:36:32 OCA PAD AMENDMENT - PROJECT HEADER INFORMATION 11/22/93

Active

oject #: E-25-585 Cost share #: Rev #: 4
nter # : 10/11-6-P5038-6A0 Center shr #: OCA file #:
Work type : INST
ntract#: LTR DTD 920807 Mod #: ADM. REVISION Document : AGR
ime #: Contract entity: GTRC

bprojects ? : N CFDA: N/A
in project #: PE #: N/A

oject unit: MECH ENGR Unit code: 02.010.126
oject director(s):
BRAZELL J W MECH ENGR (404)894-3218

onsor/division names: UNIV SPACE RESEARCH ASSOC /
onsor/division codes: 500 / 037

ard period: 920616 to 940730 (performance) 940730 (reports)

onsor amount	New this change	Total to date
Contract value	12,688.64	47,388.64
Funded	12,688.64	47,388.64
st sharing amount		0.00

es subcontracting plan apply ?: N

itle: NASA/USRA UNIVERSITY ADVANCED DESIGN PROGRAM

PROJECT ADMINISTRATION DATA

A contact: Jacquelyn L. Tyndall 894-4820

Sponsor technical contact	Sponsor issuing office
ICKIE S. JOHNSON	SUE MCCOWN
713)244-2000	(713)244-2000
ASA/USRA ADVANCE DESIGN PROGRAM	UNIVERSITIES SPACE RESEARCH ASSOC.
600 BAY AREA BOULEVARD	SENIOR PROJECT ADMINISTRATOR
OUSTON, TX 77058-1113	ADVANCED DESIGN PROGRAM
	3600 BAY AREA BOULEVARD
	HOUSTON, TX 77058-1113

curity class (U,C,S,TS) : U ONR resident rep. is ACO (Y/N): N
fense priority rating : N/A N/A supplemental sheet
uipment title vests with: Sponsor X GIT
PRIOR SPONSOR APPROVAL REQUIRED FOR EQUIPMENT PURCHASES OVER $1,000.
ministrative comments -
ISSUED TO ADD FUNDS IN THE AMOUNT OF $12,688.64. FUNDS WERE TRANSFERRED FROM PROJECT E-25-546.

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ject Title NASA/USRA UNIVERSITY ADVANCED DESIGN PRO     Status T             |
rd Period: From 16-JUN-1992  To 30-JUL-1994                                  |
I BRAZELL                JAMES          Sponsor UNIV SPACE RESEARCH ASSOC/   |
tract No LTR DTD 920807                                                      |
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E25-28?

17A+4

ENABLER: A CONCEPT MODEL FOR A LUNAR WORK VEHICLE

Georgia Institute of Technology
Mechanical Engineering
Atlanta, Georgia

James Brazell

Students at The Georgia Institute of Technology designed an earth-bound concept model for a lunar work vehicle with all-terrain capabilities. This vehicle, named "Enabler," features a tubular chassis with rotating, non-orthogonal joints. The body consists of two cylindrical T-sections, a center cylindrical cross, and a pair of composite wheels attached to each element. A boom, capable of reaching well below the wheel plane into a crevasse or crater, extends from the center section and provides balance when the vehicle traverses steep slopes.

An equal-size static model of the vehicle was exhibited in March at the National Design Engineering Show. Of 42 schools participating in the program, only Georgia Tech was honored with an invitation to the show. In this exhibit, the vehicle was flanked with two exhibit areas, one with a video solid model animation of Enabler's maneuvers, and another supporting the man-machine interface. This exhibit was located in Design News Magazine's booth at their invitation; attendees expressed interest in both the exhibit and in the Design News article "Mobility on the Moon."

During its design process, the Enabler project was the subject of a masters thesis by Georgia Tech student Clemens Saur. The thesis, "Evaluation of Manipulator Boom Design for Lunar Vehicles," was published in March 1993.

With the design stages completed, the Enabler is being transferred from a concept into a working machine. A major portion of the chassis elements have been completed, and the rest are presently being fabricated. The current goal is to have the chassis assembled and functioning mechanically with a preliminary control system by June 1993.

Four student teams are designing and constructing machines for a June 1993 "in house" competition using the rules from the First Lunar Shelter Student Contest competition. This international competition is being sponsored by the American Society of Civil Engineers and will be held in conjunction with Space '94 in Albuquerque. This class effort, while limited in time and scope, will establish a baseline design for an entry in the international competition.

ENABLER POWER SYSTEM

ME 4192

SPRING '93

RICK CANFIELD
JOHN CHAMLEE
MICHAEL JOHNSON

ENABLER POWER SYSTEM

ME 4192

SPRING '93

RICK CANFIELD

JOHN CHAMLEE

MICHAEL JOHNSON

Introduction

As a note to the reader, it is advised that the reader first read the Final Report for the Winter '93 ME4182 Power Unit Group. Many of our components are contained in this report and only slight model and configuration changes have been made to the bulk of the selected components. (a copy of the 4182 report can be found in Appendix D of this report)

Engine

The engine selected, as stated in last quarters report, was a Briggs and Stratton model 130202 type 0015. This engine has 5 Hp and will soon be converted to propane power. The engine will be placed on a engine mount bracket on the rear, "cross" section of the enabler and can be seen in the accompanying drawings.

Flexible Coupling

The flexible couping serves as a semi-rigid connecton between the shaft of the engine and pump. The coupling chosen is manufactured by Lovejoy style number L075, 3/4" for the motor and 7/8" for the pump. Both sides of the coupling will be separated by a Hytrel polymeric spider, Grainger stock number 1A924. A side view of this coupling can be seen in the drawing titiled "Sectional View of Tail".

Reservoir

The hydraulic reservoir for the power unit of the Enabler vehicle is to be custom made. The body is a stainless steel beaker from Cole Parmer, catalog number L-07206-80. The beaker is made from medium gauge 304 stainless steel. It has an 8 qt. capacity and it is 8 3/4" in diameter and is 9 5/8" tall. The beaker has a 1/4" lip with no pourout.

The faceplate of the reservoir is to be made from an aluminum block 10 3/4" X 10 3/4" X 1/2". A circular groove 8 7/8" in diameter 1/8" deep and 1/8" wide is to be machined from the center of the plate. This groove will hold a rubber O-ring which will seal the faceplate to the beaker. One inlet hole 1/2" in diameter and one outlet hole 1/2" in diameter are drilled at the bottom. A vent hole of 1/2" in diameter is placed at the top center of the plate. This will hold the pressure relief / vacuum breaker. Two holes 3 3/4" apart and connected by a clear plastic tube will serve as a fluid level indicator. 8 bolt holes 1/8" in diameter are placed on a circle with a diameter of 9 1/8" from the center of the plate. These bolts will hold the plate to the reservoir by bolting to a retaining ring, 1/2" thick with an outside diameter of 9 1/4" and an inside diameter of 8 3/4". This retaining ring is also to be made of aluminum.

The inside of the reservoir contains three baffles placed vertically. These allow the fluid to slow down and for the air to be removed. Also, the inside of the reservoir contains a 2" square of 100 mesh wire screen. This screen helps remove air from the fluid.

Pump

The pump is the Vickers PVB5 pressure compensated variable displacement piston pump. It is similsr to the Hagglands-Dennison PV6. But it only pumps 5 gpm as opposed to 6 gpm at 1800 rpm. If the pump is run at 3600 rpm (speed at which the engine produces the most power) an inlet pressure 8 psi is required. The reservoir will produce a pressure of 3 psi after the system has warmed up. This will allow the pump to be run at about 3200 rpm.

Vacuum Breaker

The vaccum breaker is available from Hycon inc. The model number is ELF 3RV- 40 - 6 . This pressure breather has a relief pressure of 6 psi and a reseat pressure of 3 psi.

Filter

The hydraulic filter to be used is a low pressure return line filter and is available from Parker Fluid products through Grainger. It has a 10 micron filter element. The filter can handle up to 20 gallons per minuet. The filter has two ports that are 3/4" in diameter and will require fittings for these. The model number is 12AT10CN15BBLI and the Grainger catalog number is 4Z618. The price is $16.83

VALVES

The selection of the valves was conducted taking into account all the parameters required by our system. Equipment required by the updated design are listed in Appendix C under Valve Request Form, and Figures detailing mounting are included in Appendix A. Two proportional, three position valves (MEV6SLFHA3.2/3.2 12) in conjunction with two cartridge valves (DS102CD012P4P) will be used to operate the wheel drive. The cartridge valves will be supported by the two proportional valve mounting and will serve to allow each side of the Enabler to lock wheels or to allow free wheel (the valves will be normally closed, locked, and activated to open, free wheel). The proportional valves allow ease of motion and varying speed control (mounting plates included in order).

Three of the joint motors will be controlled via two three position valves (D1VW1CP10K). The three valves will be supported by a three station manifold (MANIFOLD 1-1112-03) and a plate. The plate was designed to allow secure mounting of the manifold. This setup will also represent the other three joint motors in the opposite end of the Enabler.

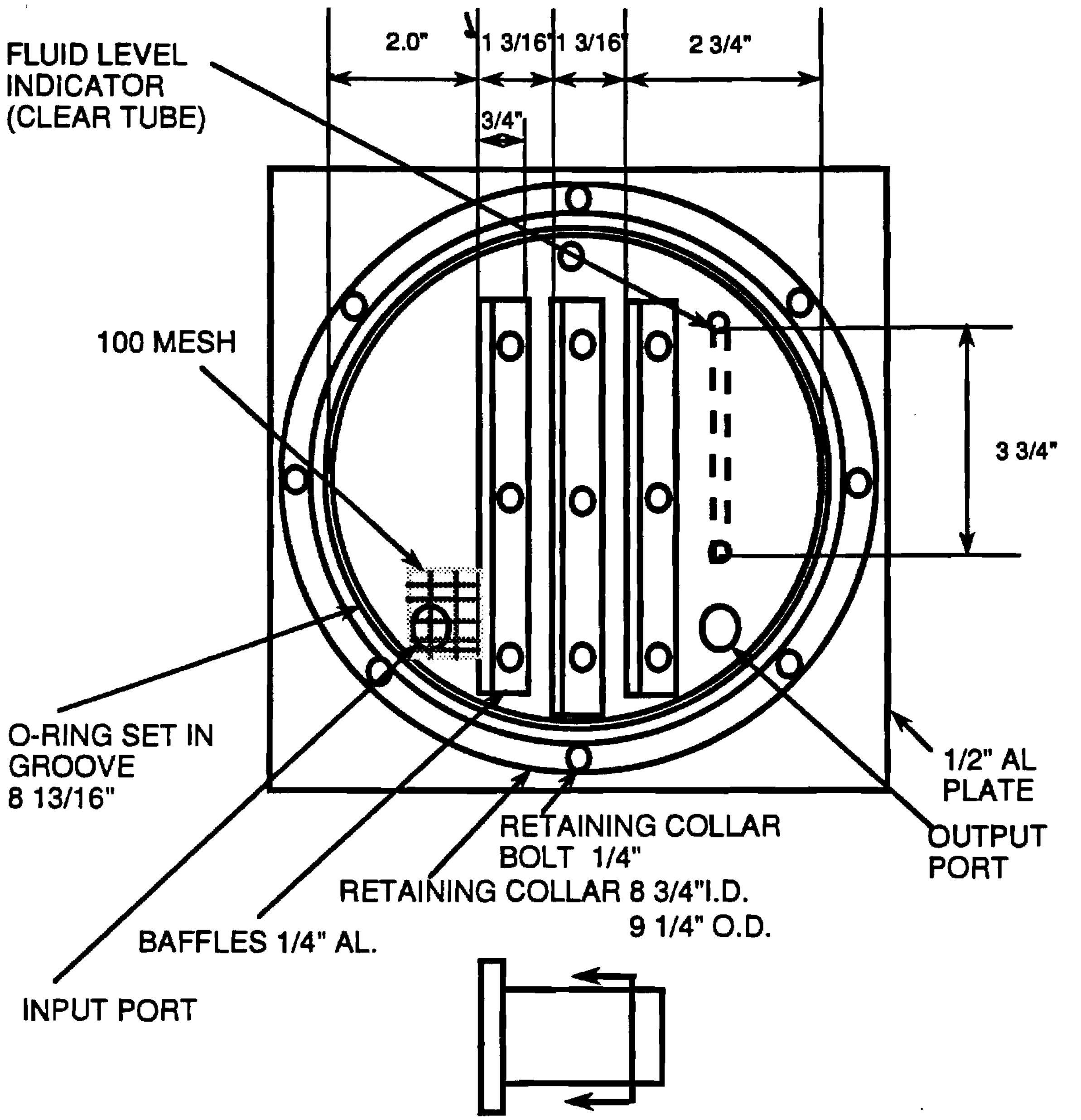
LINES

The hose selection remains the same (4182 Power Group - Appendix D) with variance in hose size at certain locations, and the number of adapters and female-swivel ends. The brand hose will be Wheatherhead, type H104, in sizes of 1/4", 3/8", and 1/2". The adapters and ends will be Aeroquip brand. A complete material list (Line Request Form - Appendix C) and line design (Appendix A) are included. The original hose design of coiling the hydraulic hose through the body of the Enabler will still be used with two coils per each rotating section.

