPEED} = \frac{2800(60)}{496(8.2)} = 41.3$$

BETTER MAKE HIGH GEAR
IN DROP BOX

$$41.3/38 = 1.086$$

LOW GEAR SHOULD BE ABOUT

$$161/8.2 = 1.96$$

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DRIVESHAFT AND U-JOINTS

$$\begin{aligned} \text{MAX TORQUE} &= .85 (259 \text{ LB-FT}) (7.48) (1.984) \\ & \qquad \qquad \qquad \text{(TRANS)} \quad \text{(DRAPBOX)} \\ &= 3270 \text{ LB-FT} \end{aligned}$$

TIP DIFFERENTIAL AND USE SINGLE HOOKE JOINT AT THAT END OF DRIVESHAFT

USE CONSTANT VELOCITY JOINT AT TRANSFER CASE END WITH TRANSMISSION ANGLE OF $13^{\circ}30'$ @ NOM STATIC SOLO

FOR THE SINGLE HOOKE JOINT A MECHANICS UNIVERSAL SIZE 7 IS SUITABLE - 3100 FT-LB MAX OPERATING TORQUE (ENDURANCE LIMIT, 1/2 LIFE 30,000 HRS)

THIS SHOULD BE MATCHED TO A MECHANICS CV DOUBLE CARDAN JOINT SIZE 7CV, 3100 FT-LB MAX OPERATING TORQUE

THESE ARE USED WITH $3\frac{1}{2}$ ODX .098 WALL TUBE
MECHANICS LISTS A TORQUE CAPACITY OF 2550 LB-FT FOR A SHEAR STRESS OF 16,000 PSI
THIS COMBINATION SHOULD BE MORE THAN ADEQUATE \leftarrow

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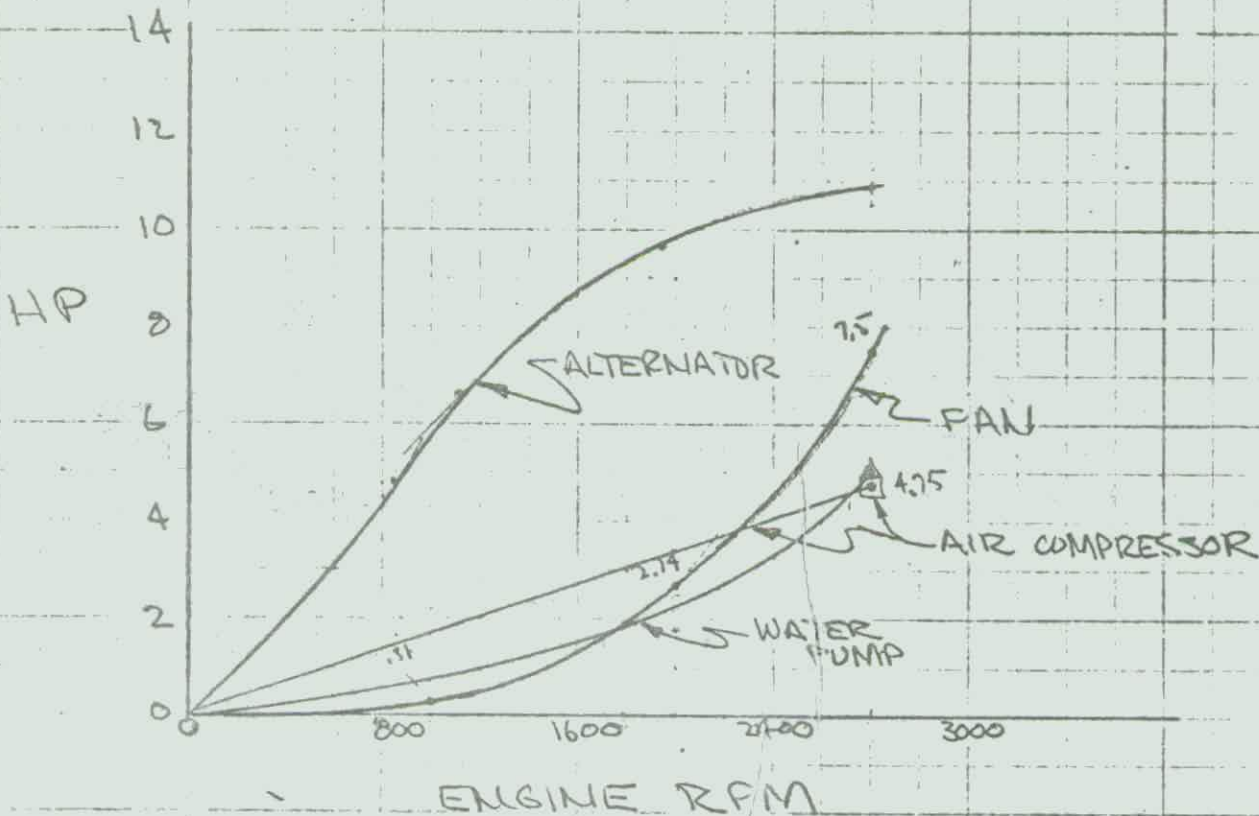
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ACCESSORY DRIVES

FAN - AIR COMPRESSOR

HP LOAD
FAN - (24 IN DIA - 6 BLADE)
7.5 HP @ 2850 RPM

AIR COMPRESSOR - MS - 12 CFM, 100 PSI
4.75 HP @ 2400 RPM



ALTERNATOR - WATER PUMP DRIVE

ALTERNATOR
4.8 HP @ 1500 (840 " " " ")
6.6 HP @ 2000 (1120 ENG RPM)
9.76 HP @ 3500 (1955 " " " ")
10.85 HP @ 5000 (2800 ENG RPM)

WATER PUMP
ASSUME 5 HP MAX

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LOAD-LIFE RELATIONS

FAN AND WATER PUMP RUN AT FULL LOAD ALL THE TIME

AIR COMPRESSOR RUNS AT FULL LOAD ON FOR SHORT BURSTS DURING OPERATION OF THE VEHICLE - IT MIGHT RUN AT FULL LOAD WHEN SUPPLYING POWER FOR OPERATING TOOLS (NO MORE THAN 5% OF LIFE)

ALTERNATOR WILL RUN ABOUT 5 AMPS NORMALLY WHEN LIGHTS ARE NOT IN USE, WITH LIGHTS IT WILL RUN ABOUT 30 AMPS, WHEN LIFTING THE HOWITZER TRAILS WITH THE ELECTRIC CRANE IT WILL RUN 84 AMPS (START) 70 AMPS (RUN) PICKING UP A 2000# LOAD. THE FULL CAPACITY OF 205 AMPS WILL BE USED ONLY WHEN THE ATLAS IS USED AS A GENERATOR SET (4900KVA).

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DUTY CYCLE FOR BELT DRIVE

VEHICLE DUTY CYCLE

A	5%	AIR POWER SOURCE (2800 ENG RPM)
B	5%	ELECTRIC GENERATOR (2800 ENG RPM)
C	5%	OPERATING CRANE (1000 RPM)
D	5%	ENGINE IDLE (600 RPM)
E	15%	MAX ENGINE SPEED (2800 RPM)
F	35%	NORMAL ROAD SPEED (2200 RPM)
G	20%	CROSS COUNTRY (1800 RPM)
H	10%	MANEUVERING (1200 RPM)

HP LOADS ON PULLEYS

	AIR COMP	FAN	CRANK	ALT	W.PUMP	CAM
A	4.75	7.5	12.25	6.5	.5	11.5
B	—	7.5	7.5	10.85	.5	15.9
C	3	1.5	3.5	3.8	1.8	4.6
D	1	.5	1.5	1.4	.5	1.9
E	1.2	7.5	8.7	1.6	5	6.6
F	.9	3.5	4.4	1.25	2.75	4.0
G	.75	1.8	2.6	1.0	1.8	2.8
H	.44	.5	.94	.5	.8	1.3

FOR A ASSUME ALTERNATOR PRODUCING 30 AMP
 FOR E, F, G, H, ASSUME AIR COMPRESSOR AVE HP
 IS .25 X HP FROM CURVE, P. 1 & ALTERNATOR
 USING 30 AMPS



Worksheet For Automotive Accessory Drives

(Use in conjunction with Design Manual 18575)

Sheet _____ of _____

Company PACIFIC CAR & FOUNDRY

Drive ATLAS ENGINE ACCESSORIES

Date _____

Address _____

Model No. _____

Designed By _____

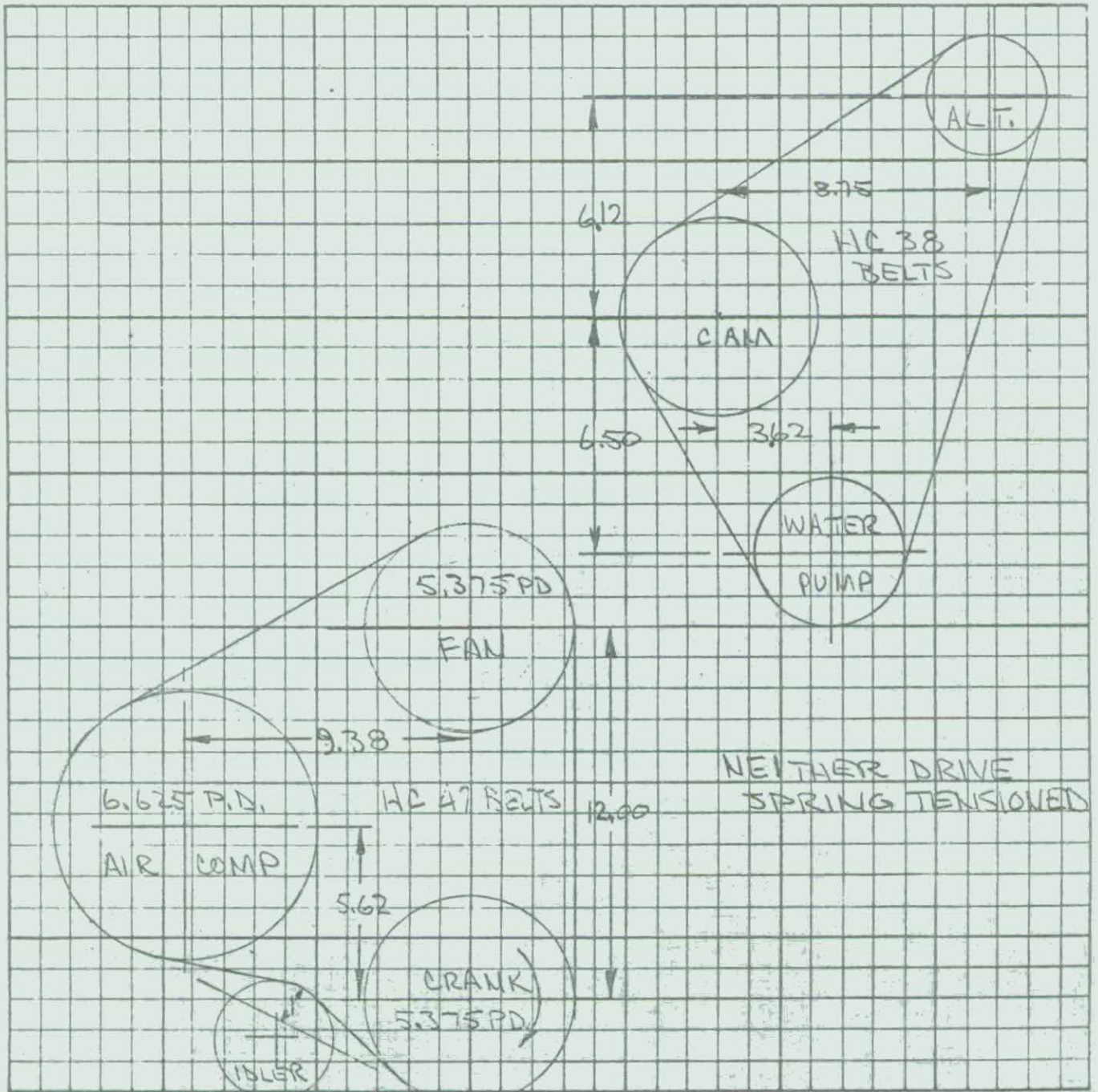
B. P. No. _____

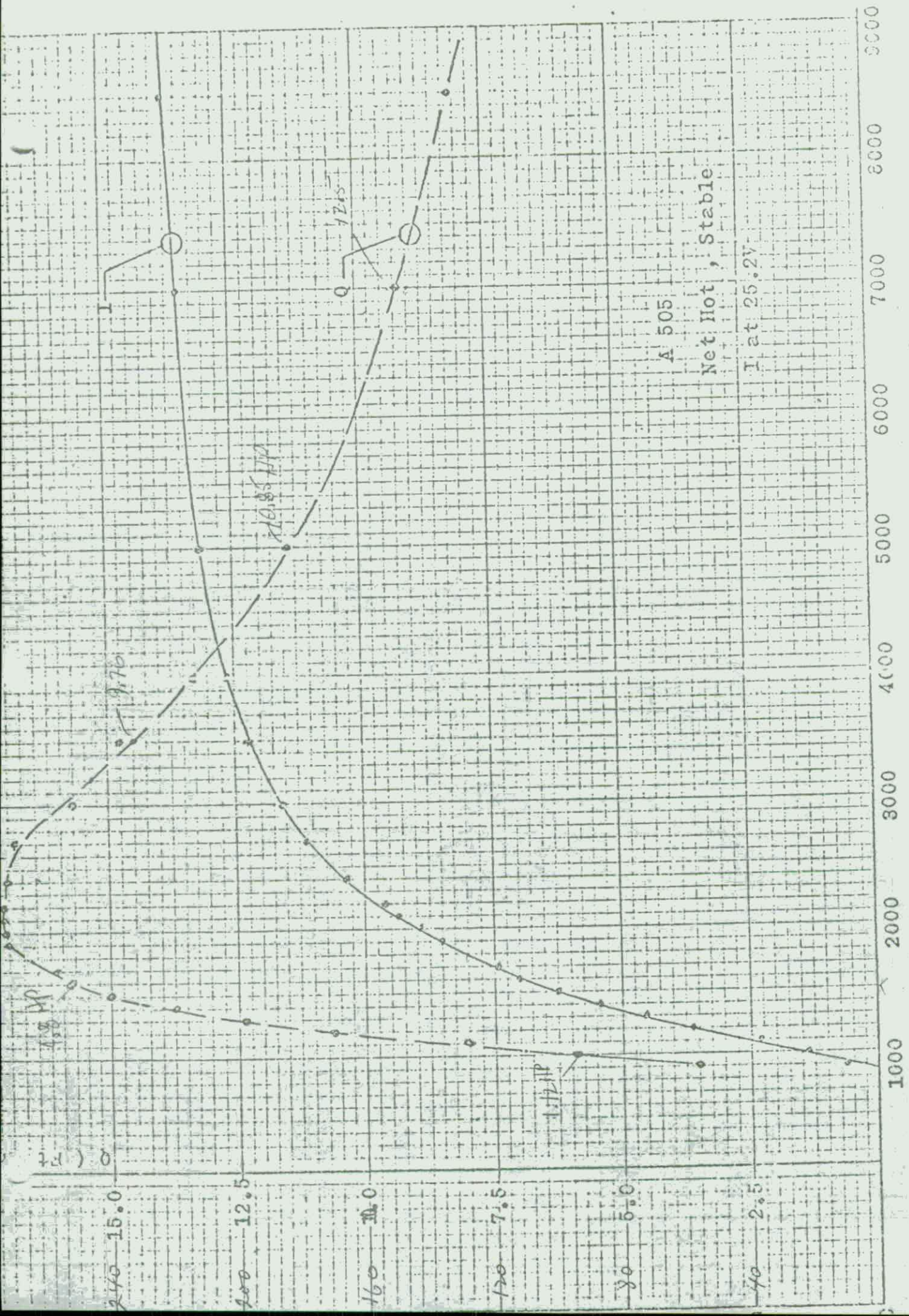
Part No.

(experimental)

(production)

Gates Prod. No. _____





3/6/72

NIEHOFF

ALTERNATOR RPM

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Reqd $\mu = \frac{12.81}{17.605} = .727$
 for grade

Actual μ must be increased to surmount rolling resistance

Resistance @ $\frac{17^{\#}}{1000} = 17 (28.7) = .487^{\#}$

Total resistance = $12.81 + .487 =$

DBT $(= 13,297 \text{ K})$

$\mu_{\text{reqd}} = \frac{13.297}{17.602} = .755$



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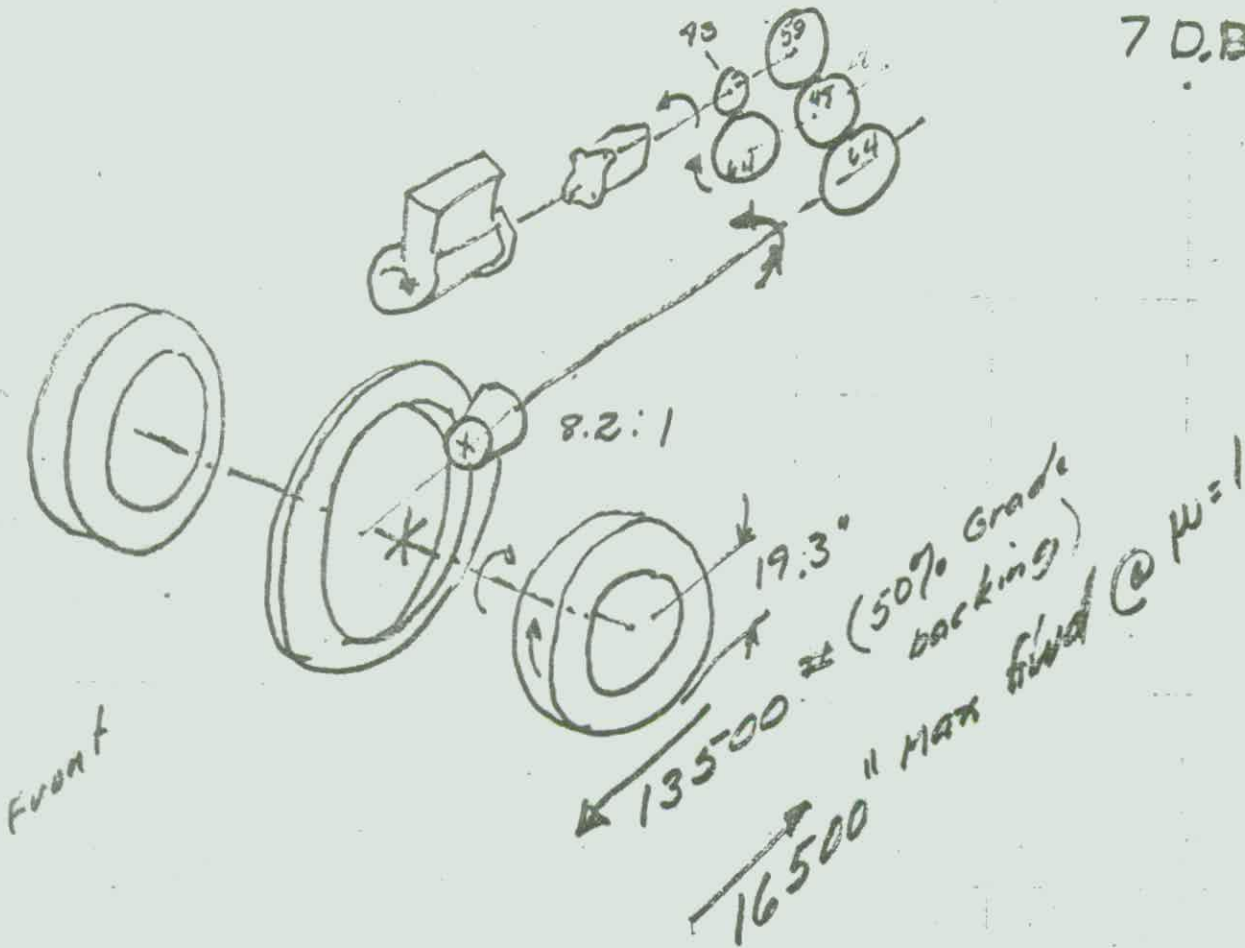
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F.W. = 3 C.P.

rear

20° STUB TOOTH

7 D.B



Rotations are for backing up 50% grade.
Force direction is for tire (Not ground reaction)

Altho 13500 # is the force required for 50% grade, the normal design point for power trains assumes a coefficient of traction of $\mu = 1.00$

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and the maximum normal axle loading. The axle load is 16500 lbs. The gearing, ^{strength} will be designed for this tractive force although the gear ratios and engine torque will not attain this tractive effort.

Axle design tractive force = 16500 #

" " torque = 19.3 (16,500)
= 318450

" Input torque = $\frac{318450}{8.2}$ = 38835.4

Transfer gear box output T_g = 38835.4 lb

" 1st gear torque = $\frac{48}{64}$ (38835.4)
= 29126.5 lb in

" Input shaft T_g (Low gear) ^{Transfer} = $\frac{43}{64}$ (29126.5)
= 19569.4 lb in

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$$\begin{aligned} \text{Transfer input } T_q \text{ (High Gear)} &= \left(\frac{59}{64}\right) 38835.4 \\ &= 35801.4 \text{ lb in} \end{aligned}$$

Transmission gear ratios

Reverse ~~7.48:1~~ 6.30:1

1st ~~6.30:1~~ 7.48:1

Reqd engine torque in ^{1st gear} reverse, Low transfer ratio

$$T_{q \text{ req}} = \frac{19569.4}{7.48} = 2616.23 \text{ lb in}$$

$$= 218.02 \text{ lb-ft}$$

Reqd engine torque in Reverse, Low transfer ratio

$$\frac{19569.4}{6.30} = 3106.25 \text{ lb in}$$

$$= 258.85 \text{ lb-ft}$$

Gross rated engine torque = 259 ft lbs

GMC 4-53N (N40 high economy)

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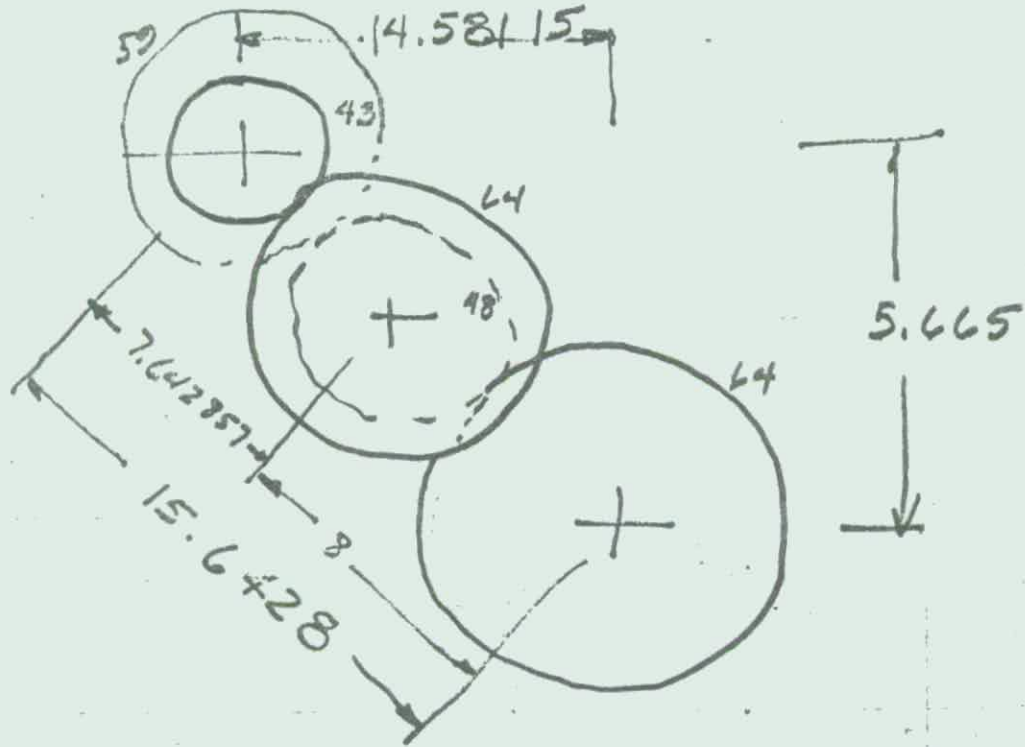
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7 DP 20° PA STUB Fillet Root

No Teeth	Pitch Diam	Pitch Radius
43	6.142857	3.07142857
48	6.8571429	3.4285714
64	9.142857	4.5714286
59	8.428571	4.21428571

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Tooth load @ gross ong tg, 1st gear (7.48:1)

	7.48(3106)		7564.2	
43T	Wt =	$\frac{19569.4}{3.07142}$	=	6371.45 16 in
48T	Wt =	$\frac{29126.5}{3.42857}$	=	8495.23 16 in *
58T	Wt =	$\frac{3106 \cdot 7.48}{4.214286}$	=	$\frac{23232.88}{4.214286}$ = 55 12.89

$$S_b = \frac{P W_t}{F Y}$$

face = $1\frac{1}{8}$

$$S_b = \frac{8495 \text{ (7)}}{1.125 \cdot \frac{.471}{.471}} = 112,300 \text{ psi} \rightarrow$$

@ Face = 1.5"

$$S_p = 84200 \text{ psi OK} \rightarrow \text{486}$$

$$S_p = \frac{7564.2 \cdot 7}{1.5 \cdot .462} = 76200 \text{ psi} \rightarrow \text{432}$$

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$$\text{TORQUE} = 23,000 \text{ IN-LB}$$

TOOTH LOAD ON 43T GEAR:

$$23,000 / 3.071 = 7,490 \text{ lbs} = W_t$$

$$S_b = \frac{P W_t}{F Y}$$

$$Y = .471$$

$$F = 1.75$$

$$P = 7$$

$$W_t = 7,490$$

$$S_b = \frac{(7)(7,490)}{(1.75)(.462)} = 65,000 \text{ PSI} \quad 43T$$

TOOTH LOAD ON 48T GEAR:

$$T = 23,000 \left(\frac{64}{43} \right) = 34,200 \text{ IN-LB}$$

$$W_t = 34,200 / 3.428 = 10,000 \text{ lb}$$

$$S_b = \frac{(7)(10,000)}{(1.75)(.471)} = 85,000 \text{ PSI} \quad 48T$$

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2-SPEED TRANSFER CASE

DESIGN PARAMETERS

LOW GEAR RATIO 1.984 : 1.0

HIGH GEAR RATIO 1.085 : 1.0

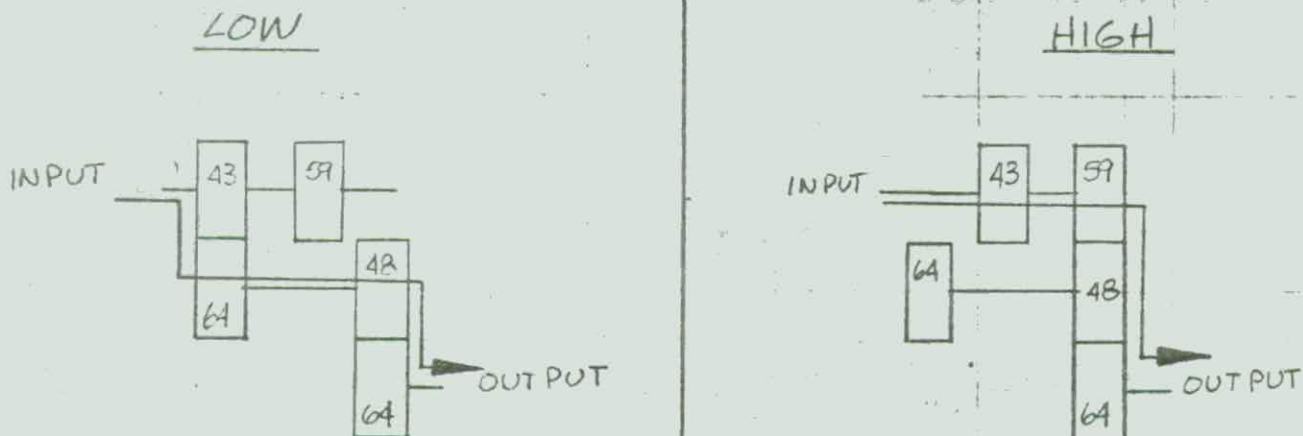
INPUT TORQUE: 23,000 IN-16 @ 240 RPM

DIAMETRAL PITCH - 7

PRESSURE ANGLE - 20°

TOOTH FORM - STUB TOOTH, FILLET ROOT

GEAR TRAIN SCHEMATIC



$$\frac{64}{48} \frac{64}{43} = 1.984496$$

$$\frac{64}{59} = 1.084745$$

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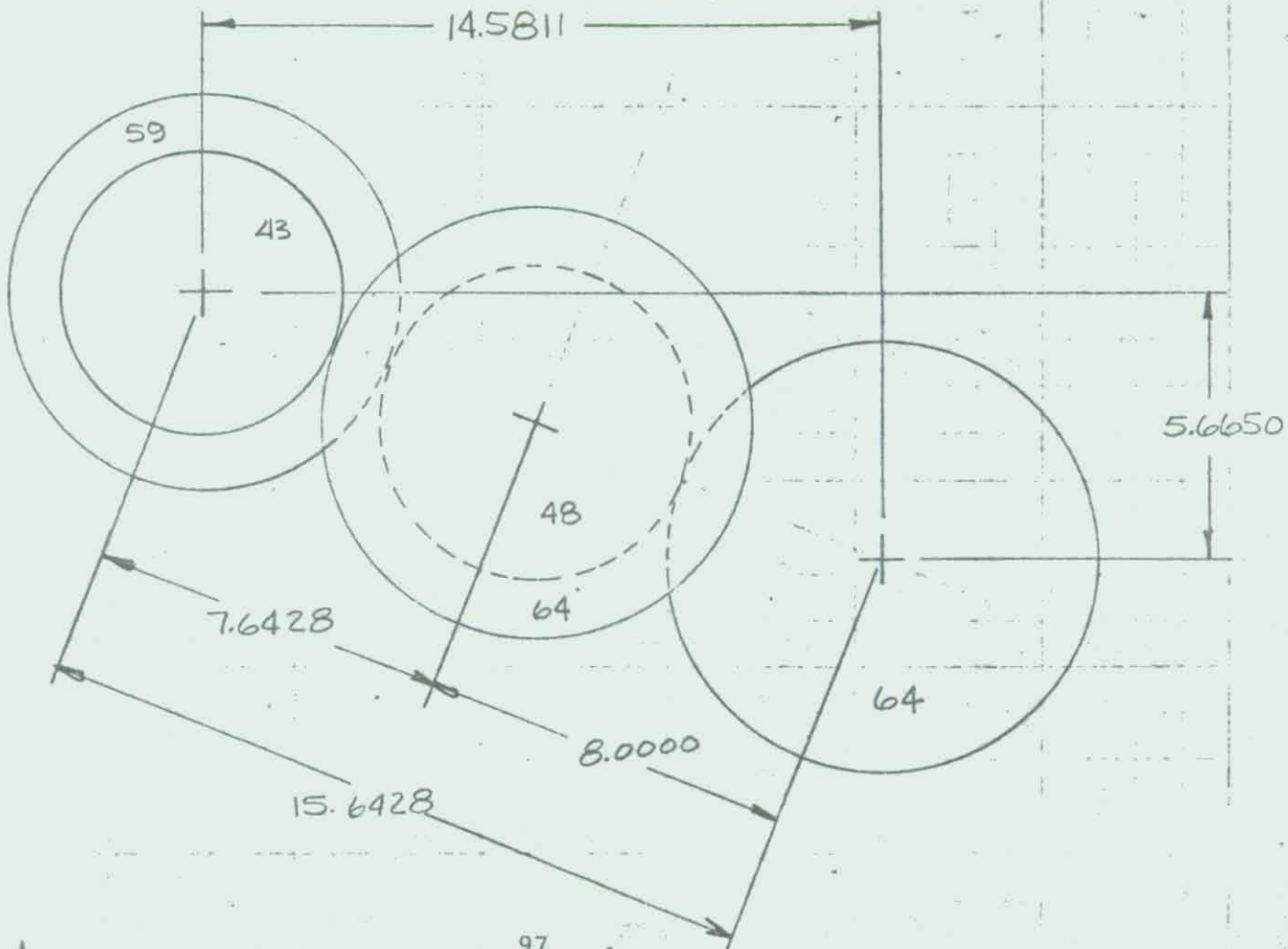
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GEAR DATA

<u>NO.</u>	<u>TEETH</u>	<u>PITCH DIA</u>
43		6.142857
48		6.857142
64		9.142857
59		8.428571



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FIND HERTZ STRESS FOR 48 TOOTH GEAR.

FROM "PRACTICAL GEAR DESIGN" - DUDLEY P. 50

$$S_c = 5,715 \sqrt{\frac{W_t}{F_d} \left(\frac{m_G + 1}{m_G} \right)}$$

$$W_t = \text{TANGENTIAL FORCE} = 10,000 \text{ lbs}$$

$$F = \text{FACE WIDTH} = 1.50$$

$$d = \text{PITCH DIAMETER} = 6.86$$

$$m_G = \text{GEAR TEETH/PINION TEETH} = 64/48 = 1.33$$

$$S_c = 5,715 \sqrt{\frac{(10,000)(1.75)}{(1.5)(6.86)}} = 235,682 \text{ lb/in}^2$$

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LIFE & LOAD CALCULATION

FAFNIR 9117K @ 7500 lbs^{*}

& 240 RPM^{**}

RADIAL LOAD (RE) = 7500

FROM FAFNIR CATALOG 68: (P.13)

$$C_N = A_f \times C_B = (.51)(8530) = 4350 \text{ lb}$$

$$L_{10} = 1500 \left(\frac{C_N}{RE} \right)^3 = 1500 \left(\frac{4350}{7500} \right)^3$$

$$L_{10} = 292 \text{ HRS}$$

* 23,000 in-lb (INPUT) / 3.07

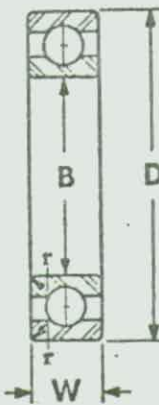
** 1800 / 7.48 = 240 RPM

M-R-C SINGLE-ROW DEEP-GROOVE BEARINGS

TYPE S

MRC 1900-S

Extremely Light Series



Shaft Fits—see pages 150-155.

Housing Fits—see pages 156-161.

Shaft Shoulders } see page 146.
Housing Shoulders }

* Radius "r" indicates maximum radius on shaft or in housing bearing corner will clear.

† Also available with one shield, type SFF and two shields, type SFF.

‡ Also available with one seal, type SZZ and two seals, type SZZ.

§ This size not presently made but be available as production requirements justify tooling.

¶ See page 168.

Note: Tolerances shown on this are ABEC-1 standard.

M-R-C Brg. No.	BORE—B			OUTSIDE DIA.—D			WIDTH W Individual Rings			RAD. r	BALLS	
	MM	Inch	Tol. (Ave.) +.0000	MM	Inch	Tol. (Ave.) +.0000	MM	Inch	Tol. (Ave.) +.000		Inch	No.
1900-S	10	.3937	-.0003	22	.8661	-.0004	6	.2362	-.005	.012	9	3/4
1901-S	12	.4724	-.0003	24	.9449	-.0004	6	.2362	-.005	.012	9	3/4
1902-S	15	.5906	-.0003	28	1.1024	-.0004	7	.2756	-.005	.012	10	5/8
1903-S	17	.6693	-.0003	30	1.1811	-.0004	7	.2756	-.005	.012	11	5/8
1904-S	20	.7874	-.0004	37	1.4567	-.0005	9	.3543	-.005	.012	9	3/4
1905-S	25	.9843	-.0004	42	1.6535	-.0005	9	.3543	-.005	.012	11	5/8
1906-S	30	1.1811	-.0004	47	1.8504	-.0005	9	.3543	-.005	.012	13	5/8
1907-S	35	1.3780	-.0005	55	2.1654	-.0005	10	.3937	-.005	.025	13	3/4
1908-S	40	1.5748	-.0005	62	2.4409	-.0005	12	.4724	-.005	.025	13	3/4
1909-S	45	1.7717	-.0005	68	2.6772	-.0005	12	.4724	-.005	.025	16	13/16
1910-S	50	1.9685	-.0005	72	2.8346	-.0005	12	.4724	-.005	.025	16	3/4
1911-S	55	2.1654	-.0006	80	3.1496	-.0005	13	.5118	-.005	.04	16	3/4
1912-S	60	2.3622	-.0006	85	3.3465	-.0006	13	.5118	-.005	.04	17	3/4
1913-S	65	2.5591	-.0006	90	3.5433	-.0006	13	.5118	-.005	.04	18	3/4
1914-S	70	2.7559	-.0006	100	3.9370	-.0006	16	.6299	-.005	.04	17	13/16
1915-S	75	2.9528	-.0006	105	4.1339	-.0006	16	.6299	-.005	.04	17	13/16
1916-S	80	3.1496	-.0006	110	4.3307	-.0006	16	.6299	-.005	.04	17	3/4
1917-S	85	3.3465	-.0008	120	4.7244	-.0006	18	.7087	-.005	.04	16	15/16
1918-S	90	3.5433	-.0008	125	4.9213	-.0008	18	.7087	-.005	.04	17	15/16
1919-S	95	3.7402	-.0008	130	5.1181	-.0008	18	.7087	-.005	.04	17	15/16
1920-S	100	3.9370	-.0008	140	5.5118	-.0008	20	.7874	-.005	.04	17	1/2
1921-S	105	4.1339	-.0008	145	5.7087	-.0008	20	.7874	-.005	.04	18	1/2
1922-S	110	4.3307	-.0008	150	5.9055	-.0008	20	.7874	-.005	.04	19	1/2
1924-S	120	4.7244	-.0008	165	6.4961	-.0010	22	.8661	-.005	.04	18	3/4
1926-S	130	5.1181	-.0010	180	7.0866	-.0010	24	.9449	-.005	.06	18	3/4
1928-S	140	5.5118	-.0010	190	7.4803	-.0012	24	.9449	-.005	.06	19	3/4
1930-S	150	5.9055	-.0010	210	8.2677	-.0012	28	1.1024	-.005	.08	17	3/4
1932-S	160	6.2992	-.0010	220	8.6614	-.0012	28	1.1024	-.005	.08	18	3/4
1934-S	170	6.6929	-.0010	230	9.0551	-.0012	28	1.1024	-.005	.08	19	3/4
1936-S	180	7.0866	-.0010	250	9.8425	-.0012	33	1.2992	-.005	.08	17	3/4
1938-S	190	7.4803	-.0012	260	10.2362	-.0014	33	1.2992	-.010	.08	18	3/4
1940-S	200	7.8740	-.0012	280	11.0236	-.0014	38	1.4961	-.010	.08	17	1



RATED RADIAL LOAD CAPACITY in POUNDS

(Based on 500 hours minimum life; for calculation of life expectancies see page 173.)
(For ratings at speeds not listed, see page 172; for limiting speeds, see page 187.)