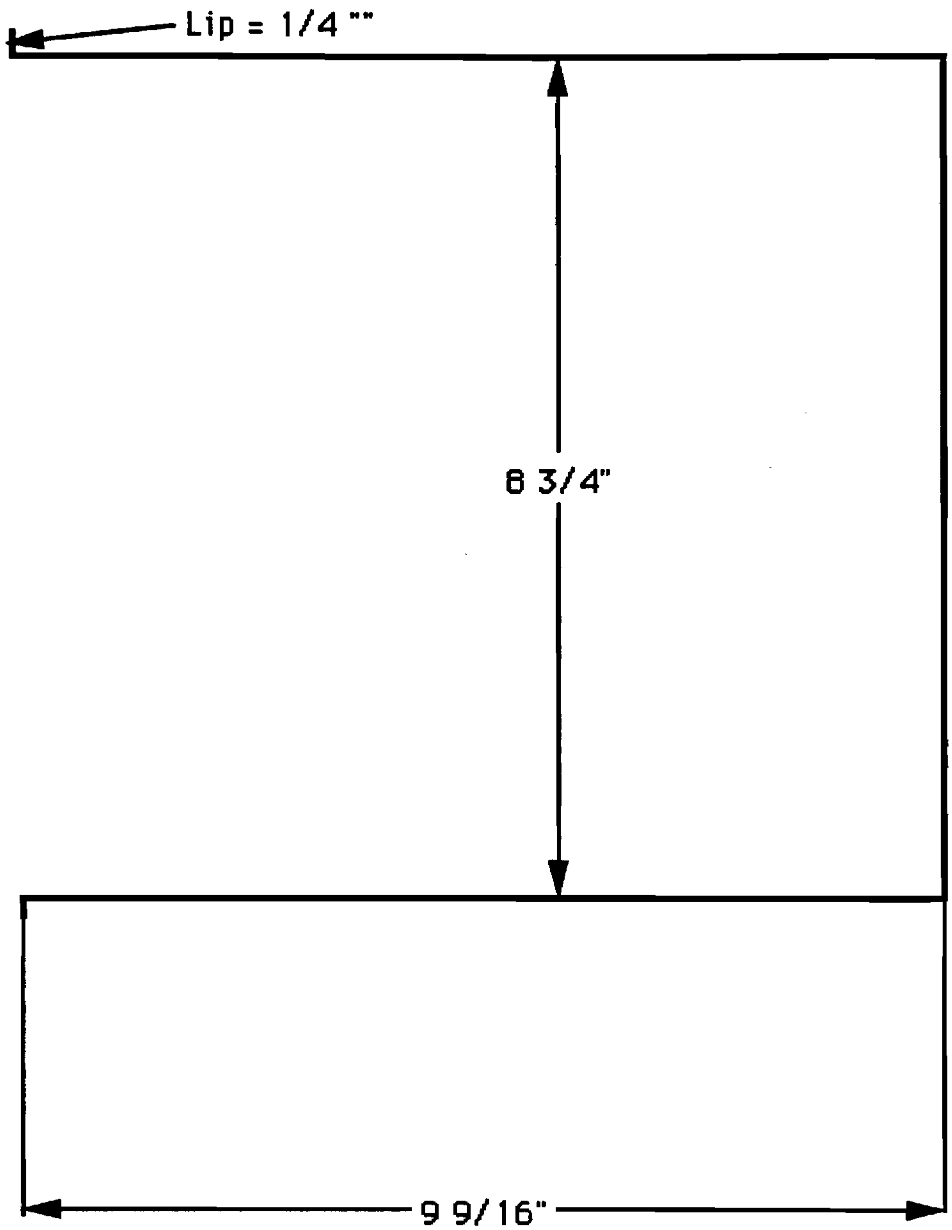
Appendix A
Detailed Drawings by Group Members

Reservoir Drawings

RESERVOIR DETAIL
SECTIONAL VIEW

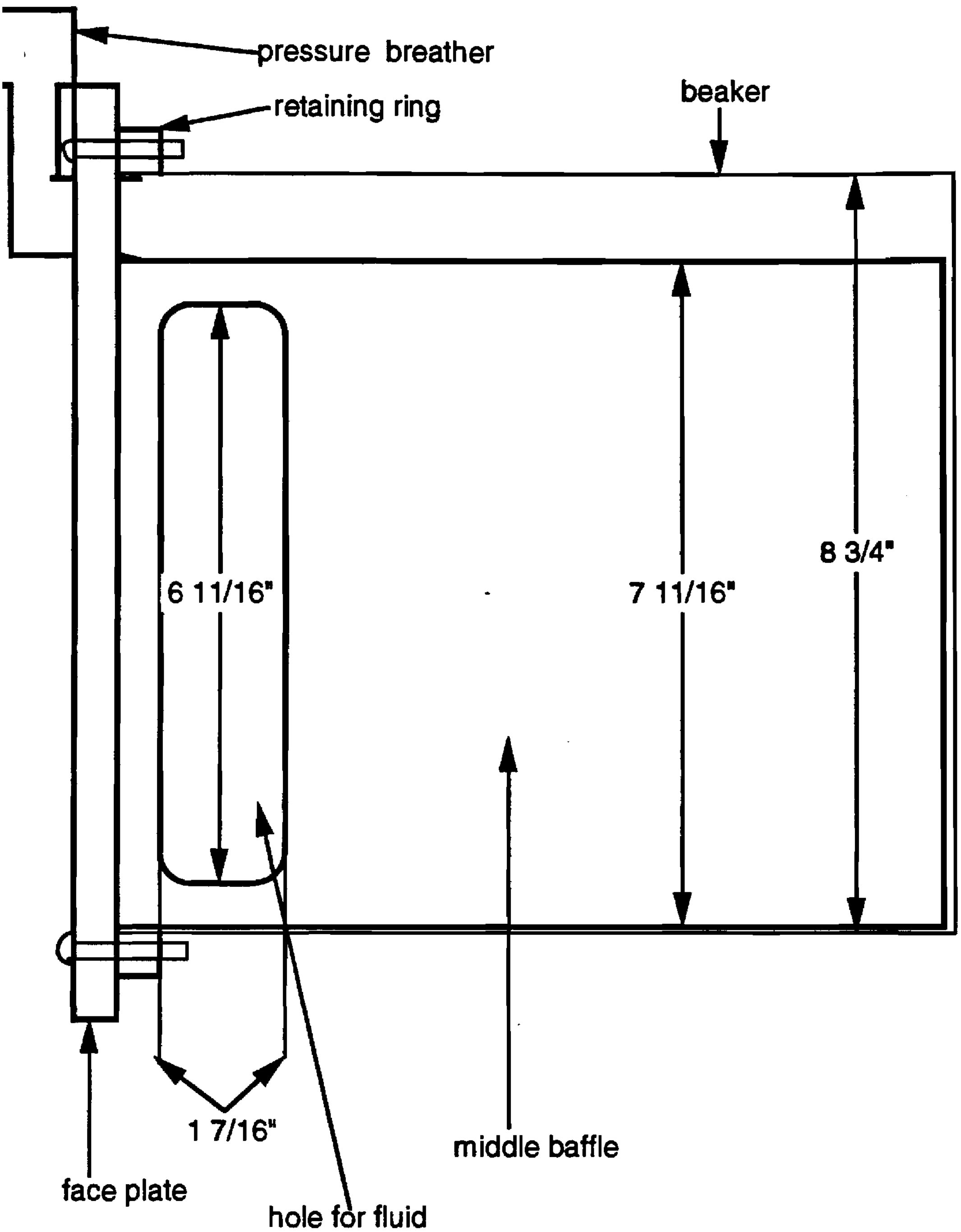


ME 4192 - ENABLER
NAME: RESERVOIR
BY: J. CHAMLEE
DATE: 6-1-93
SCALE: NONE APPX: 1/2

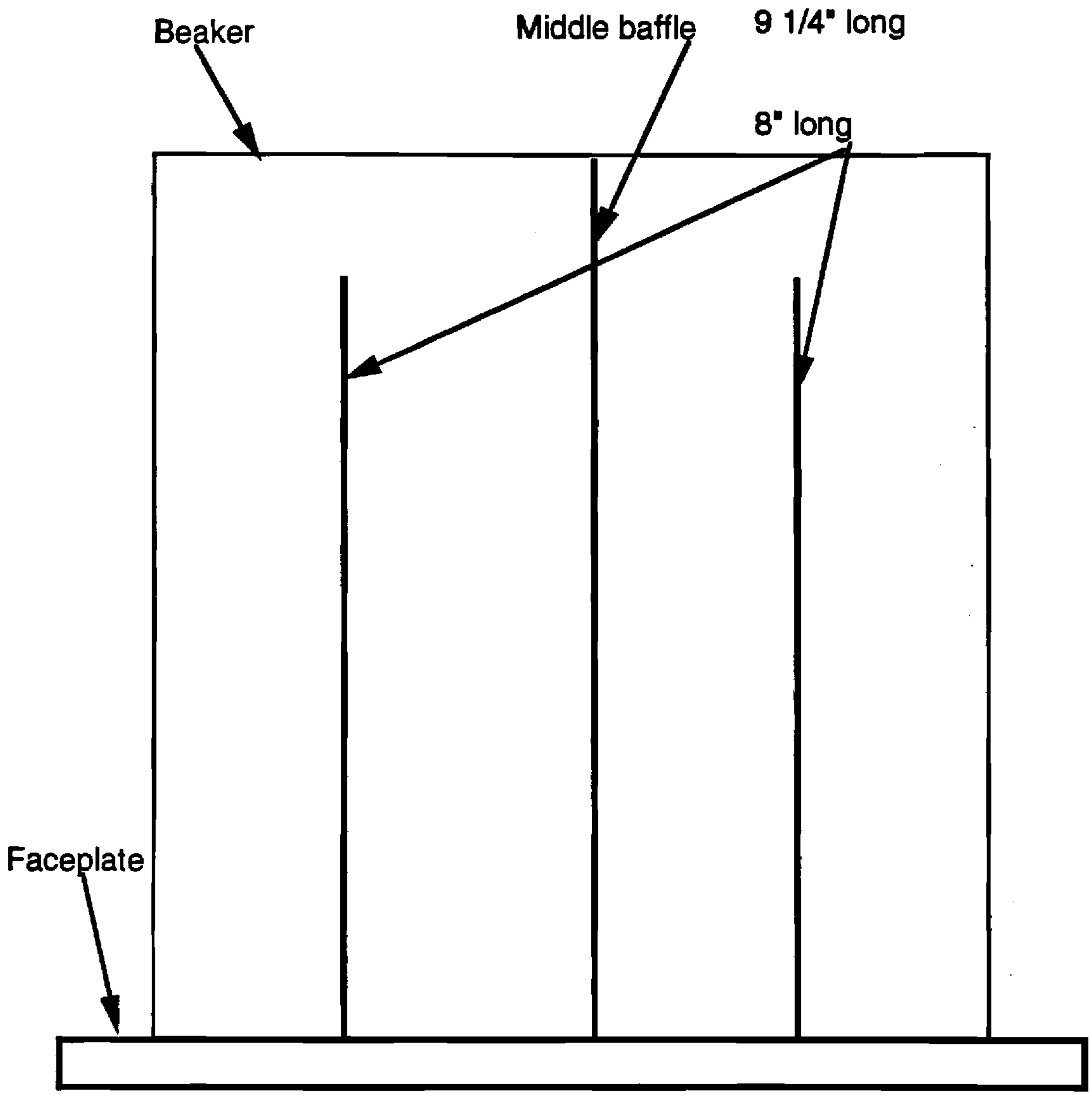


Side View of Plain Beaker

Scale : 1 to 2



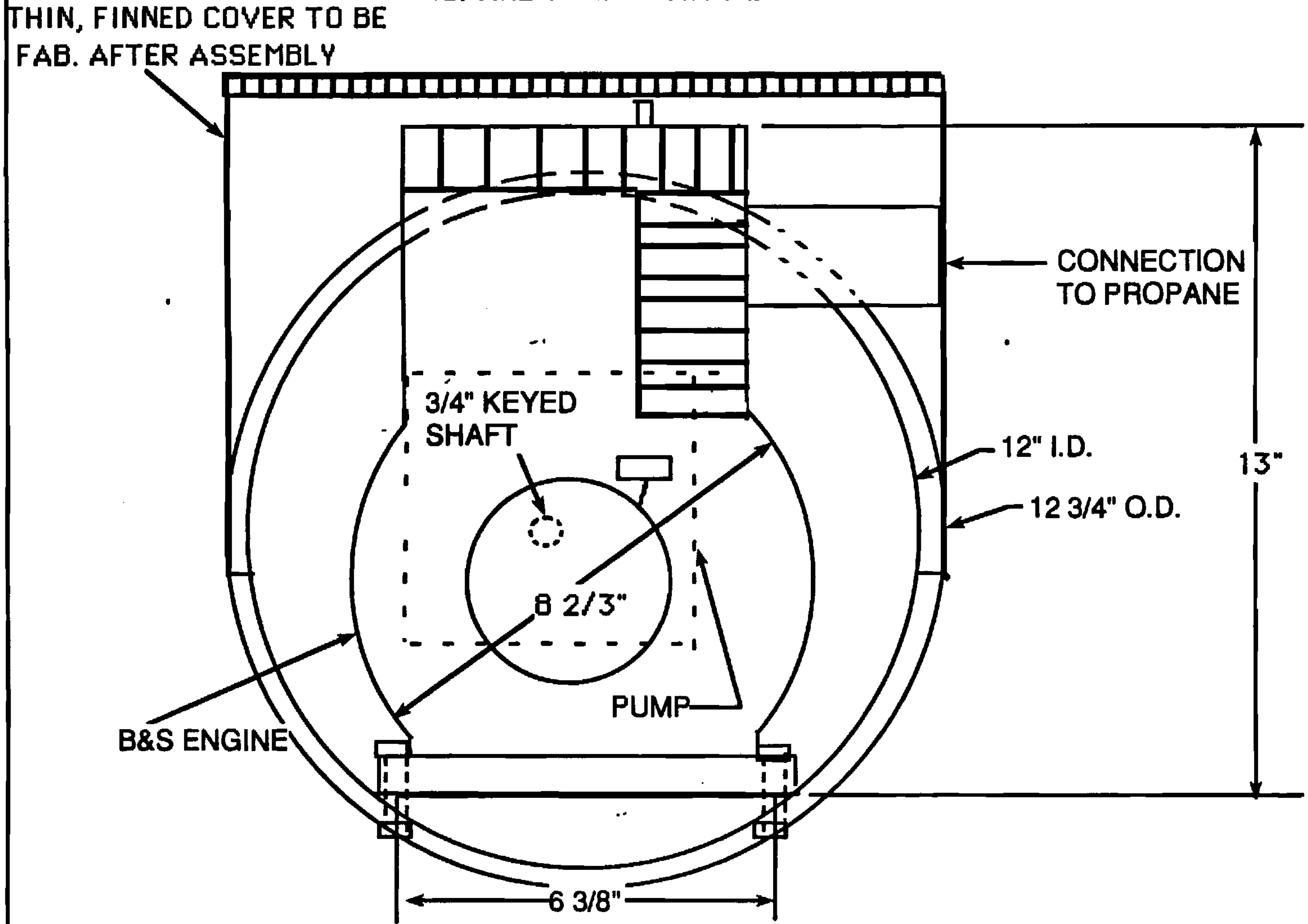
Middle baffle of reservoir
scale 1 to 2

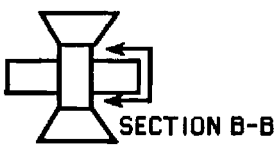


Overhead viev of reservoir showing length of baffles
Scale =1 to 2

TAIL SECTIONAL VIEW B-B

PROPANE TANKS OMITTED





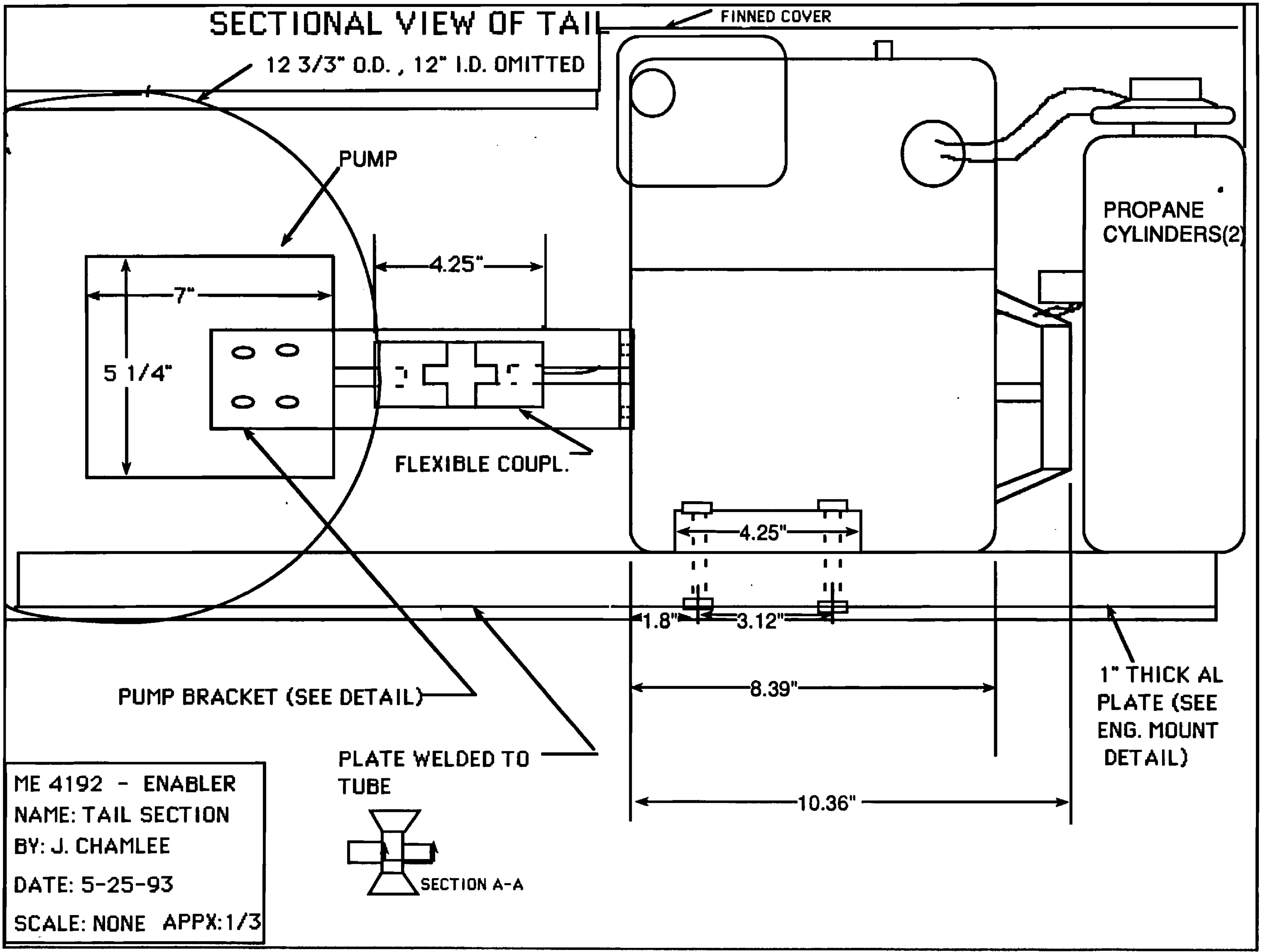
ME 4192 - ENABLER

NAME: TAIL (BACK)

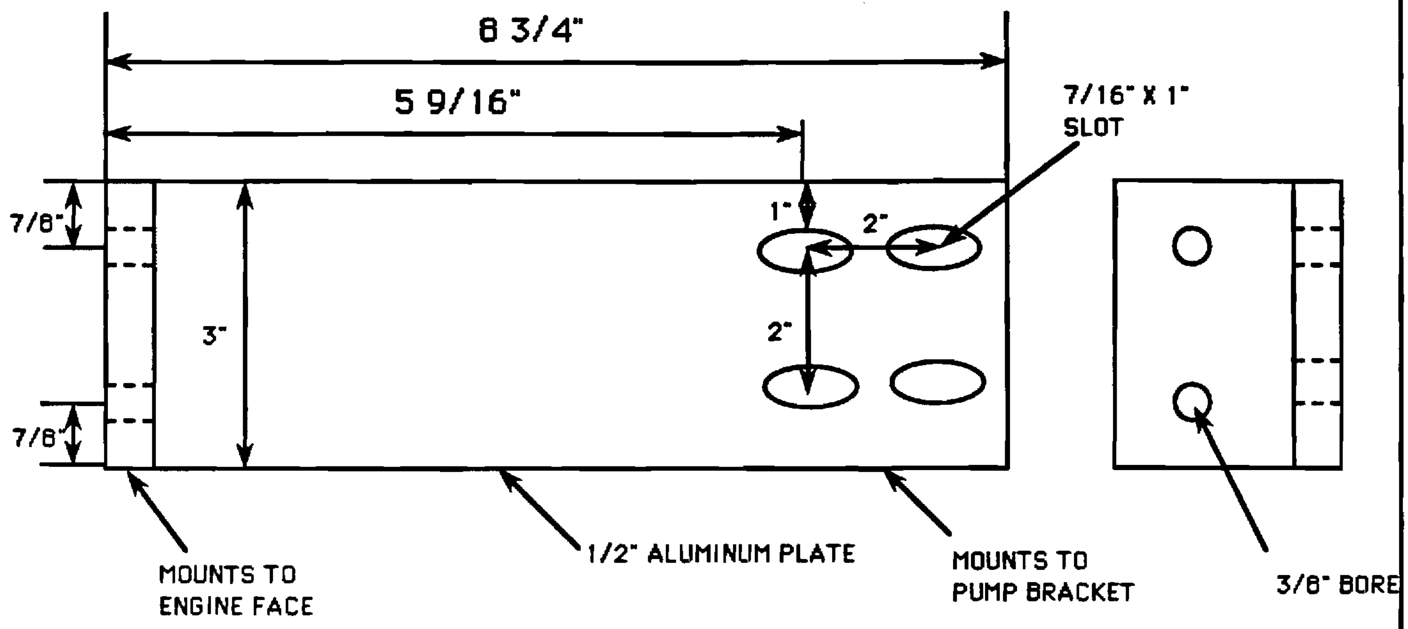
BY: J. CHAMLEE

DATE: 5-22-93

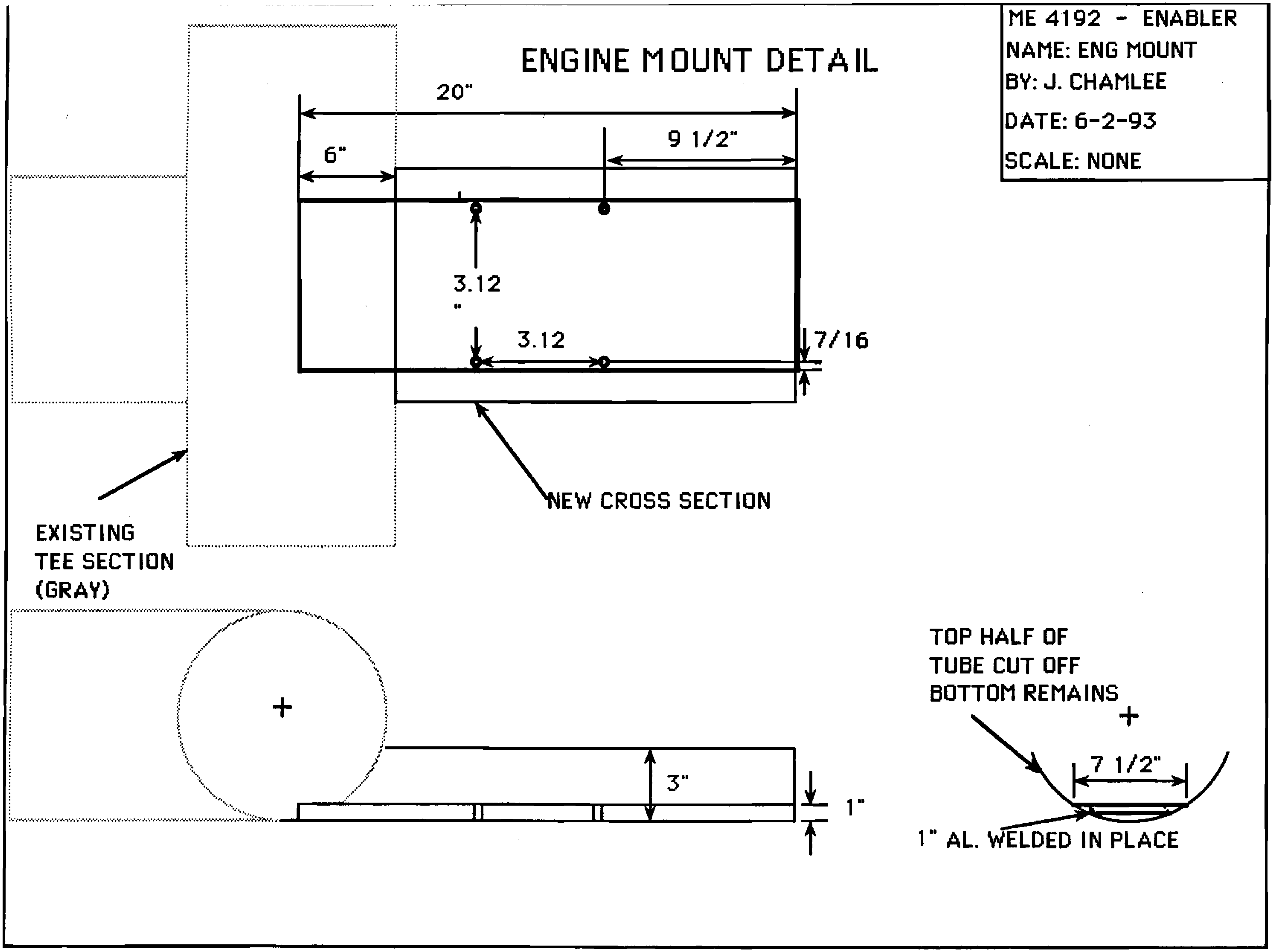
SCALE: NONE

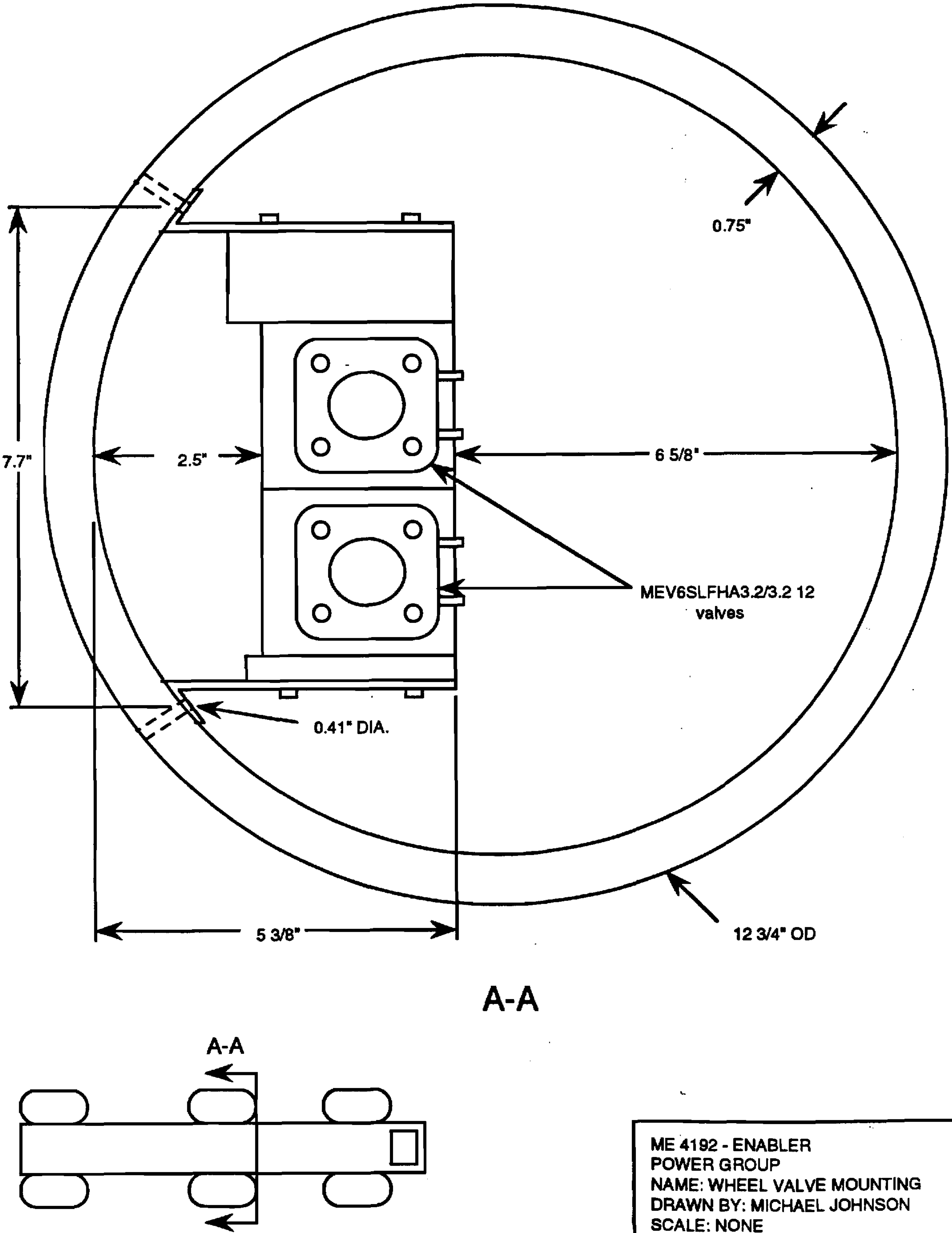

SECTIONAL VIEW OF TAIL
FINNED COVER
12 3/3" O.D. , 12" I.D. OMITTED
PUMP
PROPANE CYLINDERS(2)
4.25"
7"
5 1/4"
FLEXIBLE COUPL.
4.25"
1.8"
3.12"
8.39"
10.36"
PUMP BRACKET (SEE DETAIL)
1" THICK AL PLATE (SEE ENG. MOUNT DETAIL)
PLATE WELDED TO TUBE
SECTION A-A
ME 4192 - ENABLER
NAME: TAIL SECTION
BY: J. CHAMLEE
DATE: 5-25-93
SCALE: NONE APPX:1/3