M-R-C Bearing Number	Revolutions per Minute																		
	(For ratings at speeds not listed, see page 172; for limiting speeds, see page 187.)																		
	33 1/3*	50	100	200	300	400	500	600	700	800	900	1000	1200	1500	1800	2000	2500	3000	3600
1900-S	465	405	325	255	225	205	190	180	170	160.	155	150	140	130	125	120	110	105	100
1901-S	580	505	400	320	280	250	235	215	210	200	195	185	175	160	155	150	135	130	120
1902-S	750	655	520	410	360	325	305	285	270	260	250	240	225	210	200	190	175	165	155
1903-S	795	695	550	435	380	345	320	305	290	275	265	255	240	225	210	205	190	175	165
1904-S	1280	1120	885	705	615	560	520	490	465	445	425	410	390	360	340	325	305	285	27
1905-S	1460	1270	1010	800	700	635	590	555	530	505	485	470	440	410	385	370	345	325	30
1906-S	1610	1410	1120	885	775	705	655	615	585	560	535	520	490	455	425	410	380	360	33
1907-S	2040	1780	1420	1120	980	890	830	780	740	710	680	655	620	575	540	520	485	455	4
1908-S	2520	2200	1750	1390	1210	1100	1020	965	915	875	840	810	765	710	670	645	600	565	5
1909-S	2250	1980	1570	1240	1090	985	915	865	820	785	755	730	685	635	600	575	535	505	4
1910-S	2810	2460	1950	1550	1350	1230	1140	1070	1020	975	940	905	850	790	745	720	665	630	5
1911-S	3410	2980	2360	1870	1640	1490	1380	1300	1230	1180	1140	1100	1030	955	900	870	810	760	7
1912-S	3500	3060	2430	1930	1680	1530	1420	1340	1270	1210	1170	1130	1060	985	925	895	830	780	7
1913-S	3590	3140	2490	1980	1730	1570	1460	1370	1300	1240	1200	1160	1090	1010	950	915	850	800	7
1914-S	4100	3580	2840	2260	1970	1790	1660	1570	1490	1420	1370	1320	1240	1150	1090	1050	970	915	8
1915-S	4820	4210	3340	2650	2320	2100	1950	1840	1750	1670	1610	1550	1460	1350	1280	1230	1140	1080	10
1916-S	4950	4330	3440	2730	2380	2150	2010	1890	1800	1720	1650	1590	1500	1390	1310	1270	1180	1110	10
1917-S	7050	6160	4890	3880	3390	3080	2860	2690	2560	2440	2350	2270	2140	1980	1870	1800	1670	1570	14
1918-S	7280	6360	5050	4010	3500	3180	2950	2780	2640	2530	2430	2340	2210	2050	1930	1860	1730	1630	12
1919-S	7240	6320	5020	3980	3480	3160	2930	2760	2620	2510	2410	2330	2190	2030	1910	1850	1720	1610	11
1920-S	8160	7070	5620	4460	3890	3540	3280	3090	2940	2810	2700	2610	2450	2280	2140	2070	1920	1810	11
1921-S	8350	7300	5790	4600	4020	3650	3390	3190	3030	2900	2780	2690	2530	2350	2210	2130	1980	1860	11
1922-S	8590	7500	5950	4730	4130	3750	3480	3280	3110	2980	2860	2760	2600	2410	2270	2190	2040	1920	11
1924-S	10280	8980	7130	5660	4940	4490	4170	3920	3730	3560	3430	3310	3110	2890	2720	2630	2440	2290	2
1926-S	12480	10900	8650	6870	6000	5450	5060	4760	4520	4330	4160	4020	3780	3510	3300	3190	2960	2790	2
1928-S	12780	11160	8860	7030	6140	5580	5180	4880	4630	4430	4260	4110	3870	3590	3380	3260	3030	2850	2
1930-S	16800	14680	11650	9250	8080	7340	6810	6410	6090	5830	5600	5410	5090	4720	4520	4290	3980	3750	1
1932-S	17250	15070	11960	9490	8290	7540	7000	6580	6250	5980	5750	5550	5230	4850	4650	4410	4090	3800	1
1934-S	17720	15480	12290	9750	8520	7740	7190	6760	6420	6140	5910	5700	5370	4980	4780	4530	4200	3900	1
1936-S	22080	19290	15310	12150	10610	9640	8950	8420	8000	7650	7360	7110	6690	6210	5840	5640	5240	4930	1
1938-S	22730	19860	15760	12510	10930	9930	9220	8670	8240	7880	7580	7320	6880	6390	6010	5810	5390	5070	1
1940-S	28700	24640	19550	15520	13560	12320	11430	10760	10220	9780	9400	9080	8540	7930	7460	7200	6690	6370	1

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LIFE AND LOAD CALCULATION FOR

TIMKIN ROLLER BEARING

39520 CUP

39590 CONE

$$\text{RPM} = 240 \frac{43}{64} = 161 \text{ RPM}$$

RADIAL LOAD (RE):

$$23,000 \left(\frac{64}{43} \right) \frac{1}{3.43} = 9,950$$

$$\text{RRR} = (\text{RE})(\text{LF}) \times (\text{AF}) / \text{SF} \quad (\text{P. B-19})$$

$$\text{RE} = 9,950 \text{ lb}$$

$$\text{LF} = .719 \quad 1000 \text{ HRS} \quad (\text{P. B-13})$$

$$\text{AF} = 1.0$$

$$\text{SF} = 1.41 \quad (\text{P. B-12})$$

$$\text{RRR} = (9,950)(.719) / 1.41 = 5060 \text{ lbs}$$

① 500 RPM

$$\text{BRR} = 6100 \text{ lbs} \quad \text{① 500 RPM}$$

available in types

TS
TSF
TDI

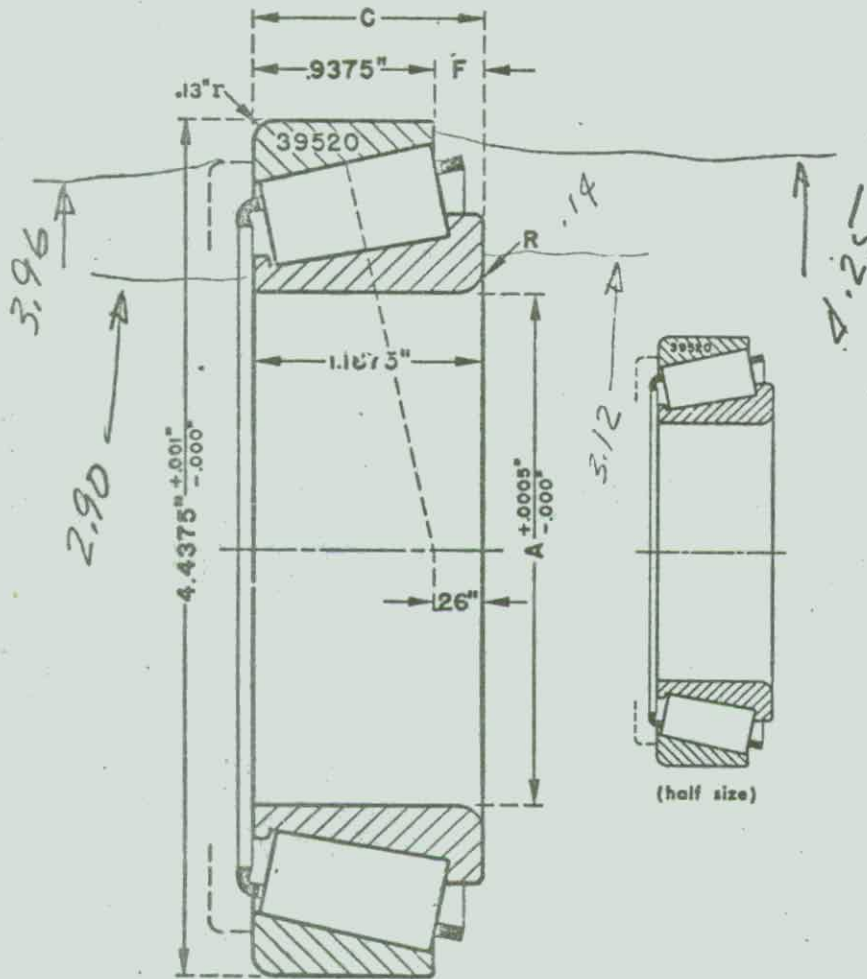
cone			cup no. 39520		bear- ing width C	cup no. 39522		bear- ing width C	cup no. 39528		bear- ing width C
bore A	radius R	number	outside diameter	standout F		outside diameter	standout F		outside diameter	standout F	
2.0000	.03	39573	4.4375	.2500	1.1875	4.4375	.2500	1.4375	4.7238	.2284	1.2894
2.0000	.14	39575									
2.1250	.14	39578									
2.2500	.14	39580									
2.2500	.31	39581									
2.5000	.14	39585									
*2.5585	.09	39586									
2.6250	.14	39590									

length = 1.1875"
radius = .13"

length = 1.0610"
radius = .03"

*cone length = 1.2175"

any cone in this series may be used with any cup in this series.



basic rating @ 500 rpm

radial (BRR)	6100 lbs.
thrust (BTR)	3550 lbs.
K	1.72

use modifying factors to compare
Timken bearing capacity-ratings with
other makes of bearings.

REV. PER MINUTE SPEED FACTOR

9. Compensation of misalignment

Self-aligning ball bearings, barrel roller bearings, radial spherical roller bearings, and spherical roller thrust bearings allow, on assembly, for the correction of initial misalignment. The outer ring raceway of these bearings is of a spherical form which allows the inner ring/roller set assembly to undergo swivelling motions. The permissible angular misalignment depends on bearing design and size.

It is also possible for the deep groove ball bearings to accept some misalignment; however, this is limited and dependent on the amount of radial clearance; the greater the clearance the greater the self-aligning ability. Cylindrical roller bearings and tapered roller bearings may achieve a limited amount of self-alignment by the provision of crowned or cambered raceways and rollers.

Table 6 presents information on the amounts of permissible angular misalignment.

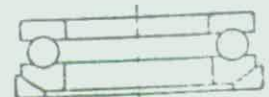
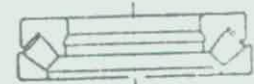
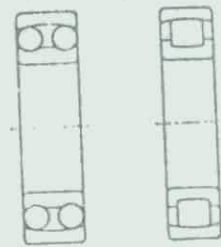
6. Permissible angular misalignment

Bearing type	Misalignment	
self-aligning ball bearings	4°	
barrel roller bearings	4°	
spherical roller bearings	0.5°	
spherical roller thrust bearings	3°	
deep groove ball bearings, standard clearance	} mating parts machined to k 5/J 6	
deep groove ball bearings, C3 clearance		8'
deep groove ball bearings, C4 clearance		12'
cylindrical roller bearings types N and NU, series 10, 2, 3, 4, RLS, RL L, RMS and RM L	16'	
cylindrical roller bearings, any other types and series	4'	
tapered roller bearings	2'	
	2'	

The prime causes of initial misalignment are the differences in height between the mounting seats of the housing support structures, for instance between the base plates of plummer blocks, or misalignment of housing bores if they are not machined in one setting.

Self-aligning bearings are also chosen in cases where dynamic misalignment through major shaft deflections or housing deformation is to be expected. The spherical basis of design automatically allows the bearing to correct for the misaligned condition. If rigid bearings were provided in such applications they would be subject to a tilting pressure with consequent additional stressing.

Under load a bearing ring should only be swivelled when the bearing rotates. Rotation is essential to ensure adequate lubrication between rolling elements and raceways. Adequacy of lubrication is in its turn required to neutralize the effects of sliding occurring between rollers and raceways during swivelling. Thrust ball bearings are sensitive to out-of-square-ness between abutment surface and bearing axis. Machining errors of this sort can be corrected by the provision of spherical housing washers and seating washers. Thrust ball bearings with spherical housing washers and seating washers are of the single or double-acting type.



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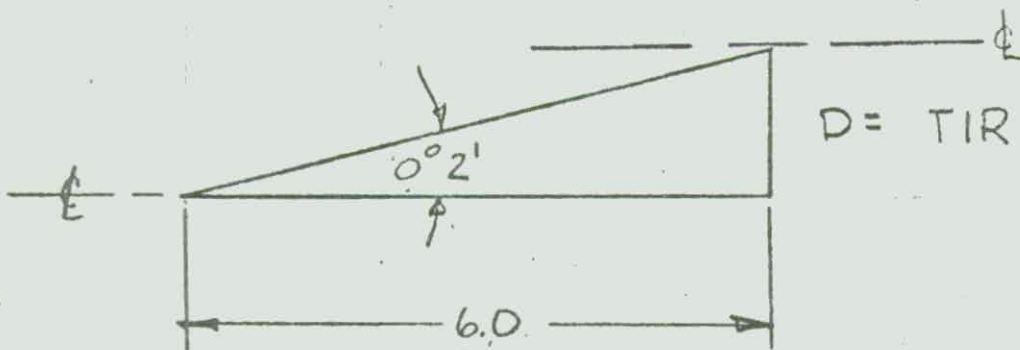
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$$\text{TAN } 0^{\circ} 21' = \frac{\text{TIR}}{6}$$

$$\text{TIR} = (\text{TAN } 21') 6 = .00058(6) = .00348$$

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LIFE AND LOAD CALCULATION FOR
SKF 6213

$$2P = 34,200 / 3.43 = 9,950 \text{ lbs}$$

$$P = 4970 \text{ lbs}$$

$$\text{RPM} = 161 \frac{48}{64} = 121$$

$$C/P \text{ FOR } 1000 \text{ HRS} = 1.95$$

C = BASIC DYNAMIC LOAD RATING
FROM TABLE

$$C = (1.95)(4970) = 9,700 \text{ lbs}$$

$$C \text{ FOR } 6213 = 9,900 \text{ lbs}$$

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FIND STRESS IN OUTPUT SHAFT

$$\tau = \frac{T r}{J} = \frac{32(T)(r)}{(3.14)(d_o^4 - d_i^4)}$$

$$\tau = \frac{(32)(45,632)(1.0)}{(3.14)(16)} = 29,064 \text{ lb/in}^2$$

$$T = 23,000(1.984) = 45,632 \text{ IN-16}$$

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SPLINE $\frac{8}{16} = \frac{P}{P_s}$

$P =$ DIAMETRAL PITCH = 8

$P_s =$ STUB PITCH = 16

$1.000/P_s =$ APPENDUM = .062

SIDE FIT (loose) CLASS 1

(MODIFY DIMS DEPENDING ON FINISH (phosphate coating, etc.))

		INTERNAL		EXTERNAL	
		MAJ DIA	MINOR DIA	MAJ DIA	MINOR DIA
PITCH DIA	2.50				
NO. TEETH	20	2.6250	2.3750	2.5960	2.3460
PITCH DIA	3.50				
NO. TEETH	28	3.6250	3.3750	3.5960	3.3460

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LEWIS STRESS FOR 43T PINION :

$$S_L = \frac{PW_t}{FY} = 65,000 \text{ PSI}$$

$$\text{ALMEN STRESS} = \frac{S_L}{(C.R.)(\cos \phi)}$$

$$\text{ALMEN STRESS} = \frac{65,000}{(1.437)(.939)} = 48,500 \text{ PSI}$$

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HERTZ STRESS: 43T PINION

$$S_c = 5715 \sqrt{\frac{W_t}{F_d} \frac{(m_G + 1)}{m_G}}$$

$W_t = 7,500$

$F = 1.75$

$d = 6.143$

$m_G = 69/43 = 1.49$

$$S_c = 5715 \sqrt{\frac{7500}{(1.75)(6.143)} \frac{2.49}{1.49}}$$

$S_c = 195,139 \text{ PSI}$

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$$\rho = d \sin \phi / 2 \quad \sin \phi = \sin 20^\circ = .34202$$

$$\rho = (6.143)(.34202)/2 = 1.050 = R_1$$

$$\rho = (9.143)(.34202)/2 = 1.564 = R_2$$

$$B = \sqrt{\frac{16 F (K_1 + K_2) (R_1) (R_2)}{L (R_1 + R_2)}}$$

$$F = 7500 \text{ lbs}$$

$$L = 1.75 \text{ IN}$$

$$K_1 = K_2 = \frac{1 - \nu^2}{\pi E} = \frac{1 - .3^2}{(3.14)(29)(10^6)} = 1 \times 10^{-8}$$

$$K_1 + K_2 = 2(10^{-8})$$

$$B = \sqrt{\frac{(16)(7500)(2)(10^{-8})(1.05)(1.564)}{(1.75)(2.614)}}$$

$$B = \sqrt{.000861} = .029$$

$$z = .393 B$$

$$z = .011 \text{ INCHES}$$

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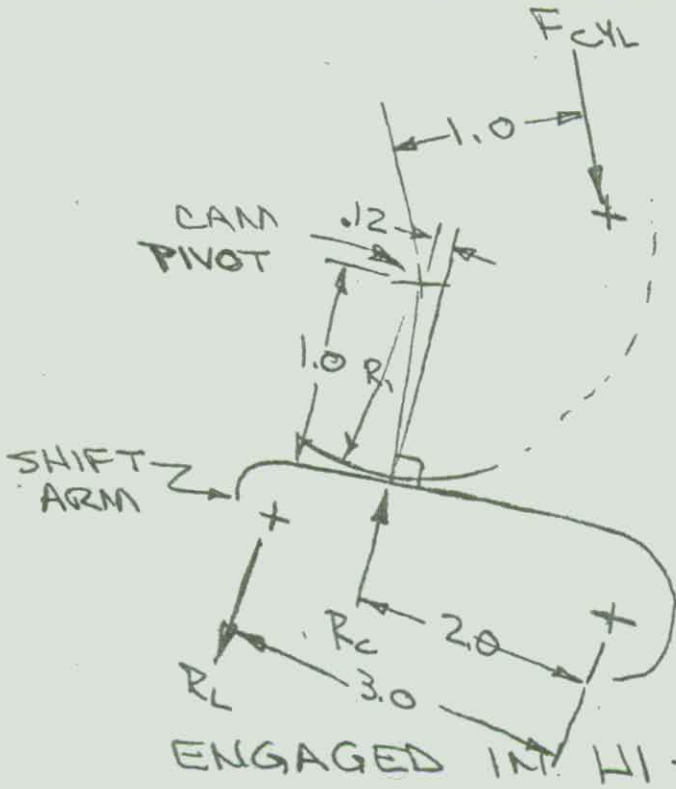
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TRANSFER CASE HI-RANGE LOCK-OUT



FOR 1.06 BORE X .313
ROD DIA

ROD END FORCE MAX

$$F_{ROD} = \frac{\pi}{4} (D_B^2 - D_R^2) P$$

@ 60 PSI

(FOR OPERATING MARGIN)

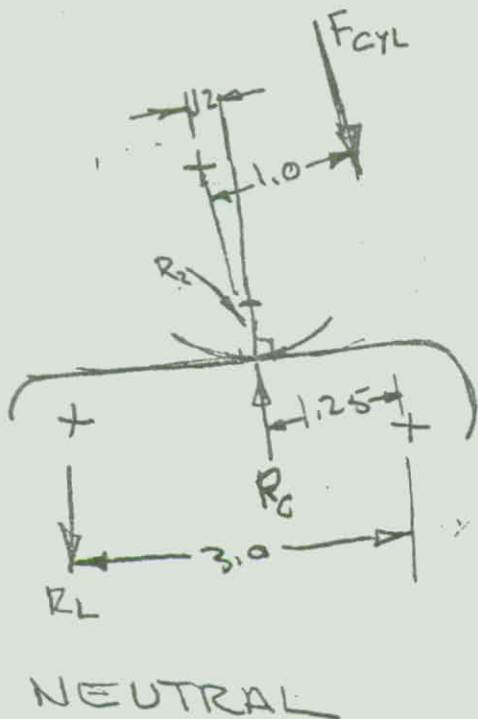
$$F_{ROD} = \frac{\pi}{4} (1.063^2 - .313^2) (60)$$

$$= 48.6 \#$$

IF THIS IS STORED IN
THE SPRING

$$R_L = \frac{48.6(1.0)}{.12} \times \frac{2.0}{3.0}$$

$$= 270 \#$$



@ 60 PSI

$$R_L = \frac{48.6(1.0)}{.12} \times \frac{1.25}{3}$$

$$= 169 \#$$

SHOULD PROVIDE
ADEQUATE RESISTANCE
TO SHIFTING BACK
INTO HIGH RANGE

(PRESS NORMALLY 100 PSI)

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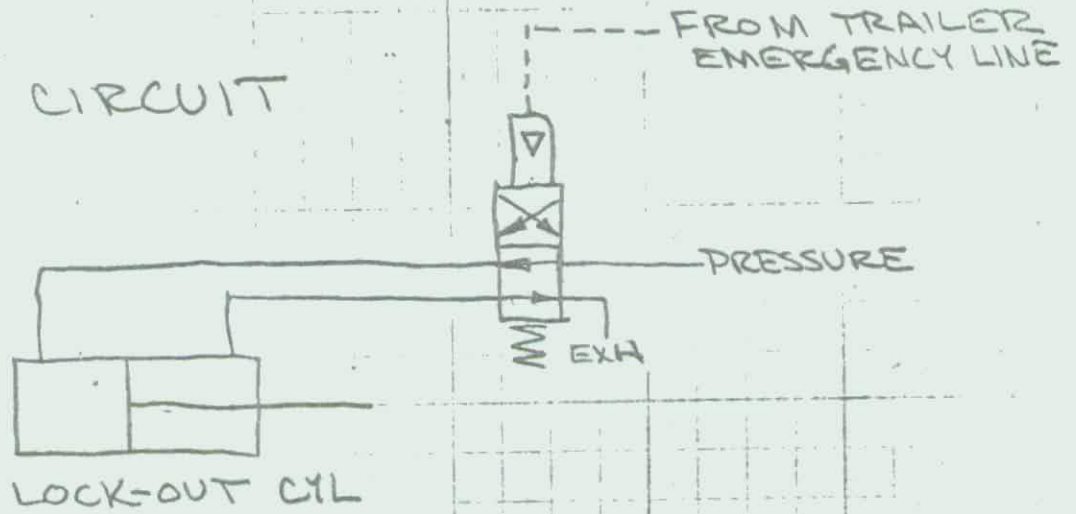
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AIR CIRCUIT

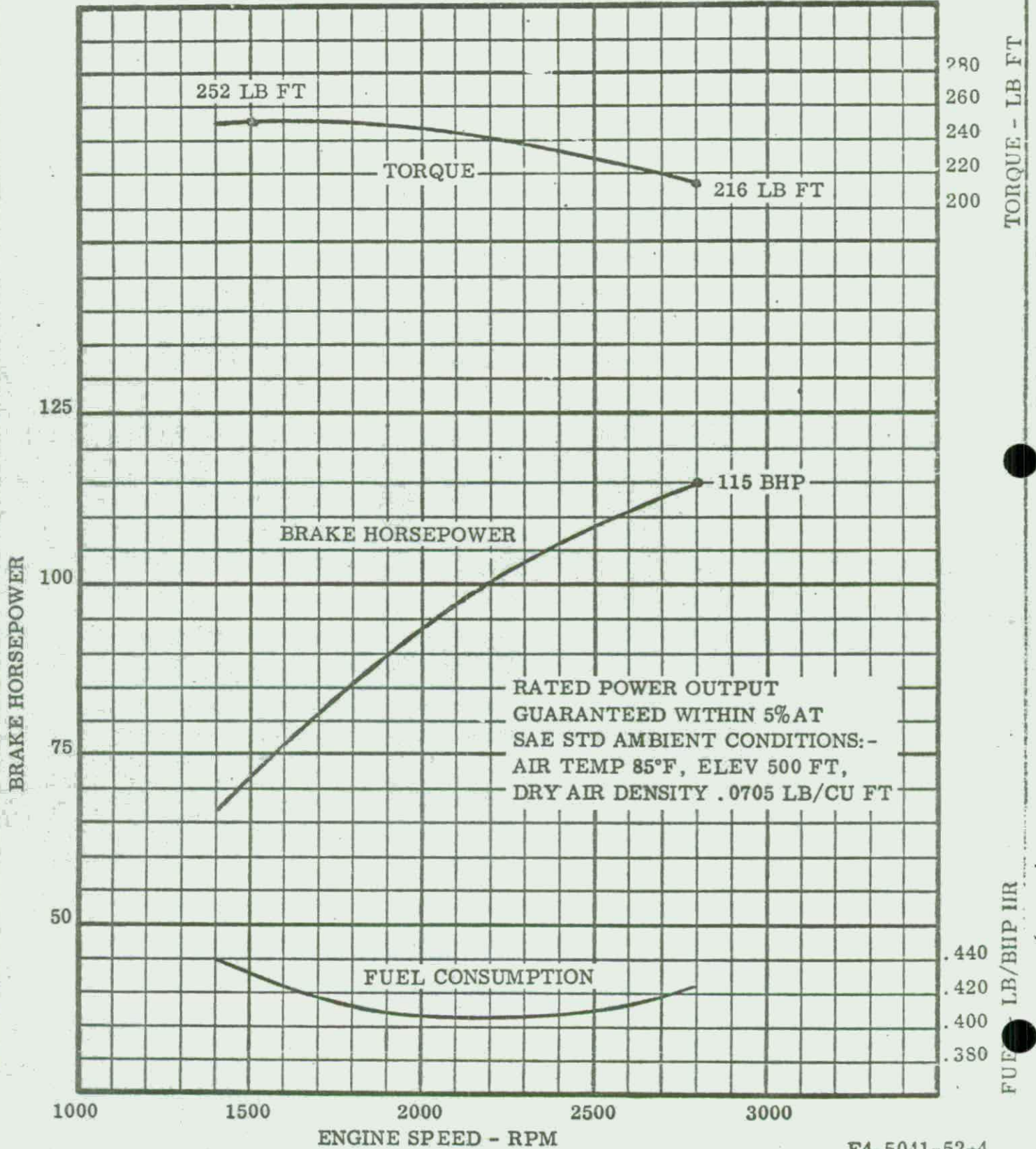




DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

BASIC ENGINE
MODEL 4-53N DIESEL
(4 VALVE HEAD)
N40 INJECTORS



DETROIT DIESEL ALLISON COOLING FANS

Page No.	Fan Part No. GM	Blade Description Dia. - Blades x Projected Width	Used As				(1) Drive Ratio	(2)	Characteristic Curve Number	Model Usage
			Blwr.		Suct.					
			R. H.	L. H.	R. H.	L. H.				
1	5128457	14 - 6 x 1.34	*		*	1.25	1	F1-0000-00-46	2-53	
1	5128457	14 - 6 x 1.34	*		*	1.70	2	F1-0000-00-46	2-53	
2	5146844	17 - 6 - 1.75		*	*	1.25	2	F1-0000-00-47	53	
2	5146844	17 - 6 - 1.75		*	*	1.0	1	F1-0000-00-47	53	
2	5146845	17 - 6 - 1.75	*		*	1.25	2	F1-0000-00-47	53	
2	5146845	17 - 6 - 1.75	*		*	1.0	1	F1-0000-00-47	53	
4	5112830	18 - 6 - 2.38	*		*	1.54	7	F1-0000-00-54	2-71	
4	5145773	18 - 6 - 2.38	*		*	1.25	4	F1-0000-00-54	53, 71	
4	5147710	18 - 6 - 2.38	*		*	1.25	4	F1-0000-00-54	53, 71	
3	5147711	18 - 6 - 2.38	*		*	1.54	5	F1-0000-00-52	2-71	
5	5145211	20 - 6 - 2.27	*		*	1.00	3	F1-0000-00-56	53, 71	
5	5145212	20 - 6 - 2.27	*		*	1.00	3	F1-0000-00-56	53, 71	
6	5100158	22 - 5 x 2		*	*	1.0	15	F1-0000-00-58	4-53	
6	5119011	22 - 5 x 2		*	*	1.25	17	F1-0000-00-58	3, 4-53	
6	5119011	22 - 5 x 2		*	*	1.0	15	F1-0000-00-58	3, 4-53	
6	5119014	22 - 5 x 2	*		*	1.0	15	F1-0000-00-58	3, 4-53	
7	5119012	22 - 5 x 2 3/4		*	*	1.0	16	F1-0000-00-59	3, 4-53	
7	5119013	22 - 5 x 2 3/4	*		*	1.0	16	F1-0000-00-59	3, 4-53	
7	5145904	22 - 5 x 2 3/4	*		*	1.0	16	F1-0000-00-59	53	
7	5162887	22 - 5 x 2 3/4		*	*	1.0	16	F1-0000-00-59	53	
20	5173871	22 - 6 x 2 1/4	*		*	1.1	17	F1-0000-00-73	6V-53	
20	5173872	22 - 6 x 2 1/4		*	*	1.1	17	F1-0000-00-73	6V-53	
8	3292286	22 - 6 x 2 3/8	*		*	1.25	8	F1-0000-00-60	3, 4, 6-71	
8	3223656	22 - 6 x 2 3/8		*	*	1.3	9	F1-0000-00-60	4-71	
9	5124701	24 - 6 x 2 3/8		*	*	1.25	10	F1-0000-00-61	4-53	
9	5128508	24 - 6 x 2 3/8		*	*	1.25	10	F1-0000-00-61	6V-71	
9	5100183	24 - 6 x 2 3/8	*		*	1.1	7	F1-0000-00-61	6V-53	
9	5140100	24 - 6 x 2 3/8	*		*	1.0	6	F1-0000-00-61	3, 4-53, 3, 4-71	
28	5139962	24 - 6 x 2.50			*	1.0	6	F1-0000-00-90	6-71	
16	5173429	26 - 4 x 2 3/8		*	*	1.25	10	F1-0000-00-62	3, 4-71, V71	
11	5173430	26 - 4 x 2 1/2	*		*	1.25	10	F1-0000-00-63	3, 4-71	
12	5171228	26 - 6 x 2 3/4		*	*	1.25	12	F1-0000-00-64	71, V-71, 6V-53, 8V-53	

(1) Fan Speed - Engine Speed x Drive Ratio.

(2) Under this number horsepower is on curve No. F1-0000-00-45, Sheet 1 & 2.

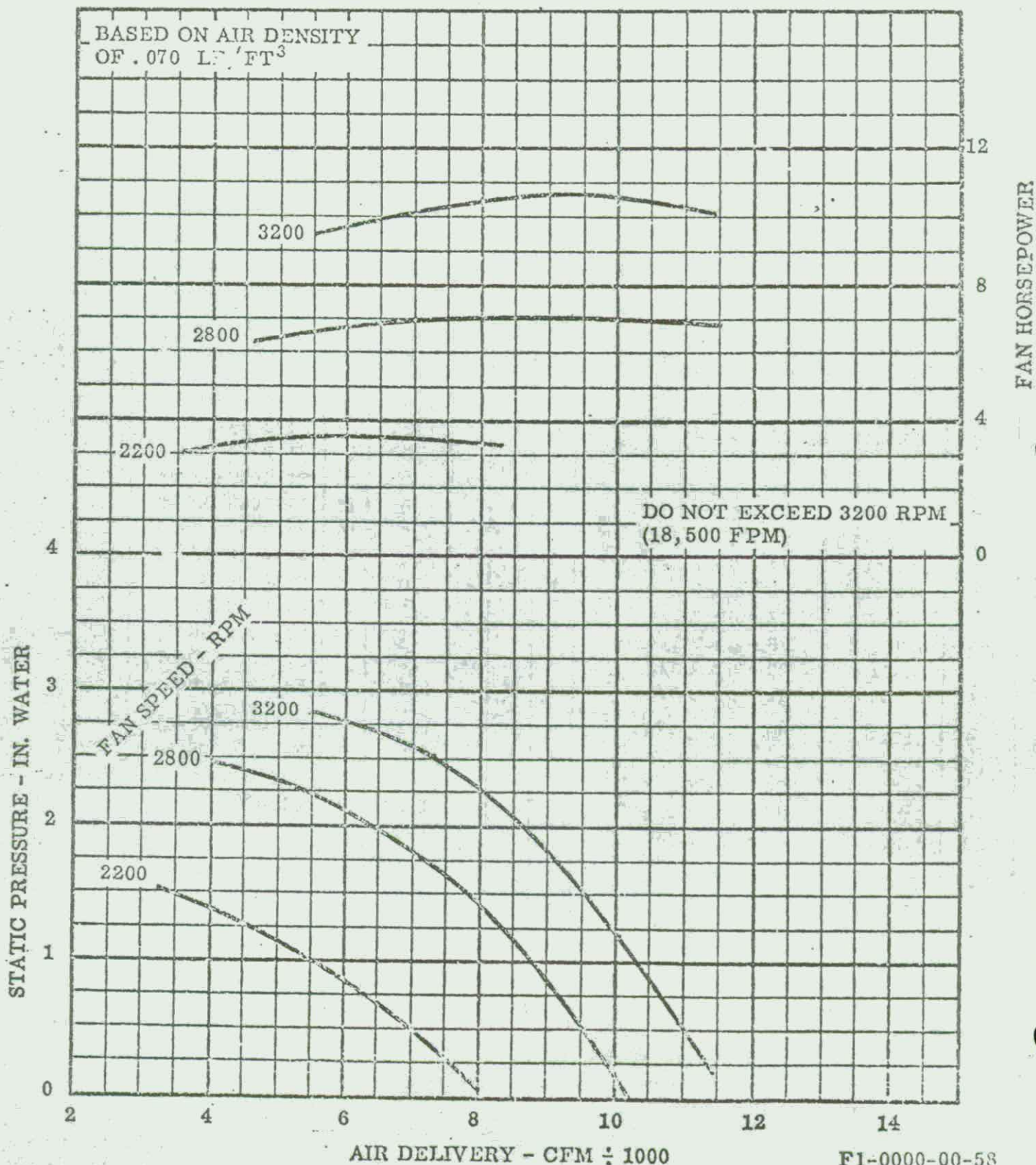
(3) Air Delivery at Fan RPM.



DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

FAN CHARACTERISTICS
22 INCH - 5 BLADE x 2 INCH PROJECTED WIDTH
PROBABLE INSTALLATION PERFORMANCE

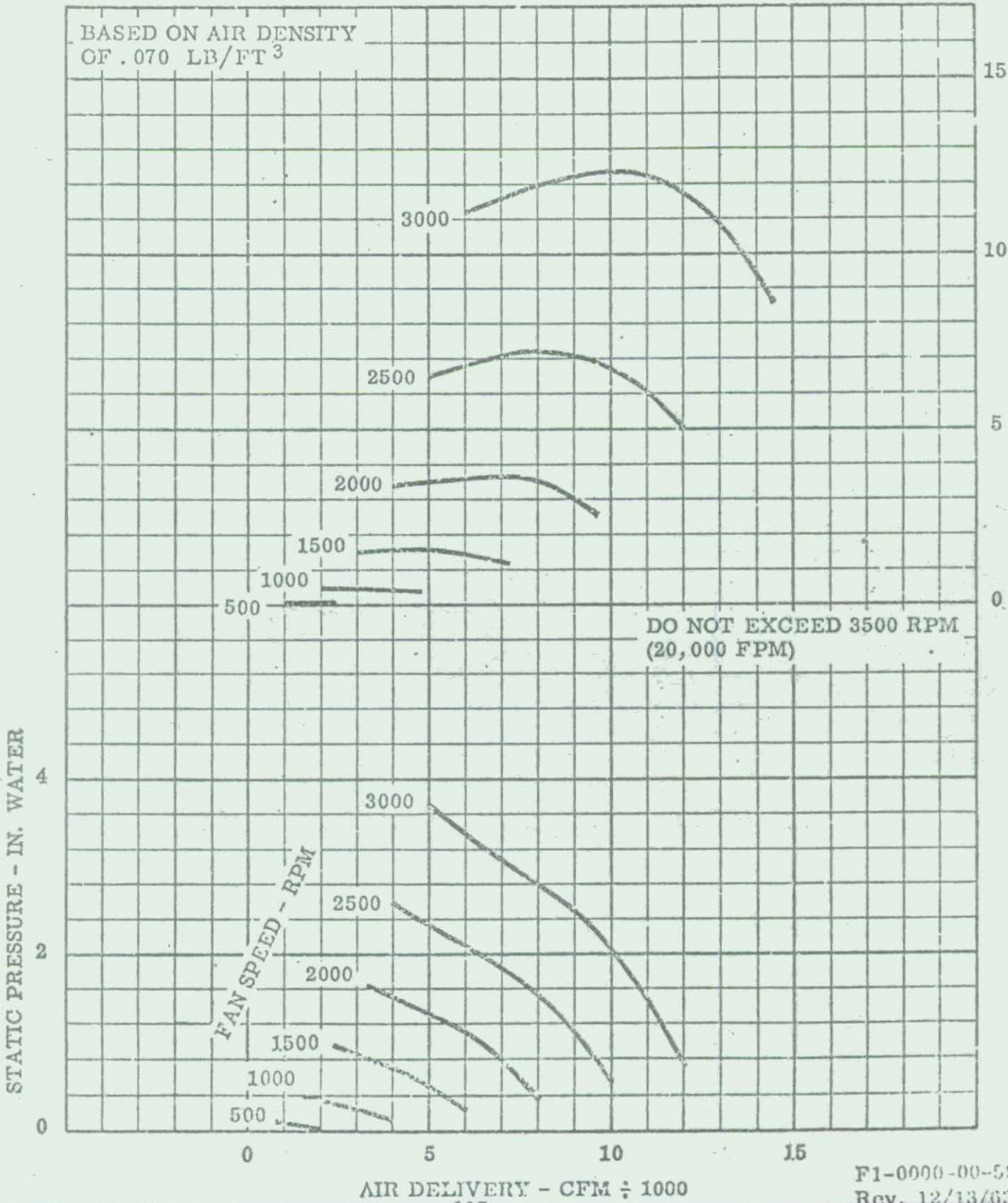




DETROIT DIESEL ENGINE DIVISION

GENERAL MOTORS CORPORATION

FAN CHARACTERISTICS
22 INCH - 5 BLADE x 2 3/4 INCH PROJECTED WIDTH
PROBABLE INSTALLATION PERFORMANCE



PACIFIC CAR AND FOUNDRY COMPANY

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ENGINE

DETROIT DIESEL 4V-53N 33.0 BTU/MIN/HP
140 HP = 4620 BTU/MIN

TRANSMISSION

CLARK 5-SPEED 96% EFF IN LOW

HYDRAULIC SYSTEM

15 HP @ 50% EFF 318 BTU/MIN

COOLANT FLOW

RPM	FLOW (GPM)
1200	29
2800	59

TOTAL HEAT REJECTION 4938 BTU/MIN

AMBIENT AIR TEMP = 125 °F

WATER TEMP TO RADIATOR = 220 °F

LIMIT 230 °F

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MASS FLOW OF COOLANT @ 220°F

$$\rho_{H_2O} = 59.5 \text{ lb/ft}^3 = 7.97 \frac{\text{lb}}{\text{GAL}}$$

$$\Delta T \text{ COOLANT} = (4938) / (7.97)(59)(1.0)$$

$$\Delta T \text{ COOLANT} = 10.5^\circ \text{F}$$

MAXIMUM AIR TEMP = COOLANT MINIMUM
TEMP MINUS 15°F

$$\text{MAX. AIR TEMP} = 210 - 15 = 195^\circ \text{F}$$

$$\Delta T \text{ AIR} = 195 - 125 = 70^\circ \text{F}$$

$$\dot{m}_{\text{AIR}} = 4938 / (241)(70) = 293 \text{ lb/MIN}$$

$$\text{@ } 195^\circ \text{F} \quad \rho_{\text{AIR}} = .060 \text{ lb/ft}^3$$

$$\text{REQUIRED CFM} = 293 / .060 = 4855$$

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RADIATOR CORE SIZE ALLOCATED:

$$24 \times 24 = 576 \text{ IN}^2 = 4.00 \text{ FT}^2$$

$$\text{FACE VELOCITY} = 4883 / 4.0 = 1220 \frac{\text{FT}}{\text{MIN}}$$

CORRECTED TO STD AIR (SFPM) =

$$1220 \left(\frac{530}{655} \right) = 988$$

ASSUME SFPM = 1000

$$\text{MTD} = 215 - 125 = 90^\circ \text{F}$$

$$Q_{\text{RAD}} = (H)(A)(\text{MTD}) = (14.0)(4.0)(90)$$

$$Q_{\text{RAD}} = 5050 \text{ BTU/MIN (NEGLECTING TUBE VELOCITY CORRECTION)}$$

4938 BTU/MIN REQUIRED.

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FOR 1000 SFPM FACE VELOCITY IS

$$1000 \frac{655}{530} = 1240 \text{ FT/MIN}$$

$$\text{CFM} = (1240)(4) = 4950$$

FOR .75 % FAN EFFICIENCY:

$$\text{CFM} = 4950 / .75 = 6600 \text{ CFM}$$

FAN REQUIREMENT IS 6600 CFM @

"X" INCHES H₂O

DP ESTIMATE: RADIATOR @ 1240 FT/MIN .75

SYSTEM TOTAL 2.75 IN H₂O

24" SCHWITZER # 24201 w/VENTURI @ 2800

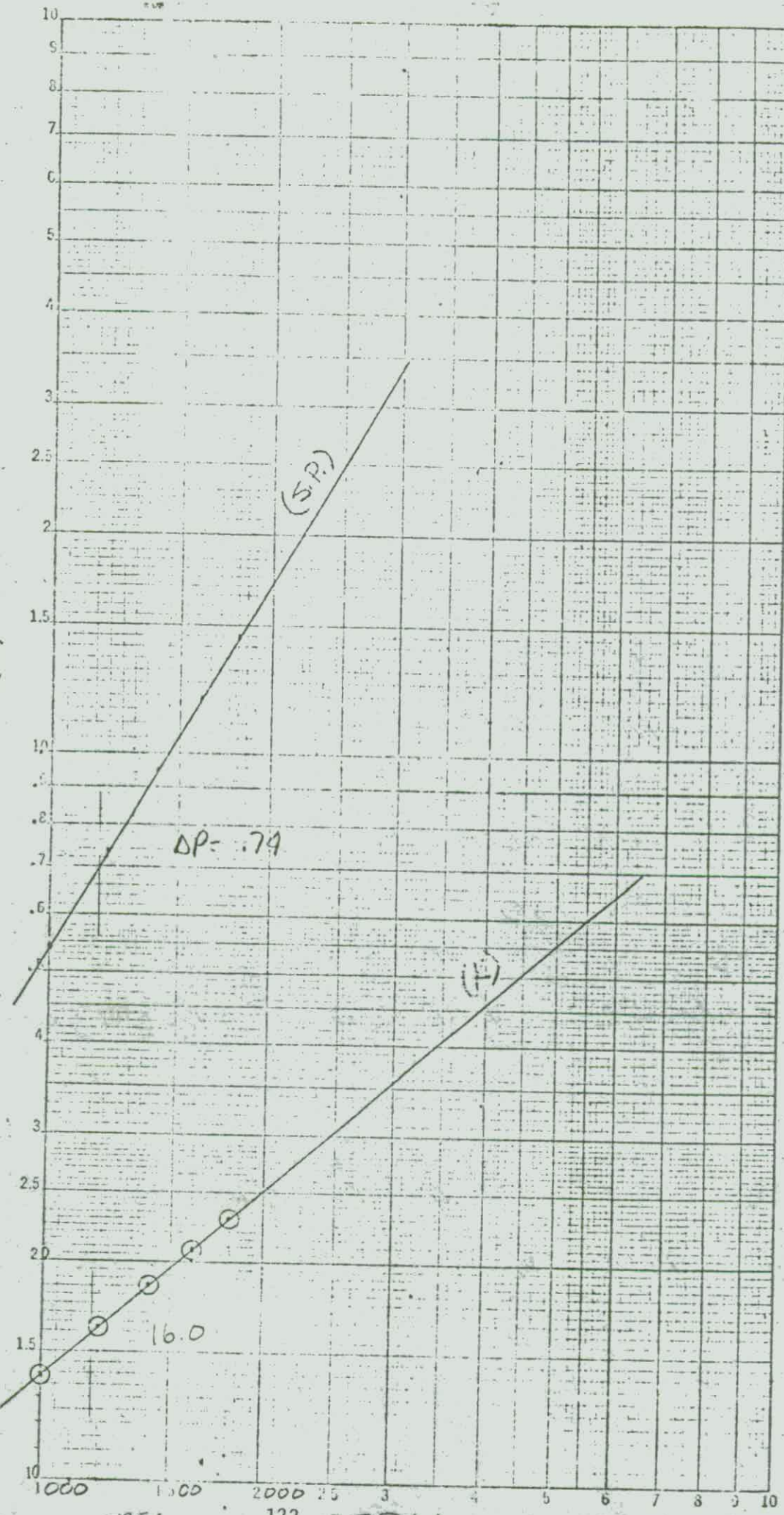
RPM: 7000 CFM @ 2.75 IN-H₂O

$$H_p = 6.8$$

GRDW
CORE

K (BTU/MIN-FT²-OMTD)
S.P. (IN-H₂O)