BRACKET TO FACE-MOUNT PUMP

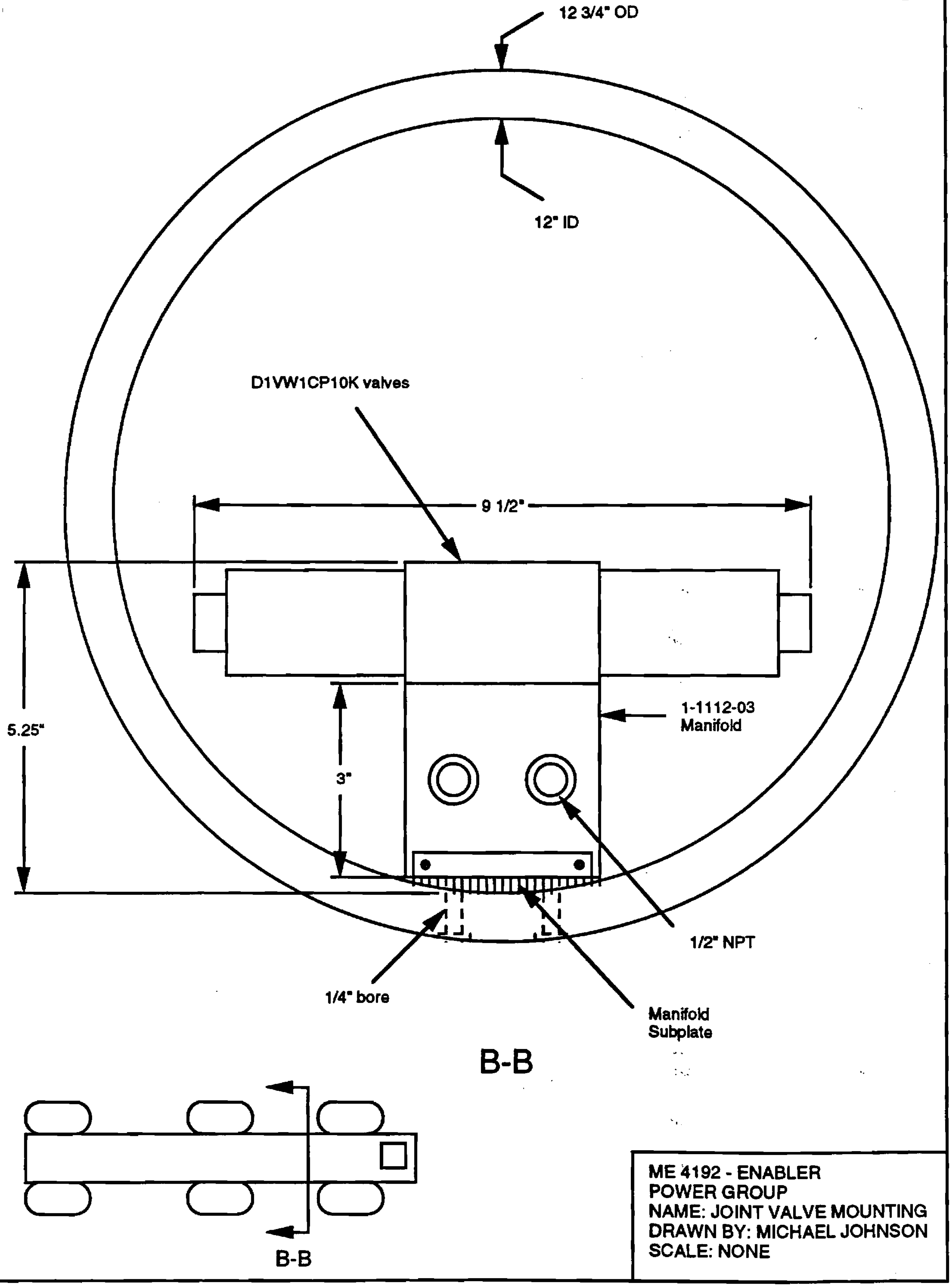


ME 4192 - ENABLER
NAME: PUMP MOUNT
BY: J. CHAMLEE
DATE: 5-20-93
SCALE: NONE

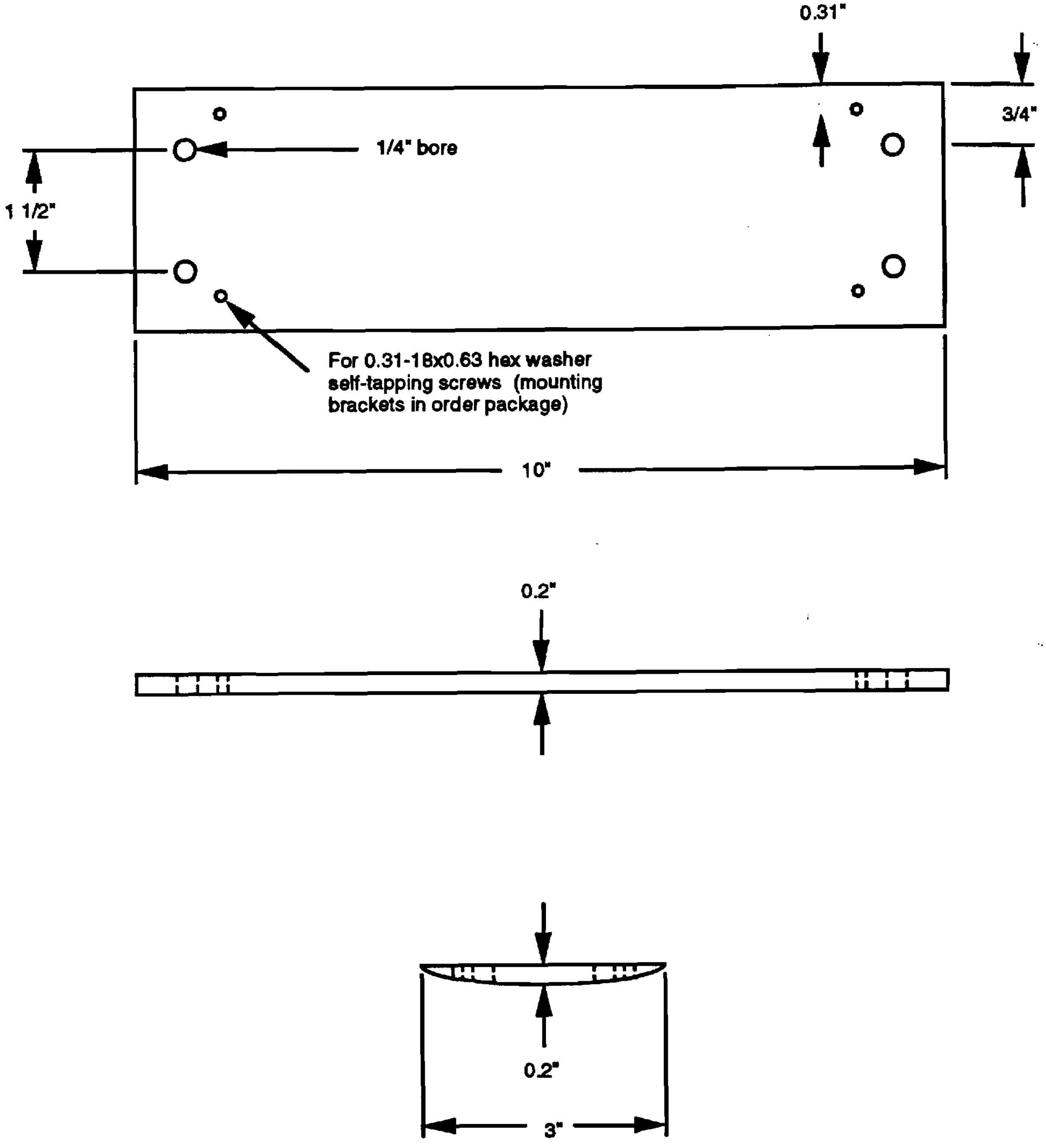

ME 4192 - ENABLER
NAME: ENG MOUNT
BY: J. CHAMLEE
DATE: 6-2-93
SCALE: NONE
ENGINE MOUNT DETAIL
20"
9 1/2"
6"
3.12"
3.12
7/16
NEW CROSS SECTION
EXISTING TEE SECTION (GRAY)
3"
1"
TOP HALF OF TUBE CUT OFF BOTTOM REMAINS
7 1/2"
1" AL. WELDED IN PLACE



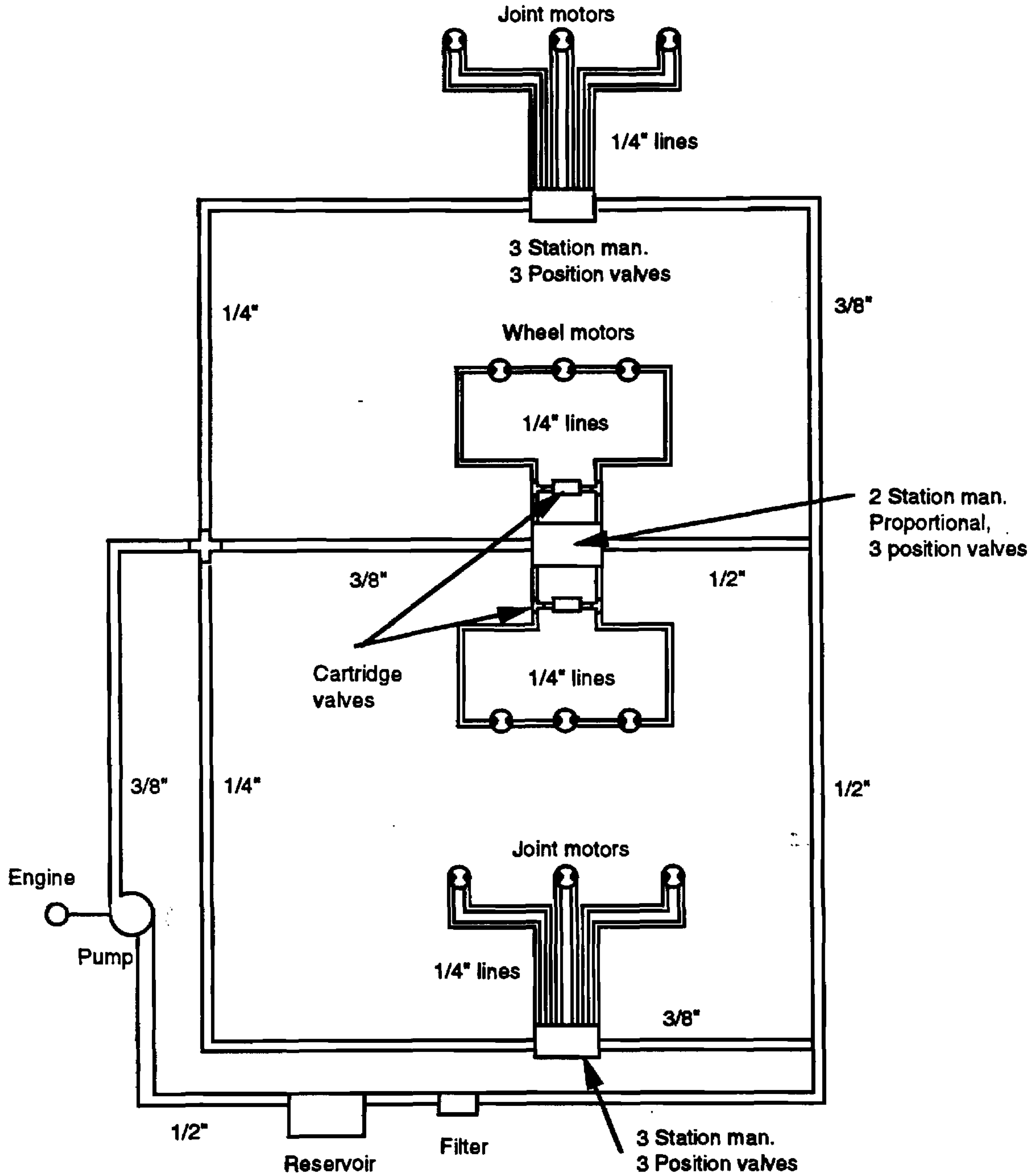
ME 4192 - ENABLER
POWER GROUP
NAME: WHEEL VALVE MOUNTING
DRAWN BY: MICHAEL JOHNSON
SCALE: NONE



ME 4192 - ENABLER
POWER GROUP
NAME: JOINT VALVE MOUNTING
DRAWN BY: MICHAEL JOHNSON
SCALE: NONE



ME 4192 - ENABLER
POWER GROUP
NAME: MOUNTING PLATE
DRAWN BY: MICHAEL JOHNSON
SCALE: NONE



ME 4192 - ENABLER
POWER GROUP
NAME: LINE CONFIGURATION
DRAWN BY: MICHAEL JOHNSON
SCALE: NONE

Appendix B

Manufacturers Drawings

Manufacturer's Drawings for:

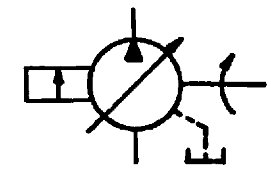
Pump

Printed in U.S.A.

PISTON PUMPS

MODEL SERIES PVB5/6

VARIABLE DISPLACEMENT INLINE TYPE



STANDARD GRAPHICAL SYMBOL FOR FLUID POWER DIAGRAMS

GENERAL DATA
These units are of the axial piston, variable displacement, inline design. Displacement is varied by means of either a pressure compensator, handwheel, or lever control.

INSTALLATION
Horizontal mounting is recommended to maintain necessary case fluid level. The case drain line must be full size unrestricted and connected from the uppermost drain port directly to the reservoir in such a manner that the case remains filled with fluid. Piping of drain line must prevent siphoning. Pipe drain line so that it terminates below reservoir fluid level. No other lines are to be connected to this drain line. Caution must be exercised to never exceed 5 PSI unit case pressure.

Starting
Before starting, fill case with system fluid through uppermost drain port. Case must be kept full at all times to provide internal lubrication.

RATINGS
See chart for operating specifications.

When first starting, it may be necessary to bleed air from pump outlet line to permit priming and reduce noise. Bleed by loosening an outlet connection until solid stream of fluid appears. An air-bleed valve is available for this purpose. See drawing 521601.

To assist initial priming, control setting must be at least 40 percent of maximum flow position.

Shaft Rotation
Shaft rotation is not reversible and must be specified when ordering. See drawing 517010 for indirect drive data.

Case Pressure
Not to exceed .... 5 PSI

Handwheel Data
Approximate displacement for one turn of handwheel is .16 in.^{3}/rev. for 5 size, and .21 in.^{3}/rev. for 6 size.

WEIGHT
Flange Mounting.................. 16 lbs.
Foot Mounting...................... 20 lbs.

APPLICATION GUIDANCE
This unit is designed to meet specifications as outlined. To insure maximum unit performance in conjunction with your specific application, consult your Vickers application engineer if your:
- Speed is above maximum RPM.
- Fluid does not meet the specifications shown on data sheet I-286-S.
- Mounting attitude is other than horizontal.
- Needs require application assistance.

FILTRATION
For satisfactory service life of these components in industrial applications, use full flow filtration to provide fluid which meets ISO cleanliness code 16/13 or cleaner. Selections from Vickers OFP, OFR and OFRS series are recommended. Refer to Section "P"-"Filters" and Section "S"- "Technical Information" of Vickers Catalog 400A, or contact your Vickers representative for further filtration advice.

FLUIDS AND TEMPERATURE DATA
Use clean petroleum antiwear industrial hydraulic oil, water-in-oil emulsions, HWBF, water glycol, phosphate esters, or automotive crankcase oil designated SC, SD, or SE. Running viscosity range 70-250 SUS. Operating temperature 1206F recommended, 1506F usual maximum.

Refer to Vickers data sheet I-286-S, "Hydraulic Fluid and Temperature Recommendations for Industrial Machinery," and drawing 507000, "Inline Piston Pump Life and Fluid Data" (included in the Technical Information section of Vickers Catalog 400A) for hydraulic fluid and temperature recommendations.

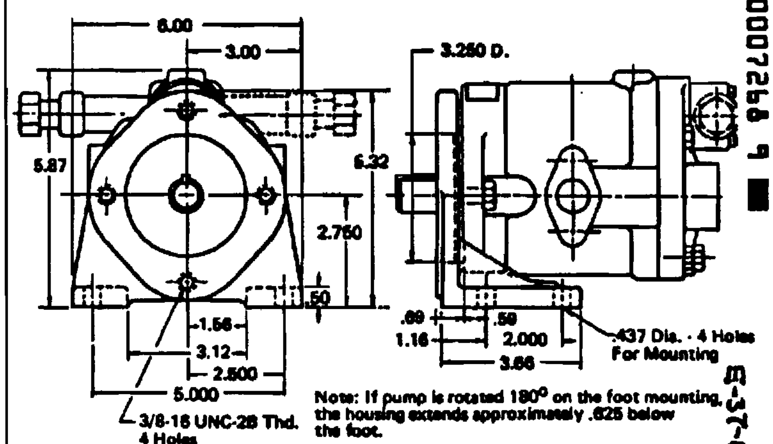
SERVICE INFORMATION
Refer to specific Vickers part drawing or overhaul manual for service information. A complete "Index to ndustrial Products Service Information" is included in the Technical Information section of Vickers Catalog 400A.

OPERATING SPECIFICATIONS

Model Number	Theoretical Displacement In.^{3}/Rev.	Delivery GPM at 1800 RPM	Delivery GPM at Max. RPM	Operating Speed RPM (Maximum)	Pressure* PSI (Maximum)	■ Sound Data dB(A)	Input Horsepower at Max. PSI & 1800 RPM
PVB5	.643	5	10	3600	3000	59	10
PVB6	.843	6.5	11.4	3200	2000	59	8.75

* Refer to curve for inlet pressure requirements.
■ Rear port model at cutoff, 1200 RPM, 2000 PSI pressure, SAE 10W oil at 1206F, 56 Hg inlet vacuum per NFPA standard T3.9.70.12.

FOOT MOUNTING - ALL MODELS



Note: If pump is rotated 180° on the foot mounting, the housing extends approximately .625 below the foot.

MODEL CODE

F3 - P V B 5 - F R S W - Y - 21 - *C C G L - 10 - *

1 2 3 4 5 6 7 8 - 9 10 11 12 13 14 15 16 17

1 Special seals

Omit if not required.

2 Pump

3 Variable displacement

4 Inline type design

5 US gpm rating

5 - 5 GPM (at 1800 RPM)
6 - 6 GPM

6 Mounting

Blank - Omit for flange. (Standard)
F - Foot bracket. (Order separate "Foot bracket kit", model FB-A-10, for best price and availability.

7 Rotation

(viewed from shaft end)
R - Right hand (CW), Standard
L - Left hand (CCW), Optional.

8 Displacement

S - One side of center (pressure compensator models).
D - Both sides of center (handwheel & lever models).

9 Optional ports & thru - shaft

Blank - Rear ports (standard)
W - Side ports.
X - Side ports and thru-shaft (standard drive shaft).

10 • Drive shaft

Y - Standard 1.75 extension .750 diameter, str. key.
Blank - 1.25 extension .563 diameter, str. key (replacement for 10 design pump).

11 Design number

Subject to change. Installation dimensions remain as shown for designs 20 through 29.

12 *Control type

C - Pressure compensator (250-3000 PSI for PVB5; 250-2000 PSI for PVB6). See compensator control information.
CM - Pressure compensator (250-1500 PSI).
CVP - Load sensing with pressure limiting. (160 PSI Δ P setting) cup 3 - load sensing with pressure limiting (350 PSI Δ P setting)
H - Handwheel.
M - Lever.

13 Control option

C - Adjustable maximum displacement stop (with compensator). Not available with thru shaft.

14 Compensator variations

D - Dual range (electric control). Not available with thru shaft.
G - Remote compensator (use CCG with right hand models).

15 Other controls or options

L - Left hand location (viewed from shaft hand). H M only.

16 Control design number

Subject to change. Installation dimensions remain as shown for design numbers 10 through 19 20 through 29 for "D" control.

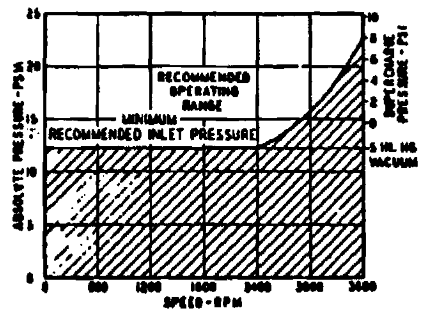
17 Special suffix

S124 - Add for spline shaft **.
*Note: Pressure shown defines the minimum adjustable range. It might be possible to make settings outside of this range. 530 - Extra drain port to permit vertical shaft-up installation.

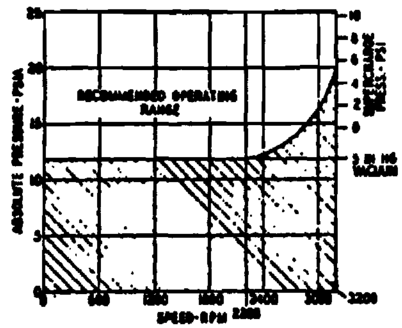
** When selecting S124 shaft, leave code 10 blank.

TYPICAL PERFORMANCE CURVES

MODEL PVB5
Inlet Pressure Curve
Based On Oil Temperature of 120°F (100 SUS)
Viscosity of 150 SSU @ 100°F
Inlet Pressure Requirement Versus Pump Speed

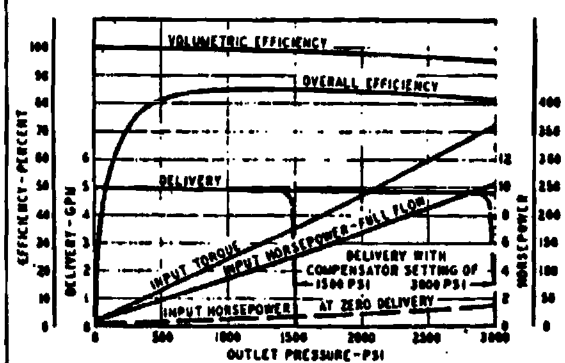


MODEL PVB6
Inlet Pressure Curve
Based on Oil Temperature of 120°F (100 SUS)
Viscosity of 150 SSU @ 100°F
1800 RPM Input - Atmospheric Inlet

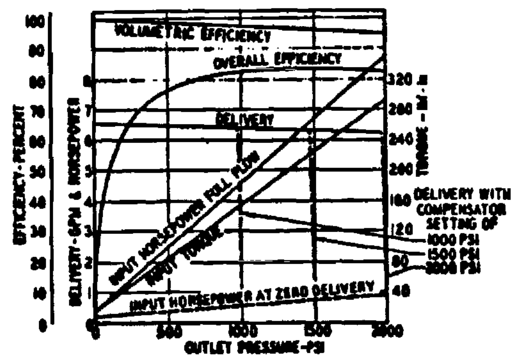


PERFORMANCE CHARACTERISTICS

MODEL PVB5
Based on Oil Temperature of 120°F
(100 SUS)
1800 RPM Input - Atmospheric Inlet



MODEL PVB6
Based on Oil Temperature of 120°F
(100 SUS)
1800 RPM Input - Atmospheric Inlet

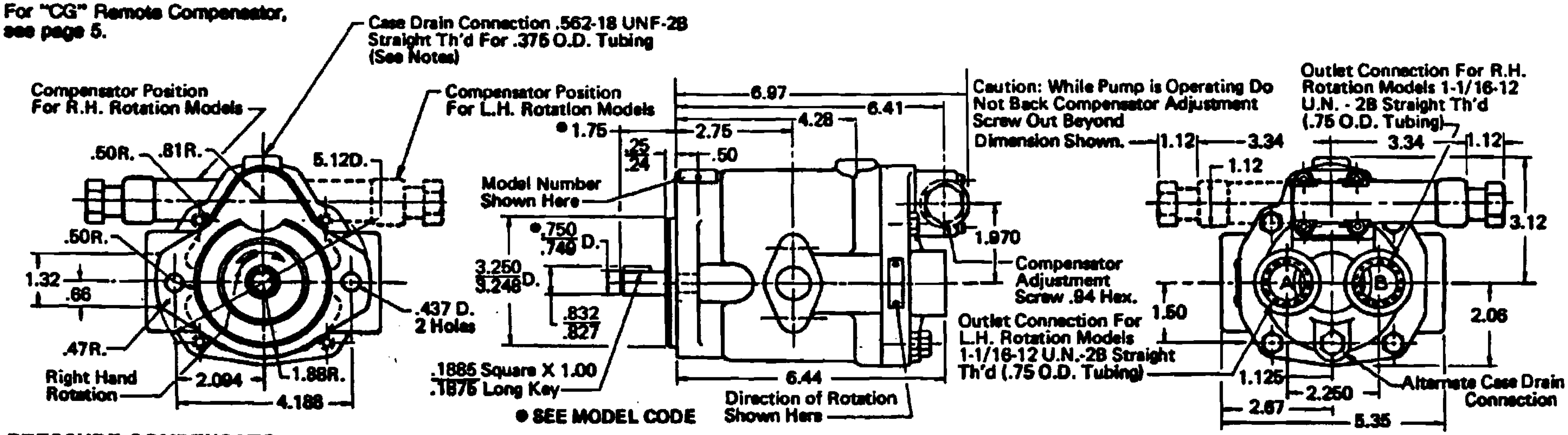


SOUND DATA AS PER NFPA T3.9.70.12

Model	Speed RPM	Pressure — PSI			
		Full Flow			Cutoff
		1000	2000	3000	2000
PVB5	1200	60	63	66	60
	1800	68	69	71	66
PVB6	1200	59	64	—	60
	1800	67	69	—	62

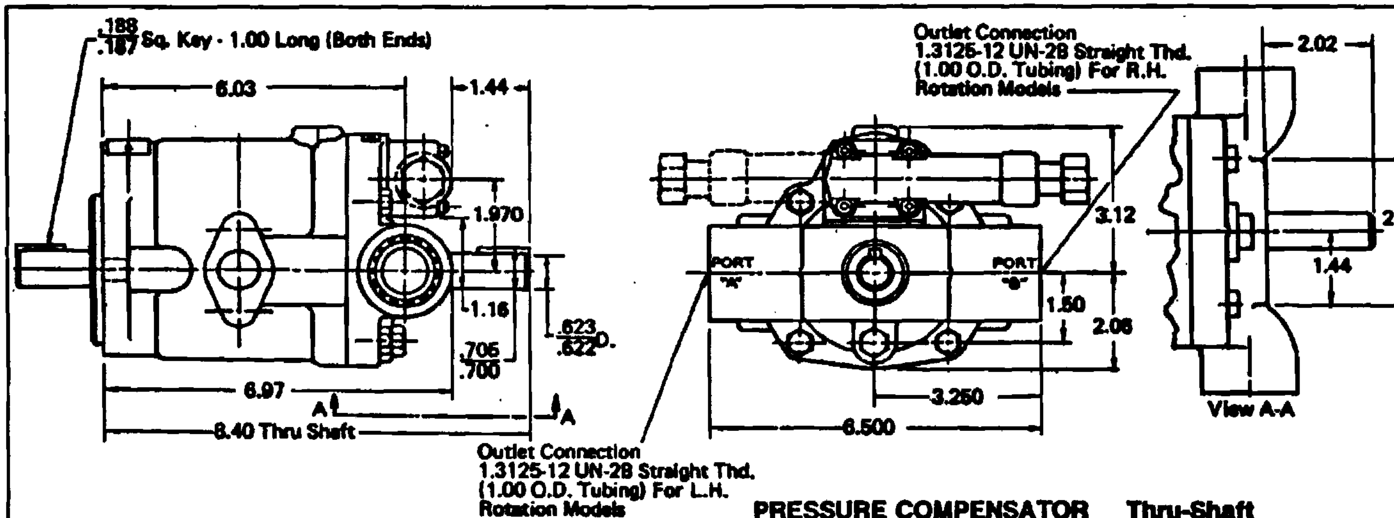
I-508300-2

For "CG" Remote Compensator, see page 5.



PRESSURE COMPENSATOR CONTROLS

The pressure compensator control automatically adjusts pump delivery to maintain volume requirements of the system at a preselected operating pressure. Maximum pump delivery is maintained to approximately 50 PSI below the pressure control setting before being reduced. The pressure compensator control operates on one side of center and has an adjustment range of 250 to 1500 PSI on "CM" models and 250 to 3000 on "C" models as designated in the model coding. Customer must limit PVB6 to 2000 PSI when using "C" compensator.



PRESSURE COMPENSATOR CONTROL - THRU-SHAFT AND SIDE PORTS

Side Ports

Rated and maximum speeds. Pressure and general performance of the side port pumps are identical to the standard units.

Thru-Shaft

Rated and maximum speeds. Pressure and general performance of the thru-shaft pumps are identical to the standard units, except the input torque is limited to 354 lb f-in.

Note:

Both shafts must have a direct drive only.

I-508300-3

Manufacturer's Drawings for:

Breather / Vacuum Breaker

Ordering Codes: Filler-Breather-Filters and Breather-Filters

215-266-0100

HYCON

Filler-Breather-Filter

ELF 3 - 20 – 1 . 0 / SO 165

Model
ELF
ELF L (lockable, size 3 only)

Size
3
3 RV = with check/relief valve
4
5
7

Filtration Rating of Air Filter
3 = 3 µm (sizes 3, 4, and 5 only)
10 = 10 µm (sizes 3, 4, and 5 only)
20 = 20 µm
40 = 40 µm

Gauge Options (ELF 7 only)
– = without gauge
K = with gauge (Gauge range = -14.5 to 9 PSI/-1 to 0.6 BAR)

Identification Number
1 = ELF 3, 4, and 7: standard units
2 = ELF 5 with weld-on fitting $2^1/_2$" BSP] Fittings supplied by HYCON
3 = ELF 5 with weld-on fitting 3" BSP] Fittings supplied by HYCON
4 = ELF 3 RV; relief cracking pressure 10 PSI/0.7 BAR, Reseat pressure 6 PSI/0.4 BAR
5 = ELF 3 RV; relief cracking pressure 13 PSI/0.9 BAR, Reseat pressure 10 PSI/0.7 BAR
6 = ELF 3 RV; relief cracking pressure 6 PSI/0.4 BAR, Reseat pressure 3 PSI/0.2 BAR

Modification Number

Supplementary Details
SO 165 = For fire resistant fluids (HWBF, HWCF)
SO 175 = Metal filler screen 4"/100 mm long] Sizes 3 & 7 only
SO 148 = Metal filler screen 8"/200 mm long] Sizes 3 & 7 only
V = Viton seals (size 5 only)

Breather-Filter

BF 3 - 10 – 1 . 0 /

Model
BF = Threaded Connection
BL = Flange Connection (size 160 only)
BLG = NPT Connection (sizes 80 and 160 only)
BLS = Weld-on Connection (size 160 only)

Size
3] BF only
3 RV = with check/Bypass Valve] BF only
4] BF only
5] BF only
7
080 = $^3/_4$" Connection] BLG only
160 = $1^1/_4$" Connection] BLG only
161 = Weld-on Connection (BLS only)
162 = Flange Connection (BL only)