TUBE VEL = 100 FT/MIN



1254

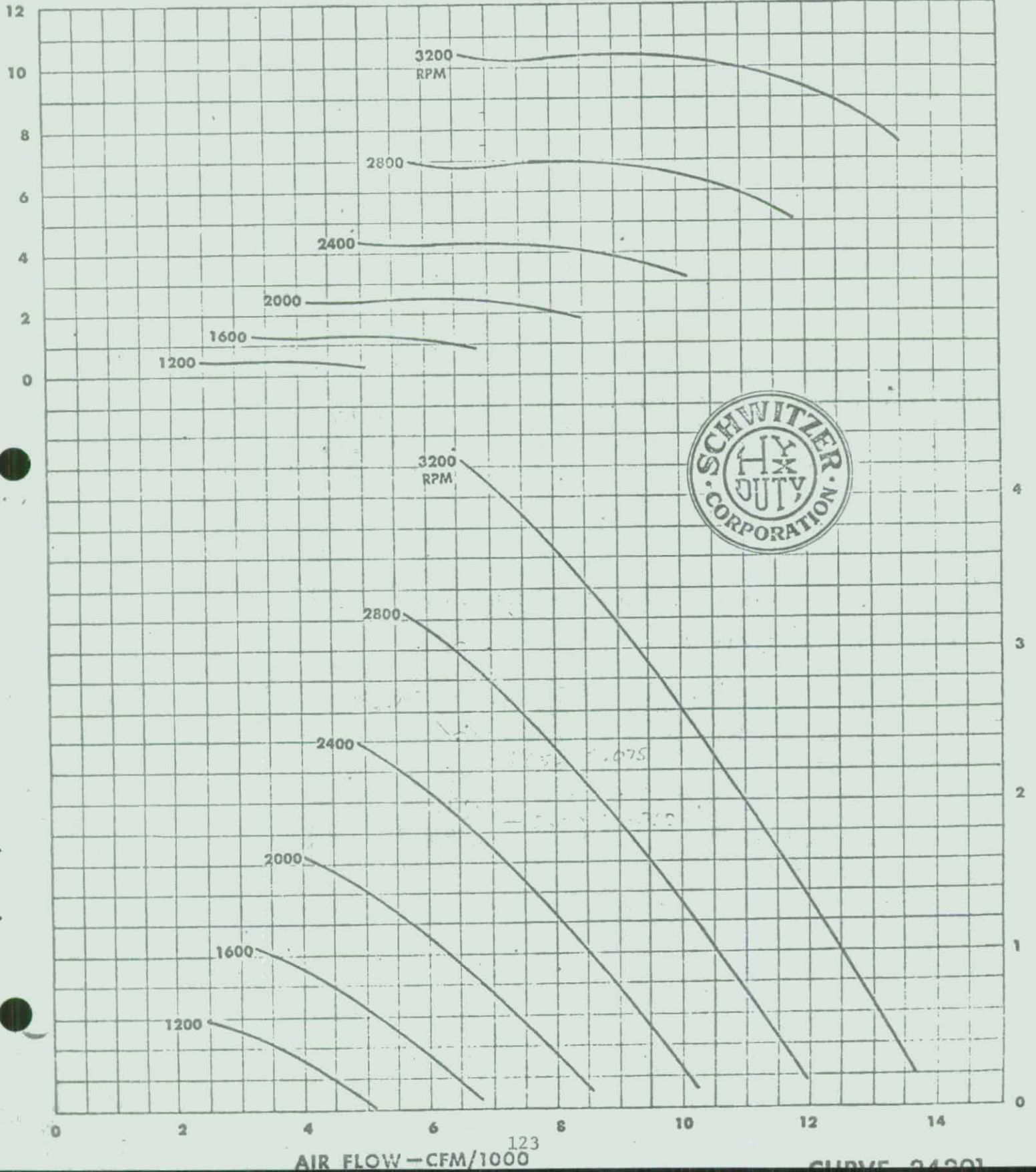
1254 122 SEDM



24
inch
Fan Diameter

Performance in venturi shroud

Standard Air Density .075



AIR FLOW - CFM/1000

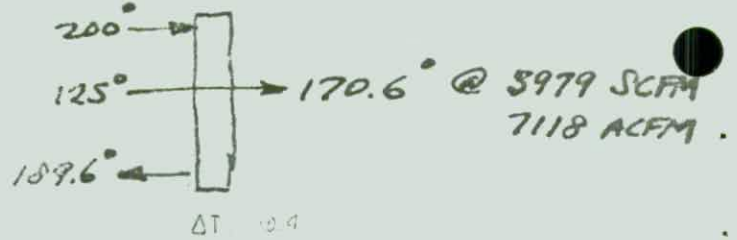
CURVE 24901

FLBBB* 17:09CST 11/16/73

ATLAS
D 23552
STEVE BLACK, DC PAR

INPUT?0

PRI74940,59.0,200,125.0,2.0
)
SEC74.2,2.19,1,24,0,0
S
TER732,1,1
)



Q= 4940 GPM= 59 PERGLY= 0 TT1=200.0 TAMB=125.0 ALT= 0 PP= 2.0 ADM=0
FA= 4.2 LEN= 2.2 FANS=1 DIA= 24 MXHP= 0 MXDPT= 0.

CC=32 NPCR=1 NPCO=1 NT= 260 VT= 79
AAH115 5 ROW CORE 4 7/16" FIN WIDTH (SEE 0235552.)
TTOUT TAOUT HP SFPM SCFM SSP ACFM TF EE DPT U CORR LMTD CF
189.6 170.6 1 1423 5979 0.86 6673 0.1 0.0 0.6 27 0.99 44.7 .964

INPUT?3 Δ = 0.84 DENSITY RATIO AT SUCKER FAN

TER?4,1,1 $\frac{5979}{0.84} = 7118 \text{ ACFM}$

CC= 4 NPCR=1 NPCO=1 NT= 312 VT= 66
AAH116 6 ROW CORE 5 5/16" FIN WIDTH
TTOUT TAOUT HP SFPM SCFM SSP ACFM TF EE DPT U CORR LMTD CF
189.6 175.3 1 1291 5424 0.83 6054 0.1 0.0 0.6 29 0.98 41.5 .956

INPUT?0

USED .22 UNITS
BYE
0000.25 CRU 0000.06 TCH 0000.84 KC

OFF AT 17:12CST 11/16/73

FLBBB*

17:27CST

11/16/73

ATLAN

D2 35552

STEVE BLACK, PACEAR

INPUT? 0

PRI? 4940, 59, 0, 210, 125, 0, 2, 0

S

SEC? 4.2, 2.2, 19, 1, 22, 0, 0

)

TER? 32, 1, 1

S

Q= 4940 GPM 59 PERGLY= 0 TTIN=210.0 TAMB=125.0 ALT= 0 PP= 2.0 ADM=0

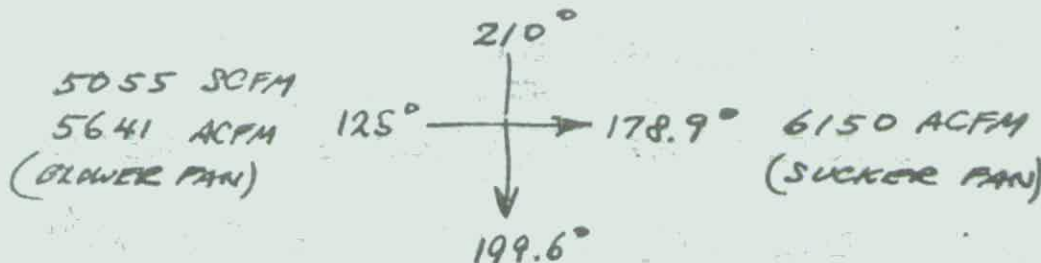
FA= 4.2 LEN= 2.2 FANS=1 DIA= 22 MXHP= 0 MXDPT= 0.

CC=32 NPCR=1 NPCO=1 NT= 260 VT= 79

AAH115

TTOUT	TAOUT	HP	SFPM	SCFM	SSP	ACFM	TF	EE	DPT	U	CORR	LMTD	CF
199.6	178.9	1	1203	5055	0.65	5641	0.1	0.0	0.6	24	0.99	49.7	.963

INPUT?



3

TER? 4.2, 1, 1

CC= 4 NPCR=1 NPCO=1 NT= 312 VT= 66

AAH116

TTOUT	TAOUT	HP	SFPM	SCFM	SSP	ACFM	TF	EE	DPT	U	CORR	LMTD	CF
199.6	184.0	1	1100	4623	0.63	5160	0.1	0.0	0.5	26	0.98	46.1	.954

INPUT? 4

PROGRAM STOP AT 2520

USED .22 UNITS

BYE

0000.25 CRU 0000.03 TCH 0000.87 KC

OFF AT 17:29CST 11/16/73

PACIFIC CAR AND FOUNDRY COMPANY

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AXLE LOADS

FROM PRELIMINARY WEIGHT & C.G.
STUDY

TOWING HOWITZER
DRIVE AXLE LOAD = 17,000#
STEER AXLE - UNLOADED

SOLO OPERATION
DRIVE AXLE LOAD = 9600#
STEER AXLE LOAD = 4800#

MAX DRAWBAR PULL = 13,500#

MAX BRAKING FORCE
DRIVE AXLE / W HOWITZER = 11,000#
STEER AXLE / SOLO = 4,790#

MAX DRIVE AXLE LOAD
PANIC STOP W/HOWITZER = 21,200#

MAX STEER AXLE LOAD
PARKED UPHILL ON 20° SLOPE = 7980#

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TORSION BAR SUSPENSION
DRIVE AXLE
 FOR 12" TOTAL TRAVEL (6 UP, 6 DOWN)

@ 8250# WHEEL LOAD

$$\text{TORQUE} = 8250 \times 25 \text{ IN} = 206,300 \text{ LB-IN}$$

(FROM LAYOUT)
 $\phi = 16^\circ$

@ MAX TRAVEL @ $\phi = 31^\circ 45'$

$$\text{MAX TORQUE} = 206,300 (31.75/16) = 409,000 \text{ LB-IN}$$

$$\text{MAX WHEEL LOAD} = 409,000 / 25.38 = 16,130\#$$

WITH 140 KSI SHEAR LIMIT
 (SHOT PEENED & PRESET)

$$d = \sqrt[3]{\frac{16 T}{\pi S_s}} = \sqrt[3]{\frac{16 (409,000)}{\pi (140,000)}} = 2.46 \text{ IN} \leftarrow$$

$$L = \frac{\phi G d^4}{584 T} = \frac{31.75 (11 \times 10^6) (2.46)^4}{584 (409 \times 10^3)}$$

$$= 53.5 \text{ " } \leftarrow$$

@ SOLO NOMINAL STATIC LOAD (8900#)

$$\text{MOMENT} = 8900 (15) = 133,500 \text{ LB-IN}$$

$$\text{WIND UP ANGLE} = \frac{133,500}{206,300} \times 16^\circ = 10.36^\circ$$

CHECK - FOR ACTIVE LENGTH = 54.92 ϕ d = 2.46

$$\phi = \frac{\pi S_s L 584}{16 G d} = \frac{\pi (140,000) (54.92) (584)}{16 (11 \times 10^6) (2.46)} = 32.52^\circ$$

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TORSION BAR PRESET

RECOMMENDED PRESET IS .022 STRAIN

FOR LARGE BAR THIS IS

$$\gamma = \frac{\phi d}{2L}$$

$$\phi = \frac{2\gamma L}{d} = 114.6 \frac{\gamma L}{d} = \frac{114.6(.022)(53.5)}{2.46}$$

$$= 54.8^\circ \leftarrow$$

FOR SMALL BAR

$$\phi = 114.6 \frac{\gamma L}{d} = \frac{114.6(.022)(40.7)}{1.275} = 80.5^\circ \leftarrow$$

MAXIMUM SET ANGLE DURING
PRESETTING (.008 STRAIN)

LARGE BAR $\frac{8}{22} (54.8) = 19.9^\circ \leftarrow$

SMALL BAR $\frac{8}{22} (80.5) = 29.25^\circ \leftarrow$

MAX ALLOWABLE SET DURING ENDURANCE

TEST = 10% OF MAX ALLOWABLE TWIST

LARGE BAR = 10% (31.45') = 3.145°

SMALL BAR = 10% (46.25') = 4.625°

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TORSION BAR SERRATIONS

$$\begin{aligned} \text{MAJOR DIA} &= 1.2 + (\text{BAR DIA}) \\ \text{LENGTH} &= .4 (\text{MAJOR DIA}) \end{aligned}$$

LARGE BAR

SMALL END

$$\text{MAJ DIA REQD} \geq 1.2 (2.46) = 2.95$$

USE 16/32 SIDE FIT FILLET ROOT

49 TOOTH 3.1250 MAJ DIA. ←

$$\text{LENGTH} = .4 (3.1250) = 1.25$$

USE 1.25 ←

LARGE END

USE 16/32 SIDE FIT FILLET ROOT

51 TOOTH 3.2500 MAJ DIA ←

$$\text{LENGTH} = 1.300$$

USE 1.312 ←

SMALL BAR

SMALL END

$$\text{MAJ DIA REQD} \geq 1.2 (1.275) = 1.56$$

USE 16/32 SIDE FIT FILLET ROOT

26 TOOTH MAJ DIA 1.6250 ←

$$\text{LENGTH} = .4 (1.625) = .6500$$

USE .68 ←

LARGE END

USE 16/32 SIDE FIT FILLET ROOT

28 TOOTH MAJ DIA 1.7500 ←

$$\text{LENGTH} = .4 (1.7500) = .75 ←$$

USE COILS

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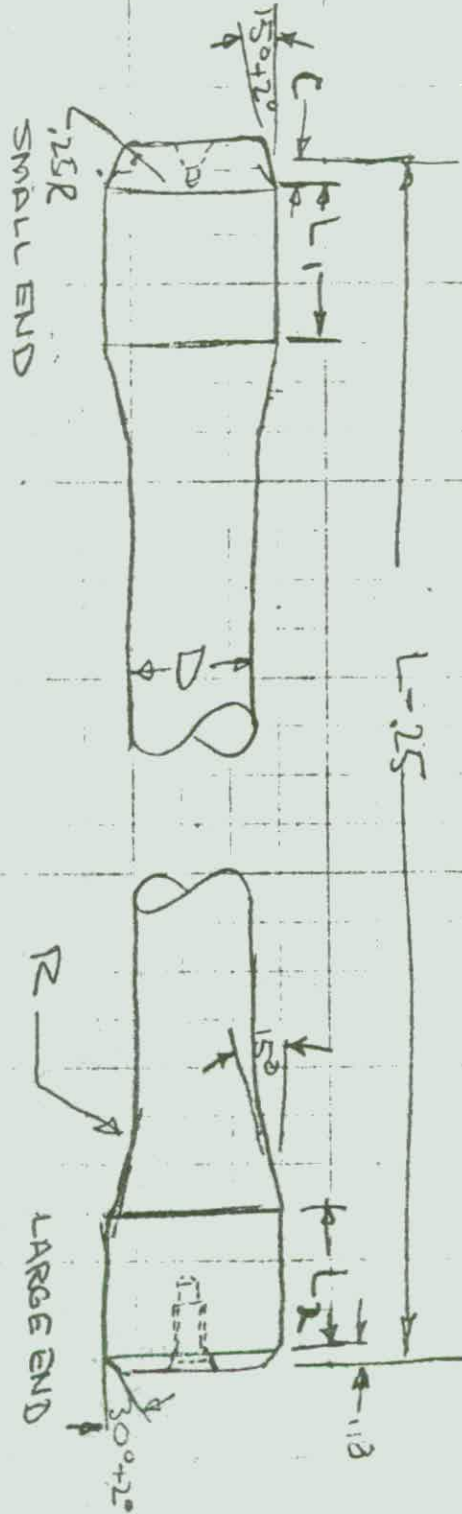
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MAX DEPARTMENT PRESET (X°)
 SET (N°)
 MAX WINDUP ANGLE
 MAX ALLOWABLE SET
 DURING FATIGUE TEST

LENGTH L	L1	L2	C	D	R	PRESET (X°)	WINDUP ANGLE (N°)	ALLOWABLE SET
59.88	1.25	1.31	.88	2.460+0.020	4.0	54.8°	31°45'	3°
59.88	1.25	1.31	.88	2.460+0.020	4.0	54.8°	31°45'	3°
44.50	.68	.75	.62	1.225+0.010	2.0	30.5°	46°25'	4.5°
44.50	.68	.75	.62	1.225+0.010	2.0	30.5°	46°25'	4.5°

use coils

55.813
1.25

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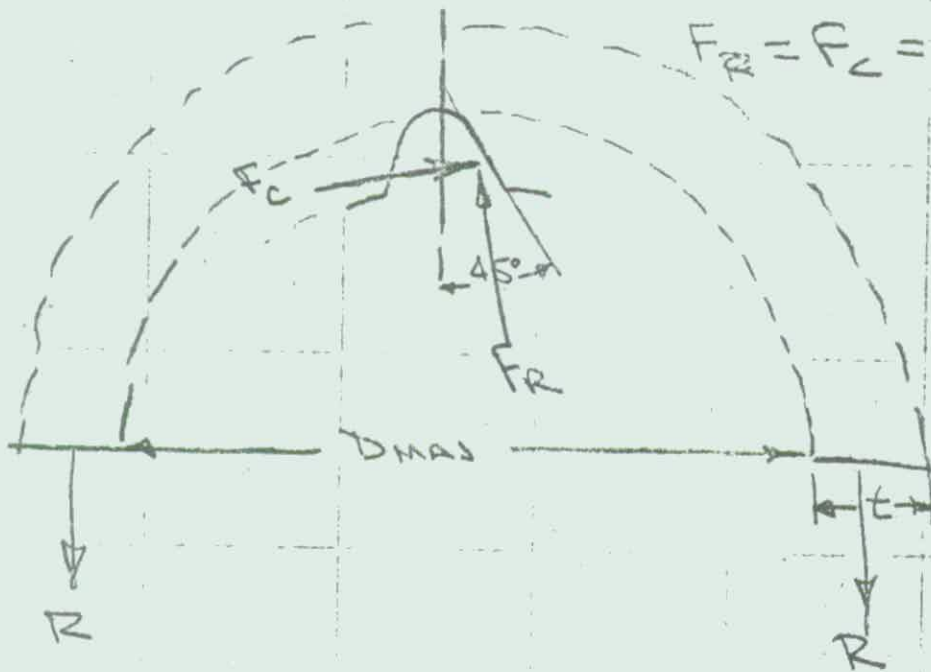
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TORSION BAR ANCHOR INTERNAL SERRATION RADIAL FORCE



$T_{MAX} = 55200 \text{ LB-IN}$
 $N = 26 \text{ TESTS}$
 $R_p = 1.5625/2$
 $D_{MAJ} = 16500$

$$R = F_R D_{MAJ}$$

$$\sigma = \frac{F_R D_{MAJ}}{2t} = \frac{T_{MAX} D_{MAJ}}{N R_p 2t}$$

FOR $\sigma = 100,000 \text{ PSI}$ W/FS = 2 ($\sigma_{UL} = 50,000$)

$$t = \frac{T_{MAX} D_{MAJ}}{N R_p 2\sigma} = \frac{55,200(165)}{26(1.5625)(2)(50,000)}$$

$= .0272 \text{ IN}$
 USE .25 PLUS

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TORSION BAR

WHEEL LOAD VS WHEEL TRAVEL

DRIVE AXLE

TORSION BAR TORQUE @ NOM STATIC LOAD

$$T_N = 8250 \times 25 = 206,000 \text{ LB-IN}$$

$$@ \theta = 16^\circ$$

$$T_{MAX} @ \theta = 31^\circ 45' = 409,000 \text{ LB-IN}$$

	FROM WHEEL MOM ARM (LAYOUT)	WHEEL LOAD #	WHEEL TRAVEL
$T_{2^\circ 30'}$ = 31,800 LB-IN	23.36	1360 #	-4.75
$T_{8^\circ 30'}$ = 114,600 "	24.12	4625 #	-2.88
$T_{13^\circ 30'}$ = 174,000 "	24.75	7025 #	-1.0
$T_{23^\circ 30'}$ = 302,700	25.38	11920 #	+2.88
$T_{31^\circ 45'}$ = 408,700	25.44	16050 #	+6.12

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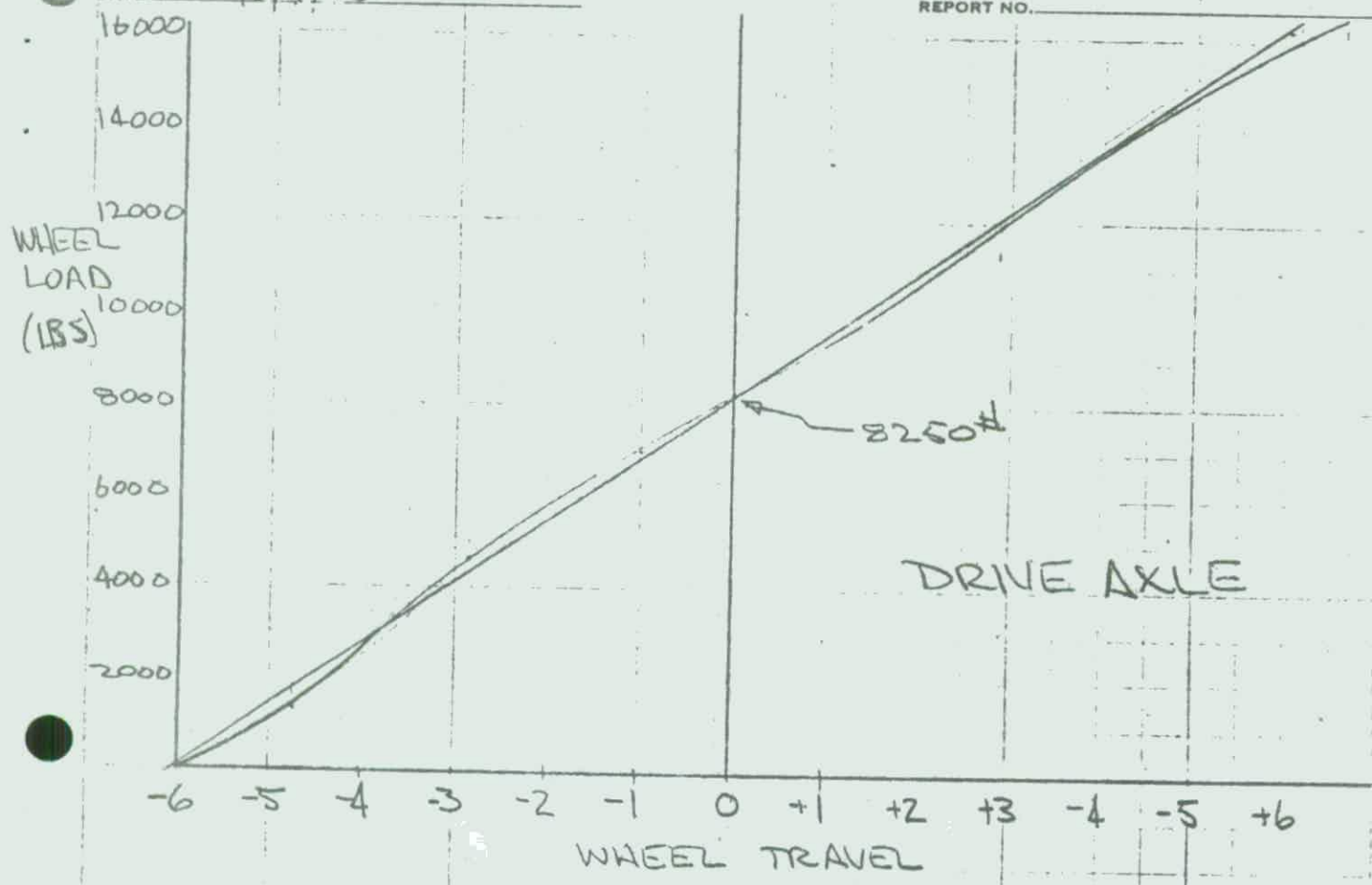
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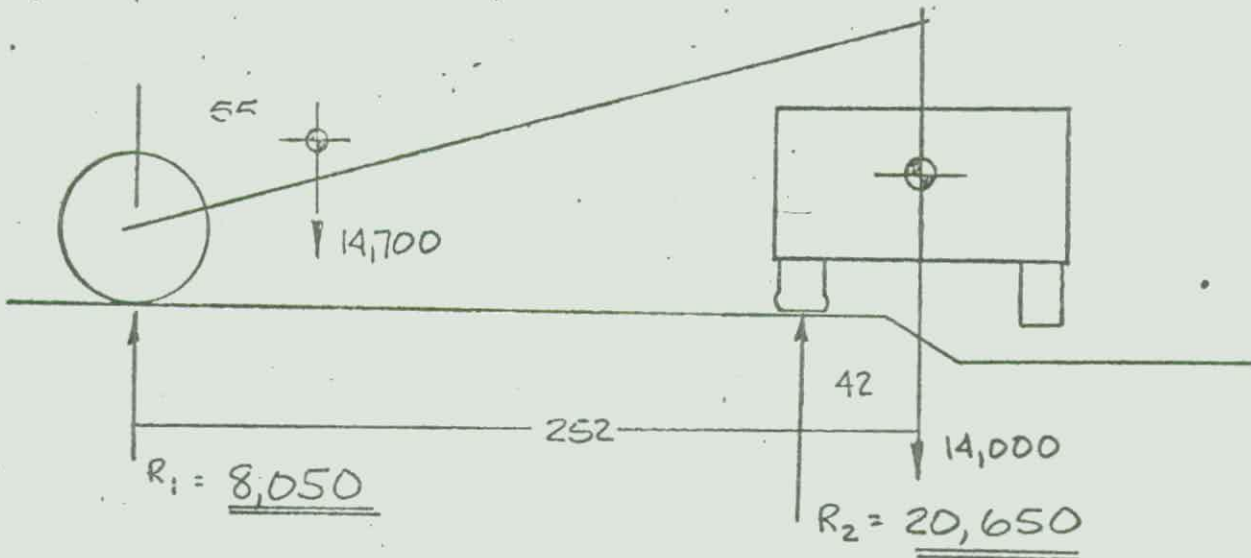
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$$\sum M_{R_1} = 0$$

$$\therefore (55)(14,700) + (252)(14,000) - 210(R_2) = 0$$

$$R_2 = 20,650$$

$$R_1 = 28,700 - 20,650 = 8,050$$

WHEEL WILL BOTTOM OUT

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

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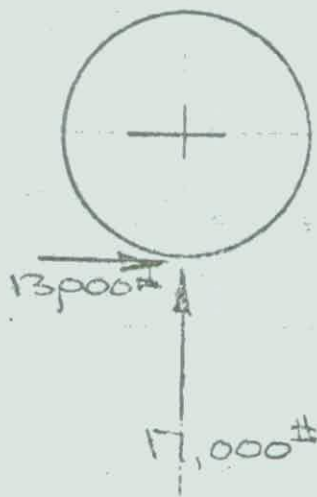
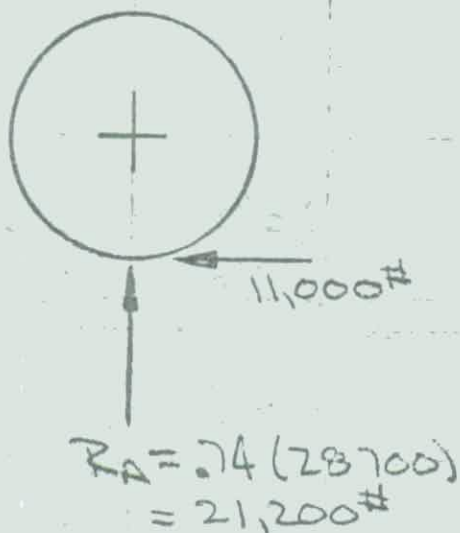
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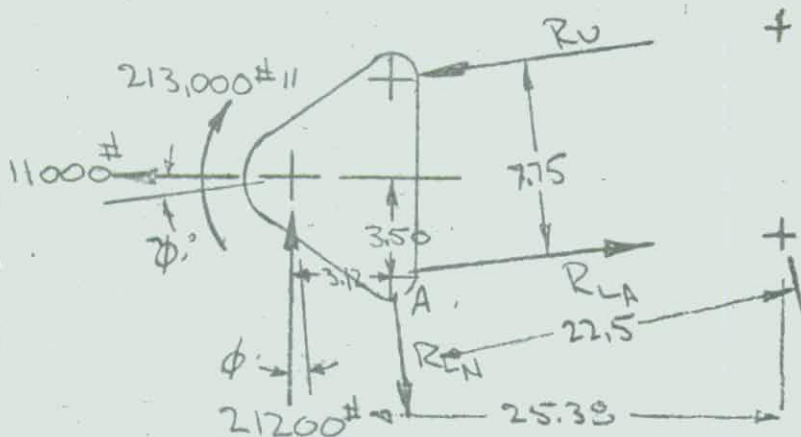
DRIVE AXLE SUSPENSION LOADS

1. BRAKING
SG FROM 35 MPH

2. MAX DBP
BACKING UP 50% SLOPE



AT WHEELS
(WORST LOAD IS COND. 1.)



AT AXLE - SUSPENSION ARM JOINT

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FOR AXLE LOAD OF 21,200#

$$\begin{aligned} \text{TOTAL SUSP ARM TRAVEL} &= 31.75^\circ @ 411,000 \text{ LBS IN} \\ \text{TRAVEL @ 21200} &= \frac{31.75}{411,000} \times \frac{21,200}{2} \times 25.38 \\ &= 20.8^\circ \end{aligned}$$

WITH SUSP RELAXED ARM IS 28.5°
 BELOW HORIZONTAL

SO AT 21,200# AXLE LOAD
 $\phi = 28.5 - 20.8 = 7.7^\circ$

TAKING MOMENTS AT A

$$\begin{aligned} 6.75 R_U + 3.5(11,000) &= 213,000 + 3.12(21,200) \\ R_U &= \frac{213,000 + 65,200 - 38,500}{7.75} \end{aligned}$$

$$= \underline{30,900\#} \text{ (TOTAL FOR BOTH WHEELS)}$$

$\Sigma F_x = 0$

$$\begin{aligned} R_{LA} &= R_U + 11,000(\cos 7.7^\circ) - 21,200(\sin 7.7^\circ) \\ &= 30,900 + 11,890 - 2,840 \\ &= \underline{39,950\#} \text{ (TOTAL FOR BOTH WHEELS)} \end{aligned}$$

$R_{LN} = \frac{\text{TORSION BAR MOMENT}}{\text{SUSP ARM LENGTH}}$

$$= \frac{21,200(25.38)}{22.50} = \underline{239,30\#}$$

(TOTAL FOR BOTH WHEELS)

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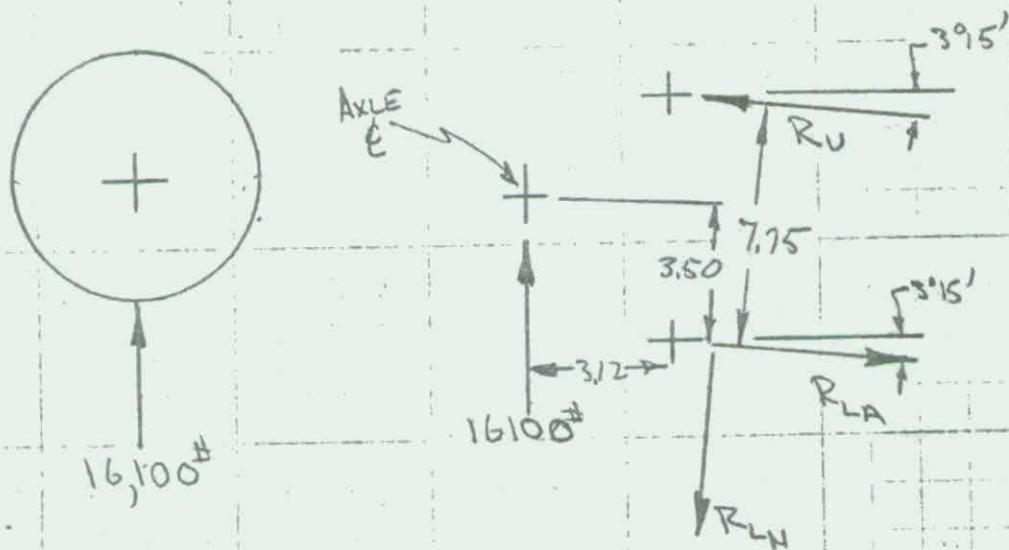
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3. LOADS AT MAX SPRING TRAVEL



NEGLECTING EFFECT OF SMALL ANGLE

$$R_U = \frac{16100(3.12)}{7.75} = \underline{\underline{6500\#}}$$

$$R_{LA} = \underline{\underline{6500\#}}$$

$$R_{LN} = \text{T-BAR TORQUE} / \text{ARM LENGTH}$$

$$= 409,000 / 22.5 = \underline{\underline{18,200\#}}$$

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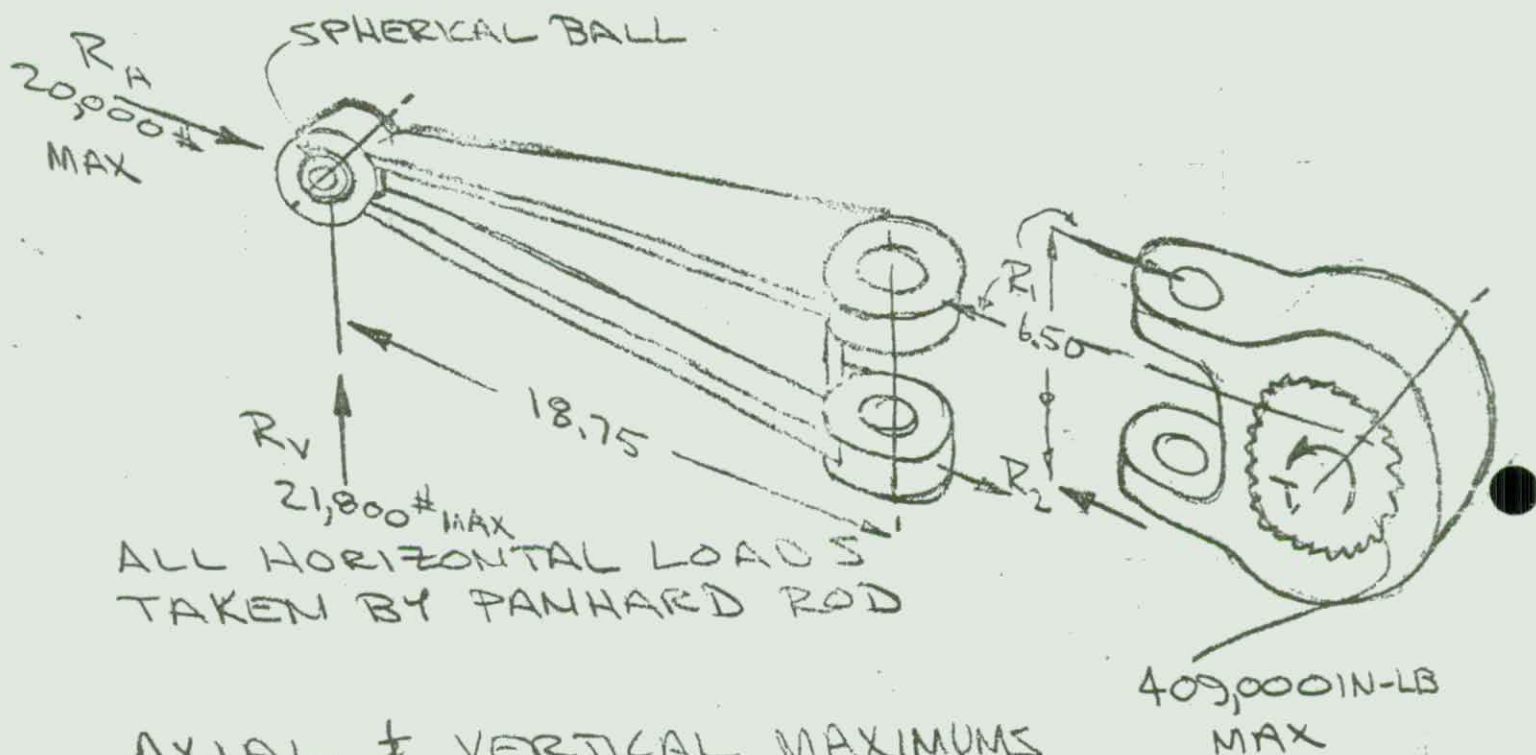
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SUSPENSION ARM - DRIVE AXLE



ALL HORIZONTAL LOADS
TAKEN BY PANHARD ROD

AXIAL & VERTICAL MAXIMUMS
DONT OCCUR AT SAME TIME

COND 1. $R_A = 20,000\#$, $R_V = 12,000\#$, $T = 225,000\text{ LB-}$

COND 2. $R_A = 6500\#$, $R_V = 21,800\#$, $T = 409,000\text{ LB-}$

CHECK COLUMN BUCKLING FOR CONDITION 1.

CHECK BENDING STRESS FOR CONDITION 2.