Filtration Rating of Air Filter
3 = 3 µm (BF only)
10 = 10 µm (BL, BLG, and BLS only)
20 = 20 µm (BF only)
40 = 40 µm (BF only)

Gauge Options (BF 7 only)
– = without gauge
K = with gauge (Gauge range = -14.5 to 9 PSI/-1 to 0.6 BAR)

Identification Number
Type BF
1 = Threaded Connection BF 3 = G $^3/_4$ (ISO 228), BF 4 = G $^1/_4$ (ISO 228), BF 5 = G $2^1/_2$ (ISO 228), BF 7 = G 1 (ISO 228)
2 = Threaded Connection BF 3 = G $^3/_8$ (ISO 228)
3 = Threaded Connection BF 3 = G $^1/_2$ (ISO 228)
4 = Threaded Connection BF 3 = G $^3/_4$ (ISO 228) with RV; Bypass Cracking Pressure 10 PSI/0.7 BAR
5 = Threaded Connection BF 3 = G $^3/_4$ (ISO 228) with RV; Bypass Cracking Pressure 13 PSI/0.9 BAR
6 = Threaded Connection BF 3 = G $^3/_4$ (ISO 228) with RV; Bypass Cracking Pressure 6 PSI/0.4 BAR
Type BL, BLG, and BLS
1 = G $1^1/_4$ (ISO 228) (replacement element MFE 160-10/1)
2 = $1^1/_2$-16UN-2B (replacement element MFE 160-10/2)

Modification Number

Supplementary Details
SO 165 = For fire resistant fluids (HWBF, HWCF)] BF only
V = Viton seals] BF only
A = Adaptor for BF 80 or BF 160 to thread into Filler-Breather-Filter

Engineering Data and Weights

<table>
<tr><td>Design:</td><td></td><td colspan="3"></td></tr>
<tr><td></td><td>ELF</td><td colspan="3">Filler-Breather-Filter</td></tr>
<tr><td></td><td>BF</td><td colspan="3">Breather-Filter</td></tr>
<tr><td></td><td>BL</td><td colspan="3">Breather-Filter with Spin-on Filter Element</td></tr>
<tr><td></td><td>BLG</td><td colspan="3">Breather-Filter with Spin-on Filter Element</td></tr>
<tr><td></td><td>BLS</td><td colspan="3">Breather-Filter with Spin-on Filter Element</td></tr>
<tr><td>Mounting Position:</td><td></td><td colspan="3"></td></tr>
<tr><td></td><td>ELF & BF</td><td colspan="3">Vertical (max. 30° off vertical axis)</td></tr>
<tr><td></td><td>BL, BLG, & BLS</td><td colspan="3">Vertical or Horizontal</td></tr>
<tr><td rowspan="2" colspan="2">Mounting Method:</td><td>Threaded Coupling</td><td>Weld-on Fitting</td><td>Bolt-on Flange</td></tr>
<tr><td>ELF 5, BF, & BLG</td><td>BLS</td><td>ELF 3, 4, 7 & BL</td></tr>
<tr><td colspan="2">Fluid Temperature Range:</td><td colspan="3">+15°F........+210°F</td></tr>
<tr><td colspan="2">Fluid Compatibility:
(ISO 2943)</td><td colspan="3">Compatible with all petroleum-based oils.
Contact HYCON office for information.</td></tr>
<tr><td colspan="2">Re-seating Pressure of
Check/Bypass Valve:
(for ELF 3 RV & BF 3 RV only)</td><td colspan="3">ΔP = 2.9 PSID/0.2 BAR,
ΔP = 5.8 PSID/0.4 BAR or
ΔP = 10.1 PSID/0.7 BAR</td></tr>
</table>

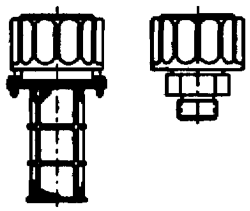
Weights

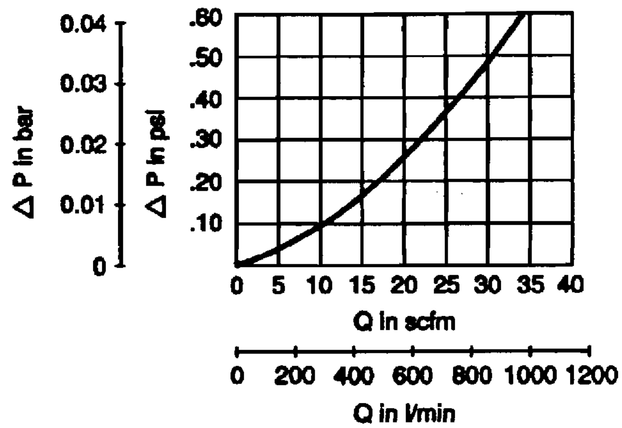
MODEL	lbs.	kg.
ELF 3	0.55	0.25
ELF 3 RV	0.66	0.30
ELF 4	0.22	0.10
ELF 5 (Type No. 2)	5.95	2.70
ELF 5 (Type No. 3)	6.83	3.10
ELF 7	0.84	0.38
BF 3	0.62	0.28
BF 3 RV	0.73	0.33
BF 4	0.18	0.08
BF 5	4.41	2.00
BF 7	0.88	0.40
BL 162	4.63	2.10
BLG 80	1.4	0.60
BLG 160	2.60	1.20
BLS 161	3.86	1.75

Pneumatic Data

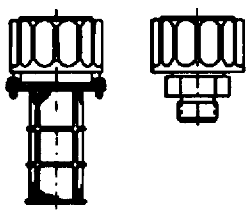
Air Flow Rates:
1 GPM = .1337 SCFM
1 L/min = .035 SCFM
(Flow rates apply to all micron ratings.)

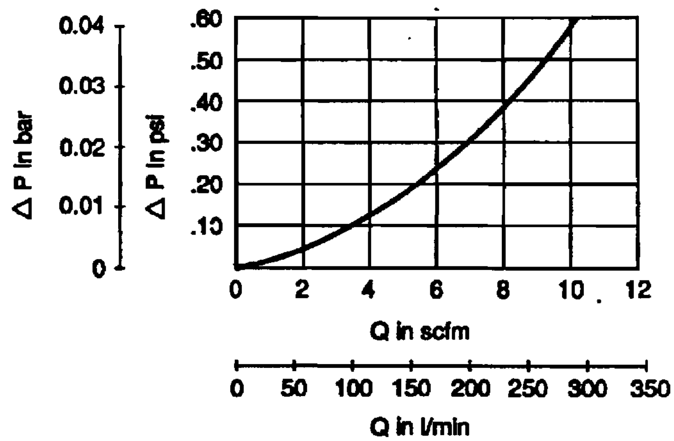
ELF 3/BF 3



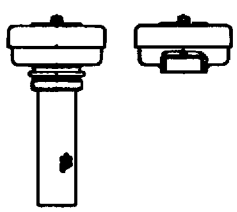


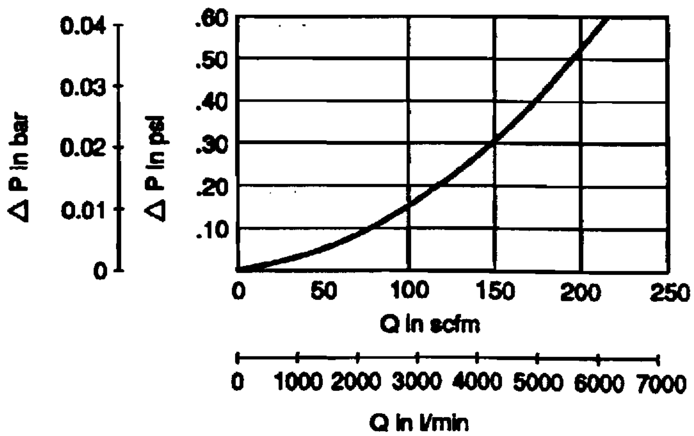
ELF 4/BF 4





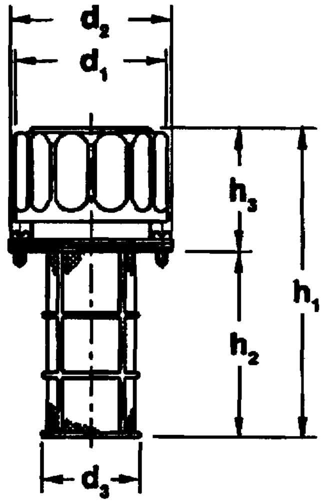
ELF 5/BF 5



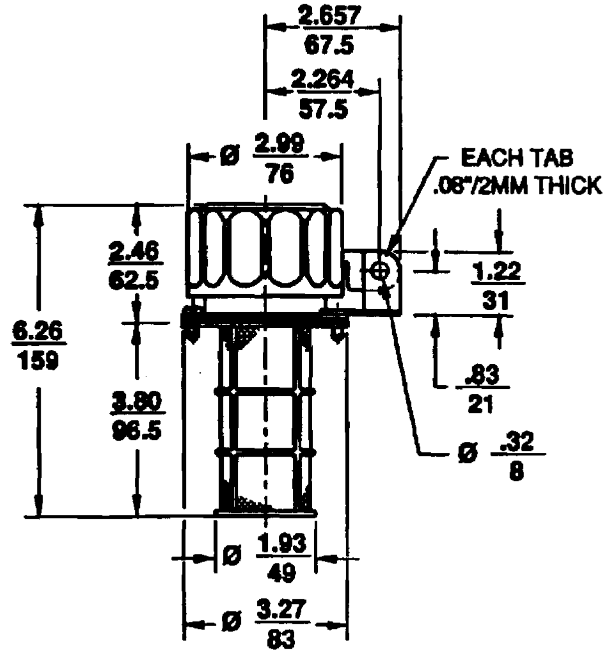


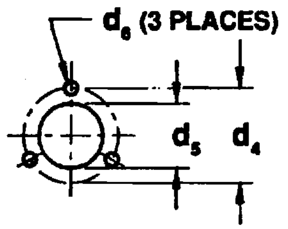
Dimensions

ELF 3
ELF 3 RV
ELF 4

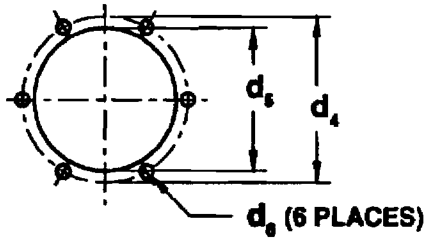


ELF L 3, lockable

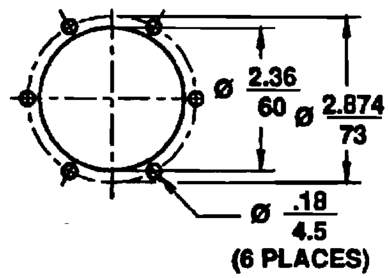




ELF 4
MOUNTNG HOLE PATTERN



ELF 3 & ELF 3 RV
MOUNTNG HOLE PATTERN
(Bore Diagram to DIN 24557/T2)



ELF L 3
MOUNTNG HOLE PATTERN
(Bore Diagram to DIN 24557/T2)

		ELF 3		ELF 3 RV		ELF 4
d_1	INCH / MM	Ø	2.99 / 76	Ø	2.99 / 76	Ø	1.73 / 44
d_2	INCH / MM	Ø	3.27 / 83	Ø	3.27 / 83	Ø	1.97 / 50
d_3	INCH / MM	Ø	1.93 / 49	Ø	1.93 / 49	Ø	1.10 / 28
d_4	INCH / MM	Ø	2.874 / 73	Ø	2.874 / 73	Ø	1.626 / 41.3
d_5	INCH / MM	Ø	2.36 / 60	Ø	2.36 / 60	Ø	1.18 / 30
d_6	INCH / MM	Ø	.18 / 4.5	Ø	.18 / 4.5	Ø	.18 / 4.5
h_1	INCH / MM		6.26 / 159		6.26 / 159		5.32 / 135
h_2	INCH / MM		3.80 / 96.5		3.80 / 96.5		3.21 / 81.5
h_3	INCH / MM		2.46 / 62.5		2.46 / 62.5		2.11 / 53.5

Dimensions in Inches / Millimeters

Dimensions are for general information only. Due to constant development and updating of details, we ask that all critical dimensions be verified by requesting a certified print.

BRIGGS & STRATTON

5 HP

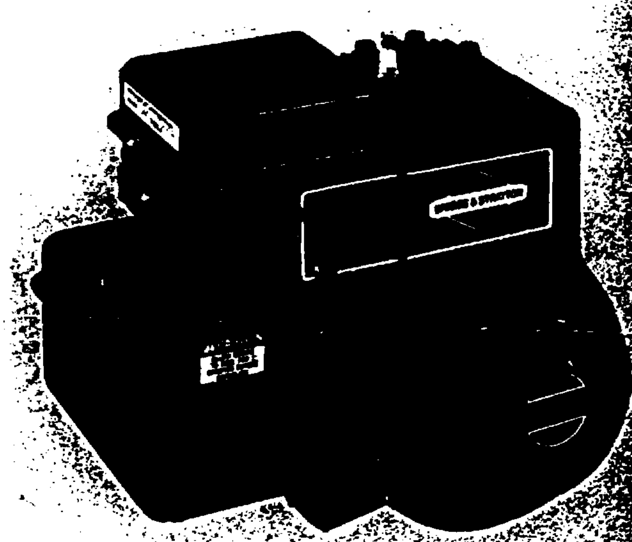
Model Series 130200

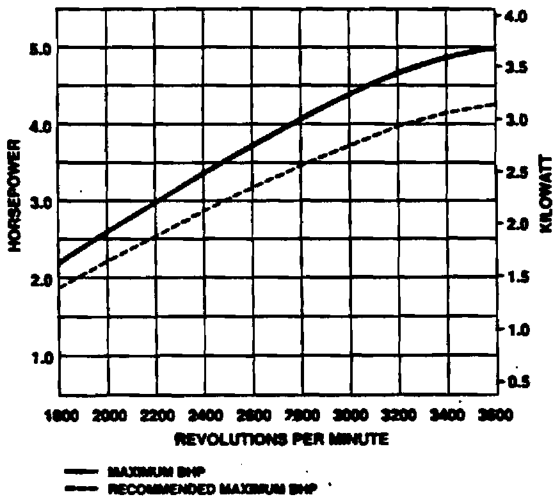
Dependability and Lasting Value

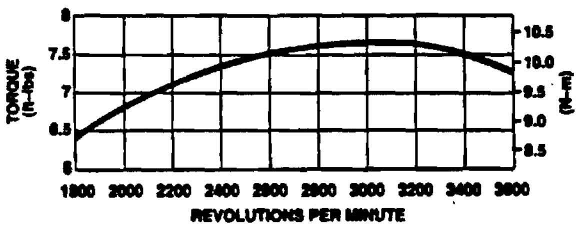
- Magnetron® Electronic Ignition
- Positive Lubrication
- Pleated Paper Element Air Cleaner

Startability and Simple Operation

- Automatic Compression Release
- Centrifugal Governing
- Heavy Flywheel for Increased Inertia
- Front Mount Control Panel





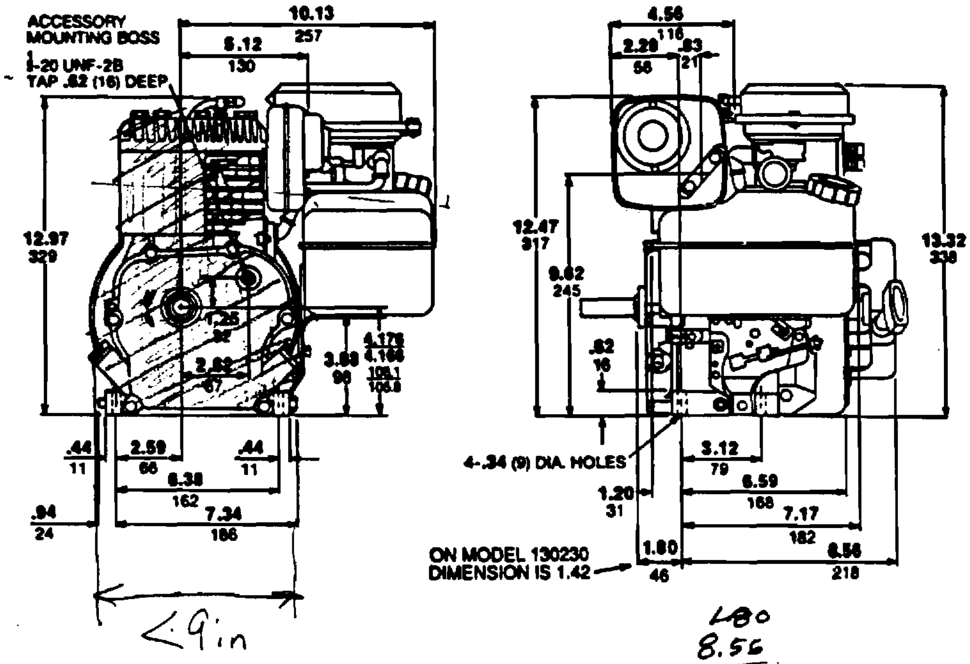


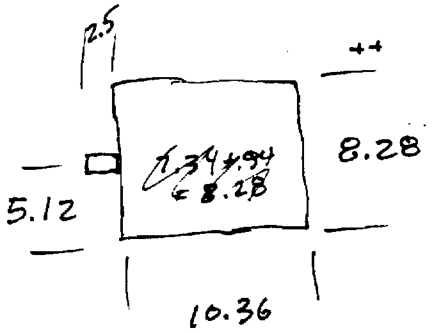
For exact or additional specifications, please contact the factory.

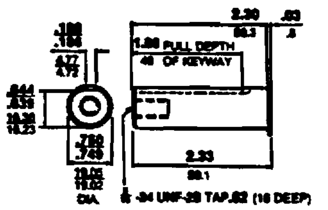
Bore 2.56" (65.1 mm)
Stroke 2.44" (61.9 mm)
Displacement 12.57 cu. in. (206 cc)
Weight* 29.75 lbs. (13.49 kg)

Note: engine photo includes optional equipment.
*Standard engine weight.
For shipping weights, please contact factory.

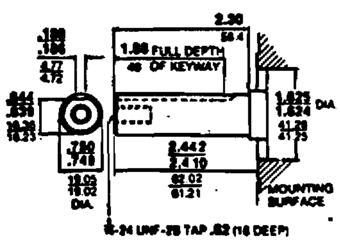
Model Series 130200—5 HP



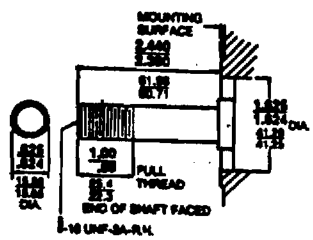




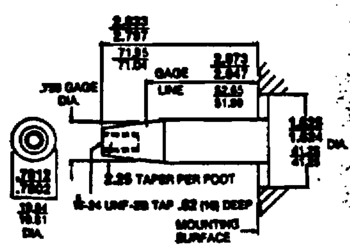
MODEL 130202 - C/S NO. 281679
3/4" DIA. .844" KEYWAY 1-7/8" LONG D&T 5/16-24



MODEL 130202 - C/S NO. 281730
3/4" DIA. .844" KEYWAY 1-7/8" LONG D&T 5/16-24



MODEL 130232 - C/S NO. 281732
5/8" DIA. 1" THREAD 5/8-18



MODEL 130232 - C/S NO. 281733
3/4" DIA. TAPERED D&T 5/16-24

Appendix C
(PURCHASE REQUESTS)

POWER GROUP
PURCHASE REQUEST

FLEXIBLE COUPLING COMPONENTS

COMPONENT NAME: LOVEJOY 3/4" COUPLING BODY FOR ENGINE

STYLE #: L075

GRAINGER STOCK #: 4X183

QTY: 1

PRICE: 2.79

COMPONENT NAME: LOVEJOY 7/8" COUPLING BODY FOR PUMP

STYLE #: L075

GRAINGER STOCK #: 4X184

QTY: 1

PRICE: 2.79

COMPONENT NAME: HYREL SPIDER

STYLE #: NONE LISTED

GRAINGER STOCK #: 1A924

QTY: 1

PRICE: 7.09

TOTAL QTY: 3

TOTAL PRICE: 12.67

ALL ITEMS AVAILABLE AT GRAINGER

1721 MARIETTA BLVD 30318-3646

PHONE # (404) 3551984

POWER GROUP

PURCHASE REQUEST

This is a request for the conversion to Propane of the Briggs and Stratton engine model 130202 to be used to power the hydraulic system of the Enabler. The conversion to Propane will be done by **Combustion Labs** in Riverdale, Ga. This conversion will include conections to two propane bottles and all other necessary components to make the engine fully operational using propane power. The complete conversion will cost $241.10. Additionally, John Chamlee will be glad to remove the gasoline tank from the existing engine and deliver the engine to Combustion Labs by 6/4/93 if it is so desired.

DESCRIPTION OF ITEM: PROPANE CONVERSION OF B&S ENGINE

PRICE: ~~$241.10~~ 260.10

COMBUSTION LABS

RIVERDALE, GA

(404) 997-0425 SPEAK WITH MIKE

POWER GROUP
PURCHASE REQUEST (VALVES)

Order #	Description	Quantity	Indiv. Cost
MEV6SLFHA3.2/3.2 12	Proportional, three position valve for wheel drive.	2	
AD6ST	Inlet plate for MEV6 valves.	1	$739.84
XRD6S02	Cross over relief valve for MEV6 valves.	2	
DS102CD012P4P	Cartridge valve with body cavity to allow free wheel.	2	$200.00
D1VW1CP10K	Three position valve for joint control.	6	$118.50
BK209	Bolt kit for D1VW valves.	6	$4.00
MANIFOLD 1-1112-03	Three station manifold for D1VW valves.	2	$72.30

TOTAL COST $2019.44

CONTACT:

Mr. Joe Howard 956-8994
Parker Hannifin Corporation
2264 Northwest Pkwy., Suite G
Marietta, GA 30067 USA
404/956-0881

POWER GROUP
PURCHASE REQUEST (LINES/ADAPTORS)

Order #	Description	Quantity	Indiv. Cost
H10404	1/4" hydraulic hose	105 ft.	$1.67 per ft.
H10406	3/8" hydraulic hose	10 ft.	$2.00 per ft.
H10408	1/2" hydraulic hose	10 ft.	$2.30 per ft.
2020 6-6S	3/8" cross section	1	$11.16
2020 8-8S	1/2" cross section	1	$12.50
2021 4-4S	1/4" adaptor	4	$1.00
2021 6-6S	3/8" adaptor	3	"
2021 8-8S	1/2" adaptor	4	"
2021 4-6S	1/4"-3/8" adaptor	18	"
2021 4-8S	1/4"-1/2" adaptor	2	"
202702 10-4S	1/4"-7/8" adaptor	24	$2.26
2021 6-8S	3/8"-1/2" adaptor	6	$1.00
2021 8-10S	1/2"-1" adaptor	1	$1.50
2021 6-10S	3/8"-1" adaptor	1	$1.50

Female ends:	Size	#	Cost
	1/4"	36	$3.94
	3/8"	6	$4.21
	1/2"	10	$4.62

Total Cost $549.55

CONTACT:

Mr. Dick Snowden 876-8657
AAA Hose, Fittings & Accessories
654 8th Street, NW
P.O. Box 93362
Atlanta, GA 30318

Material Request Form
Power Group

Hydraulic Filter	
Manufacturer-	Parker- Hannifin
	10 micron spin on line filter
Model number-	12AT10CN15BBLI

Vendor- Grainger
1721 Marrietta Bvld.
Atlanta, Ga, 30318-3646
(404) 355-1984

Catalog Number- 4Z618
Price- $16.83

Appendix D

4182 PAPER

Power Unit and Distribution System for the Lunar Enabler

ME 4182 Group 5

John Chamlee

Rick Canfield

Michael Johnson

Doug Kanipe

James Melchiors

Teresa Powell

March 10, 1993

Letter of Transmittal

March 10, 1993

Mr. J. W. Brazell
School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, GA 30332

Dear Mr. Brazell:

With this letter we are transferring the final report on our ME 4182 project to you. The report gives our recommendations for the power unit and power distribution system for the Enabler. We would like to thank you, Jeff Donnell, and Scott Pierce for your encouragement and assistance.

Sincerely,

John Chamlee, Group Leader

Rick Canfield

Michael Johnson

Doug Kanipe

James Melchiors

Teresa Powell

Executive Summary

The purpose of this paper is to describe the method of supplying power to the Enabler recommended by Group 5, **Power Unit and Distribution System.** The goal of Group 5 was to design a hydraulic system to provide desired pressure and flow to sufficiently power the enabler.

The power unit consists of a five horsepower Briggs and Stratton model 130200 internal combustion engine and a Hagglunds Denison PV6 Variable Piston Pump. This engine would be converted from gasoline to propane to reduce offensive emissions. The power unit is capable of supplying 2000 psi at a constant pressure and a maximum flow rate of 5.372 gallons per minute.

The power distribution system consists of a network of 1/4", 3/8" and 1/2" Weatherhead H104 wire braided Neoprene hydraulic hoses that supply hydraulic pressure to the wheels, the articulating joints, and the boom. Each boom or joint motor requires a Parker Fluidpower D1VW solenoid operated, three-position, directional control valve to control the flow direction. Each wheel motor needs a Parker Fluidpower MEV6 proportional directional control valve for smooth operation in forward and reverse and a Parker D31VW free-wheel / connect valve to control the free wheel and lock positions.

The auxiliary equipment required for the hydraulic system is a reservoir, a filter, and a relief valve. The reservoir is a custom fabricated 1.4 gallon box with internal baffles, a magnet, and a screen. The filter used is a 10 micron Parker 4Z618 and it is inline before the reservoir. The relief valve is a Parker RA 12005. It is located after the pump and it is set at 2300 psi for safety pressure relief.

The 2/3 scale Enabler model needs modifications so that the hydraulic pump and the engine can be located on the vehicle instead of having them external and connected by an umbilical cord. This modification consists of an extension on the body of the vehicle effectively changing one of the t-shaped body sections to a cross-shape.

Table of Contents

List of Figures

Introduction

The ME 4182 class meeting Winter Quarter, 1993 continued the work of other classes to design the Enabler, a lunar work vehicle. The focus was on designing a 2/3 scale working model to be built next quarter by the ME 4192 classes for NASA.

This report describes the specifications for the power unit and power distribution system as determined by Group 5, **Power Unit and Distribution System**. Nine components were examined for the power system. They are:

- Engine
- Pump
- Lines
- Valves
- Reservoir
- Heat Exchanger
- Filter
- Relief Valve
- Accumulator

Each component is individually analyzed to see what the operating requirements are and the equipment is selected to meet these requirements. This report details the hydraulic system required to power the Enabler.

The heart of the system is a Hagglunds Denison variable displacement, axial piston pump powered by a five horsepower Briggs and Stratton internal combustion engine converted to propane fuel. The hydraulic power is distributed to the six wheel motors, the six joint motors and the four boom motors and is controlled by two and three position servo valves. The hydraulic fluid, after flowing through a ten micron filter, returns to a reservoir where air and contaminants are removed.

Engine

The engine is a key component in any hydraulic system. The engine will provide the necessary input power to the hydraulic system and all of its members. The engine which will best meet the requirements of the enabler is a five horsepower, propane powered, Briggs and Stratton Model 130200 internal combustion engine. This engine will optimally satisfy the various constraints imposed by the hydraulic power system of the enabler. The engine will provide adequate power to supply all of the system components. This engine also will be the optimum design and size to be mounted on the body of the enabler. Finally this engine will meet the requirements of the indoor demonstration.

$$hp = P(Q) / 1714$$

The Briggs and Stratton engine meets the power constraints imposed on it by the hydraulic system. The power system of the enabler consists of sixteen individual hydraulic engines each of which require a certain flow rate and pressure to optimally perform their designated tasks. Using the flow rates for the sixteen engines and the above formula it can be shown that the enabler will require at least a 4.5 horsepower engine - a constraint well within the range of the five horsepower Briggs and Stratton. This horsepower figure is generated using the maximum pressure (P) in pounds per inch squared and flow (Q) in gallons per minute under the most adverse conditions (worst case scenario) in order to insure desired performance at all times.

The enabler will be constructed from a twelve inch inside diameter aluminum tube and moving over various, sometimes steep, terrain. The enabler will require its engine to be versatile enough to adapt to the constraints of this environment. Since the Briggs and Stratton is an internal combustion engine, it requires no bulky external power connections or potentially dangerous voltage (Aluminum being an excellent conductor). The size of the

Briggs and Stratton (see Appendix D) makes it easy to place in a engine compartment located at the rear of the craft (see Appendices A-5 to A-8). During operation, the enabler will be required to traverse various objects in its path. While traversing these objects the body of the enabler will be angled, requiring the engine, specifically the oil sump, to operate in this angled position. The Briggs and Stratton engine can be effectively operated at a constant angle of 25 degrees, reaching angles of up to 35 degrees intermittently.

A gasoline powered engine gives off many unpleasant exhaust gasses that can be decreased by conversion to propane power. The Briggs and Stratton can easily be converted from gasoline to propane operation, making it ideal for the indoor demonstration. The propane power system will be obtained at no charge from Combustion Labs Inc. in Riverdale, Ga. The propane power system will consist of a propane cylinder, a series of valves, a regulator, and finally an output port into the intake of the engine (Appendix A-9). For more detailed specifications, drawings and decision information see Appendix D.

Pump

The pump for the enabler is a Hagglunds Denison PV6 Variable Piston Pump. This pump supplies 2000 psi at a chosen variable flow rate of 5.4 gal/min. The Displacement is 0.88 inches^3/rev. The dimensions of the pump (see Appendix E) are small enough so that the pump fits inside the enabler chassis. The weight of the pump is 24 lbs. Additional features include Nine-piston rotating group, fast compensator response and quiet operation.