CHECK EYE STRESS ON BOTH PARTS

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BENDING STRESS

CHECK @ SECTIONS	A	1"	from small end	
	B	4"	"	"
	C	6 1/2"	"	"
	D	17 1/4"	"	"
	E	9 1/4"	"	"

FOR .38 THICK FLANGES & .25 THK WEB

$$I_B = \frac{.25(3.25)^3}{12} + 2 \left[\frac{(1.312)(.38)^3}{12} + 1.312(.38)(1.813)^2 \right]$$

$$= 1.03 + 2 [.056 + 1.1635] = 4.412 \text{ in}^4 \leftarrow$$

$$I_C = \frac{.25(3.75)^3}{12} + 2 \left[\frac{(1.625)(.38)^3}{12} + 1.625(.38)(2.063)^2 \right]$$

$$= 1.10 + 2 [.0714 + 2.59] = 6.323 \text{ in}^4 \leftarrow$$

$$I_D = \frac{.25(5.813)^3}{12} + 2 \left[\frac{3.04(.38)^3}{12} + 3.063(.38)(3.09)^2 \right]$$

$$= 4.09 + 2 [.1347 + 10.96] = 26.28 \text{ in}^4 \leftarrow$$

$$I_E = \frac{.25(4.313)^3}{12} + 2 \left[\frac{2.03(.38)^3}{12} + 2.03(.38)(2.344)^2 \right]$$

$$= 1.668 + 2 [.0893 + 4.18] = 10.21 \text{ in}^4 \leftarrow$$

MAX BENDING & AXIAL STRESS COMBINED
SECTION B

$$\sigma_{\text{MAX}} = \frac{4(21,800)(2)}{4.412} \pm \frac{6500}{3.25(.25) + 2(1.312)(.375)}$$

$$= 39,500 \text{ psi} \pm 360 \text{ psi} = 35,900 \text{ TENSION}$$

$$43,100 \text{ COMPRESS}$$

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SECTION C

$$\sigma_{MAX} = \frac{6.5(21,800)(2.25)}{6.323} + \frac{6500}{.25(3.75) + 2(1.625)(.375)}$$

$$= 50,400 \pm 3000 = \begin{matrix} 47,400 \text{ psi TENSION} \\ 53,400 \text{ psi COMP} \end{matrix}$$

SECTION E

$$\sigma_{MAX} = \frac{9.688(21,800)(2.524)}{10.21} + \frac{6500}{.25(4.713) + 2(2.03)(.375)}$$

$$= 52,300 \pm 2500 = \begin{matrix} 49,800 \text{ psi TENS} \\ 54,800 \text{ psi COMP} \end{matrix}$$

SECTION D

$$\sigma_{MAX} = \frac{17.25(21,800)(3.27)}{26.28} + \frac{6500}{.25(5.813) + 2(3.063)(.375)}$$

$$= 48,600 \pm 1700 = \begin{matrix} 46,900 \text{ psi TENS} \\ 50,300 \text{ psi COMP} \end{matrix}$$

PLOTTING BENDING STRESS VS DIST. FROM LOAD SHOWS THE VALUE AT SECTION E TO BE VERY NEARLY THE MAXIMUM

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COLUMN BUCKLING

FIRST APPROXIMATION:
 ASSUME STRAIGHT I BEAM WITH I =
 I_x & S_x & E
 HINGED AT BOTH ENDS
 $A = 2.59 \text{ in}^2$

$$I = \frac{4.315(2.5)^3}{12} + 2 \left(\frac{.38(2.03^3)}{12} \right)$$

$$= .262 \text{ in}^4$$

$$r = \sqrt{I/A} = \sqrt{.262/2.59} = .318 \text{ in}$$

$$L/r = 18.15/.318 = 59.0$$

SHORT COLUMN

BUCKLING IN HORIZONTAL PLANE
 (FROM AXIAL LOAD ONLY)

$$\frac{P}{A} = S_y - 1.172 \left(\frac{L}{r} \right)^2 \quad \text{NEED } S_y \approx 115000 \text{ psi FOR BENDING}$$

$$P = A \left[S_y - 1.172 \left(\frac{L}{r} \right)^2 \right]$$

$$= 2.59 \left[115,000 - 1.172(59)^2 \right] = 287,000 \text{ lb}$$

BUCKLING IN VERTICAL PLANE
 WON'T BUCKLE UNTIL BENDING
 & DIRECT COMPRESSION STRESS EXCEEDS
 YIELD STRESS

$$\sigma_B + \sigma_c = \frac{9,688(12000)(2524)}{10,21} + \frac{20000}{2.59} \quad 141$$

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$$\sigma_B + \sigma_C = 28,700 \text{ psi} + 7,730 \text{ psi}$$

② $\sigma_y = 115,000 \text{ psi}$. Arm won't
Buckle

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EYE LOADS & STRESS - ARM

⊙ MAX. TORQUE IN TORSION BAR

$$R_1 = R_2 = 409,000 / 6.50 = 63,000 \#$$

FROM H. MUELLER'S WORK (PCF)

$$\sigma = K \frac{P}{(d_o - d_i)h}$$

USING K FOR .002 d_i clearance
WITH $d_i = 1.5$
 $d_o = 3.375$
 $K = 3.7$

$$\sigma = \frac{3.7 (63,000)}{(3.375 - 1.5)(1.0)} = 124,200 \text{ PSI}$$

FOR $d_i = 1.875$ (TIGHT FIT BUSHING)

$$K = 3.0$$

$$\sigma = \frac{3.0 (63,000)}{(3.375 - 1.875)(1)} = 126,000 \text{ PSI}$$

1.125 THICK REDUCES TO
112,000 PSI

CAN MAKE THICKNESS 1.25 MAX

$$R_1, R_2 = 409,000 / 6.75 = 60,700 \#$$

$$\sigma = \frac{3.0 (60,700)}{(3.375 - 1.875)(1.25)} = 97,000 \text{ PSI}$$

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USING 4130 Q&T RHC 36-41
 $\sigma_y = 137,500 \text{ psi MIN FOR } 1'' \text{ THICKNESS}$
 $110,000 \text{ psi } " " 2'' "$
 YIELD STRESS SHOULD BE 133,000 psi

$$FS = \frac{133,000}{97,000} = 1.37$$

THIS IS ADEQUATE SINCE THE TORSION BAR TORQUE IS LIMITED BY THE BUMP STOP

EARS ON TORSION BAR FITTING

IT IS ADVANTAGEOUS TO KEEP THE CONNECTING PIN FROM HAVING BENDING APPLIED. TO DO THIS THE PIN MUST HAVE A TIGHT FIT AT THE CENTER OF THE FITTING. THEREFORE THE PAIR OF EARS MUST BE ONE SOLID BLOCK AND OBVIOUSLY THERE WILL BE NO PROBLEM WITH THE EYE STRESS.

PIN SHEAR

$$\text{SHEAR LOAD} = 409,000 \text{ LB-IN} / 5.50$$

$$= 74,500 \# \text{ (SINGLE SHEAR ON EACH END)}$$

$$\tau = P/A = 74,500 / \pi/4 (1.5) = 63,200 \text{ PSI}$$

NEEDS $63,200 / .67 \times 2 \text{ (F.S.)} = 189,000 \text{ psi}$
 (TENSILE YIELD)

USE S160 Q&T RHC 52-57 $\sigma_{y \leq 2} = 205,000 \text{ PSI}$
 (1.5 ROUND)

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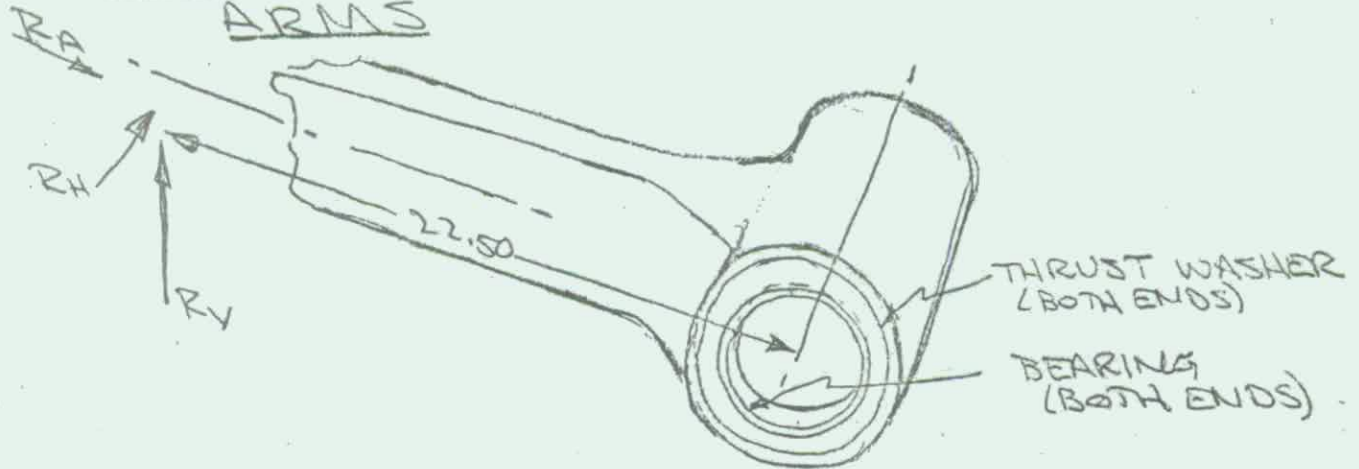
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BEARINGS - DRIVE AXLE SUSPENSION ARMS



MIN BEARING SPREAD = 2.38 IN @ $\frac{1}{2}$ s
THRUST WASHER MEAN RADIUS = 2.62

<u>MAX LOADS</u>	<u>PANIC STOP</u>	<u>MAX AXLE TIP</u>
RH	0	1170#
RV	11,965#	18,200#
RA	19,975#	6,500#

RESULTANT
RADIAL LOAD (R_R) 20,800# 19,300#

WHEN A HORIZONTAL LOAD EXISTS
ASSUME 75% OF MOMENT FROM IT
IS REACTED BY BEARINGS

FROM PANIC STOP
 $R_R = 20,800\#$ (ON BEARINGS)

$$PSI = \frac{20800}{(4.25 + 4.50)(.75)} = 2850 \text{ PSI}$$

(ON PROJECTED AREA)

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FROM MAX AXLE TIP

$$R_R \text{ (HALF ON EACH BRG)} = 19,300$$

$$R_M = \frac{R_H (22.50)}{2.38} = \frac{1170(22.5)}{2.38} = 11,070 \#$$

(USE 75%)

$$\begin{aligned} \text{TOTAL RADIAL LOAD} &= R_R/2 + R_M(.75) \\ &= 9650 + 11,070(.75) = 17,940 \# \end{aligned}$$

$$\text{PSI} = \frac{17940}{(4.250)(.75)} = 5510 \text{ PSI} \leftarrow$$

(SMALLEST BRG)

THRUST WASHERS

FROM MAX AXLE TIP

$$R_H = 1170 \#$$

$$R_M = \frac{R_H(22.5)}{2(2.62)} = \frac{1170(22.5)}{2(2.62)} = 5030 \# \text{ (USE ONLY 25\%)}$$

ASSUMING R_M IS APPLIED TO ONLY .25 OF AREA

$$\begin{aligned} \text{PSI} &= \frac{R_H}{\frac{(5.50+4.68)}{2}(.41)} + \frac{.25R_M/2}{(.25 \frac{(5.50+4.68)}{2})} \\ &= \frac{1170}{\frac{5.50+4.68}{2}(.41)} + \frac{.25(5030/2)}{.25 \left(\frac{5.50+4.68}{2} \right)} = 6094 \text{ PSI} \leftarrow \end{aligned}$$

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GARLOCK DU BUSHINGS HAVE ULTIMATE LOAD CAPACITY OF 20,000 PSI WITH LITTLE OR VERY SLOW MOVEMENT

THE PRESSURES CALCULATED ARE FOR WORST LOAD CONDITIONS - NORMAL OR NOMINAL STATIC GVW BEARING PRESSURES WOULD BE ON THE ORDER OF 1/4 OF THOSE CALCULATED

IF MAX WHEEL TRAVEL IS ASSUMED TO OCCUR IN .33 SEC THE SLIDING SPEED IN THE BUSHINGS IS

$$\frac{\frac{31^{\circ}45' / 2}{360^{\circ}}}{.33/60} \times \frac{\pi(4.50)}{122} = 340 \text{ FPM}$$

THIS WOULD OCCUR AT AN ^{APPROX} AVERAGE PSI OF

$$\frac{\text{MAX AXLE TIP } 5510 + \text{NOMINAL } 5510 + \text{MAX TRAVEL NO TIP } 5510}{3 \times 4 \times 2} = 3210 \text{ PSI}$$

THIS WOULD PRODUCE A VERY HIGH PV (109,000) BUT THIS DOESNT OCCUR CONTINUOUSLY, ONLY 1 CYCLE AT A TIME SO PV FACTOR HAS LITTLE MEANING

THE UNIT LOAD (PSI) IS A BETTER MEASURE AND THESE VALUES ARE ACCEPTABLE

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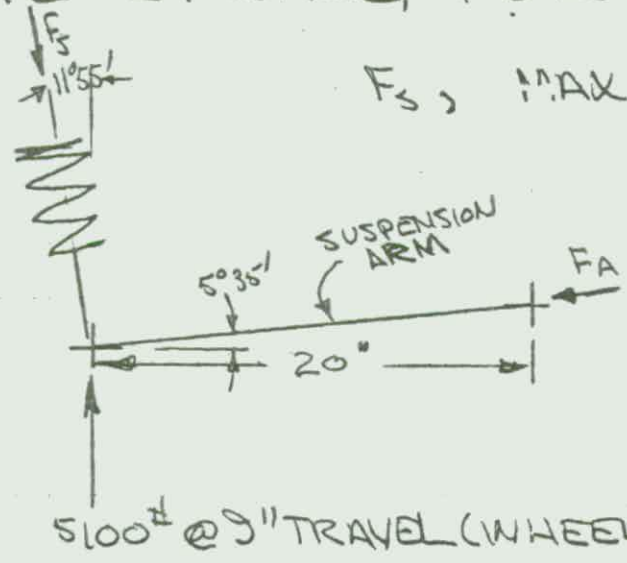
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COIL SPRING FOR STEER AXLE



$$F_s, \text{ MAX SPRING FORCE} = 5100 / \cos 11^\circ 55'$$

$$= 5210 \#$$

$$F_A = \frac{F_s \sin 11^\circ 55'}{\cos 5^\circ 35'}$$

$$= \frac{5210 (.2065)}{.9953}$$

$$= 1080 \#$$

5100# @ 9" TRAVEL (WHEEL)

SPRING RATE = $5210 / 9 = 579 \text{ LB/IN}$

AT STATIC MAX LOAD STRESS CAN GO TO 125,000 PSI (UNCORRECTED) USING ABOUT .8 DIA WIRE, HOT COILED, PRESET ALLOY STEEL (SAE HELICAL SPRING HANDBK) SPRING COULD BE SHOT PEENED FOR FATIGUE LIFE IMPROVEMENT

FOR 13/16 (.813) DIA ROD

$$S_s = \frac{8PD}{\pi d^3} \quad \text{where } D = (15.813 - .813)$$

$$S_s = \frac{8(5210)(5.00)}{\pi (.813)^3} = \frac{123,200 \text{ PSI}}{\text{OK}}$$

$$N(\text{ACTIVE}) = \frac{Gd^4}{8P/kD^3} = \frac{11.4 \times 10^6 (.813^4)}{8(579)(5.00^3)} = 8.57$$

SOLID HEIGHT = $d(N_A + 2 - k/2) = .813(8.5 + 2 - .5)$
 $= 8.13 \text{ \"}$

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$$C = D/d = 5/0.813 = 6.15$$

$$K \text{ (WAHL FACTOR)} = 1.246$$

$$h/d = 10, Z \text{ (ECC. FACTOR)} = 1.054$$

NOMINAL WHEEL TRAVEL FOR 10^5 CYCLES
 = ± 1.5 IN

STRESS RANGE

$$S_s = \frac{8 P D}{\pi d^3} K Z \quad P = 3(579) = 1737 \#$$

$$= \frac{8(1737)(5)}{\pi(0.813^3)} (1.246)(1.054) = \underline{41,100 \text{ psi}}$$

THIS LOW RANGE STRESS SHOULD NOT
 REQUIRE SHOT PEENING FOR
 ADEQUATE FATIGUE LIFE
NO DECARB ON SURFACE ALLOWABLE
 TO MAINTAIN ALLOY ENDURANCE
 LIMIT

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STEER AXLE SUSPENSION ARM

THE SUSPENSION GEOMETRY SHOWS THAT IF ALL LINKS WERE RIGID THE ARM WOULD HAVE TO UNDERGO A MAX 2° TWIST AND A MAX 1.00 HORIZONTAL DEFLECTION WHEN THE AXLE IS AT FULL TIP. IF THE VEHICLE WERE BACKED INTO SOMETHING AND THE WHEEL HIT IT A HIGH COLUMN LOAD WOULD ALSO BE APPLIED.

THEREFORE THE TWIST & DEFLECTION MUST BE ABSORBED MOSTLY IN RUBBER MOUNTINGS, SINCE THESE CAN ALSO STORE ENERGY AND PROVIDE ROLL RESISTANCE.

THE WORST LOADING WILL BE
A COLUMN LOAD OF 4800 LB ←
(2G X NOMINAL STATIC WHEEL LOAD)
VERTICAL LOAD OF 1345 LB ←
(REACTION OF PANIC STOP TORQUE)
COMBINED WITH 3° TWIST ←
.25 IN HORIZ DEFLECTION ←
(ASSUMING RUBBER ABSORBS 75% OF DEFLECTION
AND TWIST)

Uniform Thickness = .5"

1/2

Torsion stress $\phi = 3^\circ = .05236 \text{ rad}$, $L = 16$, $G = 10,000,000$, $P = 4800$

$$b/h = b_{av}/h = 2.72/.5 = 5.44, \quad \beta = .294$$

$$T = \frac{\phi G \beta b h^3}{L} = \frac{.05236 \times 10,000,000 \times .294 \times 2.72 \times .5^3}{16} = 3271$$

T_{max}

at $x=0$ $b/h = 3.4/.5 = 6.8$, $\alpha = .3016$

$$\tau_T = \frac{T}{\alpha b h^2} = \frac{3271}{.3016 \times 3.4 \times .5^2} = 12,759$$

$x=16$ $b/h = 2.45/.5 = 4.9$, $\alpha = .290$

$$\tau_T = \frac{3271}{.290 \times 2.45 \times .5^2} = 18,415$$

Column buckling

$r_c = .1443$, $P = 4800$, $A_{av} = 1.358$, $e = .25$

Using secant formula

$$f_a = \frac{P}{A_{av}} = \frac{4800}{1.358} = 3535$$

$$f_y' = 220,000 - 60,490 = 159,510$$

f.s. = 4.30508

Check - $N = f.s.$

$$f_a = \frac{f_y}{N \left[1 + \frac{ec}{r_c^2} \sec \left(\frac{L}{2L} \sqrt{\frac{N f_a}{E}} \right) \right]} = \frac{159,510}{4.30508 \left[1 + \frac{.25 \times .25}{.1443^2} \sec \left(\frac{16}{2 \times .1443} \sqrt{\frac{4.30508 \times 4800}{30,000,000}} \right) \right]}$$

$3535 = 3535$

Assume simple column

$$K = .9$$

$$\frac{KL}{r} = \frac{.9 \times 16}{.1443} = 99.8$$

$$f'_y = 159,510$$

Failure inelastic if $\frac{KL}{r} \leq \sqrt{\frac{\pi^2 E}{f'_y}} = \sqrt{\frac{\pi^2 \times 30,000,000}{159,510}} = 43.1$

Since $\frac{KL}{r} > 43.1$ Failure occurs elastically and is given by Euler formula

$$f_c = \frac{\pi^2 E}{(KL/r)^2} = \frac{\pi^2 \times 30,000,000}{99.8^2} = 29,728$$

$$P_c = A_{gr} f_c = 1.358 \times 29,728 = 40,371$$

$$f.s. = \frac{P_c}{P} = \frac{40,371}{4800} = 8.4$$

GHSBI 13:17 02/01/74 FRIDAY 104

1 DATA 30000000,10000000,16,.25,.05236,1345,1,6,7

11 DATA 2.4,.5,2.667

12 DATA 2.3,.5,2.667

13 DATA 2.6,.5,2.667

14 DATA 2.55,.5,2.667

15 DATA 2.5,.5,2.667

16 DATA 2.45,.5,2.667

20 DATA 0,2.667,5.333,0,10.667,13.334,16

100 DIM F

BREAK
RUN

GHSBI 13:20 02/01/74 FRIDAY 104

Sequence	H	t	A	I ₁	Z ₁	$v_1 \sqrt{I_1/4}$	I ₂	Z ₂	$v_2 \sqrt{I_2/4}$
1	2.4000	0.5000	0.0000	0.0000	0.0000	0.0000	1.7000	1.6377	0.9633
2	2.6000	0.5000	0.0000	0.0000	0.0000	0.0000	1.4000	0.9147	0.6533
3	2.6000	0.5000	0.0000	0.0000	0.0000	0.0000	1.3000	0.7323	0.5633
4	2.5500	0.5000	0.0000	0.0000	0.0000	0.0000	1.2750	0.6909	0.5419
5	2.5000	0.5000	0.0000	0.0000	0.0000	0.0000	1.2500	0.6510	0.5200
6	2.4500	0.5000	0.0000	0.0000	0.0000	0.0000	1.2250	0.6120	0.5002

H = 2.12

A = 1.358

I₂ = 0.1283

1	5.9514E-06	3.2472E-06	4.4629E-05	3.3467E-05	5.0590E-05	2.6915E-05	3.3472E-06	2.5101E-06
2	7.2267E-06	4.0645E-06	4.3252E-05	3.2510E-05	1.9934E-04	7.3289E-05	1.4106E-05	5.5591E-06
3	7.7826E-06	4.3772E-06	3.5013E-05	2.6256E-05	4.2787E-04	1.0402E-04	3.2307E-05	2.8406E-06
4	7.2352E-06	4.4630E-06	2.3707E-05	1.7845E-05	7.4703E-04	1.2633E-04	6.1348E-05	1.2197E-05
5	8.0939E-06	4.5523E-06	1.2132E-05	9.0977E-06	1.1042E-03	1.3999E-04	9.2404E-05	1.5601E-05
6	8.2591E-06	4.6452E-06	*****	*****	1.4858E-03	1.4462E-04	1.4466E-04	1.9085E-05

F₁=1345. F₂=642. M=4863.

T = 3271

Point	X	G _F	T _{F₁}	G _{F₂}	T _{F₂}	$\frac{G_{F_1} + G_{F_2}}{T_{F_1} + T_{F_2}}$	T _{max} slope	D _{max}
1	0.	22339.	791.	30161.	277.	12,2540.	60400.	60400.
2	3.	27448.	961.	31657.	459.	0.	59105.	59105.
3	5.	25469.	1035.	19299.	404.	0.	43768.	43768.
4	8.	19957.	1055.	2550.	503.	0.	22407.	22407.
5	11.	13772.	1076.	13929.	513.	0.	27601.	27601.
6	13.	7169.	1098.	30877.	524.	0.	38046.	38046.
7	16.	0.	1098.	47636.	524.	18,4150.	47636.	47636.

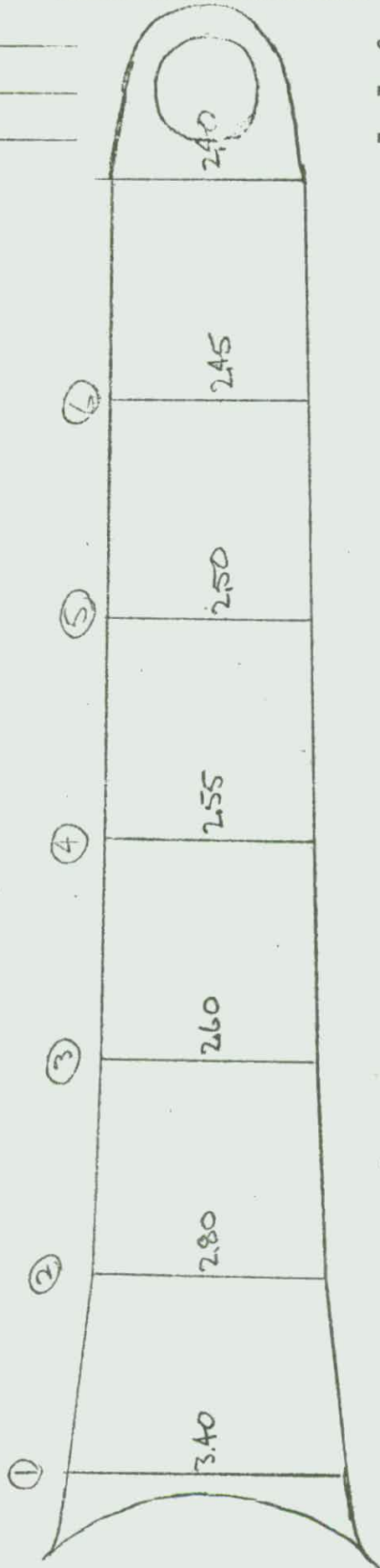
PROCESSING 2 UNITS

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY AK
CHECKED BY _____
DATE 1-31-74

ORDER NO. _____
PAGE _____ OF _____
REPORT NO. _____



ALL THICKNESSES .150

PACIFIC CAR AND FOUNDRY COMPANY

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CHECKED BY _____

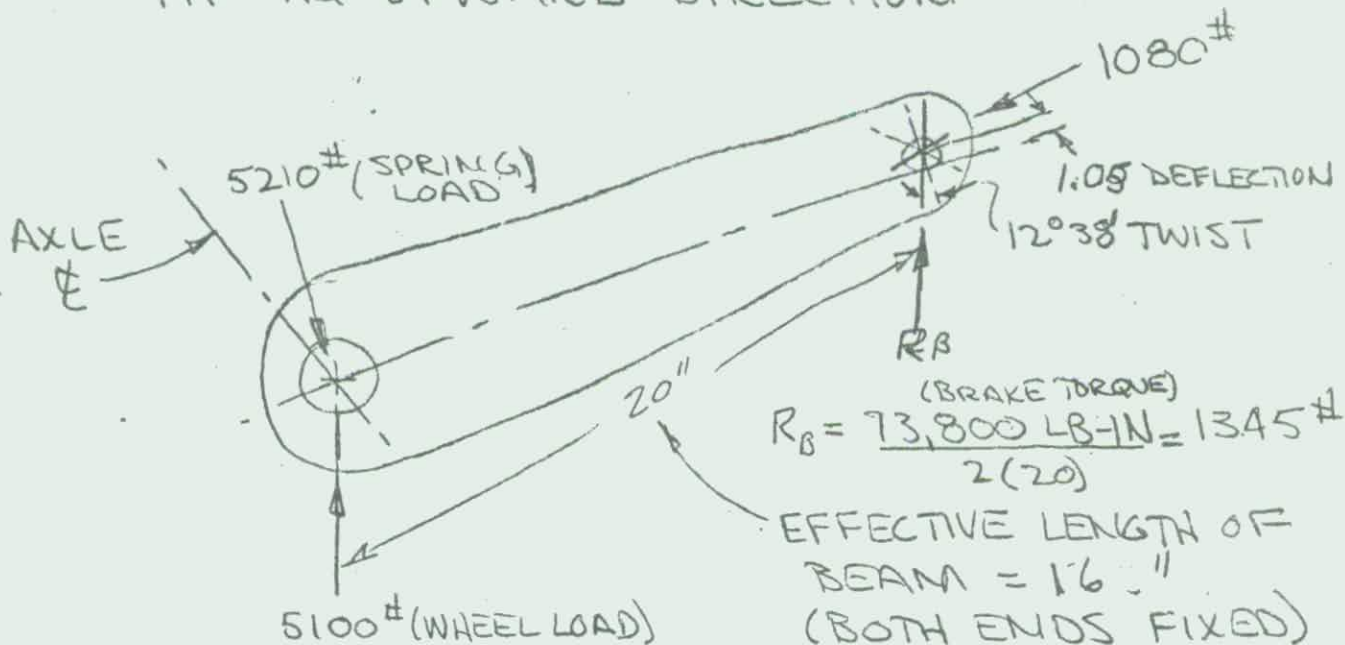
PAGE _____ OF _____

DATE 11/29/73

REPORT NO. _____

STEER AXLE SUSPENSION ARM LOADING

WORST LOADING IS AT MAX AXLE TIP IN UPWARD DIRECTION



IF MAX BRAKING ALSO OCCURRED AT MAX AXLE TIP THE 1080# WOULD INCREASE TO 2400#
 R_B OCCURS AT THIS TIME ALSO

RIVET JOINT W/AXLE

USING MS 20613 .4375 DIA RIVETS
 MIN SHEAR STRENGTH (YIELD) = 25000 PSI
 TORQUE PER JOINT (FROM PANIC STOP)
 = $73,800 / 2 = 36,900 \text{ LB-IN}$
 WITH 5 RIVETS & RIVETS IN DOUBLE
 SHEAR ON 2.313 R

$$FS = \frac{S_{sy}}{P/A} = \frac{25000}{\frac{36900/2.313}{5(2)(\pi/4(.4375)^2)}} = 2.32$$

PACIFIC CAR AND FOUNDRY COMPANY

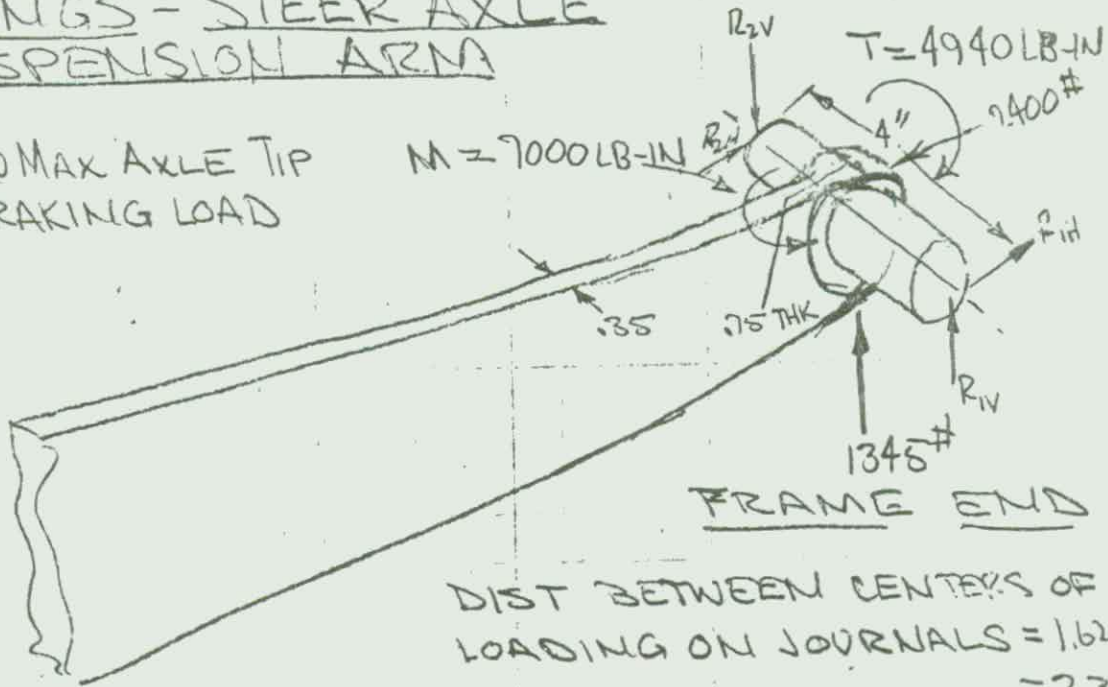
ENGINEERING DEPARTMENT

PREPARED BY WJK
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 DATE 12/12/73

ORDER NO. _____
 PAGE _____ OF _____
 REPORT NO. _____

BEARINGS - STEER AXLE SUSPENSION ARM

LOADING @ MAX AXLE TIP
 W/MAX BRAKING LOAD



$$R_{1V} = R_{2V} = 4940 / 2.37 = 2080 \#$$

$$R_{1H} = 7000 / 2.37 - 2400 = 460 \#$$

$$R_{2H} = 7000 / 2.37 + 2400 = 5350 \#$$

$$R_R \text{ MAX} = \sqrt{R_{1V}^2 + R_{2H \text{ MAX}}^2}$$

$$= \sqrt{2080^2 + 5350^2} = 5740 \#$$

FOR THE SAME UNIT PRESSURE AS THE
 DRIVE AXLE ARM BUSHINGS - 5500 PSI

$$\text{JOURNAL DIA} = \frac{R}{l \text{ PSI}} = \frac{(5740)}{1.62(5500)} = .645$$

USING $d = 1.5$ & $l = .75$

$$\text{PSI} = \frac{5740}{1.5(.75)} = 5100 \text{ PSI} \quad \text{THIS IS COMPATABLE W/DRIVE AXLE}$$

PACIFIC CAR AND FOUNDRY COMPANY

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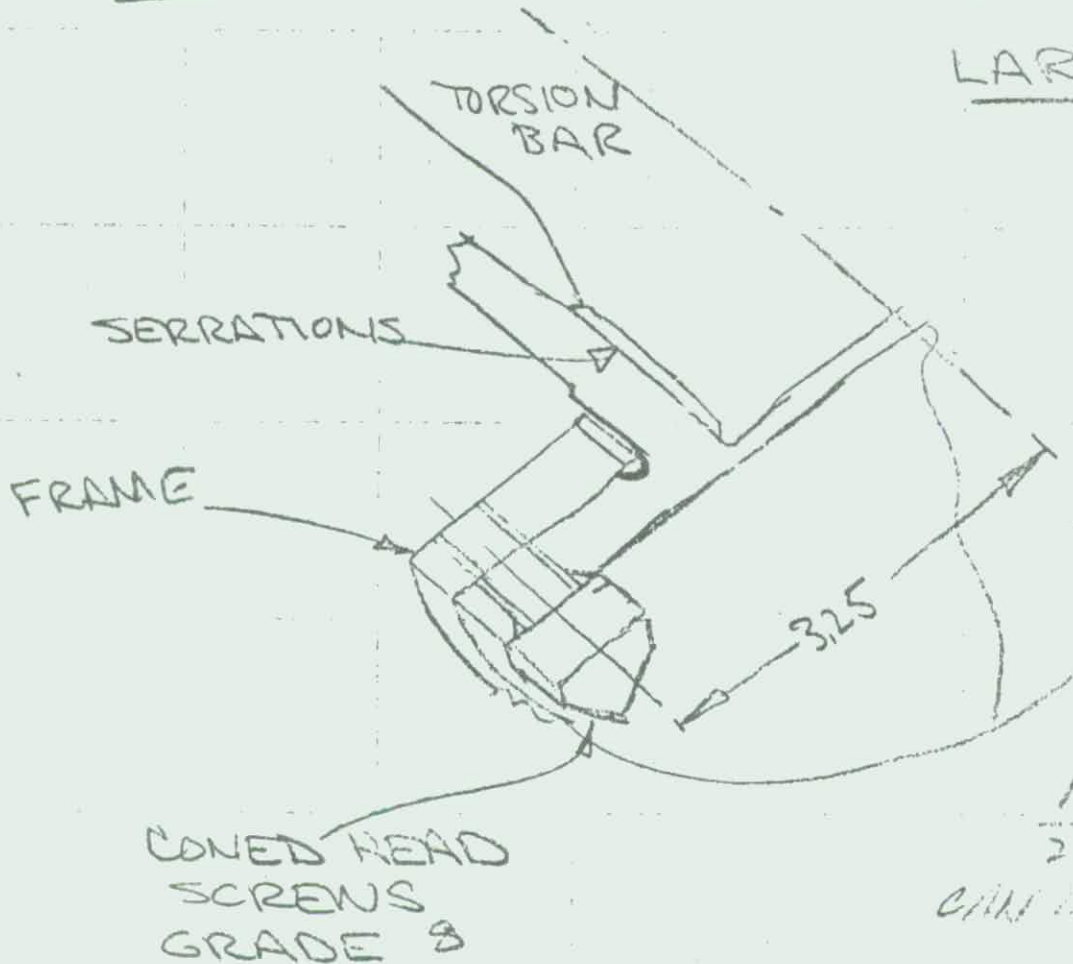
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TORSION BAR ANCHOR



LARGE BAR

7/8-14 (5)
 CUMULATIVE =
 5(45,000) =
 225,000 #
 CONCENTRIC
 W/ 3.25 IN

Head of 1/4"

$$\frac{125,800}{225,000} = 1.55$$

CAN DO W/ 5/8 IN

MAX TORQUE ON BAR = 409,000 LB-IN

SHEAR FORCE TO BE SUPPLIED
 BY SCREENS = $409,000 / 3.25$
 = 125,800 #

FOR GRADE 8 SCREENS, FS=2 ON YIELD
 10 SCREENS

AREA REQD = $\frac{125,800}{120,000} \times 2 = .21 \text{ IN}^2$

5/8-18 IS .2560 IN²

PREPARED BY

HK

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OF

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PANHARD ROD - DRIVE AXLEROD TRANSFERS SIDE LOADS ON
AXLE TO FRAME

$$\begin{aligned} \text{DESIGN LOAD} &= 3 (\text{NOM STATIC LOAD}) \\ &\quad \text{WHEEL} \\ &\quad \text{W/HOWITZER} \\ &= 3(8250) = 24750 \# \\ &\quad (\text{CAN BE TENSION OR COMPRESSION}) \end{aligned}$$

COLUMN LOADING

$$\text{LENGTH} = 49.18''$$

FOR A 2 OD X .125 WALL TUBE

$$L/r = 49.18 / .6644 = 74.2$$

FOR TUBULAR COLUMN - NO END RESTRAINT
(ROARK - p. 237)

$$\frac{Q}{A} \text{ ALLOWABLE} = \frac{P}{1.5 A}$$

 $(S_y = 60,000 \text{ psi})$
 $(4120, 8630)$
 (NORMALIZED)

$$P/A = 60000 - 1.172 \left(\frac{L^2}{r^2} \right)$$

$$= 60000 - 1.172 (74.2)^2$$

$$= 60000 - 5490 = 54,510 \#/\text{IN}^2$$

$$A_{\text{REQD}} = \frac{Q(1.5)}{P/A} = \frac{24750(1.5)}{54,510} = .68 \text{ IN}^2$$

$$A \text{ OF TUBE} = .736 \text{ IN}^2$$

OK

IN TENSION (FOR FACTOR OF SAFETY = 2)

$$\sigma = P/A = \frac{24750(2)}{.736} = 67,200 \text{ PSI}$$

HEAT TREAT RHC 31-36

$$\sigma_{\text{yield}} = 95-110 \text{ KSI FOR } 8630$$

4120 HIGHER

PACIFIC CAR AND FOUNDRY COMPANY

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PANHARD ROD - STEER AXLE

LENGTH = 23.4

LOAD = 3 x WHEEL LOAD = 3 (5100#)
= 15,300#

TRY 1.50 OD x .093 WALL (4130, 8630)
r = .498 l/r = 23.4 / .498 = 47

$$\frac{P}{A} \text{ ALLOWABLE} = \frac{P}{1.5A}$$

$$P/A = S_y - 1.172 \left(\frac{l}{r}\right)^2$$

$$= 60,000 - 1.172(47)^2 = 60,000 - 2590$$

$$= 57,410 \text{ psi}$$

$$A_{REQ'D} = \frac{Q(1.5)}{P/A} = \frac{15,300(1.5)}{57,410} = .400 \text{ in}^2$$

1.50 x .093 TUBE A = .4142 OK

TENSILE LOAD (FOR FS=2)

$$\sigma = \frac{2P}{A} = \frac{2(15,300)}{.414} = 73,900 \text{ psi}$$

HEAT TREAT RbC 31-38

PACIFIC CAR AND FOUNDRY COMPANY

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STEERING SYSTEM - STEER AXLE

1. Kingpin Torque

$$T = Wf \sqrt{\frac{B^2}{B} + E^2}$$

$$W = \text{Vehicle weight}^* = 5100 \text{ lb}$$

$$B = \text{Nom. tire width} = 12.00 \text{ in}$$

$$E = \text{Kingpin eccentricity} = 10.30 \text{ in}$$

$$f = \text{Coeff. of friction based on } E/B (= .86)$$

$$= .12$$

$$T = (5100)(.12) \sqrt{\frac{(12.00)^2}{B} + (10.30)^2}$$

$$= \underline{\underline{6817.4 \text{ in-lb}}}$$

* on rear axle

2. Required Force

$$F = \frac{T}{r}$$

$$T = \text{Kingpin Torque} = 6817.4 \text{ in-lb}$$

$$r = \text{Effective Radius Arm} = 4.60 \text{ in}$$

$$F = \frac{6817.4}{4.60} = 1482 \text{ lb}$$

PACIFIC CAR AND FOUNDRY COMPANY

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$$P_{max} = 2200 \text{ psi}$$

$$A_{req'd} = \frac{F}{P} = \frac{1482}{2,200} = .674 \text{ in}^2$$

$$A = A_{rod\ end} + A_{piston\ face}$$

$$= \pi [R^2 - r^2] + \pi [R^2]$$

$$= \pi [2R^2 - r^2]$$

Try a 1.50 bore with a .75 in rod

$$A = \pi [2(.75)^2 - (.375)^2] = 3.09 \text{ in}^2$$

Req'd pressure to create a 1482 lb force

$$P = \frac{1482}{3.09} = 479 \text{ psi}$$

Set r at 3.40 inches (a 6.00 radius of action)

$$F = \frac{6817.4}{3.40} = 2,050 \text{ lb}$$

$$A = \frac{2,050}{2,200} = .913 \text{ in}^2$$

A of cyl is 3.09 in² - effective

$$P_{used} = \frac{2,050}{3.09} = 650 \text{ psi}$$

PACIFIC CAR AND FOUNDRY COMPANY

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The selected cylinder is a 1.50 ID x 6.00 stroke, 2 used per axle

$$A = \frac{\pi (D)^2}{4} = \frac{\pi (1.50)^2}{4} = 1.767 \text{ in}^2$$

$$V = AS = (1.767)(6.00) = 10.60 \text{ in}^3$$

steering wheel turns

$$\text{Orbital Displacement} = \frac{2V}{N}$$

Set N at 4.0 turns, Lock to Lock

$$\text{Orbital Displacement} = \frac{2(10.60)}{4} = 21.20 \frac{\text{in}^3}{\text{rev.}}$$

The cylinder selected for the rotary actuator (Loaded condition) is the YU-12 that has a displacement of 47.16 in³/rev.

$$N = \frac{21.2 \text{ in}^3}{47.16 \text{ in}^3/\text{rev.}} = .45$$

As this is much quicker and needs less power than the rotary actuator, the loaded condition must take precedence in sizing calculations.

PACIFIC CAR AND FOUNDRY COMPANY

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STEER AXLE - TIE ROD

TIE ROD MAX LOAD WILL OCCUR WHEN AXLE IS TIPPED AND ALL AXLE LOAD IS ON ONE WHEEL

THEN ONE CYLINDER WILL BE TRANSMITTING ITS FORCE THROUGH THE TIE ROD TO THE OTHER WHEEL

WORST CASE WILL BE IF THE CYLINDER FORCE RESULTS FROM THE PISTON AREA SO THE TIE ROD FORCE WILL BE

$$F_T = \frac{M_T (@ 5100 \# \text{ AXLE LOAD})}{2 R_{MIN}} \frac{A_P}{A_R}$$

M_T = TURNING MOMENT
 R_{MIN} = MIN TIE ROD MOMENT ARM
 A_P = AREA - PISTON END OF CYL
 A_R = AREA - ROD END OF CYL

$$F_T = \frac{1620}{2(3.40)} \times \frac{.913}{.605} = \underline{360 \#}$$

PUSH OR PULL

FOR COLUMN LOADING

$$Q/A \text{ ALLOWABLE} = \frac{P/A}{1.5} \quad \text{FOR TUBULAR, PIN END COLUMN}$$

$$\frac{P}{A} = 36000 - 1.172 \left(\frac{L}{r} \right)^2 \quad (\text{ROARK - p 237})$$

$$L = 52.88 \text{ IN}$$

$$r = .3217, \text{ FOR } 100 \times .095 \text{ WALL}$$

$$\frac{P}{A} = 36000 - 1.172 \left(\frac{52.88}{.3217} \right)^2 = 4300 \text{ psl}$$

PACIFIC CAR AND FOUNDRY COMPANY

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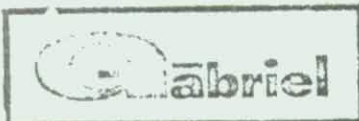
DATE 12/6/73

REPORT NO. _____

$$\begin{aligned} \text{MIN A READ} &= \frac{1.5Q}{P/A} & Q &= 350^\# \\ &= \frac{1.5(350)}{4300} = .122 \text{ IN}^2 \end{aligned}$$

FOR 1.00 OD x .095 WALL TUBE, A = .2701 IN²

OK
W



22371 Newman Avenue, Dearborn, Michigan -- 48124
Phone: Area 313 -- LO. 1-7937 TWX-313-278-9756

December 21, 1973

Mr. H. G. Kirchner
PACIFIC CAR & FOUNDRY
1400 4th St. North
Renton, Washington 98055

Dear Mr. Kirchner:

Supplementing our letter of December 4th we are attaching the following:

I. Rear Shock Data - 1-5/8" Bore

- A. Typical unit outline drawing 680055 which indicates recommended 2.5 dia. mounting ends and method of dimensioning.
- B. Section Drawing 422512 illustrating unit construction and minimum dead length of 4.00 plus mounting ends. Strokes can be specified in .250 increments.
- C. Installation drawing 420212 which shows Type I mounting we recommend for rear.

II. Front Shock Data - 1-3/8" Bore

- A. Typical unit outline drawing 651018 illustrating integral mounting which uses .75 dia. bolt.
- B. Section Drawing 422517 describing unit and minimum dead length 3.952 plus mounting ends. Strokes available in .25 increments.

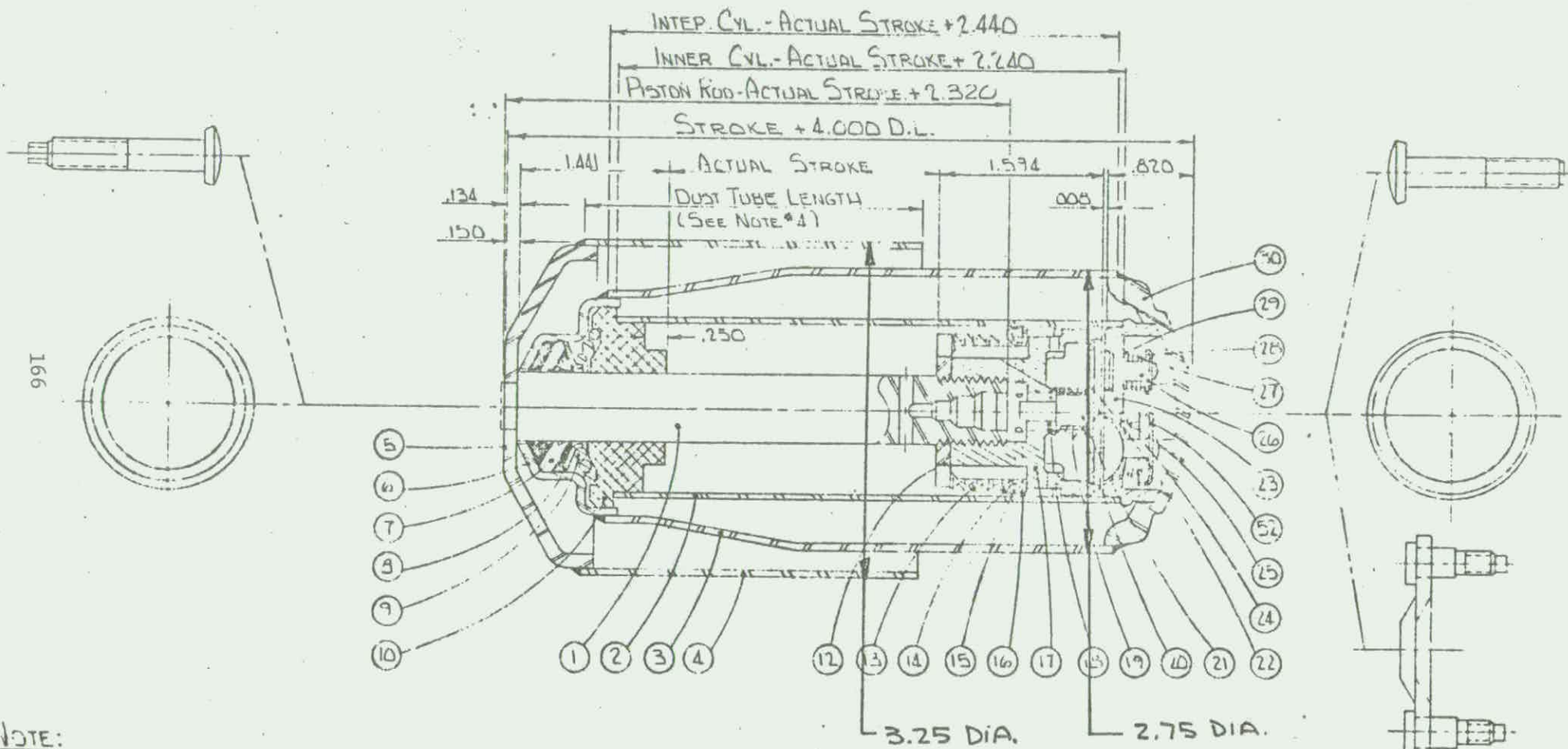
We trust this information will be of aid to you.

Very truly yours,

GABRIEL OF CANADA

Ral F. Homoyec
General Sales Manager

RFH:md



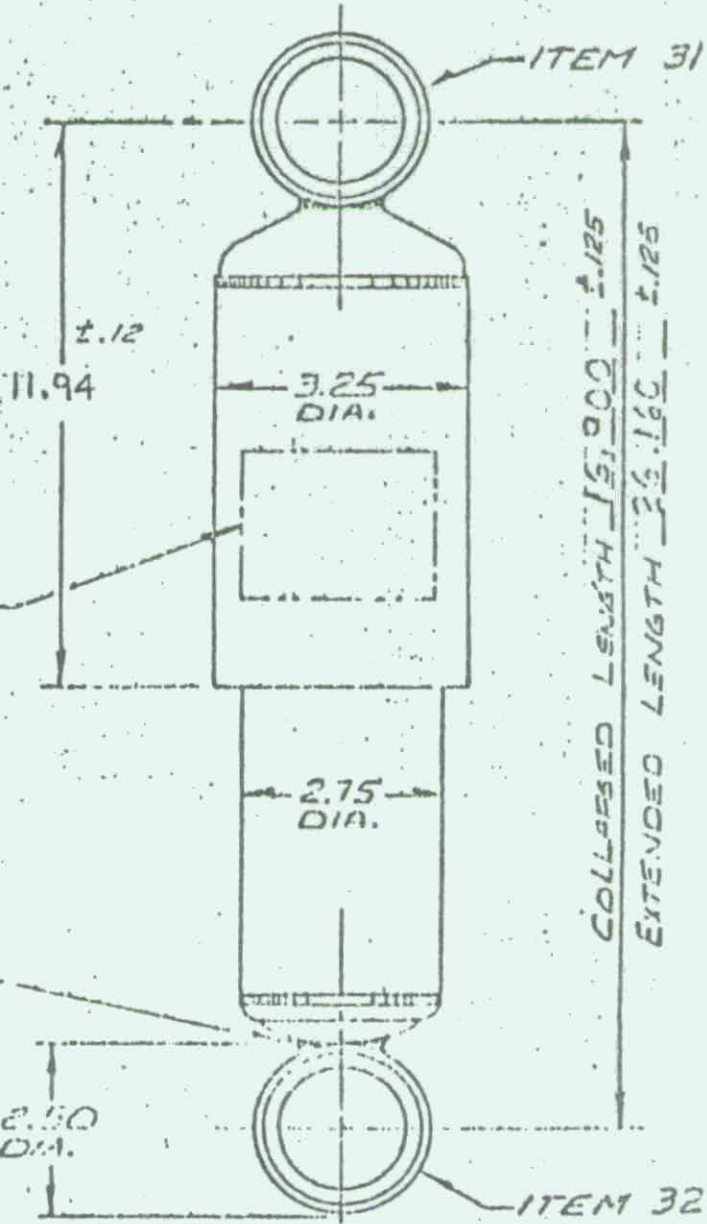
166

- NOTE:**
- * SYMBOL DENOTES AFTER WELDING DIMENSIONS.
 - USE ACTUAL STROKE FOR CALCULATIONS. ACTUAL STROKE IS .010 LONGER THAN NOMINAL STROKE. EXAMPLE: 5.00 NOM. STROKE IS 5.01 ACTUAL STROKE.
 - FLUID QUANTITY (IN CC) 61 TIMES NOMINAL STROKE + 110 CC.
 - DUST TUBE LENGTH 1.650 LESS THAN PISTON ROD LENGTH.

TOL DECIMAL S.		ANGULAR =		WT.	
SCALE	1 To 1	MATERIAL			
DRAWN	W. M. GIBNEY	NAME STOCK REQUISITION			
DATE	JAN 24 1951	SECTION DRAWING			
CHECKED	AJN	1 1/2" HEAVY DUTY			
APPROVED		PARTIAL DRAFT			
DATE	LCY	ALTERATIONS		D.C. No	
D.R. No 55		SUPERSEDES			
		No 422512			
GABRIEL OF CANADA LTD.					
TORONTO, CANADA					

NOTE:-

- 1. VALVE CODE F8
- 2. STROKE 9.26 D.L. 7.64
- 3. SECTION DWG. 422512 (RECOIL CUT-OFF)
- 4. ASSEMBLY SPEC. 430020
- 5. 2 PLACE DIMS. (XX) ARE REF.
- 6. ITEMS 2, 3 & 70 USE 10.51 STROKE



(SEE NOTE 6)

SEE ITEM 4 DWG. FOR STAMPING

N

QTY. REQD

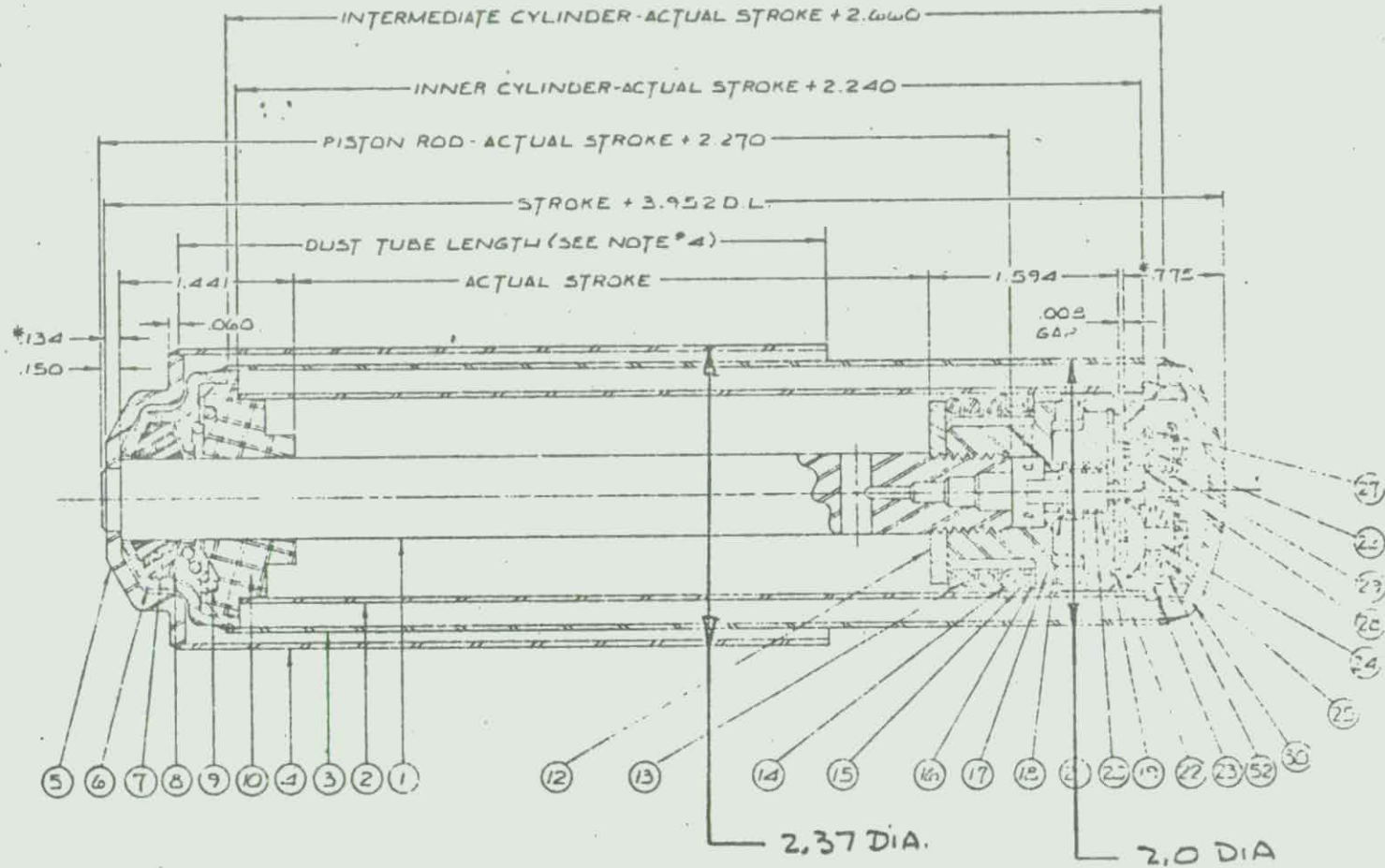
1

75/

AR

HEAVY SHOCK RESISTANT
 (1.625 INCH) HEAVY DUTY
 10.51 STROKE

TX 680051



NOTES:

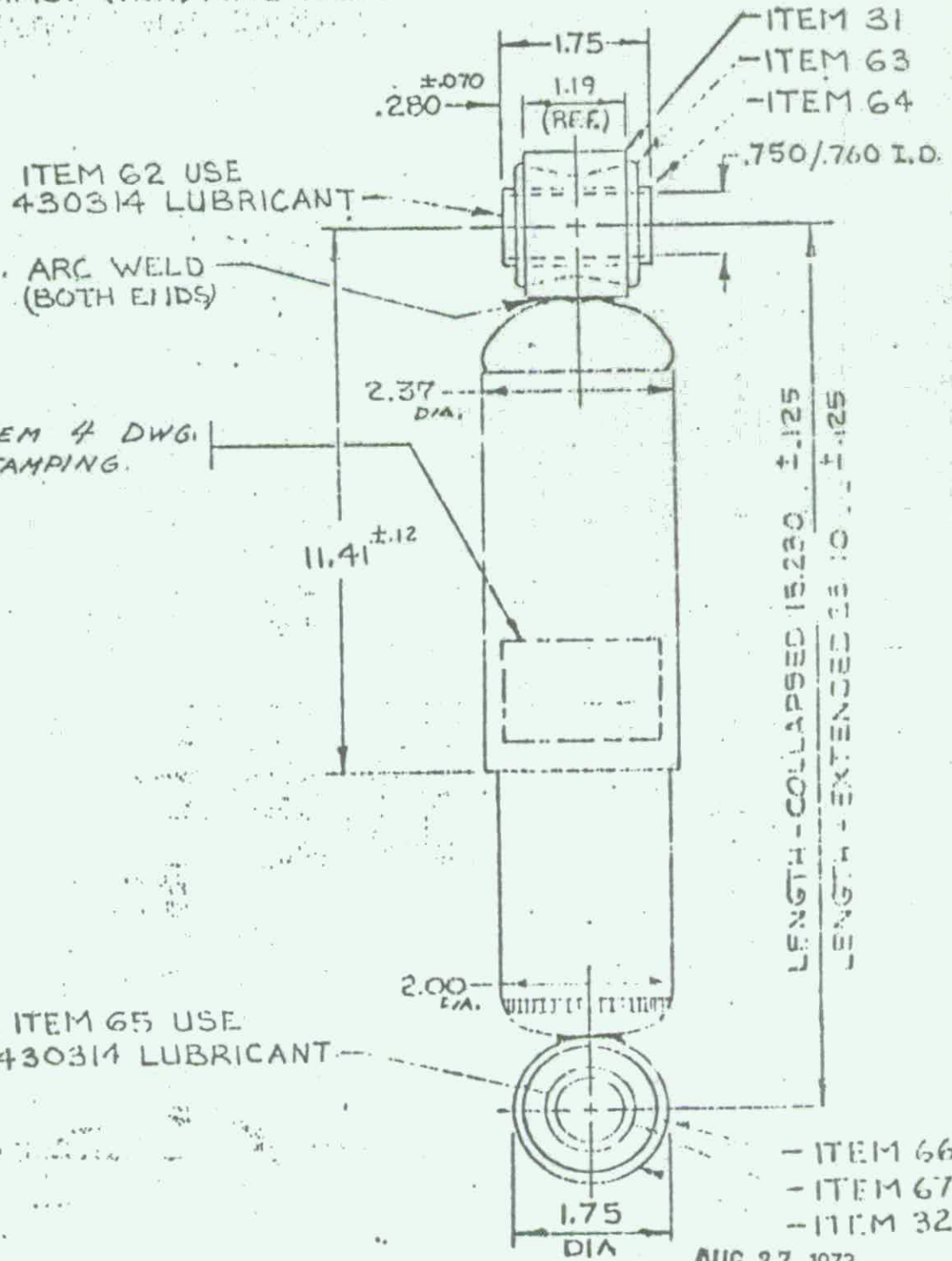
- 1 * SYMCL DENOTES AFTER WELDING DIMENSIONS.
- 2 USE ACTUAL STROKE FOR CALCULATIONS - ACTUAL STROKE IS .010 LONGER THAN NOMINAL STROKE
Example: 5.00 NOMINAL STROKE IS 5.01 ACTUAL STROKE
- 3 FLUID QUANTITY (INC.L.) 40% NOMINAL STROKE + 7% .C
- 4 DUST TUBE LENGTH IS 1.650 LESS THAN PISTON ROD LENGTH.

TOL DECIMAL = ANGULAR = WT.

SCALE	MAT'L		
1/2" = 1"			
DRAWN			
DATE	NAME		
1/12/70	SECTION ENGINEER		
CHECKED	DESIGN OFFICER		
	WELDED ASSEMBLY		
APPROVED			
DATE LET	ALTERATIONS	D.C. No	GABRIEL OF CANADA LTD. TORONTO CANADA
D 11 No 5427	SUPERSEDES		

- 1 DRAWING CODE: 74/
- 2 PART NO: 988 D.I. 5.35
- 3 SECTION DWG.: 422002 (RECOIL CUT-OFF)
- 4 ASSEMBLY SPEC.: 438020
- 5 NOM. OVERALL COLL. LENGTH:
- 6 2 PLACE DIMS. (.XX) ARE REF.

SSY
 SSY
 116
 A.R.

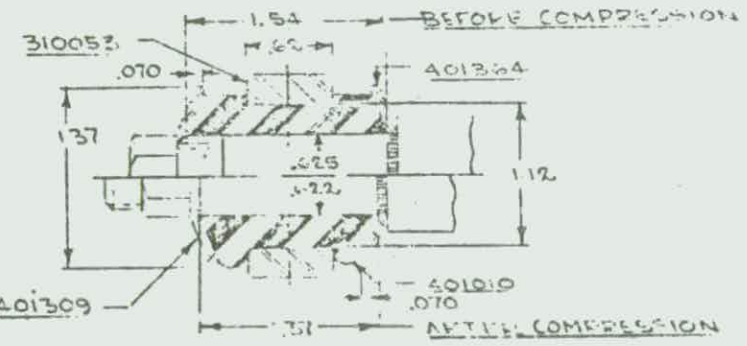
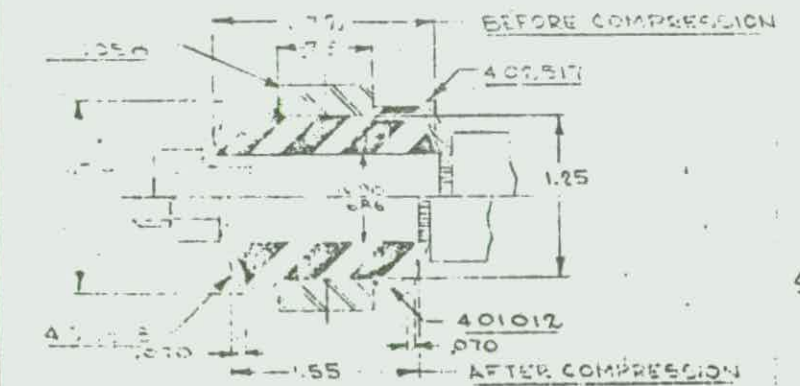
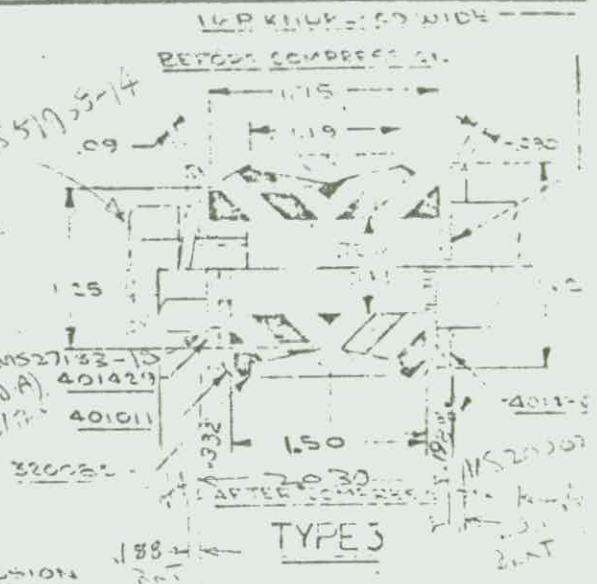
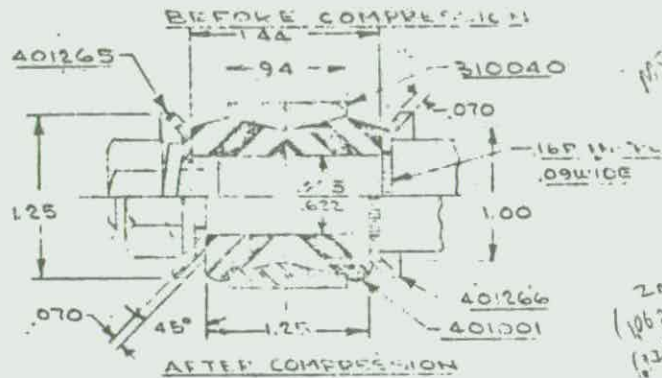
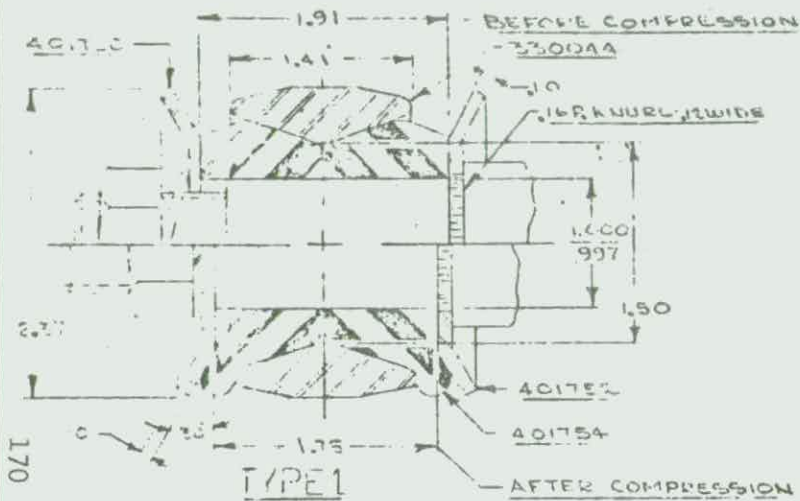


AUG 27 1973

SHOCK ASSEMBLY
 (L&R BOTH)

TX 651018

INMAY 28 1942



TYPE 4

TYPE 5

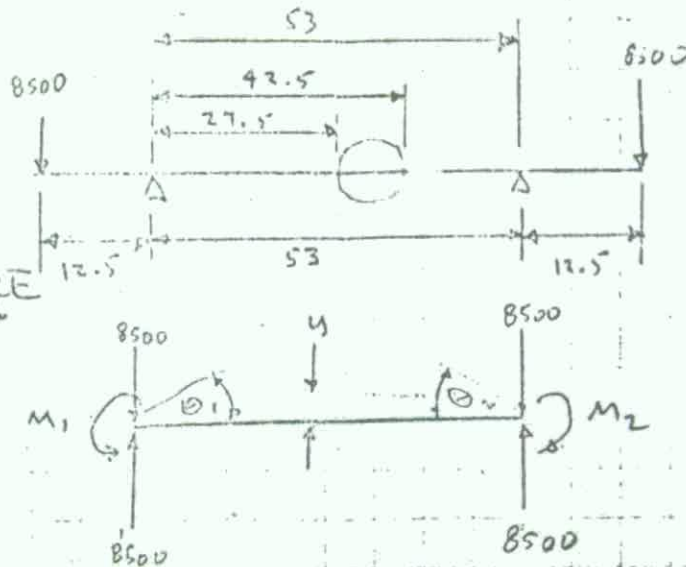
NOTES:
A) INDIVIDUAL DETAIL DWG. FOR OTHER DIMENSIONS
B) TOLERANCE

		TOL. DECIMAL ±	ANGULAR ±	WT.
SCALE:				MAT'L:
DRAWN:				
DATE:	11 23 41			NAME:
CHECKED:				WKT OF SPEC. LITH
APPROVED:				STYLING DEPARTMENT
				No. 425212
DATE	REV	ALTERATIONS	D. C. No.	
D. R. No.		SUPERSEDES - 47		

THE GABRIEL COMPANY
CLEVELAND OHIO

Afflas axle housing 1/24/77

1. STATIC
LOADING
ASSUMING
LOAD ~~RECT~~ SQUARE
TO AXLE
SECTION



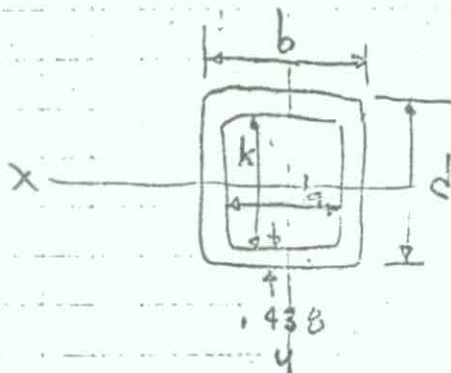
9750 VERT
5300 HOR

$$M_1 = M_2 = 8500 \times 12.5 = 106,250$$

$$M_x = M_1 = M_2 = 106,250$$

$$I = 14.99, \quad E = 7.05$$

x	M	y/I	y^2/I	$\sigma = \frac{M}{I} y$
27.5	106,250	1.28	.085	15,071
42.5	106,250	.82	.055	15,071



$$b = 3.813$$

$$d = 4.25$$

$$h = 3.813 - 2 \times 0.438 = 2.937$$

$$k = 4.25 - 2 \times 0.438 = 3.374$$

$$I_{x-x} = \frac{bd^3 - hk^3}{12} = \frac{3.813 \times 4.25^3 - 2.937 \times 3.374^3}{12} = 14.99$$

$$E_{x-x} = \frac{2I_{x-x}}{d} = \frac{2 \times 14.99}{4.25} = 7.05$$

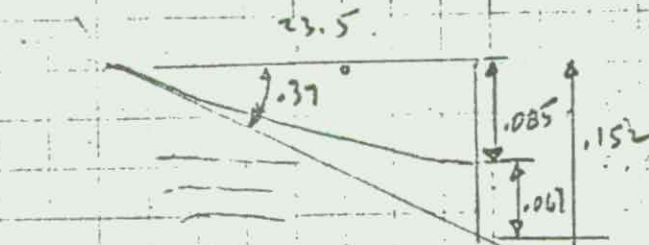
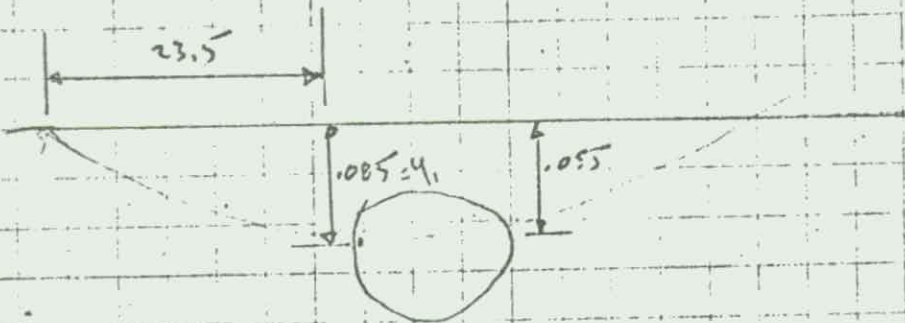
$$I_{y-y} = \frac{db^3 - kh^3}{12} = \frac{4.25 \times 3.813^3 - 3.374 \times 2.937^3}{12} = 12.51$$

$$E_{y-y} = \frac{2I_{y-y}}{b} = \frac{2 \times 12.51}{3.813} = 6.56$$

$$\theta_1 = \theta_2 = \frac{ML}{EI} \left(\frac{1}{3} + \frac{1}{6} \right) = \frac{ML}{EI} \cdot \frac{1}{2} = \frac{ML}{2EI}$$

$$M = 106,250, \quad I = 14.99$$

$$\theta_1 = \frac{106,250 \times 53}{2 \times 29,000,000 \times 14.99} = .0065 \text{ rad} = .37^\circ$$



$$y_{\text{diff}} = y_1 - y_2 = .152 - .085 = .067$$

THIS IS THE DEFLECTION THAT THE AXLE SHAFT WILL UNDERGO BECAUSE OF THE MOMENT AT THE SPLINED END OF THE AXLE SHAFT.

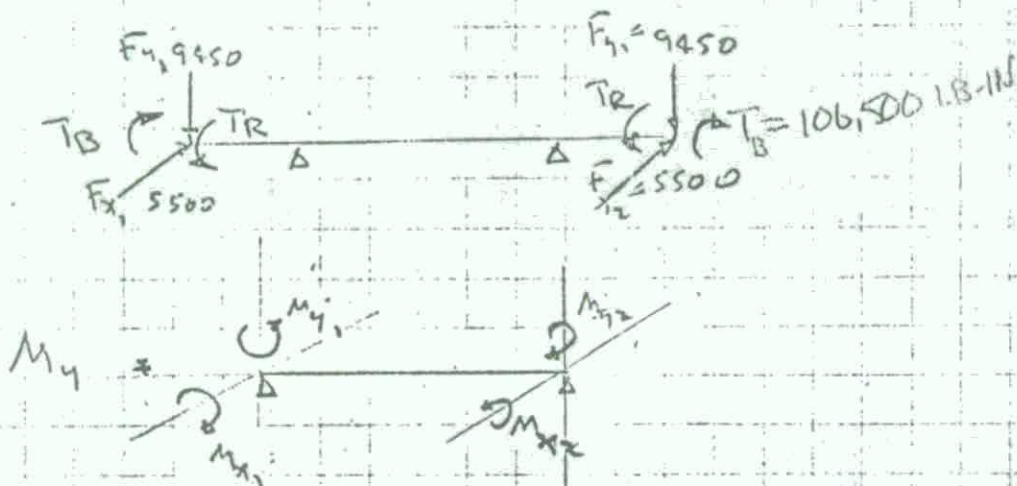
Combined loading due to braking

BRAKING TORQUE IS REACTED AT SUSP. ARMS

NO TORSIONAL STRESS IN ~~INNER~~ ^{CENTRAL} PART OF HVIS

At wheel: $F_y = 9450$

$F_x = 5500$



$M_{x1} = M_{x2} = 12.5 \times 9450 = 118,125$

$M_{y1} = M_{y2} = 12.5 \times 5500 = 68,750$

SHEAR STRESS WILL BE MAX AT MID POINT OF LONG SIDE OF RECT

LONG SIDE:

$Q_x = \frac{M_{x1}}{Z_{x-x}} = \frac{118,125}{7.05} = 16,755$

$\tau = \frac{T}{2t(b-t)(l-t)}$
 $= \frac{106,500}{2(44)(4.25-44)(3.81-44)}$
 $= 9420 \text{ PSI}$

$Q_y = \frac{M_{y1}}{Z_{y-y}} = \frac{68,750}{6.56} = 10,480$

SHORT SIDE:

$Q_{max} = Q_x + Q_y = 16,755 + 10,480 = 27,235$
 (AT CORNER)

$\frac{T}{2t/b}$

STRESS IS SAME BECAUSE THICKNESS IS CONSTANT

$y_1 = \frac{1.28}{106,250} \frac{M_{x1}}{I_{x-x}} = \frac{1.28 \times 118,125}{106,250 \times 14.99} = .095$

$x_1 = \frac{1.28}{106,250} \frac{M_{y1}}{I_{y-y}} = \frac{1.28 \times 68,750}{106,250 \times 12.51} = .066$

$d_u = \sqrt{y_1^2 + x_1^2} = \sqrt{.095^2 + .066^2} = .116$

$$\theta_{z_1} = \frac{M_y L}{2EI_{x-x}} = \frac{118,125 \times 53}{2 \times 29,000,000 \times 14.99} = .0072 \text{ rad} = .413^\circ$$

$$\theta_{x_1} = \frac{M_y L}{2EI_{y-y}} = \frac{68,750 \times 53}{2 \times 29,000,000 \times 12.51} = .0050 \text{ rad} = .288^\circ$$

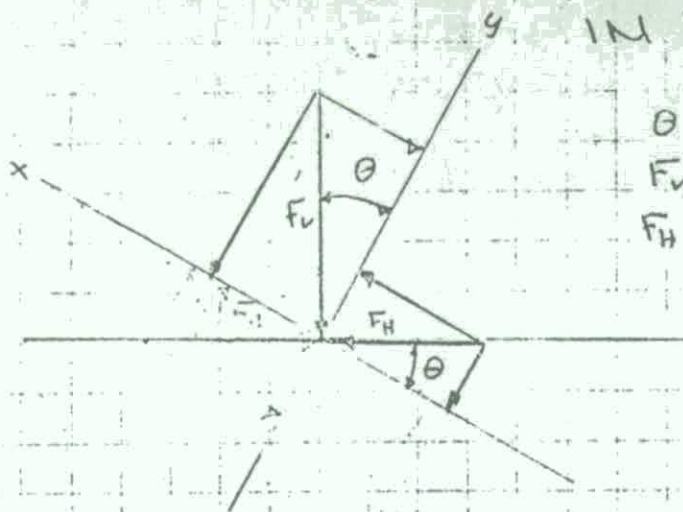
$$y_1 = 23.5 \tan \theta_{z_1} = 23.5 \times \tan .413^\circ = .167$$

$$x_1 = 23.5 \tan \theta_{x_1} = 23.5 \times \tan .288^\circ = .118$$

$$d_s = \sqrt{y_1^2 + x_1^2} = \sqrt{.167^2 + .118^2} = .206$$

$$d_{\text{net}} = d_s - d_H = .206 - .116 = .090 \quad (\text{NET AXLE SHAFT DEFLECTION FROM PANIC STOP})$$

WITH AXLE TIPPED 23.5° & LOADED
IN PANIC STOP



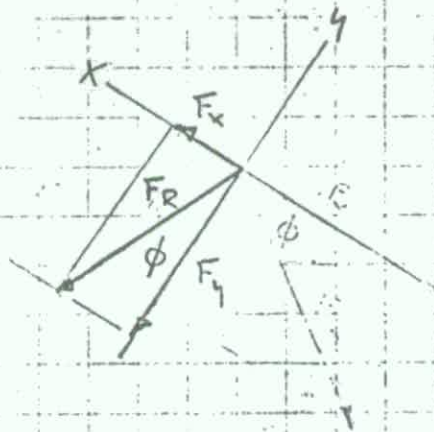
$$\theta = 23.5^\circ$$

$$F_v = 9450$$

$$F_H = 5500$$

$$F_y = F_v \cos \theta + F_H \sin \theta = 9450 \cos 23.5^\circ + 5500 \sin 23.5^\circ = 10,859$$

$$F_x = F_v \sin \theta - F_H \cos \theta = 9450 \sin 23.5^\circ - 5500 \cos 23.5^\circ = -1276$$



$$F_R = \sqrt{F_x^2 + F_y^2} = \sqrt{(-1276)^2 + 10,859^2} = 10,934$$

$$\phi = \tan^{-1} \left(\frac{-F_x}{F_y} \right) = \tan^{-1} \left(\frac{1276}{10,859} \right) = 6.7^\circ$$

Check

$$F_R = \sqrt{F_v^2 + F_H^2} = \sqrt{9450^2 + 5500^2} = 10,934$$

$$M_x = 12.5 \times F_y = 12.5 \times 10,859 = 135,738$$

$$M_y = 12.5 \times F_x = 12.5 \times 1276 = 15,950$$

Design ① (ORIGINAL SECTION)

$$k = 3.374, h = 2.937, b = 3.813, d = 4.25, t_1 = .438$$

3.374

3.374

$$I_{x-x} = \frac{3.813 \times 4.25^3 - 2.937 \times 3.374^3}{12} = 14.99$$

$$z_{x-x} = \frac{2 \times 14.99}{4.25} = 7.05$$

$$\sigma_x = \frac{135,738}{7.05} = 19,254$$

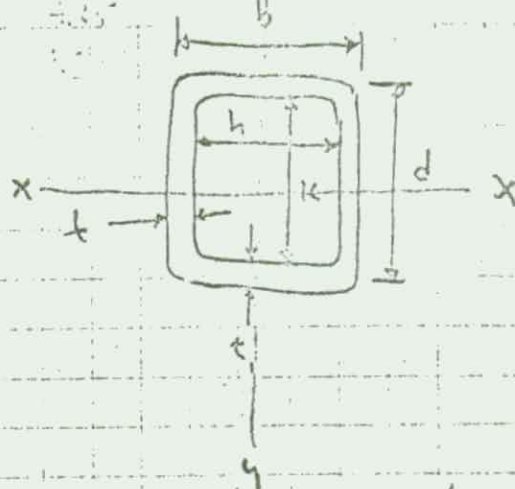
$$I_{y-y} = \frac{4.25 \times 3.813^3 - 3.374 \times 2.937^3}{12} = 12.51$$

$$z_{y-y} = \frac{2 \times 12.51}{3.813} = 6.56$$

$$\sigma_y = \frac{15,950}{6.56} = 2431$$

$$\sigma_{max} = 19,254 + 2431 = 21,685$$

(AT CORNER)



COMBINED STRESS w/ TORQUE ADDED

@ Y-Y AXIS $\sigma = 19,254$ PSI
 $\tau = 9470$ PSI

$$\sigma_{comb} = \sigma_x + \sqrt{\frac{\sigma_y^2}{4} + \tau^2}$$

$$= 19,254 + \sqrt{\frac{19,254^2}{4} + 9470^2}$$

$$= 32,724 \text{ PSI}$$

(SHEAR STRESS SAME AS BEFORE)

Design ② (ADDING .25 THICK PLATES TOP & BOTTOM)

$$k = 3.374, h = 2.937, b = 3.813, d = 4.25 + .50 = 4.75, t_1 = .438 + .25 = .688$$

$$I_{x-x} = \frac{3.813 \times 4.75^3 - 2.937 \times 3.374^3}{12} = 24.65$$

$$z_{x-x} = \frac{2 \times 24.65}{4.75} = 10.38$$

$$\sigma_x = \frac{135,738}{10.38} = 13,077$$

$$I_{y-y} = \frac{4.75 \times 3.813^3 - 3.374 \times 2.937^3}{12} = 14.82$$

$$z_{y-y} = \frac{2 \times 14.82}{3.813} = 7.77$$

$$\sigma_y = \frac{15,950}{7.77} = 2053$$

$$\sigma_{max} = 13,077 + 2053 = 15,130$$

Compute shear & combined stress with torsion load added.

$$a = 4.75$$

$$b = 3.813$$

$$r = .938 + .25 = .688$$

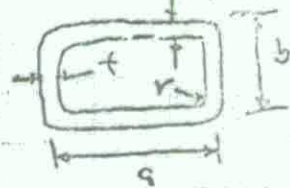
$$t_1 = .438$$

Torsion induced by braking $T = 106,500 \text{ in-lb}$.
Use formula given in ref 11, Table IV, case 11

Average $s_1 = \frac{T}{2t(a-t)(b-t_1)}$ near mid-length of short sides t_1

$$s_1 = \frac{106,500}{2 \times .688(4.75 - .688)(3.813 - .438)}$$

$$s_1 = 5646 \text{ psi}$$



Average $s_2 = \frac{T}{2t_1(a-t)(b-t_1)}$ near mid-length of long sides

$$s_2 = \frac{106,500}{2 \times .438(4.75 - .688)(3.813 - .438)}$$

$$s_2 = 8869 \text{ psi}$$

Combined stress at middle of each side

Middle of short side (b)

$$\text{Bending stress} = \sigma_x = 13,077$$

$$\text{Shear stress} = s_1 = 5646$$

$$\text{Combined stress} = \sigma_{\text{comb}} = \frac{\sigma}{2} + \sqrt{\frac{\sigma^2}{4} + s_1^2}$$

$$\sigma_{\text{comb}} = \frac{13,077}{2} + \sqrt{\frac{13,077^2}{4} + 5646^2} = 15,177$$

Middle of long side

$$\text{Bending stress} = \sigma_y = 2053$$

$$\text{Shear stress} = s_2 = 8869$$

$$\sigma_{\text{comb}} = \frac{2053}{2} + \sqrt{\frac{2053^2}{4} + 8869^2} = 9755$$

Combined stress at corner considering stress concentration under torsional loading. Stress concentration factor is theoretical maximum which may never be obtained under actual loading conditions using ductile material.

$$\tau_{max} = \tau K_{ts}$$

$$K_{ts} = f(r/c)$$

$$\text{Assume } c = \frac{t + r_i}{2} = \frac{.688 + .438}{2} = .563$$

$$r = 3/16 = .1875$$

$$r/c = .1875 / .563 = .333$$

From Fig. 113, ref. (2)

$$K_{ts} = 2.1$$

$$\text{Use } \tau = \frac{s_1 + s_2}{2} = \frac{5646 + 8869}{2} = 7258$$

$$\tau_{max} = 7258 \times 2.1 = 15,242$$

Bending stress at inside corner

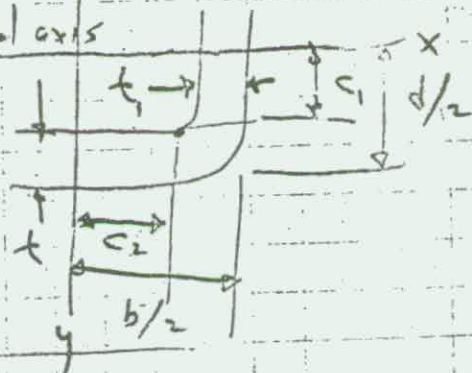
Since $\bar{\epsilon}_{x-x}$ and $\bar{\epsilon}_{y-y}$ are based on the distance from the neutral axis to the outermost fiber a reduced bending stress is computed using c_1' and c_2' .

$$\sigma_x' = \frac{c_1'}{d/2} \sigma_x = \frac{1.687}{2.375} 13,077 = 9289$$

$$\sigma_y' = \frac{c_2'}{b/2} \sigma_y = \frac{1.47}{1.91} 2053 = 1580$$

neutral axis

neutral axis



$$c_1 = d/2 - t = 2.375 - .688 = 1.687$$

$$d/2 = 4.75/2 = 2.375$$

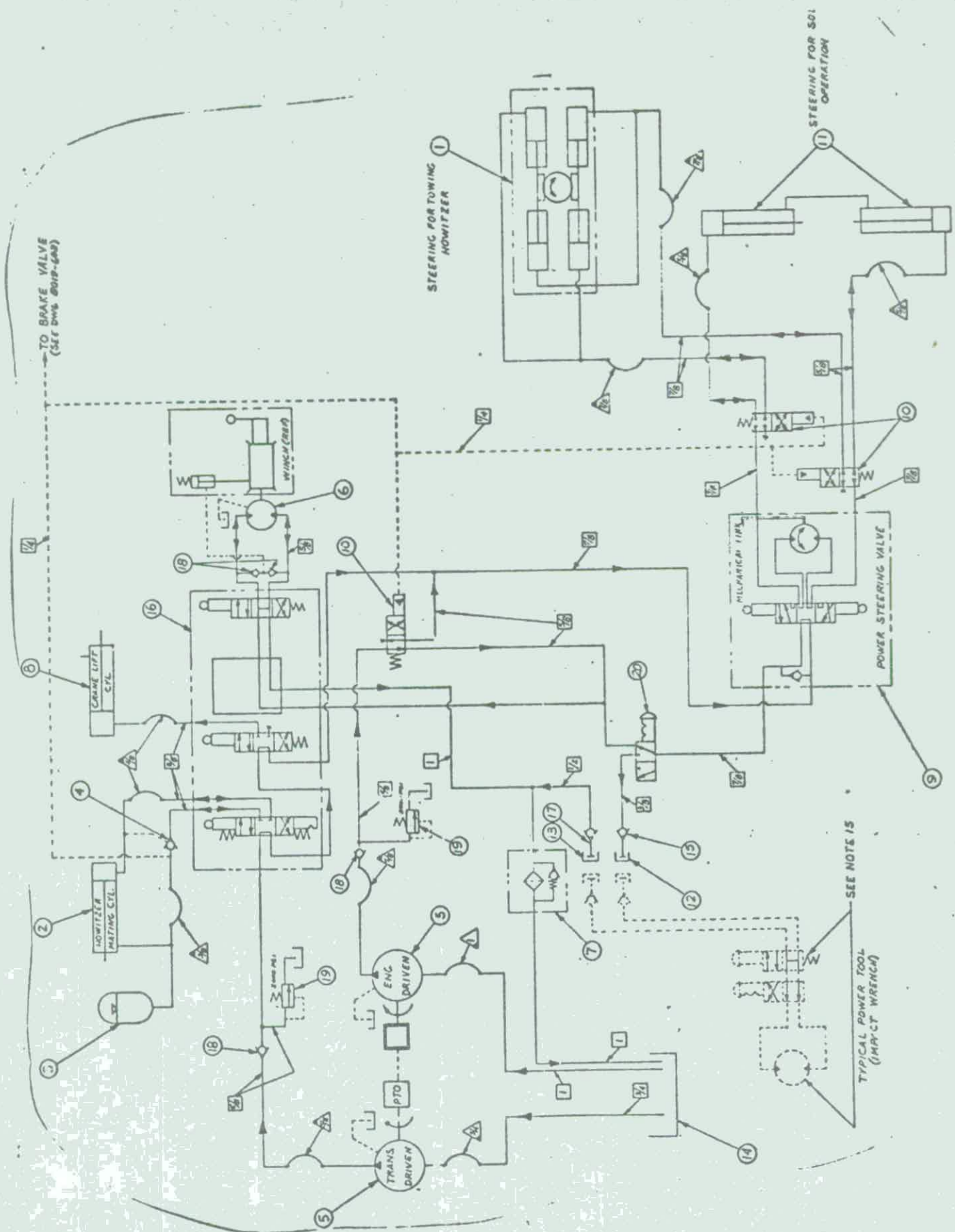
$$c_2 = b/2 - t = 1.91 - .438 = 1.47$$

$$b/2 = 3.813/2 = 1.91$$

Combined stress

$$\sigma_c = \frac{1}{2} (\sigma_x' + \sigma_y') + \sqrt{\frac{(\sigma_x' - \sigma_y')^2}{4} + \tau_{max}^2} = \frac{1}{2} (9289 + 1580) + \sqrt{\frac{(9289 - 1580)^2}{4} + 15242^2}$$

$$\sigma_c = 21,156$$



$$P_{PRESS} = 2000 \text{ PSI}$$

BOOM CYL.

$$LOAD = 2000 \text{ lb.}$$

$$\frac{LOAD}{AREA} = \frac{AREA}{F}$$
$$2000 \times 22.6 = 5.4 \times F$$

$$F = \frac{2000 \times 22.6}{5.4}$$
$$= 8360$$

$$A_{REQ} = \frac{8360}{1800} = 4.65 \text{ in}^2$$

$$\sqrt{\frac{4.65}{3.14}} = r$$

$$1.215 = r$$

$$d = 2.43$$

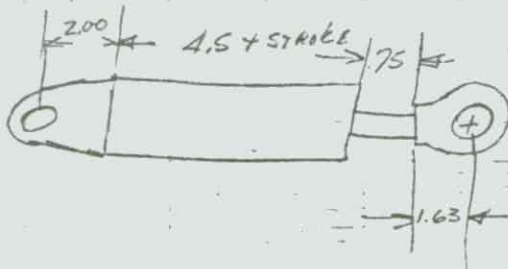
USE CYL. WITH 2 1/2 IN BORE

SELECT BRUNING 3-2.5-20.25-RE1

SAE CLEVIS END

EXTENDED LENGTH 37.75

RETRACTED LENGTH 29.13



2.00

4.50

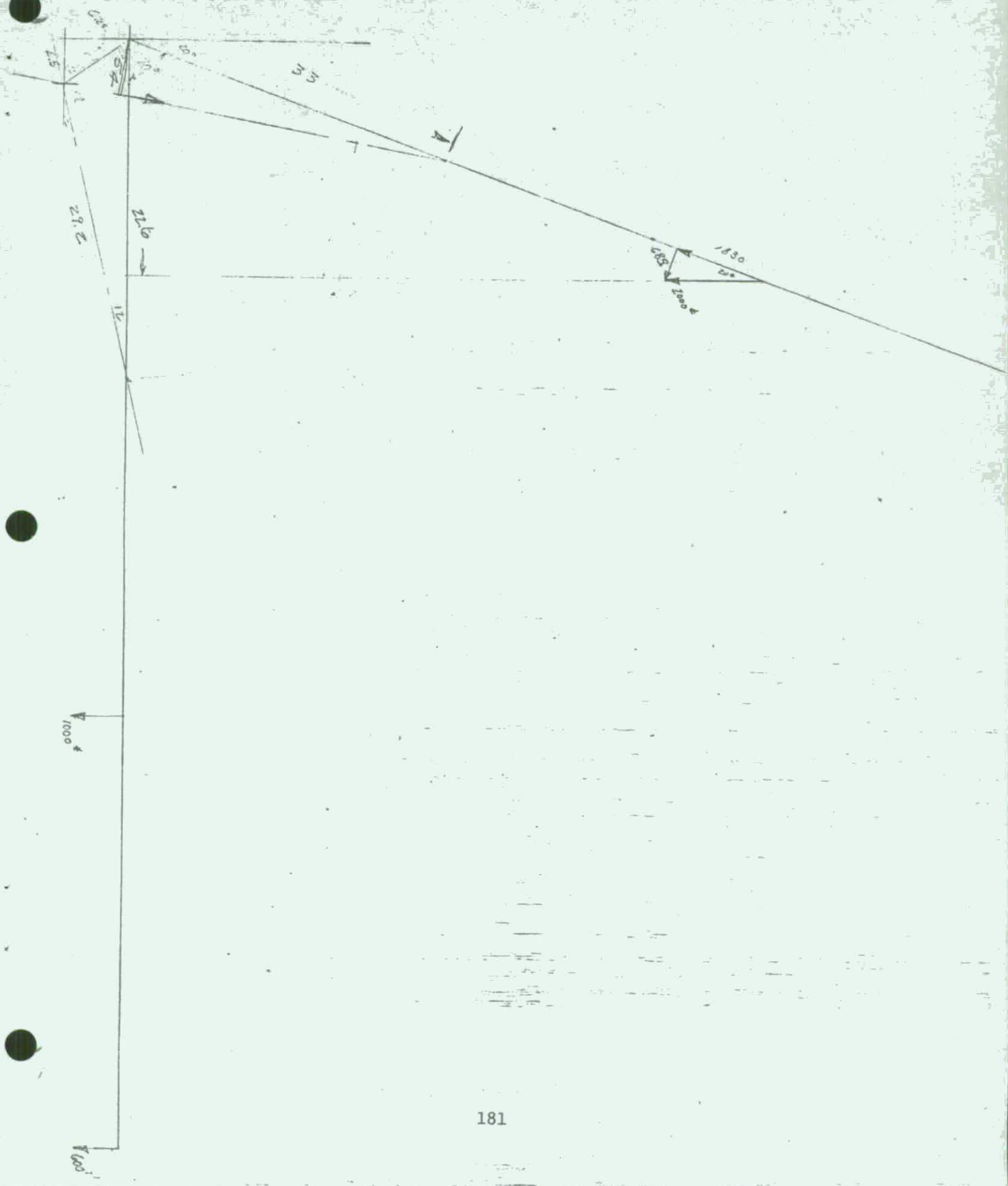
.75

1.63

8.88

$$29.13 - 8.88 = \text{STROKE}$$

$$20.25 = \text{STROKE}$$



PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY J. CRAIG
 CHECKED BY _____
 DATE 11-16-73

ORDER NO. ATLAS
 PAGE _____ OF _____
 REPORT NO. _____

Reservoir - OPEN CIRCUIT

$$206 \text{ GAL} \times 231 \text{ IN}^3/\text{GAL} = 4620 \text{ IN}^3$$

INSIDE WIDTH - 7.75"

" LENGTH - 24.50"

" HEIGHT - 22.75

$$\frac{24.50 \times 22.75 \times 7.75}{231} = 18.76 \text{ GAL}$$

LESS VOLUME TO HIGH MARK.

$$\frac{18 \times 7.75 \times 3}{2 \times 231} = .9 \text{ GAL}$$

$$18.76 - .9 = 16.8 \text{ GAL}$$

$$\text{WT} = 16.8 \text{ GAL} \times 7.5 \text{ \#/GAL} = 126 \text{ \#}$$

SEE LAYOUT 8019-503 FOR DETAILS

8019-502 FOR LOCATION

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY J. CRAIG
 CHECKED BY _____
 DATE 11-16-73

ORDER NO. ATLAS
 PAGE 1 OF 2
 REPORT NO. _____

PROBLEM — CAN ATLAS USE MANUAL STEER ON SANDY DIRT SOIL IN LOADED CONDITION

ASSUME — ROLLING RESISTANCE — $37\frac{\#}{1000\#}$ ON SANDY DIRT SOIL

AXLE LOAD IN LOADED — $17,670\#$ CONDITION

TRACK WIDTH — 78"

TORQUE REQUIRED TO TURN DRIVE AXLE IN SANDY DIRT

$$T = \frac{17,670 \times 37 \times 78}{1000} = 25,400 \text{ in-lb.}$$

ROTARY ACTUATOR — HYDRA POWER
 P/N TR180-15011 — PRODUCES 150,000 in-lb. AT 3000 PSI

$$\frac{150,000}{3000} = 50 \text{ in-lb/psi}$$

PRESSURE REQUIRED BY MANUAL OPERATION OF ORBITROL PUMP

$$\frac{25,400}{50} = 510 \text{ psi}$$

ORBITROL DISPLACEMENT SELECTION ROTARY ACTUATION

ASSUME $t = 4 \text{ SECS}$
 $N = 4 \text{ TURNS}$

CYL VOL = 184 IN³

$$\text{ORBITROL DISP} = \frac{\text{CYL VOL}}{N} = \frac{184}{4} = 46 \text{ IN}^3/\text{REV}$$

SELECT YU-12 DISP = 47.6 IN³/REV

$$\text{ACTUAL } N = \frac{184}{47.6} = 3.86 \text{ TURNS}$$

$$\text{STEERING WHEEL SPEED} = \frac{3.86}{4} = .965 \text{ REV/SEC}$$

$$\text{GPM} = \text{ORBITROL DISP} \times \text{WHEEL SPEED} \times \frac{60}{231}$$
$$= 47.6 \times .965 \times \frac{60}{231} = 11.97$$

COMBINED PUMP OUTPUT @ 1600 ENG. = 13.06

$$\text{STEER TIME} = \frac{60 \times \text{CYL DISP}}{231 \times \text{GPM}}$$
$$= \frac{60 \times 184}{231 \times 13.0} = 3.68 \text{ SECS FULL STEER}$$

LIN. ACTUATOR. 1.5 DIA PISTON - .75 DIA ROD
USING PTO PUMP ONLY - 5.75 GPM @ 1600 RPM

$$\text{DISP} = 9.86 \text{ IN}^3$$
$$\text{STROKE} = 4.62$$

$$2 \text{ CYL WITH } 1.77 \text{ IN}^2 \text{ HEAD END}$$
$$\frac{1}{2} \text{ .378 IN}^2 \text{ ROD} = 2.148 \text{ IN}^2$$

$$\text{CYL. SPEED} = \frac{4.62}{4} = 1.15 \text{ IN/SEC.}$$

$$\text{STEERING TIME} = \frac{60 \times V}{\text{GPM} \times 2.31} = \frac{60 \times 9.86}{5.75 \times 2.31} = .445 \text{ SECS.}$$

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY J. CRAIG

ORDER NO. ATLAS

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WINCH P-10

LINE SPEED (HIGH)

BARE DRUM - 127 FPM

FULL DRUM - 236 FPM

(LOW)

BARE DRUM - 27 FPM

FULL DRUM - 44 FPM

GEAR RATIO (LOW)

140 : 1

(HIGH)

26 : 1

DRUM DIA = 6 1/2"

ROPE DIA = 1/2"

$$\text{LINE SPEED (FPM)} = \frac{(.5236)(\text{MOTOR RPM})(R \text{ TO CENTER WIRE})}{\text{GEAR RATIO}}$$

$$\text{MOTOR RPM} = \frac{\text{LINE SPEED} \times \text{GEAR RATIO}}{(.5236)(R \text{ TO CENTER WIRE})}$$

$$= \frac{127 \times 26}{.5236 \times 3.5}$$

$$= 1800 \text{ RPM}$$

USING MOTOR ON DRAWING 13218E4112

DIST OF MOTOR = 1.2 IN³/REV

$$Q_{\text{INPUT}} = \frac{1800 \times 1.2}{231} = \underline{9.35 \text{ GPM}}$$

PUMP REQUIREMENT @ $\eta_0 = 90\%$

$$\frac{9.35}{.90} = 10.4 \text{ GPM.}$$

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY J. CRAIG

ORDER NO. ATLAS.

CHECKED BY _____

PAGE 2 OF 2

DATE 11-16-73

REPORT NO. _____

ASSUME 1 REV/SEC OF STEERING WHEEL
OR 60 RPM

FROM ORBITROL DATA

DISP = 23.8 IN³
T = 2,950 IN-LB.
RPM = 60

THEREFORE TORQUE @ 45.7 DISP AND 60 RPM

$$T = \frac{45.7 \times 510}{2\pi} = 3,700 \text{ IN-LB.}$$

STEERING WHEEL DIA IS 12"

WHEEL INPUT REQUIRED TO DEVELOP
3,700 IN-LB.

$$\frac{3,700}{6} = 617 \#$$

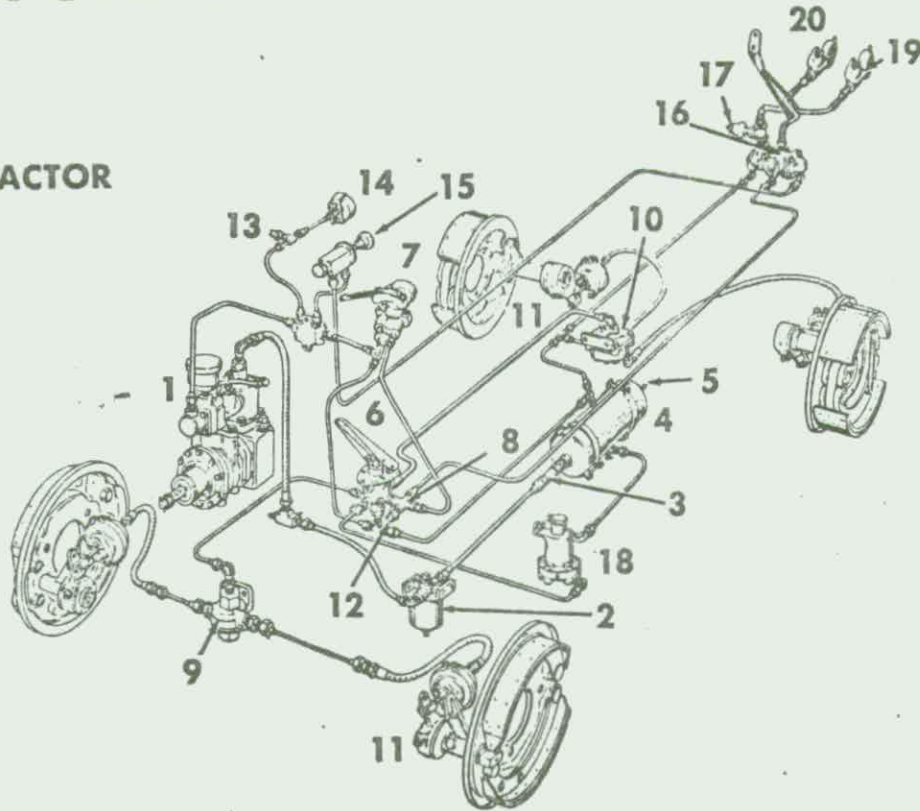
617# INPUT TO STEERING WHEEL
CAN NOT BE ACCOMPLISHED BY
DRIVER USING BOTH HANDS.

Wagner Lockheed®

STRAIGHT AIR SYSTEM

PROTECTED TRACTOR

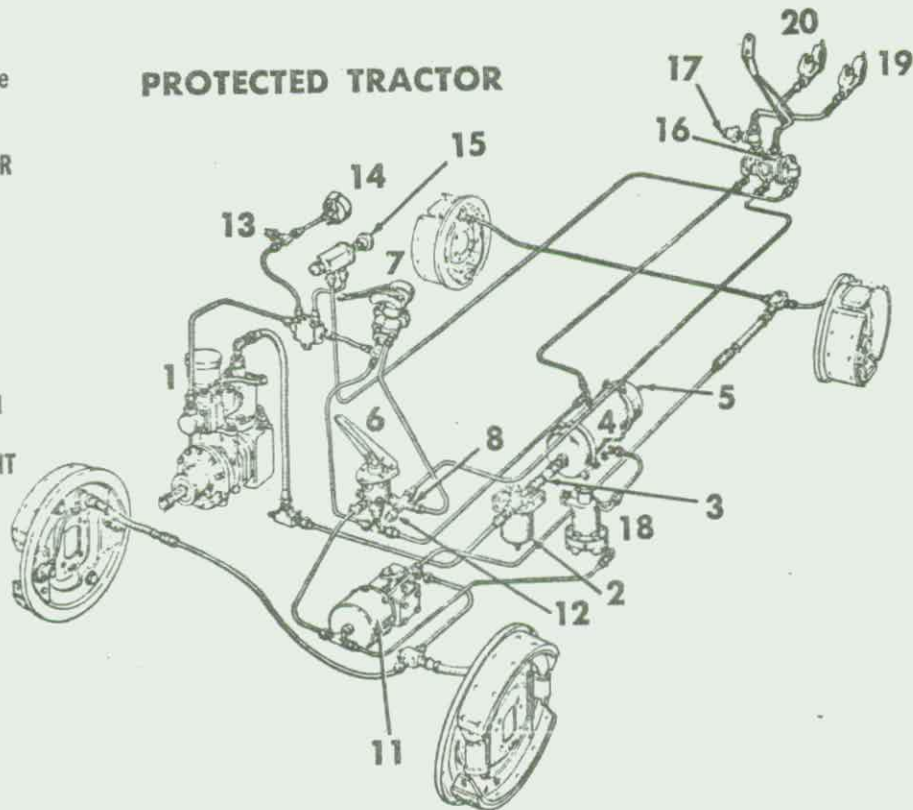
1. COMPRESSOR (includes control valve)
2. ALCOHOL INJECTOR—Accessory option
3. SINGLE CHECK VALVE
4. AIR TANK
5. RESERVOIR SAFETY VALVE
6. FOOT APPLICATION VALVE
7. HAND APPLICATION VALVE
8. TWO-WAY CHECK VALVE
9. QUICK-RELEASE VALVE (shown) or optional Tee



AIR-OVER-HYDRAULIC SYSTEM

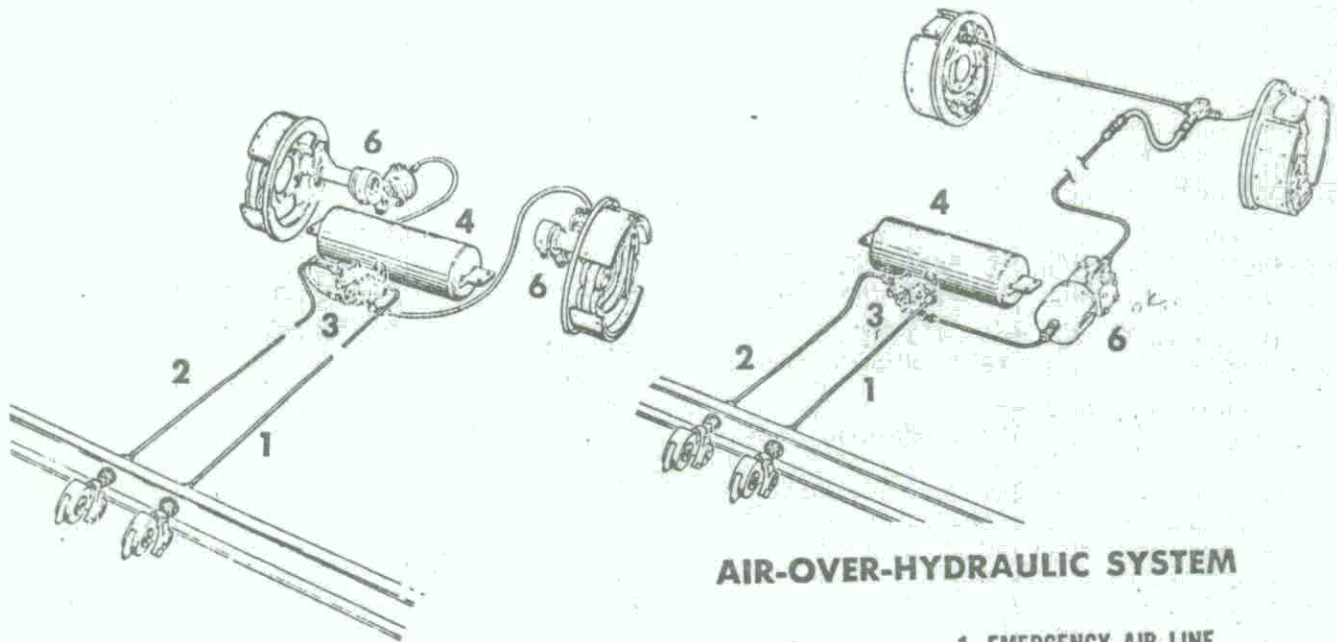
PROTECTED TRACTOR

10. RELAY QUICK-RELEASE VALVE (shown) or optional Quick-Release Valve or Tee
11. BRAKE CHAMBER and SLACK ADJUSTER (straight-air) or POWER CLUSTER (air-over-hydraulic)
12. SWITCH, NORMAL STOP LIGHT CIRCUIT
13. SWITCH, LOW PRESSURE INDICATOR CIRCUIT
14. AIR GAUGE
15. EMERGENCY BRAKE VALVE
16. TRACTOR AIR LINE PROTECTION VALVE
17. SWITCH, EMERGENCY STOP-LIGHT CIRCUIT
18. MOISTURE EJECTION VALVE
19. EMERGENCY AIR LINE and HOSE COUPLER
20. SERVICE AIR LINE and HOSE COUPLER



AIR BRAKE SYSTEMS

PROTECTED SEMI-TRAILER



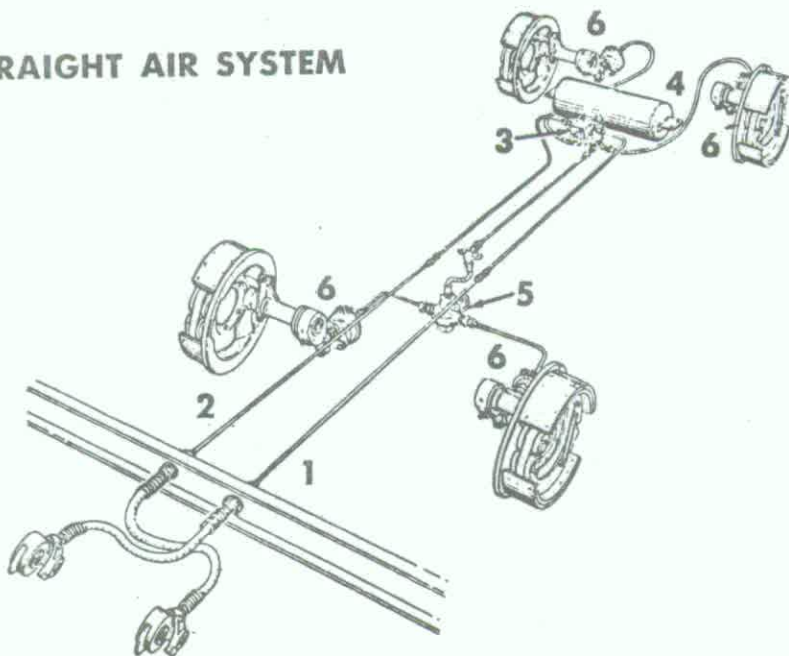
STRAIGHT AIR SYSTEM

AIR-OVER-HYDRAULIC SYSTEM

1. EMERGENCY AIR LINE
2. SERVICE AIR LINE
3. RELAY QUICK-RELEASE EMERGENCY (BREAKAWAY) VALVE

PROTECTED FULL TRAILER

STRAIGHT AIR SYSTEM



4. CLOSE-COUPLED TRAILER TANK
5. QUICK-RELEASE VALVE (shown) or optional Tee
6. BRAKE CHAMBER and SLACK ADJUSTER (straight-air) or POWER CLUSTER (air-over-hydraulic)

AIR APPLICATION VALVES

TYPE FF

TREADLE VALVE

Standard Valve

AE31770.....w/3/8"-18 Application Ports

AE31865.....w/1/2"-14 Application Ports

PUSH VALVE

Standard Valve

AE31680.....w/3/8"-18 Application Ports

AE31863.....w/1/2"-14 Application Ports

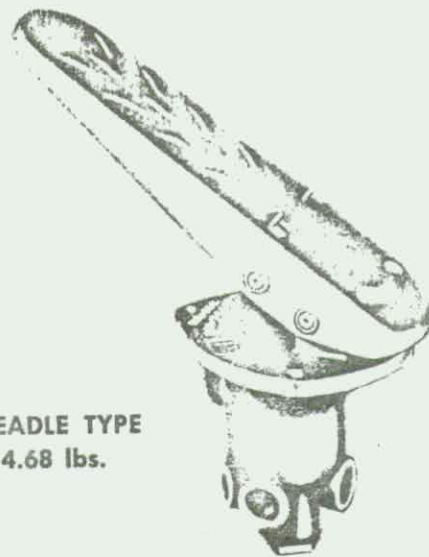
With Threaded Push Rod - 1/2"-20 Thread

AE34290.....w/3/8"-18 Application Ports

With Elliptical Mounting Flange - 2 Hole

AE33990.....w/1/2"-14 Application Ports

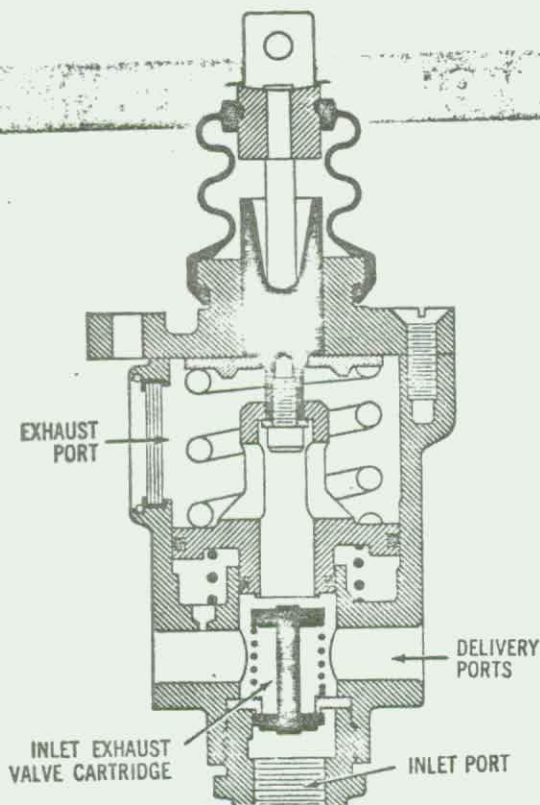
The new type FF foot application valve is a corrosion resistant unit which is both smaller and lighter than equivalent old style valves, yet maintains the same high capacity performance. The type FF application valve has a graduated metering range. Normal braking pressures in the 5 to 75 psi range are metered in exact proportion to foot pressure and pedal movement. Above 75 psi the full tank pressure is "dumped" for emergency braking.



TREADLE TYPE
4.68 lbs.



PUSH TYPE
3.25 lbs.



Valve illustrated in "released" position

Controlling force, applied at the valve push rod and transferred through a piston stem and metering spring, strokes the application piston against its return spring. During the stroke, the piston picks up the spring loaded inlet-exhaust valve cartridge. The centered exhaust passage in the piston is closed as the piston meets the cartridge exhaust (inner) poppet and then the continuing stroke unseats the inlet (outer) poppet, admitting compressed air into the application system. Applied air also by-passes to the piston through an equalization orifice. Pressure building against the piston forces it to move back upon the stem, compressing the metering spring. The piston is balanced by these opposing forces as its lapping action permits the spring load to close the inlet poppet while holding the exhaust poppet seated on the piston. The unit remains poised in this "holding" position until another movement of the push rod unbalances it, either to admit increased air pressure or to exhaust the system.

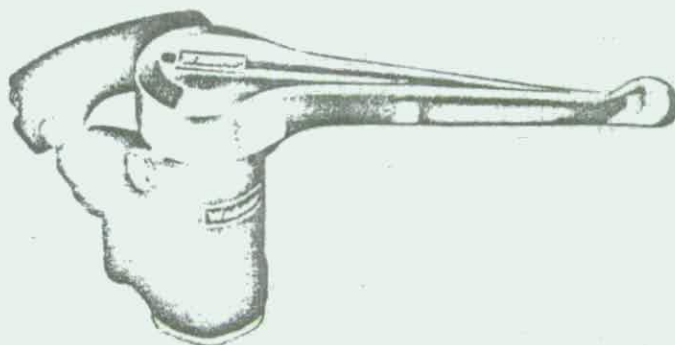
The valve meters pressure up to approximately 75 psi. Above this pressure the piston stem and piston bottom, lap is prevented, the inlet poppet remains fully open and tank pressure is applied.

HAND OPERATED AIR APPLICATION VALVE

TYPE HE—Trailer Control

Valve Part No. *AE43769, w/Bracket
Valve Part No. *AE25340, w/Bracket
 AE25816, w/o Bracket

* Different Handle positions

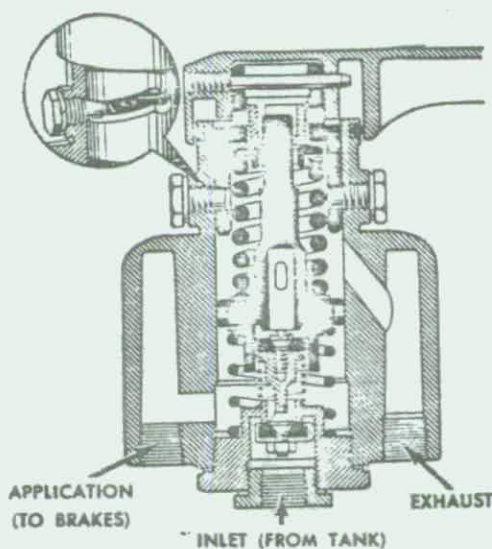


This metering type hand control valve regulates braking air pressure in direct proportion to hand pressure and movement. Braking effort balances every movement of the control valve handle. Resistance increases or decreases with the braking effort and handle "feel" provides the driver with a braking "gauge" to secure a smooth stop throughout the range from slow to emergency deceleration.

The Wagner Air Brake Trailer Control Valve provides the driver with independent metered control of the trailer brakes on combination vehicles. If the tractor is equipped with air brakes, the foot application valve also operates the trailer brakes in conjunction with the tractor brakes. In this dual system, a two-way check valve separates hand valve and foot valve so that no application pressure escapes through the exhaust port of the valve not in use.

Weight, w/o bracket ... 3.5 lb.
w/ bracket ... 3.8 lb.

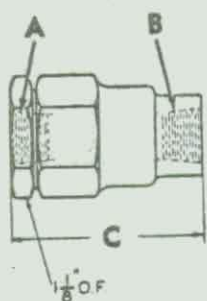
The valve is in exhaust position (handle rotated counterclockwise) and the application side of the system is open to atmosphere until the brake application cycle starts. Clockwise movement of the handle rotates the actuating cam downward as cam ramps slide on pins threaded into the valve body. Downward thrust is transferred to the metering piston through a travel adjusting nut and metering spring. The piston is forced down, closing the exhaust valve to seal the system from atmosphere. Further movement forces open the inlet valve mounted on the exhaust valve stem. Tank air pressure now flows past the inlet valve and through the outlet port, applying the brakes. Applied air pressure also builds against the face of the metering piston and cup, opposing the downward thrust. Increasing pressure returns the piston and cup until compression of the metering spring balances application pressure against pressure on the handle. As the piston returns, the valve return spring holds the exhaust valve seated while closing the inlet valve at the balanced braking pressure. Additional handle movement forces the piston and metering spring to compensate the increase or decrease by moving to relieve the unbalanced pressure condition.



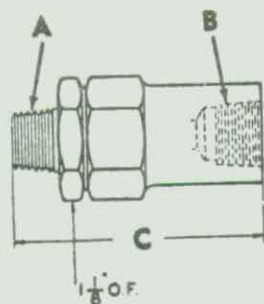
Depending upon the direction of movement, applied air pressure is either increased or exhausted to atmosphere.

SINGLE CHECK VALVES

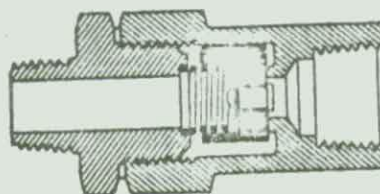
TYPE SA



Part No. AC267
Weight 0.5 lbs.



Part Nos. AC16739,
23540, 40574
Weight 0.6 lbs.

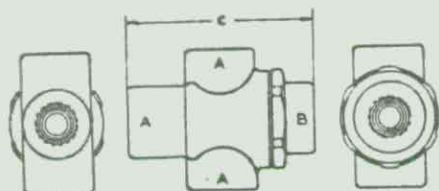


Tubing Size	Thread "A"	Thread "B"	Length "C"	Part Number
3/8"	1/4" - 18 Female	1/4" - 13 Female	2 5/32"	AC267
1/2"	3/8" - 18 Male	3/4" - 18 Female	2 15/32"	AC16739
5/8", 3/4"	1/2" - 14 Male	1/2" - 14 Female	3 3/32"	AC23540
3/4"	3/4" - 14 Male	3/4" - 14 Female	3 7/32"	AC40574

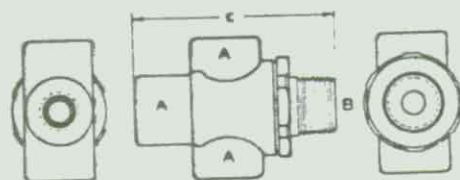
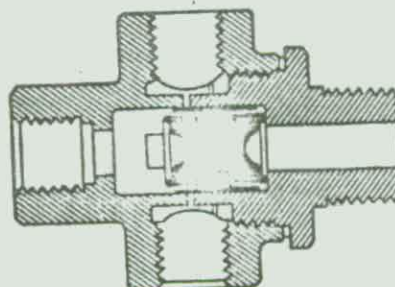
TWO-WAY CHECK VALVES

TYPE WB

Weight 0.6 lbs.



FOUR FEMALE PORTS



1 MALE • 3 FEMALE PORTS

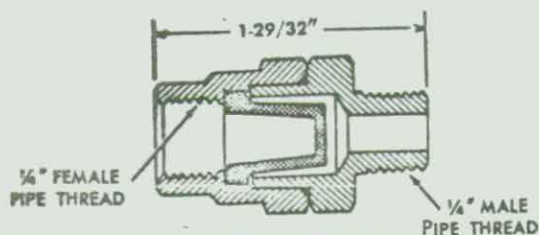
Tubing Size	Thread "A"	Thread "B"	Length "C"	Part Number
1 MALE AND 3 FEMALE PORTS				
3/8"	1/4"-18 Female	1/2"-14 Male	2 7/8"	AC32917
3/8"	1/4"-18 Female	3/8"-18 Male	2 7/8"	AC32940
3/8"	3/8"-18 Female	1/2"-14 Male	2 7/8"	AC43755
FOUR FEMALE PORTS				
3/8"	1/4"-18 Female	1/4"-18 Female	2 1/8"	AC32922
3/8"	1/4"-18 Female	3/8"-18 Female	2 1/8"	AC32938
3/8" - 1/2"	1/4"-18 Female	1/4"-18 Female	2 1/8"	AC35893
3/8"	1/4"-18 Female	1/4"-18 Female	2 1/8"	AC36709
3/8"	3/8"-18 Female	3/8"-18 Female	2 1/8"	AC40522
3/8"	3/8"-18 Female	3/8"-18 Female	2 1/8"	AC43912

* One outlet contains AC25134 Pipe Plug.

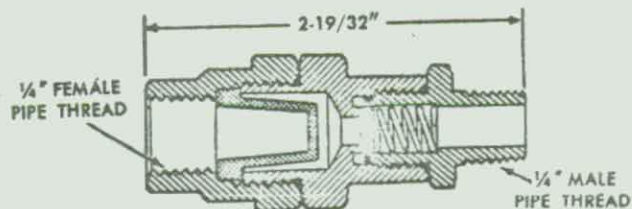
** One outlet contains AC31283 Pipe Plug.

† Outlet contains 1/4" Close Pipe Nipple—7/8" long.

AIR LINE FILTERS



AIR FILTER - AC27370
Weight 0.2 lbs.



AIR FILTER & CHECK VALVE - AC27375
Weight 0.3 lbs.

MOISTURE EJECTION VALVE

Air reservoirs are kept clean and dry when the air brake system is equipped with a moisture ejection valve preventing the accumulation of moisture and sludge in the air tanks. Fully automatic, operating in the 15-25 psi pressure range, the unit mounts in any convenient location. It is connected into the air system by two air lines—one leading from the bottom of the air tank, the other from a brake application line. Normal brake applications operate the valve, keeping the reservoir clean and moisture-free. Expulsions occur without a noticeable drop in gauge pressure.



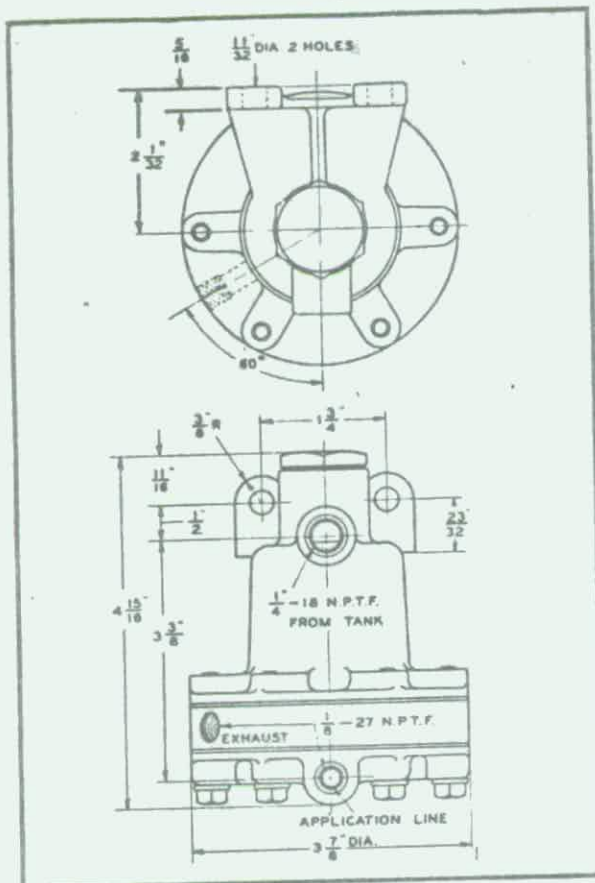
Part No. AE21857
Weight 1.85 lbs.

PERFORMANCE FEATURES

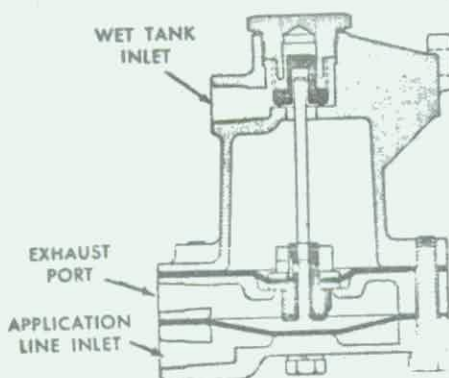
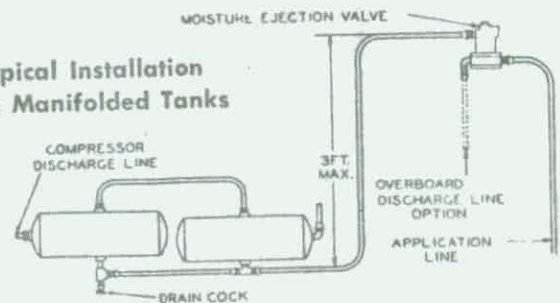
- POSITIVE OPERATION**—Average 15–25 psi brake application pressures guarantee frequent ejection to keep tank clean and dry. Valve ejects only upon pressure release. High application pressures do not harm the valve.
- HIGH CAPACITY**—Valve capacity is sufficient to eject up to four fluid ounces at one time, far more than will ever be required in any operation.
- NO AIR PRESSURE LOSS**—Ejection requires little air and does not cause a noticeable drop in gauge pressure. Working pressures cannot “balance” the valve in open position and “dump” reservoir pressure.
- OPEN TO ATMOSPHERE**—Valve fluid cavities are open to atmosphere. It is impossible to trap moisture within the unit and the valve will not freeze in open (exhaust) position.

CONSTRUCTION FEATURES

- DIAPHRAGM**—Nylon cord and neoprene rubber combined, oil resistant and long lived.
- INLET VALVE INSERT**—Resilient synthetic insert. Valve never requires lapping and seats better with use. Small particles of carbon and foreign matter are ejected without causing valve leakage because the insert seals around particles trapped on the valve seat.
- CORROSION RESISTANT**—All metal parts are made of corrosion resistant materials or are plated to prevent rust. Housing is of aluminum alloy.
- CONVENIENT MOUNTING**—Valve may be mounted in any location up to three feet higher than the air tank. It may be located inside the vehicle body with an overboard discharge line attached to the tapped exhaust port.



Typical Installation with Manifolded Tanks



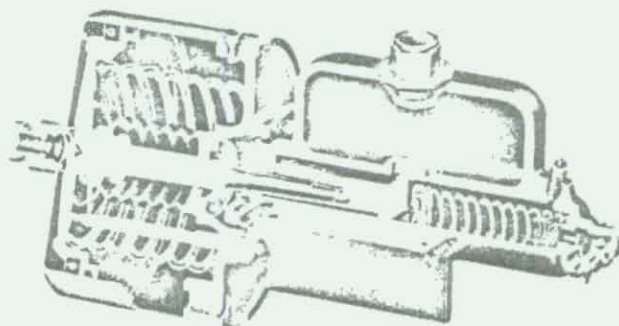
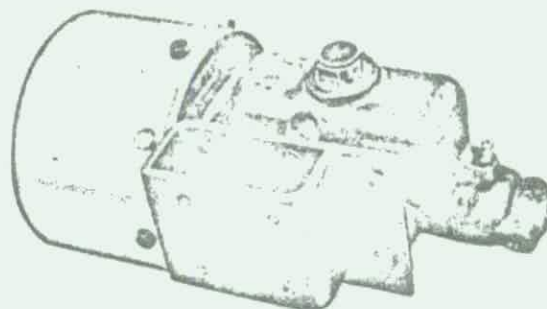
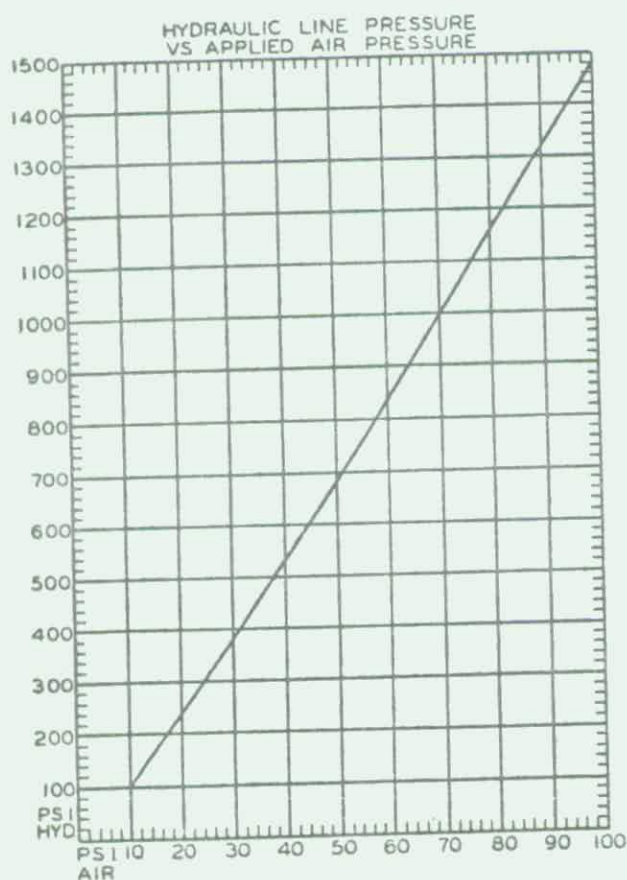
Applied air pressure forces the lower diaphragm upward to seat on and close the exhaust valve. Further upward movement raises exhaust valve and stem, opening the inlet valve. Reservoir pressure forces fluid and air mixture from bottom of reservoir to pass inlet valve and into fluid chamber where it is momentarily trapped. Release of application pressure permits pressure of trapped mixture to force exhaust valve diaphragm downward, permitting inlet valve to close. Continued releasing action moves lower diaphragm away from seat on exhaust valve and mixture is blown through valve and exhaust port. Valve remains open to atmosphere until the next application cycle.

AIR-OVER-HYDRAULIC POWER CLUSTER

AIR CYLINDER TYPE

Stroke	Cylinder I.D.		Power Cluster Part No.
	Hydraulic	Air	
2 $\frac{9}{16}$ "	1 $\frac{3}{4}$ "	7"	AF-840
2 $\frac{9}{16}$ "	1 $\frac{3}{4}$ "	7"	AF-43943
1 $\frac{1}{2}$ "	1 $\frac{3}{4}$ "	7"	AF-839
1 $\frac{1}{2}$ "	1 $\frac{1}{2}$ "	6"	AE-838
1 $\frac{1}{2}$ "	1 $\frac{1}{4}$ "	5"	AE-837

The power cluster is used in air-over-hydraulic brake systems to effect the transition from moderate applied air pressures to relatively high hydraulic pressures required for hydraulic braking. Air pressure entering the unit forces an air cylinder piston and push rod to stroke a standard hydraulic brake master cylinder. Hydraulic pressure is built in 15:1 ratio to the amount of applied air pressure. Optional use of additional power clusters provides independent hydraulic systems on specific axles and prevents complete loss of vehicle braking in the event of a hydraulic component failure.



PERFORMANCE FEATURES

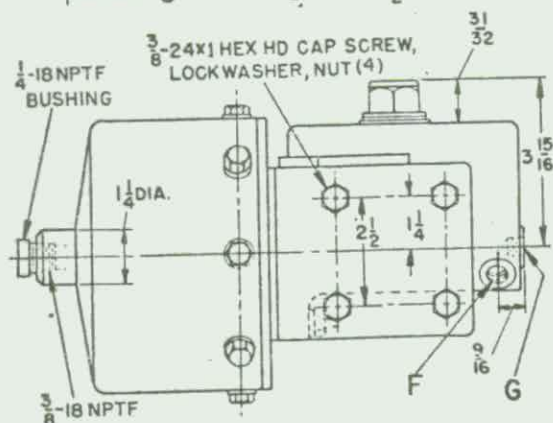
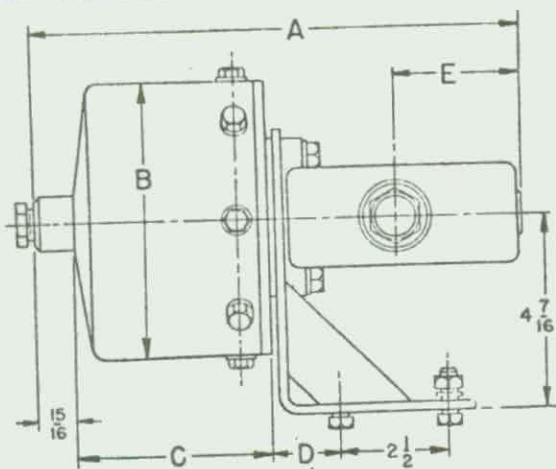
- 1500 PSI MAXIMUM PRESSURE**—15:1 pressure ratio provides 1500 psi hydraulic pressure at approximately 100 psi applied air pressure.
- GRADUATED CONTROL**—Power cluster, used in conjunction with a metering application valve, relates braking effort directly to pedal resistance. There is no "two-stage feel" regardless of load or road conditions.
- NEGLECTIBLE FRICTION LOSS**—There is no linkage to increase friction loss.
- BRAKE ADJUSTMENT INDICATOR**—Stroke travel indicator warns of the need for lining clearance adjustment. Indicator may be used to operate a warning lamp switch.

CONSTRUCTION FEATURES

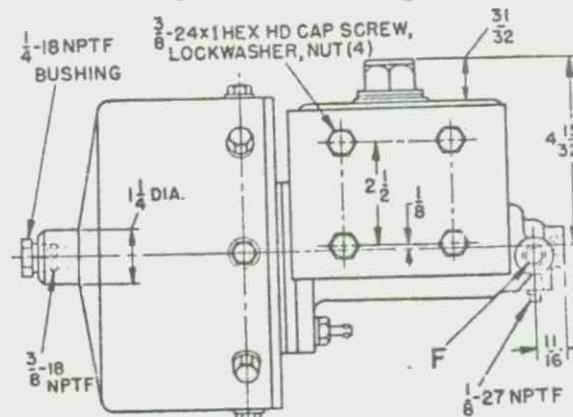
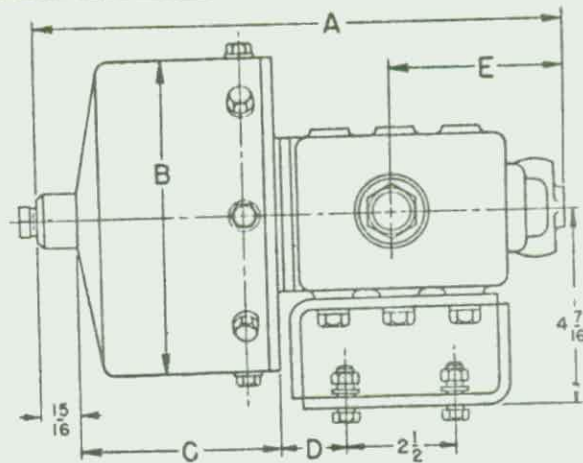
- COMPACT, PROVED UNIT**—The power cluster consists of a standard Wagner Lockheed hydraulic master cylinder and a single piston air cylinder engineered into a self contained, factory assembled and tested unit.
- NO EXTERNAL STRESSES**—Operating stresses are balanced out within the self contained unit.
- SIMPLE INSTALLATION**—No leverages or stresses to calculate. No levers or rods to lay out. Simply use power cluster of comparable size instead of usual brake master cylinder. Position unit in any accessible location.
- PROTECTED HYDRAULIC CYLINDER**—Boot stretched between the air piston and air cylinder head excludes dust and oil from hydraulic cylinder. Air displaced on the atmospheric side of the air piston is vented through a filtered breather port.
- NO CHANGE IN BRAKE BLEEDING AND MAINTENANCE**—Use of the power cluster causes no change in usual brake bleeding and maintenance procedures.

DIMENSIONS AND SPECIFICATIONS

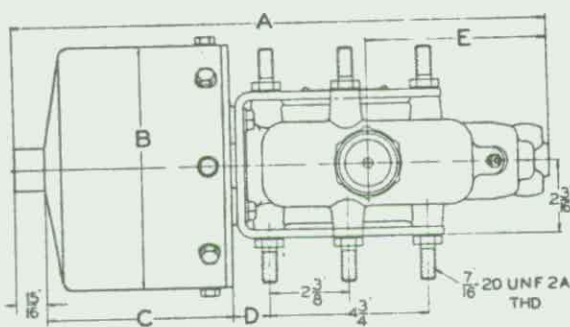
AE837 (Ord. 8017004)
 AE838 (Ord. 7763611; A-8408703;
 8955-7-24; 9969589)
 AF839 (Ord. 9968237)
 AF840 (Ord. 7763610)



**5" and 6" I.D. AIR CYLINDER
 1-1/4" and 1-1/2" I.D. HYDRAULIC CYLINDER
 Part Nos. AF837, AF838**

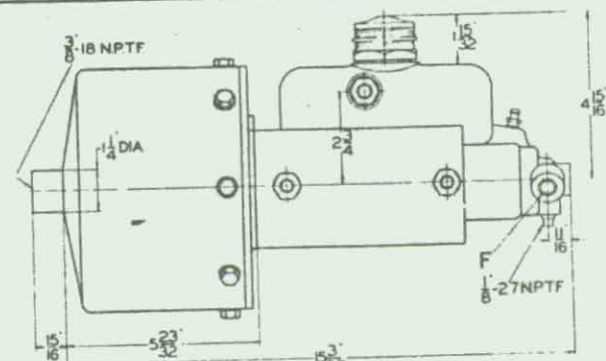


**7" I.D. AIR CYLINDER
 1-3/4" I.D. HYDRAULIC CYLINDER
 Part Nos. AF839, AF840**



PLAN VIEW

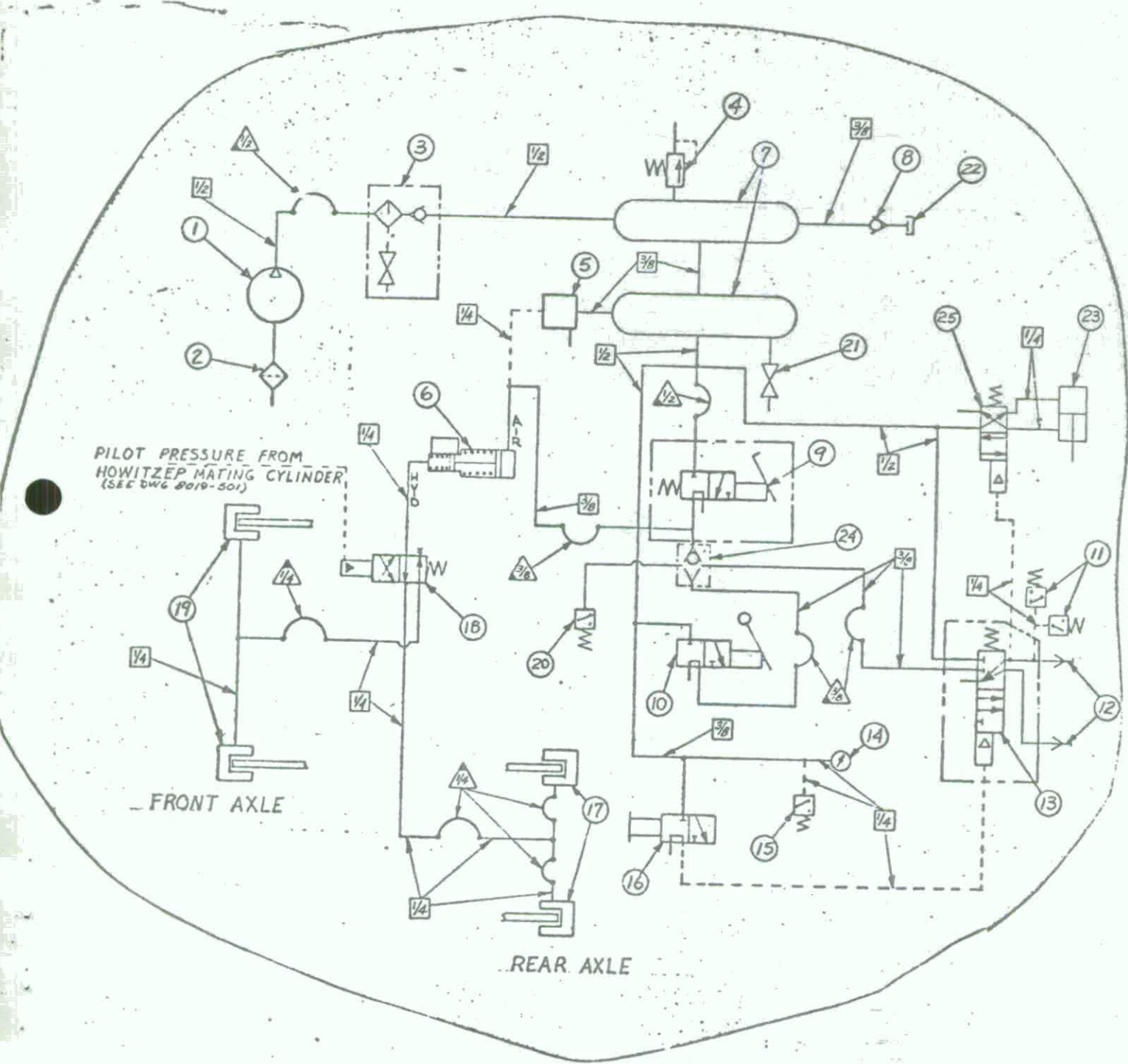
**7" I.D. AIR CYLINDER - 1-3/4" HYDRAULIC CYLINDER
 For Off-the-Road Equipment - Part No. AF43943**



SIDE VIEW

Hydraulic Fluid Displacement (Cu. In.)	Stroke	Cylinder I.D.		Charted Dimensions							Weight lbs.	Power Cluster Part No.
		Hyd.	Air	A	B	C	D	E	F (UNF-2B Thread)	G (UNF-2B or NPTF Thread)		
5.80	2 9/16"	1 3/4"	7"	16 1/32"	7 5/32"	5 9/16"	11 3/32"	5 7/16"	1/2"-20 (3)	31.75	AF-840
5.80	2 9/16"	1 3/4"	7"	15 3/32"	7 5/32"	5 9/16"	13 3/32"	5 7/16"	1/2"-20 (3)	28.00	AF-43943
3.25	1 1/2"	1 3/4"	7"	12 1/32"	7 5/32"	4 1/2"	11 3/32"	3 1 9/16"	1/2"-20 (3)	27.13	AF-839
2.40	1 1/2"	1 1/2"	6"	11 3/32"	6 5/32"	4 2 5/64"	11 7/32"	2 7/8"	1/2"-20 (2)	1/8"-27 (1)	20.25	AE-838 *
1.65	1 1/2"	1 1/4"	5"	11 7/32"	5 3/32"	4 1 7/64"	11 7/32"	3 1/8"	1/2"-20 (1)	17.25	AE-837

*With 5/8"-18 threaded filler cap, use assembly No. AE33478



PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY: W. K. Schmitt

ORDER NO. _____

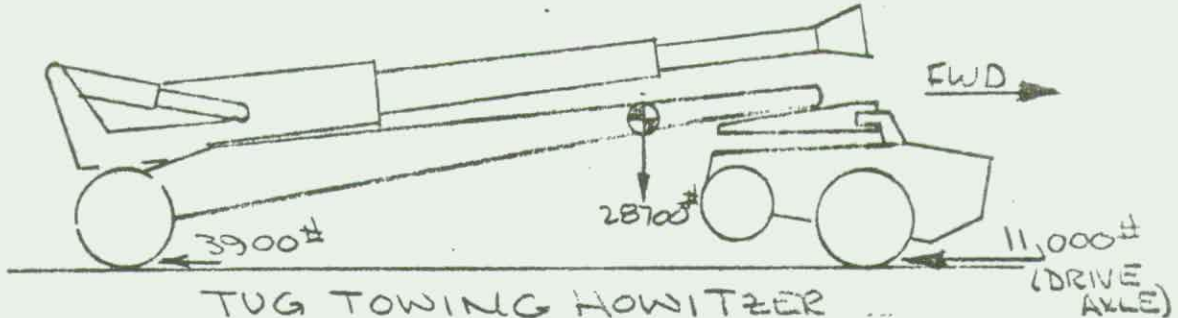
CHECKED BY: _____

PAGE _____ OF _____

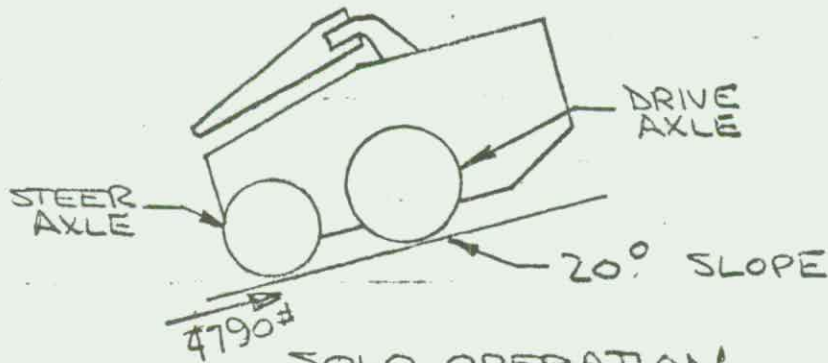
DATE: 9/20/73

REPORT NO. _____

BRAKING REQUIREMENTS



1. TUG TOWING HOWITZER ...
 TO STOP AT .5G FROM 35 MPH REQ'S
 BRAKE TORQUE (DRIVE AXLE) = $11,000 \# (19.3") = 213,000 \text{ LB-IN}$
 ($\mu = .6$, DIST = 70.7 FT)
 WHEELS - 19.5 DIA, 13.0 WIDE, 15° DROP CENTER
 MOUNTED ON ROCKWELL-STANDARD H-140 AXLE
 (WEIGHT TRANSFER TO DRIVE AXLE INCLUDED)



2. SOLO OPERATION
 TO HOLD TUG ON 20° SLOPE
 BRAKE TORQUE (STEER AXLE) = $4,790 \# (15.4") = 73,800 \text{ LB-IN}$
 WHEELS - 16.5 DIA, 9.75 WIDE, 15° DROP CENTER

BRAKES TO BE MOUNTED INSIDE WHEELS
 OPERATION BOTH ON & OFF-ROAD
 AIR OVER HYDRAULIC ACTUATION (100 PSI AIR MAX)
 (DRIVE AXLE BRAKES WILL NOT BE USED DURING
 SOLO OPERATION)

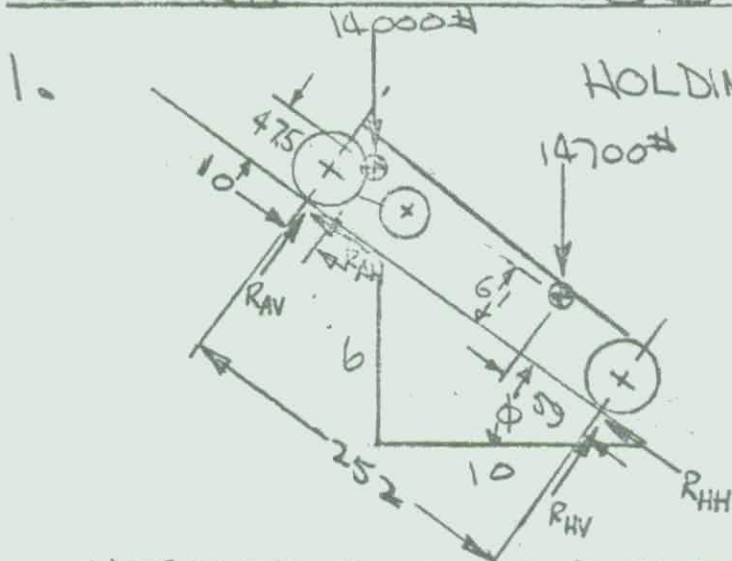
PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY WJK
 CHECKED BY _____
 DATE 5/12/73

ORDER NO. ATLAS
 PAGE 1 OF _____
 REPORT NO. _____

BRAKING LOADS



HOLDING UPHILL (60% SLOPE)

$$R_{HV} = 14,000 \cos \phi (10) + 14,000 \sin \phi (47.5) + 14,700 [\cos \phi (193) + \sin \phi (61)]$$

$$\frac{\quad}{252}$$

$$= 13,320 \# \leftarrow$$

NEGLECT ROLLING RESISTANCE

$$R_{AV} = 14,000 [\cos \phi (242) - \sin \phi (47.5)] + 14,700 [\cos \phi (59) - \sin \phi (61)]$$

$$\frac{\quad}{252}$$

$$= 11,290 \# \leftarrow$$

$$R_{HH} = R_{AV} \tan \phi$$

$$= 13,320 (0.6)$$

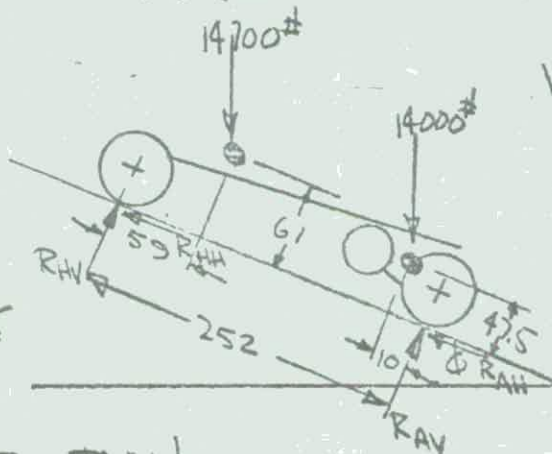
$$= 7,990 \# \leftarrow$$

$$R_{AH} = R_{AV} \tan \phi$$

$$= 11,290 (0.6)$$

$$= 6,770 \# \leftarrow$$

2.



HOLDING DOWNHILL (60% SLOPE)

$$R_{AV} = 14,700 [\cos \phi (59) + \sin \phi (61)] + 14,000 [\cos \phi (242) - \sin \phi (47.5)]$$

$$\frac{\quad}{252}$$

$$= 17,670 \# \leftarrow$$

NEGLECT ROLLING RESISTANCE

$$R_{AH} = R_{AV} \tan \phi$$

$$= 17,670 (0.6) = 10,600 \# \leftarrow$$

$$R_{HV} = 14,700 [\cos \phi (193) - \sin \phi (61)] + 14,000 [\cos \phi (10) - \sin \phi (47.5)]$$

$$\frac{\quad}{252}$$

$$= 6,940 \# \leftarrow$$

$$R_{HH} = R_{HV} \tan \phi$$

$$= 6,940 (0.6) = 4,165 \# \leftarrow$$

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY HJK

ORDER NO. ATLAS

CHECKED BY _____

PAGE 2 OF _____

DATE 9/12/73

REPORT NO. _____

HOLDING ON 60% SLOPE - WORST
BRAKE REQ FOR ATLAS IS
DOWN SLOPE - BRAKES MUST
HOLD 10,600#

BRAKE TORQUE REQD = FR
FOR 16-19.5 TIRE R = 19.3 IN

MAX BRAKE TORQUE REQD

$$= 10,600 (19.3) = \underline{\underline{205,000 \text{ LB-IN}}}$$

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY HJK

ORDER NO. _____

CHECKED BY _____

PAGE _____ OF _____

DATE 9/19/73

REPORT NO. _____

HOWITZER BRAKE CAPACITY

BF GOODRICH 419-174 CALIPER
(FIXED DISC - 4 PISTON)

$$\begin{aligned} \text{TANGENTIAL FORCE} &= 16000 \# @ 1500 \text{ PSI} \\ \text{BRAKING RADIUS} &= \sqrt{7.12^2 + 2.125^2} \\ &= 7.43 \end{aligned}$$

$$\begin{aligned} \text{MAX BRAKING TORQUE} &= 16,000 (7.43)(2) \\ &= 237,600 \text{ lb-in} \end{aligned}$$

$$\begin{aligned} \text{TO STOP @ .5G FROM 35 MPH} \\ \text{TAKES } .26 (14900) \times 19.3 &= 75000 \# \end{aligned}$$

BRAKES WILL LOCK UP - THIS IS UNSTABLE

HAVE TO LIMIT AIR PRESSURE TO
HOWITZER TO AVOID LOCK-UP
AIR-HYD BOOSTER RATIO = 15:1

AIR PRESSURE HAS TO BE LIMITED TO

$$\frac{75000}{237,600} \times \frac{1500}{15} = \underline{31.5 \text{ PSI}} \leftarrow$$

PREPARED BY S. BLACK

ORDER NO. ATLAS

CHECKED BY _____

PAGE 3 OF _____

DATE 9/12/73

REPORT NO. _____

3. STOP FROM 35 MPH @ .5G

ASSUMPTIONS:

1. MASS INCREASED BY 4% FOR ROTATING MEMBERS

$$MASS = 28,700 (1.04) / 32.2 = 926 \text{ SLUGS}$$

2. AIR RESISTANCE: NEGLECTED

3. ROLLING RESISTANCE: NEGLECTED

4. GRADE = 0

5. DYNAMIC WEIGHT TRANSFER = 0

6. ENGINE BRAKING POWER = 0

$$B_f = 16.2(m)$$

$$B_f \text{ (BRAKING FORCE @ } \frac{1}{2} \text{ "G")} = 16.2 (926) = 14,900 \text{ lb}_f$$

ASSUMING $\mu = .6$

$$S_{MIN} = \frac{1.04 (35)^2}{30 (.6)} = 70.7 \text{ ft}$$

AXLE LOADS

ATLAS 16,500[#]

CG HT = 54.7"

HOWITZER 12,200[#]

BRAKE FORCE DISTRIBUTION = $\frac{L_f + H(\mu + f)}{L_f + H(\mu + f)}$ $\mu = .6, f = .011$

L_f = DIST. REAR AXLE TO CG, $H = .217L$

$L_f = H(\mu + f)$

L_f = " FRONT "

L = WHEEL BASE = 252"

$$= \frac{.58 + .217(16 + .011)}{.42 + .217(.6 + .011)} = 2.74$$

204

74% ON ATLAS AXLE

BRAKE FORCE ON ATLAS = .74 (14,900) = 11,000

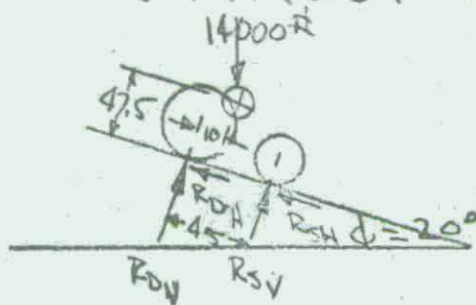
PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY HJK
 CHECKED BY _____
 DATE 9/12/73

ORDER NO. _____
 PAGE 4 OF _____
 REPORT NO. _____

4. TO HOLD ATLAS ON 20° SLOPE WITHOUT HOWITZER



$$R_{sv} = \frac{14000 [\cos \phi (10) + \sin \phi (47.5)]}{45}$$

$$= 7980 \# \leftarrow$$

(HEADED UPSLOPE)

FACED OPPOSITE WAY ON SLOPE

$$R_{sv} = \frac{14000 (\cos \phi (35) - \sin \phi (47.5))}{45} = 5160 \# \leftarrow$$

(HEADED DOWN)

$$\text{BRAKING FORCE REQD} = 14000 \sin 20^\circ$$

$$= 4790 \# \leftarrow$$

SO WITH ATLAS HEADED DOWNSLOPE
 COEFFICIENT BETWEEN GROUND &
 TIRES MUST BE $4790/5160$
 $\geq .93$

$$\text{UPSLOPE } 4790 \geq .6$$

$$\text{BRAKE TORQUE} = 4790 (15.4) = \underline{73,300 \text{ LB-IN}}$$

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY H.K. Kishner

ORDER NO. ATLAS

CHECKED BY _____

PAGE 5 OF _____

DATE 9/12/72

REPORT NO. _____

BRAKING REQ'S FOR ATLAS

1. TO HOLD ATLAS & HOWITZER ON 60% SLOPE (DOWNHILL)

DRIVE AXLE BRAKE TORQUE = 205,000 LB-IN
MINIMUM

STEER AXLE = 0

2. TO STOP ATLAS & HOWITZER FROM 35 MPH @ $> .5 G$

DRIVE AXLE BRAKE TORQUE = 213,000 LB-IN
(.6 BRAKING COEF.) AVE

DISTANCE - 70.7 FT

STEER AXLE = 0

3. HOLD ATLAS ON 20° SLOPE WITHOUT HOWITZER

STEER AXLE BRAKE TORQUE = 73,800 LB-IN
MINIMUM

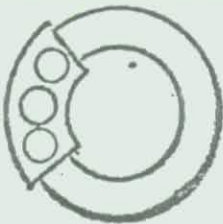
DISK BRAKES FOR PACIFIC CAR AND FOUNDRY COMPANY

CALCULATED PERFORMANCE

Date 10-16-732

WHEEL DRIVE Job No.

565Originator JCM

B. F. Goodrich Brake Series		REMARKS
DRIVE AXLE		
EMPTY WEIGHT	14,400 #	
LOADED WEIGHT	28,700 #	
TIRE SIZE AND TIRE RADIUS	16.5 x 19.5 19.3 In.	
MAX. SPEED LOADED	35 Mph.	
DISK SIZE AND WEIGHT	15.125 x 5 x 1 45 #	
BRAKE RADIUS IN INCHES	6.062 In.	
DIAMETER OF PISTONS AND NUMBER/VEHICLE	2-7/8 12	
TUG TORQUE LOADED	212,355 Lb-In.	SEE NOTE NO. 1
REQ. PRESSURE FOR STOP	1,215 PSI	SEE NOTE NO. 1
ACTUAL GROUND COEF. FOR STOP	.505	
MINIMUM STOP DISTANCE AT <u>1215</u> PSI.	96 Ft.	SEE NOTE NO. 2
MAX. GRADE HOLDING ABILITY AT <u>1215</u> PSI.	60 %	
NUMBER OF STOPS AT 500°F	12	SEE NOTE NO. 3
NUMBER OF STOPS AT 750°F	27	SEE NOTE NO. 3
DISK Δ T FOR LOADED STOP FROM MAX. SPEED	72.9 °F	

NOTES:

1. Brake Torque Requirements are Based on Stopping Loaded Vehicle from 35 mph, a .505G deceleration and with .3 Sec. System Delay.
2. Min. Stop Distance From 35 MPH with Loaded Vehicle and with .3 Sec. System Delay.
3. Stops Per Hour with a Loaded Vehicle from a Speed of 35 MPH without Exceeding the Disk Temperatures Shown.
4. Dynamic Lining Coefficient .37.

F. GOODRICH GROUND VEHICLE BRAKE ENGINEERING RECOMMENDATION

Job Number Revision _____ Job Number 553
Revision Date _____ Date 10-16-73

A. Axle Mfg. Rockwell Model H-140 Tire Size 16.5 x 19.5 R 19.3

B. Vehicle Mfg. Pacific Car and Foundry Model _____ Type Tug

Number of Driven Wheels 2
Empty Weight 14,400 Loaded Weight 28,700
Loaded Axle Weight Tug - 16,500; Howitzer 12,200
Empty Axle Weight Tug Front 9,600; Tug Rear 4,800

C. Special Application Considerations or Usage Conditions Or Limitations (if any)

D. B.F. Goodrich Components. List All; and Where Used on Vehicle _____
Use two (2) 419-213 brakes, one per wheel to be mounted on front axle of tug.

Pressure Limitations 1500 psi. If Equipment is Proposed and no drawings is available, describe by Type, Piston Size, Similar To, and Etc. _____

E. Brake Application is Approved.
Signed J.C. Moore W.L. Daniels
J.C. Moore

F. Brake Application is Approved With Qualifications as Listed Below: _____

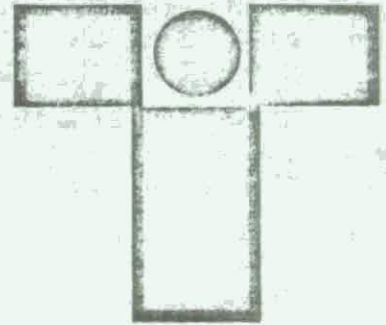
Signed _____

G. Brake Application is Not Approved Because - _____

H. Axle Manufacturer Approval (when applicable) Brake Application (is) (is not) approved. Comments _____
Signed _____

ZINK ENTERPRISES
P. O. Box 771
Bellevue, Washington 98004
Phone: 206-454-8174

Signed _____, Title _____



TOL-O-MATIC

December 17, 1973

Pacific Car & Foundry
1400 N. 4th St.,
Renton, WA 98055

Attn: Mr. H.G. Kirchner

Dear Mr. Kirchner,

In reference to our phone conversation today we recommend our H220 DXCIG caliper with a spacer for a 1" thick disc. Two such calipers/wheel.

As for the disc we recommend a 1" thick disc 12" in diameter. Such a disc will allow for proper heat dissipation and absorption.

Enclosed is the data sheet on the cast iron unit and our quotation on 400 units.

Yours truly,

W.C. Branham

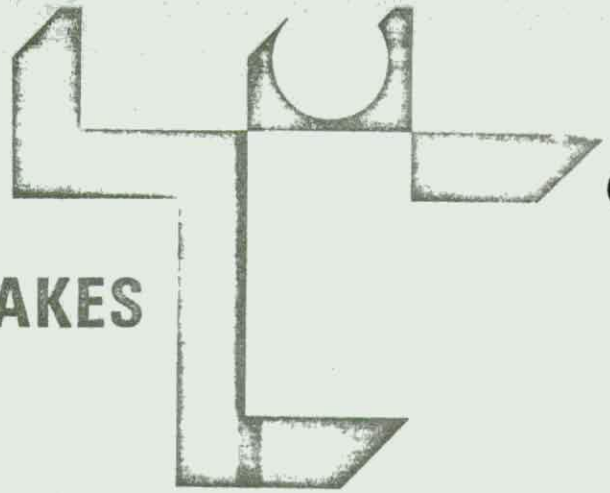
William C. Branham
Marketing Manager

WCB:gg

cc: Jack Ogle
M. Bress

TOL-O-MATIC

246 Tenth Avenue South
 Minneapolis, Minnesota 55415
 Telephone 612 335-6605



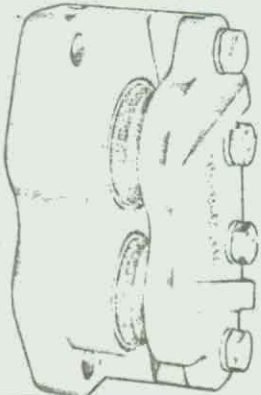
CAST IRON 220 SERIES HYDRAULIC CALIPER DISC BRAKES

For—Mining Equipment
 Mobile Off-the-Road Equipment

FEATURES:

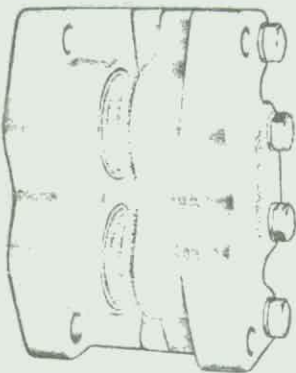
- CAST IRON HOUSINGS
- CHROMATE OVER CADMIUM RUST PREVENTIVE
- HIGH GRADE FRICTION MATERIAL
- NICKEL PLATED STEEL PISTONS
- BUNA "N" SEALS - STANDARD
 Buna "S" Seals (Available for Automotive Brake Fluids at no additional charge).
- GRADE 8 MOUNTING BOLTS, CADMIUM PLATED
- 220 SERIES PROVIDES 4 SQ. INCHES OF ACTIVE PISTON AREA AND 8 SQ. INCHES OF PUCK AREA

H 220SI



MODEL NUMBER	STOCK NUMBER	LIST PRICE	MODEL NUMBER	STOCK NUMBER	LIST PRICE	MODEL NUMBER	STOCK NUMBER	LIST PRICE
H220 SACI	0733-0301	46.86	H220 SBCI	0733-0401	49.02	H220 SECI	0733-0501	49.17
H220 SAFCI	0733-0321	55.38	H220 SBFCI	0733-0421	57.54	H220 SEFCI	0733-0521	57.69
H220 SARCI	0734-0311	47.61	H220 SBRCI	0734-0411	49.77	H220 SERCI	0734-0511	49.92
H220 SARFCI	0734-0331	57.06	H220 SBRFCI	0734-0431	59.22	H220 SERFCI	0734-0531	59.37

H 220DI



MODEL NUMBER	STOCK NUMBER	LIST PRICE	MODEL NUMBER	STOCK NUMBER	LIST PRICE	MODEL NUMBER	STOCK NUMBER	LIST PRICE
H220 DACI	0735-0401	64.00	H220 DBCI	0735-0501	64.00	H220 DECI	0735-0601	64.00
H220 DARCI	0736-0413	72.05	H220 DBRCI	0736-0511	72.05	H220 DERCI	0736-0611	72.05

LEGEND

- H - HYDRAULIC ACTUATION
- D - DOUBLE ACTING
- S - SINGLE ACTING
- A - "A" SPACER - 5/32" DISC
- B - "B" SPACER - 1/4" DISC
- R - RETRACTABLE PISTON(S)
- F - FLOATING MOUNTING BRACKET
- C - BLEEDER FITTINGS
- E - "E" SPACER 1/2" DISC
- G - BUNA "S" SEALS - (Must Specify if Required)
- I - CAST IRON CONSTRUCTION

H 220 SERIES TORQUE VERSUS PRESSURE

6 INCH DISC		8 INCH DISC		10 INCH DISC		12 INCH DISC		16 INCH DISC	
PSI	Torque - Inch Lbs.	PSI	Torque - Inch Lbs.	PSI	Torque - Inch Lbs.	PSI	Torque - Inch Lbs.	PSI	Torque - Inch Lbs.
100	685	100	907	100	1,184	100	1,463	100	2,076
200	1,371	200	1,814	200	2,367	200	2,926	200	4,153
500	3,427	500	4,536	500	5,918	500	7,315	500	10,382
1,000	6,854	1,000	9,072	1,000	11,837	1,000	14,630	1,000	20,765
1,500	10,282	1,500	13,608	1,500	17,755	1,500	21,946	1,500	31,147

ABOVE TORQUES BASED ON 288 LBS. OF FORCE PER 100 PSI X BRAKING RADIUS (INCHES)

(BRAKING RADII)

- 6 INCH DISC - 2.38
- 8 INCH DISC - 3.15
- 10 INCH DISC - 4.11
- 12 INCH DISC - 5.08
- 16 INCH DISC - 7.21

For Dimensional Data-See Other Side

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

PREPARED BY _____

ORDER NO. _____

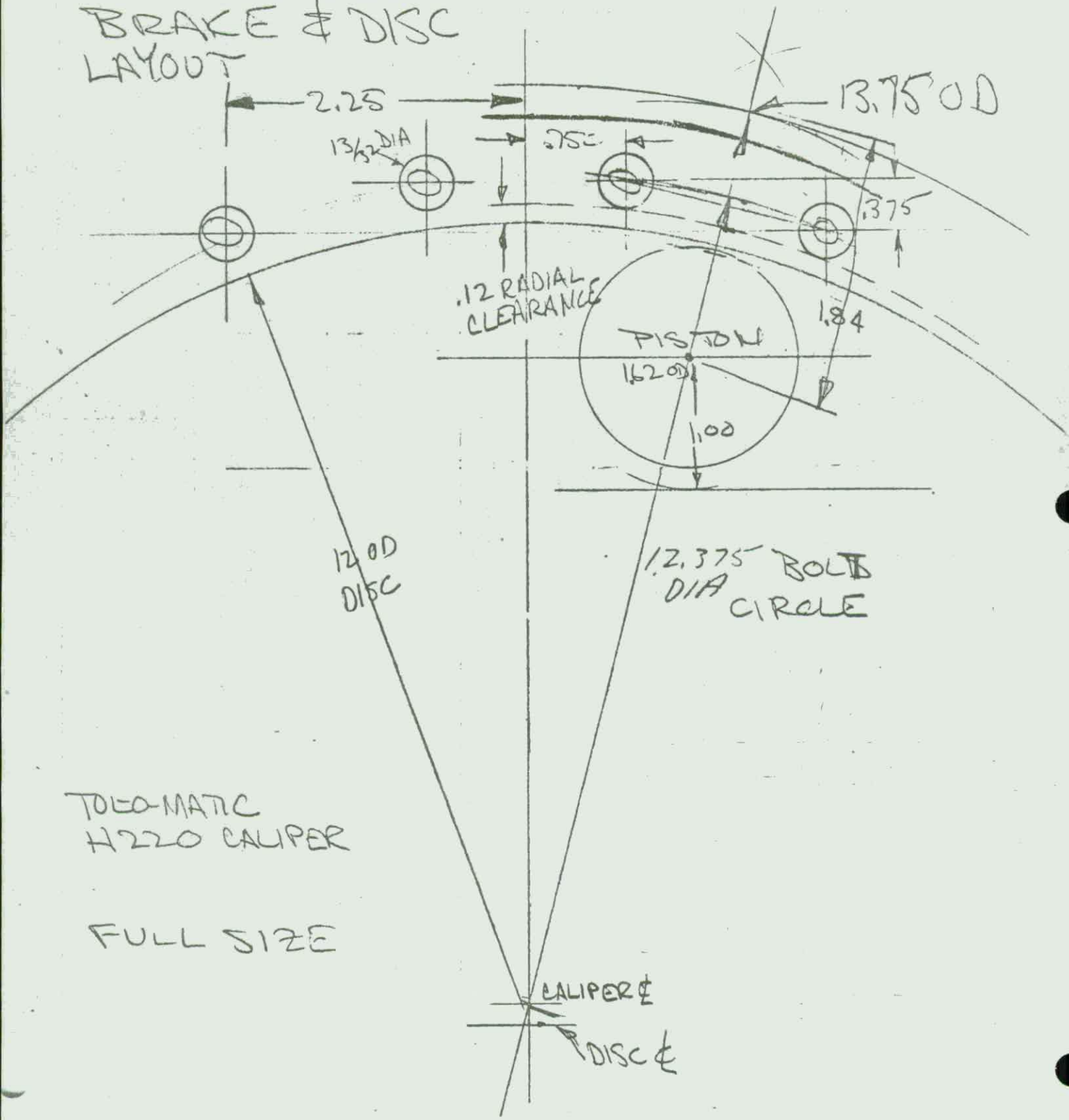
CHECKED BY _____

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DATE _____

REPORT NO. _____

BRAKE & DISC
LAYOUT



TOLE-MATIC
H220 CALIPER

FULL SIZE

CALIPER &

DISC &

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

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PAGE 1 OF _____

DATE 10/21/73

REPORT NO. _____

PARKING BRAKE

REQUIREMENT - HOLD ATLAS ON
20° SLOPE

BRAKING FORCE REQUIRED
4790#

FOR BRAKE ON DROP BOX OUTPUT

BRAKE TORQUE = $\frac{4790(19.3)}{8.2}$

= 11,300 LB-IN

MAX BRAKE DISC RADIUS = 5.5 IN

H-H PRODUCTS - KELSEY HAYES

MODEL 385 M W 10" DIA ROTOR

DEVELOPS ~14000 LB-IN TORQUE

WITH 310# PULL ON LEVER

(ORSCHERM BRAKE LEVER MAX PULL

IS 3900#)

Use B.F. Grolich instead
419-176

ON 4.90 BRAKE RADIUS (11.33 OD DISC)

@ MAX PULL = 1500#, TANG. BRAKE FORCE = 3000

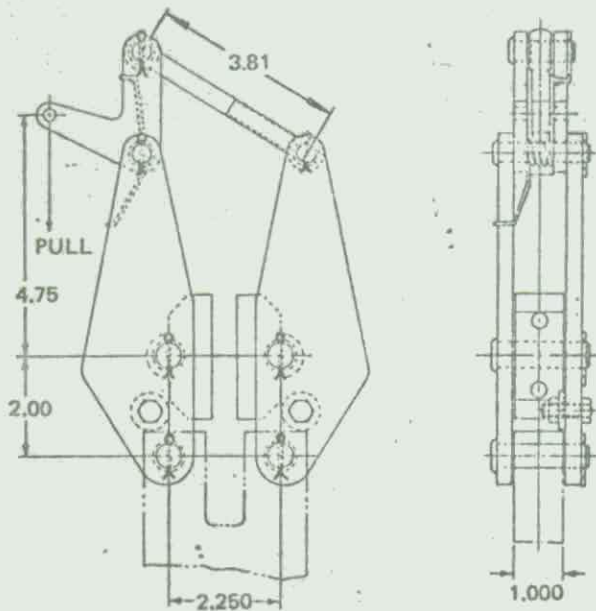
TORQUE = $3000(4.90) = \underline{14700 \text{ LB-IN}}$

BRAKE

MECHANICAL

BR-100-073

BR-100-073



BRAKE TYPE

Caliper disk, mechanically applied.
(Approximate weight = 5.5 lbs.)

RECOMMENDED USE

Parking, mount on drive shaft. Emergency use
at low energies.

LINING

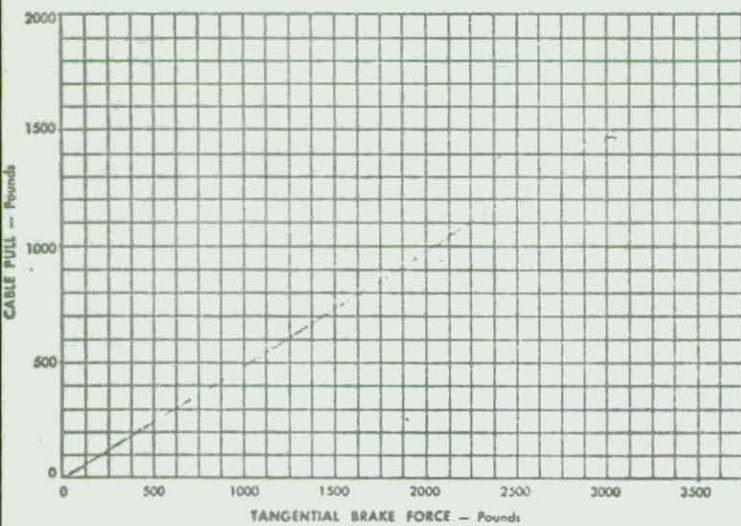
Organic, attached to lining carrier. Lining area
4 in.²

DISK

6" to 20" diameter. Thickness 1/2".

DESCRIPTION

Pin-mounted lining carriers retained by pins.
For use on all types of vehicles. Mounts ahead
of reduction. Linkage adjustable for lining
wear. Limit cable pull to 1500 lbs.



TORQUE FORMULA

Brake Radius = (.5 X Disk Dia.) - .62
Brake Torque = Tangential Brake Force X
Brake Radius

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

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W. K. Schmees

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2-20-74

REPORT NO. _____

ATLAS CRANE

ATLAS WILL USE A MODIFIED BA-2 WORKHORSE CRANE BUILT BY AUTOCRANE COMPANY. BECAUSE OF SPACE LIMITATIONS THE COMPLETE UNIT CANT BE USED. THE HOIST WINCH IS REMOVED AND MOUNTED INSIDE THE VEHICLE. A SPECIAL BASE WITH MOUNTING FOR A HYDRAULIC CYLINDER IS ADDED SO THAT THE BOOM CAN BE RAISED BY A HYDRAULIC CYLINDER. THE STANDARD BOOM IS USED WITH A CLEVIS ADDED TO CONNECT THE HYDRAULIC CYLINDER. (THE AUTOCRANE HYDRAULIC HBA-2 IS ALSO TOO LARGE.) THE STANDARD 5'4" TO 9' TELESCOPING BOOM IS USED WITH THE LOAD SENSOR KIT ADDED. THE LOAD SENSOR SHUTS OFF THE HOIST WINCH IF TOO BIG A LOAD IS LIFTED.

THE HYDRAULIC CYLINDER IS SINGLE ACTING. DEPENDING ON THE METERING ABILITY OF THE CONTROL VALVE IT MAY BE NECESSARY TO ADD RESTRICTORS TO THE CYLINDER TO CONTROL RAISE & LOWER SPEED OF THE BOOM. IT MAY ALSO BE DESIROUS TO CONTROL THE HYDRAULIC CYLINDER WITH THE LOAD SENSOR.

THE BOOM CAPACITY WILL BE GREATER THAN LISTED BY THE MANUFACTURER, SINCE THE ADDITION OF THE HYDRAULIC CYLINDER REDUCES THE CANTILEVERED LENGTH OF THE BOOM.

PACIFIC CAR AND FOUNDRY COMPANY

ENGINEERING DEPARTMENT

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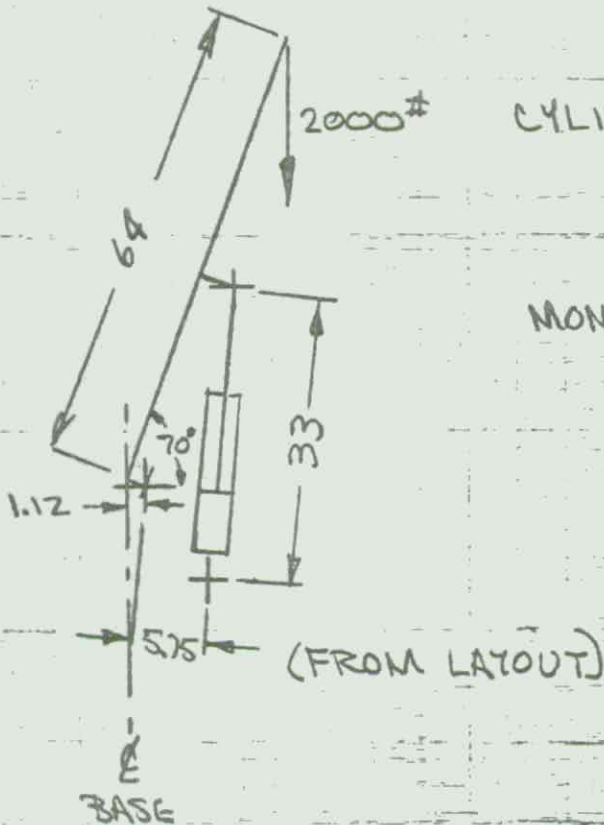
PAGE 2 OF _____

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CRANE LOADS

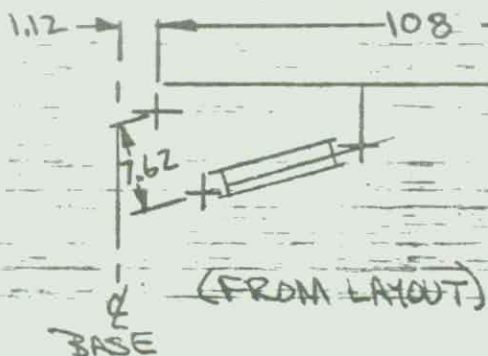
MAX LOAD - 2000# @ 70° ELEV, 5' 4" BOOM LENGTH



$$\begin{aligned} \text{CYLINDER FORCE} &= \frac{2000 \sin 70^\circ (64)}{5.75} \\ &= 7620 \# \leftarrow \end{aligned}$$

$$\begin{aligned} \text{MOMENT AROUND BASE } \& \\ &= 2000 [64 \sin 70^\circ + 1.12] \\ &= 46,100 \text{ LB-IN } \leftarrow \end{aligned}$$

MAX LOAD - 600# @ 0° ELEV, 9' BOOM LENGTH



$$\begin{aligned} \text{CYL FORCE} &= \frac{600 (108)}{7.62} \\ &= 8500 \# \leftarrow \end{aligned}$$

$$\begin{aligned} \text{MOMENT AROUND BASE } \& \\ &= 600 (108 + 1.12) \\ &= 65,500 \text{ LB-IN } \leftarrow \end{aligned}$$

PACIFIC CAR AND FOUNDRY COMPANY

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CYLINDER SIZE

ASSUMING $\mu = .25$, TORQUE ON BOOM
PIVOT PIN $\approx .31(8500\#) = 2660 \text{ LB-IN}$
(.625 DIA)

$$\text{ADDED CYL FORCE REQ'D} = \frac{2660}{7.62} = 350\#$$

FOR 2.5 IN BORE CYL

$$\text{PRESS REQ'D} = \frac{8850}{2.5^2(\pi/4)} = 1800 \text{ psi}$$

RELIEF VALVE SETTING IS 2000 psi

OK

FROM LAYOUT

MAX EXTENDED LENGTH = 33 IN

FOR BRUNING TYPE 3 DEZCTO CYL

MAX LENGTH = 8.88 + 2xSTROKE
(W/RE-1 ROD END)

$$\text{STROKE} = \frac{33 - 8.88}{2} = 12.06$$

USE 12.00 ←

(W/BOOM RESTING ON PROJECTILES MIN LENGTH
IS 26.88, OR 6.12 STROKE SO CYL IS OK)

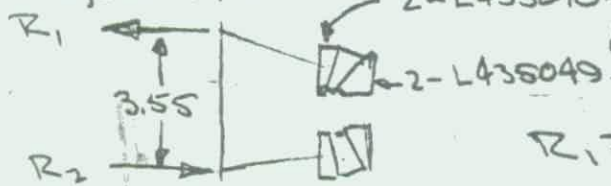
USE BRUNING CYL 3-2.5-12.00-RE1 ←

KINGPOST BEARINGS

AT HORIZONTAL POSITION MOMENT IS MAX AND IS

ONLY APPLIED LOAD (600# IS NEGLIGIBLE)

65,500 LB W



$R_1 = R_2$, NO THRUST

$\therefore R_{REQ} = R_1$ (TINKEN ENG JOURNAL)

$$R_1 = \frac{65,500}{3.55} = 18,450 \#$$

PACIFIC CAR AND FOUNDRY COMPANY

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FOR SLOWLY OSCILLATING BEARING

R_E CAN BE AS HIGH AS $4 \times BRR$

FOR THIS BEARING $BRR = 6950 \#$

$$\frac{R_E}{BRR} = \frac{18,450}{6950} = 2.66$$

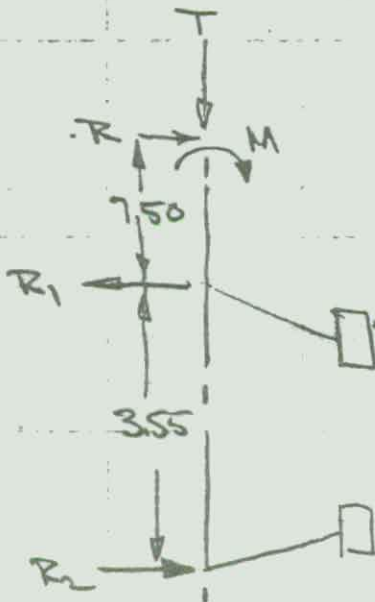
(PLENTY OF CAPACITY)

AT 70° ELEVATION

MOMENT = 46,100 LB-IN

THRUST LOAD = $2000 (\cos 70^\circ) = 1880 \#$

RADIAL LOAD = $2000 (\sin 70^\circ) = 685 \#$



$$R_1 = \frac{M + 11.05R}{355} = \frac{46100 + 11.05(685)}{355} = 15,120 \#$$

$$R_2 = R_1 - R = 14,430 \#$$

(FROM TIMKEN JOURNAL)

$$R_{E1} = .53R_1 + .47R_2 + KT$$

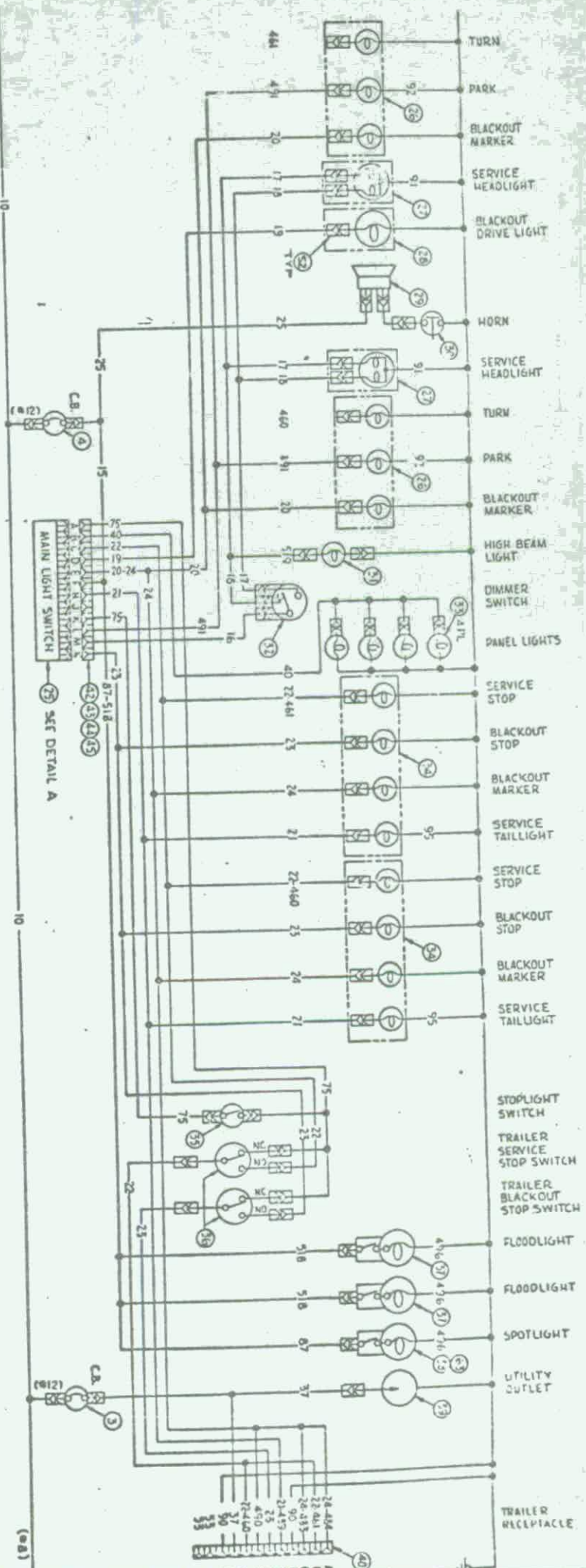
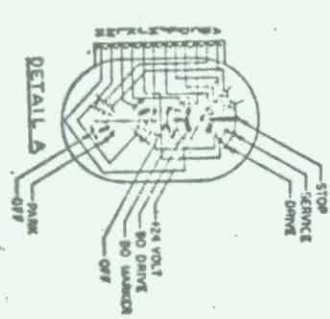
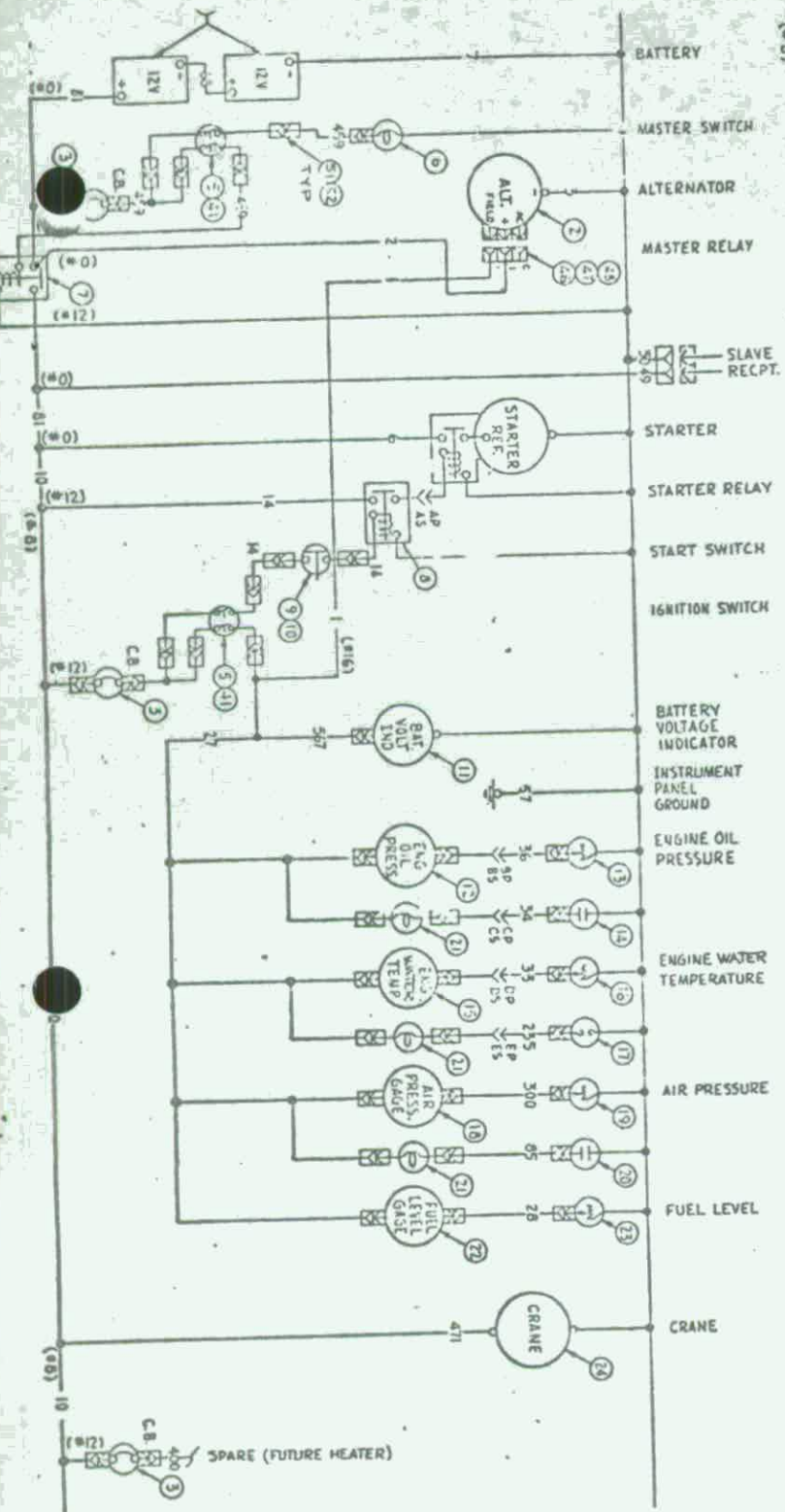
$$= .53(15,120) + .47(14,430)$$

$$+ 1.55(1880)$$

$$= 17,620 \#$$

$$R_{E2} = R_2 = 14,430 \#$$

$$R_{E1} / BRR = \frac{17620}{6950} = 2.54 \quad \leftarrow \quad \underline{OK}$$



ATLAS

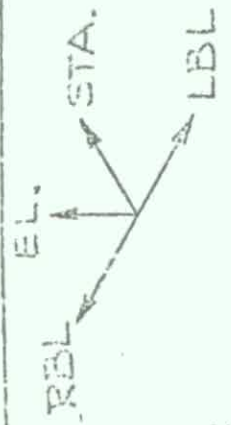
1/31/77

C. G.

MOMENTS

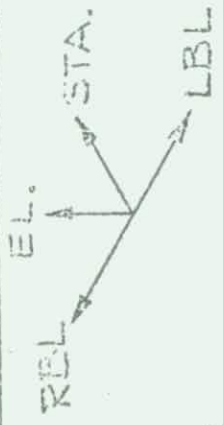
Wt.	STA	E	BL	M _{GRA}	M _{EL}	M _{BL}
9446	103.9	44.2	4.3	981,585	417,778	45,269
4325	131.7	46.0	-5.6	569,441	199,106	-24,181
13,771	112.6	44.8	1.5	1,551,026	616,884	21,088

Suspension & Drive System from Trucks.



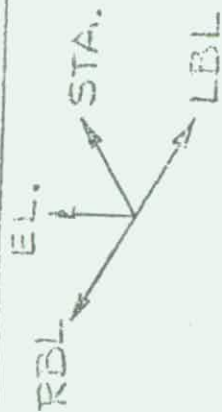
COMPONENT	QUAN.	WEIGHT	STA.	EL.	RBL / LBL	W(X)	W(Y)	W(RBL) / W(LBL)
		W	X	Y				
Front assy								
Shocks	2	30	90	28	0	2700	840	0
Torsion bars	2	131	76	27	0	9956	3537	0
2.3 x (5x25 + 4x.58) 2.25								
Suspension arms (End caps)	4	103	84	24	0	9072	2592	0
Pinhard buds	1	12	88	19.3	0	1056	232	0
201.25x47 Pin	1	67	75	27	0	5025	1809	0
Torsion Bar covers								
Brakes	2	84 2x39	100	21.5	0	16200	3423	0
TORSION BAR ANCHORS	4	NE 201.25	76	26	0	6080	2080	0
Rear assy								
Spring	2	14	148	24.5	0	2072	343	0
Suspension arms	2	14	137	18	0	1918	252	0
Pinhard buds	1	5	139	17.5	0	645	88	0
Shocks	2	20	145	23.5	0	2900	470	0
Brakes	2	28 + 30 + 30	145.5	15.4	0	21243	2248	0
Tire rods	1	8	139.5	15.4	0	1116	120	0

Air & Hydraulic



EH:	COMPONENT		QUAN.	WEIGHT W	STA. X	EL. Y	RBL /ALBL	W(X)	W(Y)	W(RBL) /W(LBL)
	RBL	STA.								
	Hydraulic Reservoir		1	8	147	39	-17.5	23,373	6201	-2783
	Hydraulic Pump		1	30	140	23	12.5	1184	184	100
	Rever. Take off		1	8	143	26	12	4290	780	360
	Hyd Pump(s)		1	4.3	135.5	46.8	9.8	1084	374	78
	Hyd Filter		1	5	132.5	51	-16.3	570	219	-70
227	Air/1 injector		1	2	86	63	19	430	315	95
	Water Ejector		1	2	101.3	38.5	19	203	77	38
	Source Cluster (Air/Hyd)		1	25	122	24.5	17	3050	613	425
	Air compressor		1	55	105.5	47.5	-17	5803	2613	-935
	Air reservoir		2	44	89.5	53.2	14	3938	2341	616
	Valve Body		1	50	125.5	34.25	17.5	6215	1713	875
	Valves (D)		2	20	131.5	43	17	3682	1204	476
			1	14	117	47.3	15.5	1638	662	217
	PIPINGS		1	10	126	57	0	1250	570	0
	Rotary Actuator		1	350	123	89	0	43,050	31,150	0

Misc.



COMPONENT	QUAN	WEIGHT		STA.	EL.	RBL / LBL		W(X)	W(Y)	W(RBL) / W(LBL)
		W	W	X	Y	RBL	LBL			
Winch Assembly	1	200		107.5	49	17		21,500	9800	3400
Winch Front (with 1/2 cable)	1	207		67	36	0		13,869	7452	0
CAGE (for both beams both)	1	200		130	56	36.75		26,000	7350	7350
Paint, Touring	1	522		141.5708	65.353	0		73,900	39,130	0
Roll Cage	1	198		65	98	32		12,870	19,404	6336
Front Powder Bin	1	80		44.2	65.4	0		3536	5232	0
CROSS BEAM Detail	1	180		106	78.25	0		19,080	14,085	0
S. Frame Detail	1	393		83	44	0		35,024	17,512	0

P.C.F. SALES NO. _____

CONTRACT NO. _____

FILE NO. (013)

DRAWING SERIES 8019-XXX

DRAWN BY	DRAWING NO.	SIZE	SHT'S	TITLE	INDEX PAGE
	1				
	-3	R	1	DRIVE AXLE BRAKE LAYOUT	
	-4	R	1	STEER AXLE BRAKE LAYOUT	
	-5	R	1	STEER AXLE LAYOUT	
	-6	K	1	TRANSFER CASE LAYOUT (CONVERTED TO 8019-216)	
	-7	K	1	ROLL CAGE CONCEPT LAYOUT	
	-8	K	1	SUSPENSION LAYOUT	
	-9				
	-10	K	4	CLASS & DV. - OVER ALL VEHICLE LAYOUT	

BY	WING NO.	TYPE	TITLE	NEXT ASSY.
	100	K	F	
	101	K	SIDE FRAME	
	102	D	SIDE FRAME, ARCH	
	103	D	BEAM, ARCH	
	104	D	BOTTOM ASSY	
	105			
RF	106	C	KING PIN	
RF	107	B	PIN, DRIVE	
	109	B	TARGET WASHERS	
	110	B	BUSHINGS, KING PIN	
	111	C	ADAPTER BOLT	
	112	K	BOLTING CYLINDER AND WASHER	
	113	B	STUB SHOCK MOUNTING	
	114	D	TRAILER ARM	
	115	B	BRACKET, RADIATOR MOUNT	
	116	C	BRACKET, RADIATOR MOUNT	
SA	117	D	SIDE BUSHING	
	118	C	BRACKET, SHOCK MOUNT	
HK	119	B	BOLT, FRAME	
	120	C	MOUNT FRONT ENGINE	
	121	C	BRACKET, FUEL TANK	
HK	122	R	FRONT BOOM LATCH LAYOUT	
RF	123	B	WASHER	
DM	124	C	BRACKET	

DRAWN BY	DRAWING NO.	REV	SHT'S	TITLE
	-200	K	1	POWERTRAIN LIST
	-201			TRANSFER CASE
				RADIATOR
	-204	D		SHROUD, RADIATOR
	-205	D		SUPPORT, RADIATOR
	-206	B		SUPPORT, RADIATOR
	-207	B		ADAPTOR (OUTLET)
	-208	C		TUBE (INLET)
	-209	C		TUBE BYPASS
	-210	C		ANGLE (RADIATOR)
	-211	B		FUEL TANK ASSY
	-212	K		HOUSING TRANSFER CASE
	-213	F		HOUSING TRANSFER CASE
	-214	F		HOUSING ASSY (MACHINE)
	-215	F		TRANSFER CASE ASSY
	-216	K	1	SPLINE, ADAPTER
	-217	D	1	NUT
	-218	C	1	FORK, SHIFT
	-219	D	1	SPLINE, FLUTING
	-220	D	1	SHIFTER ASSY
	-221	D	1	

DRAWN BY	DRAWING NO.	SIZE	SHT'S	TITLE	NEXT ASSY.
RE	-300	K	1	SUSPENSION INSTL	
AK	-301	D		DRIVE AXLE	
RF	-302	D		TORSION BAR	
RSE	-303	D		SUSPENSION ARM, DRIVE AXLE	
SA	-304	C		SUSPENSION ARM, STEER AXLE	
RF	-305	D		PANHARD ROD ASSY, DRIVE AXLE	
RF	-306	C		TORSION BAR ANCHOR, FRAME, DRIVE AXLE	
RF	-307	D		TORSION BAR ANCHOR, ARM, DRIVE AXLE	
RF	-308	E		BUSHING	
RF	-309	E		BUSHING	
	-310	B		WASHER, THRUST	
	-311	F		COVER, TORSION BAR	
RF	-312	B		SPACER	
AK	-313	D		STEER AXLE ASSY	
RF	-314	C		SPRING	
SA	-315	C		TIEROD	
	-316	C		PANHARD ROD ASSY, STEER AXLE	
RF	-317	B		PIN, SPRING BRACKET	
RF	-318	B		BRACKET, SPRING	
AK	-319	B		PAD, BUMP STOP	
RF	-320			SLEEVE	
	-321			WASHER	
DM	+322	B		SPACER	
DM	-323	B		PLUG	