The requirements of the system on the pump include three different sub systems: the wheels, the articulating joints, and the boom. The calculations for the required flowrate are based on a "worst case" or the maximum power requirement of the combined sub systems. This worst case is defined as four wheels moving along with 4 articulating joints. The wheel motors need a flowrate of 0.7 gal/min, and the articulating joints need a flowrate

0.643 gal/min. If all eight of these motors are run at full power for five minutes the flowrate required would be 5.372 gal/min. Note that the boom will operate only when the enabler is stopped. Therefore, the worst case for the wheel-joint systems is assumed to be a larger drain on the hydraulic system than the boom sub system.

The PV6 pump has two specific features which make it a good choice for this hydraulic system. These features are the Adjustable Compensator and the Standard Maximum Volume Adjustment. The pump actually is able to supply up to 3000 psi however the Adjustable Compensator allows a preset pressure in this case 2000 psi to be maintained. Then a relief valve will be set for a pressure which is approximately 15 -20 percent higher than the 2000 psi operating pressure. The Compensator actually controls the swashplate angle. When the swashplate is perpendicular to the pistons, there is no stroke and therefore no displacement. However when the pump detects a lag in the pressure the Compensator moves the swashplate angle allowing the pistons to reciprocate ,and the pump generates more fluid output, resulting in a pressure increase.

The Standard Maximum Volume Adjustment allows the control of the flowrate leaving the pump. Using the equation:

$$Q = \frac{(D)(RPM)}{231}$$

[Q is flowrate in gal/min, D is displacement in inches^3/rev]

the maximum flowrate produced by the pump may be calculated. The speed supplied by the motor is 3600 rpm. Therefore the maximum flowrate possible is 13.7 gal/min. However, the needed flowrate is 5.372 gal/min for the worst case. So the pump maximum volume adjustment may be set for 5.372 gal/min. The adjustment basically reduces the displacement of the pump to correspond to the speed of the motor and the needed flowrate. This Standard Maximum Volume Adjustment also eliminates the need for an Accumulator.

Hydraulic Lines

The type hose that will be selected for the supply line is a 1/4 inch I.D. Weatherhead H104 hydraulic hose (SAE 100R1 Type AT) with an operating temperature range from -40°F to +212°F (-40°C to +100°C). The hose has a working pressure of 2750 psi and a minimum bursting pressure of 11000 psi. It has single steel wire braid reinforcement, a bending radius of 4 inches (17/32 inch O.D.), and the inner and outer hose material is made of Neoprene - a rubber material compatable with hydraulic oil. The return lines will also be selected as a Weatherhead H104 but one will have a 3/8 inch I.D. (11/16 inch O.D.) and the other will have a 1/2 inch I.D. (13/16 inch O.D.). The working pressure of the 3/8 inch return line is 2250 psi and a minimum bursting pressure of 9000 psi. The working pressure of the 1/2 inch return line is 2000 psi and a minimum bursting pressure of 8000 psi.

The Power Group will use hydraulics as a means of powering the Enabler. The hydraulic lines will serve as a way to transport fluid from the pump to the various motors located throughout the body of the Enabler. There, fluid flow is converted to mechanical rotation and the means by which the Enabler can move is established. Hydraulic horsepower is equal to the flow rate (gallons per minute) times the pressure (pounds per square inch) divided by a conversion factor of 1714. In order for the hydraulic system to be efficient and reliable, certain requirements and a safe configuration of the lines had to be met when selecting the type of hose.

The required working pressure of the Enabler will be 2000 psi and the hydraulic lines will have to meet this requirement. A single steel wire braid reinforcement will be sufficient for our pressure requirement. A maximum flow rate of 2.3 gal/min will also be required from any one supply line. Our calculations with a 1/4 inch supply line operating at

15 ft/sec would provide 2.3 gal/min and will be sufficient for our required flow rate. Higher flow velocities are certainly possible without adverse effects to the system. Hydraulic oil (petroleum base) will be the type of fluid used, so the inner surface of the hose will have to be compatible with the oil. A Neoprene inner and outer coating will be the best option for our application.

The cost of the hydraulic lines is another selection criteria. The lowest cost depends on the vendor and the most economical line configuration possible without affecting the performance of the Enabler. Due to space constraints, the hose will have to have a small bending radius in order to allow line connections in small areas, but not too small as it will incur excessive pressure losses. To have a cost effective line configuraion, two supply lines and two return lines both branching from T's and running the length of the Enabler will be used to supply power to the wheel motors. A single 1/4 inch line will supply power to the joint motors and a single 1/4 inch line will supply power to the boom motors. Both will have there own 3/8 inch return line.

Another problem is the possible torsion of hydraulic lines due to vehicle appendage rotation. The solution imposed is to actually coil the hydraulic lines through the body cavity of the Enabler to allow for rotation by winding the lines and unwinding them. A minimum number of coils is set at 2 coils per each section or 2 coils between each motor.

For additional information and detail drawings see Appendix F.

Servo-Valves

The servo-valves for the enabler are Parker Fluidpower directional control valves. For the wheels, two different types of valves will be needed: proportional directional control valves (Series MEV6) and free-wheel / connect valves (Series D31VW). For the articulating joints and the boom joints a solenoid operated, three-position, directional control valve (Series D1VW) will be used.

The two valves selected to control the direction of the fluid to the wheels will be solenoid operated, proportional, directional control valves and two position free-wheel / connect valves. The proportional control valve will have three positions. The center position will be stop, which will not let any flow through. The left position will be the forward position, which will drive the wheels in the forward direction. The right position will be the reverse position, which will reverse the flow of the fluid and drive the wheels in the opposite direction. The Series 6 valves provide precise and variable speed control without lurching when the wheels start moving. They are controlled by proportional solenoids, which provide an output flow that is proportional to the input signal. Because of the proportional signal, these valves will provide precise metering from minimum to full speed. The maximum flow for each wheel will be 0.7 GPM and this will correspond to a 12 VDC control signal. For the valve to be halfway open, a control signal of 6 VDC is required. The two position free-wheel / connect valve will be placed between the proportional valve and the wheel motor. This valve has two positions. The first position will be the free-wheel position, which will allow the wheel to move without flow being supplied to the motor. The connect position allows the wheel to operate as it normally would. The enabler will require one proportional control valve and one free-wheel / connect valve for every wheel.

The valves for the articulating joints will be a three position, solenoid operated, directional control valve. The three positions will be the same as the proportional valve for the wheels. The center position will be stop, the left position will be forward, and the right position will be reverse. These valves supply a sudden burst of fluid to the motors and not a gradual build up to the maximum flow as the proportional valves do. This sudden flow rate will cause the joints to jerk as they try to start turning. It is preferred to use the proportional control valves to eliminate the jerking of the joints and to control the different flow rates required by the motors, but because the proportional valves cost considerably more than the regular control valves, the above valves are recommended. A total of six valves are needed for the articulating joints.

The valves for the boom joints will be the same as the valves for the articulating joints. These valves will control the direction of the fluid supplied to the motors for the joints. Again it is preferred to use the proportional control valves used for the wheels to prevent the boom from jerking as it moves, but because of monetary restrictions we will recommend the normal directional control valve. A total of four valves will be required for the boom joints.

For more detailed specifications and details on the valves see Appendix G.

Reservoir

Due to the constraints of weight and space as well as the uniqueness of the system, it has been decided that the hydraulic reservoir should be custom made at the machine shop at Georgia Tech. This will allow for the optimum design for the system as well as a competitive cost for the reservoir. The total capacity is to be 1.4 gallons and the overall dimensions are approximately: Length- 9 in. Height- 6in. Width- 6 in. The complete drawings and dimensions as well as description can be found in Appendix H.

Typically the volume of a reservoir is 2 to 3 times the mean flow rate. This would correspond to a 13 gallon reservoir, and several vendors supply reservoirs in the 10 to 15 gallon range. However, the system does not require the reservoir to act as a heat exchanger so this volume can be reduced. Since weight is a main consideration, the 1.4 gallon reservoir will be sufficient.

The hydraulic system is part of a mobile vehicle so the reservoir must be pressurized in order to keep the outlet to the pump flooded. Most vendors do not supply this type of reservoir. This is another reason why the decision to custom make the reservoir was made.

Filter

The filter used is a hydraulic return line spin-on filter. The filter has a 10 micron cellulose medium capable of up to 20 GPM. The maximum pressure rating for the filter is 150 psi and the maximum temperature rating is 225°F. The filter is manufactured by Parker and is sold through Grainger. The filter has 3/4 in ports that will require couplings to accommodate the size difference of the return lines.

The main advantage to this filter is the fact that it is inexpensive and readily available. This filter will meet all the requirements of the system and is easily maintained.

Relief Valve

The relief vâlve for the system is a Parker RA1200S direct-operated pressure relief valve. Its maximum flowrate is 20 gal/min, its maximum operating pressure of 3000 psi. the relief valve will be manually set for 2300 psi. The position of the relief valve will be directly after the pump. This pressure relief will protect the system from harmful pressure increase due to a pump malfunction.

Bill of Materials

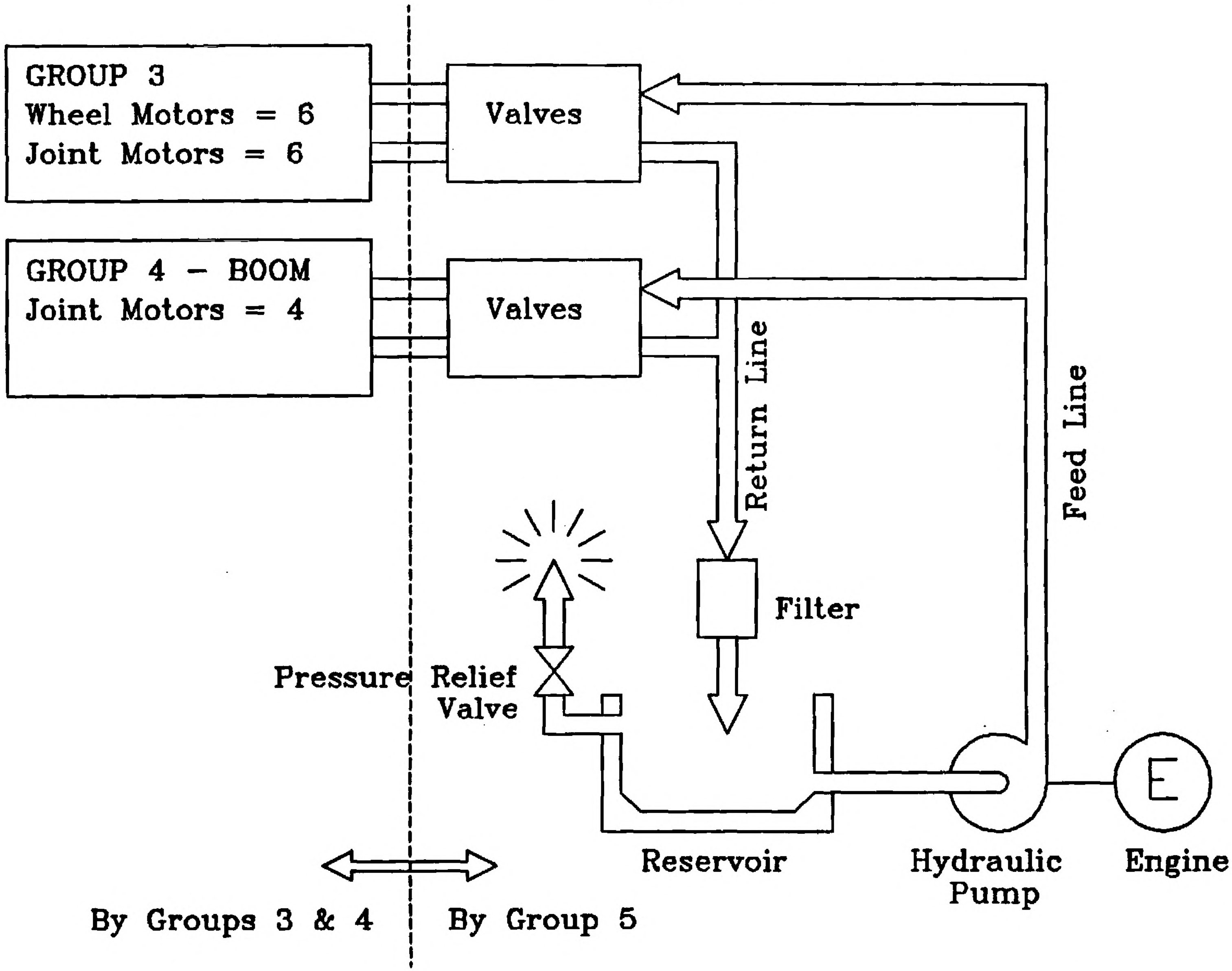
Component	#	Vendor	Model No.	Per Cost	Total Cost
Engine	1	Briggs & Stratton	130200	$253.50	$253.50
Pump	1	Hagglunds Denison	PV6	$714.00	$714.00
Proportional 3-way valves	6	Parker Fluidpower	MEV6	$700.00	$4,200.00
2-way valves	6	Parker Fluidpower	D31VW	$100.00	$600.00
3-way valve	10	Parker Fluidpower	D1VW	$150.00	$1,500.00
Relief valve	1	Parker	RA1200S	$100.00	$100.00
Filter	1	Parker	4Z618	$15.96	$15.96
Reservoir	1	Fabricated	N/A	$75.00	$75.00
1/4" Hydraulic hose	99 ft	Wheatherhead	H10404	$1.67 /ft	$158.65
3/8" Hydraulic hose	99 ft	Wheatherhead	H10406	$1.80 /ft	$171.00
1/2" Hydraulic hose	1 ft	Wheatherhead	H10408	$2.00 /ft	$2.00
3/8" cross fitting	2	Aeroquip	20806-6	$11.16	$22.32
1/4" female swival end	140	Aeroquip		$3.94	$551.60
3/8" female swival end	55	Aeroquip		$4.12	$226.60
1/2" female swival end	6	Aeroquip		$4.50	$27.00
1/4"-1/4" adapter	115	Aeroquip	2081 4-4	$1.00	$115.00
3/8"-3/8" adapter	30	Aeroquip	2081 6-6	$1.00	$30.00
1/2"-1/2" adapter	5	Aeroquip	2081 8-8	$1.00	$5.00
3/8"-1/4" adapter	19	Aeroquip	2081 6-4	$1.00	$19.00
1/2"-3/8" adapter	1	Aeroquip	2081 8-6	$1.00	$1.00
7/8"-1/4" adapter	24	Aeroquip	202702 10-4S	$3.50	$84.00
1/4" "T" fitting	13	Aeroquip	2090 4-4	$7.25	$94.25
3/8" "T" fitting	12	Aeroquip	2090 6-6	$7.78	$93.36
3/8" cross fitting	2	Aeroquip	2080 6-6	$11.16	$22.32
1/4" flow control valve	6	Aeroquip	2090 4-4	$3.60	$21.60

Appendices

APPENDIX A	**Overall System**
APPENDIX B	**Accumulator**
APPENDIX C	**Heat Exchanger**
APPENDIX D	**Engine Information**
APPENDIX E	**Pump Information**
APPENDIX F	**Line Information**
APPENDIX G	**Valve Information**
APPENDIX H	**Reservoir Information**

APPENDIX A

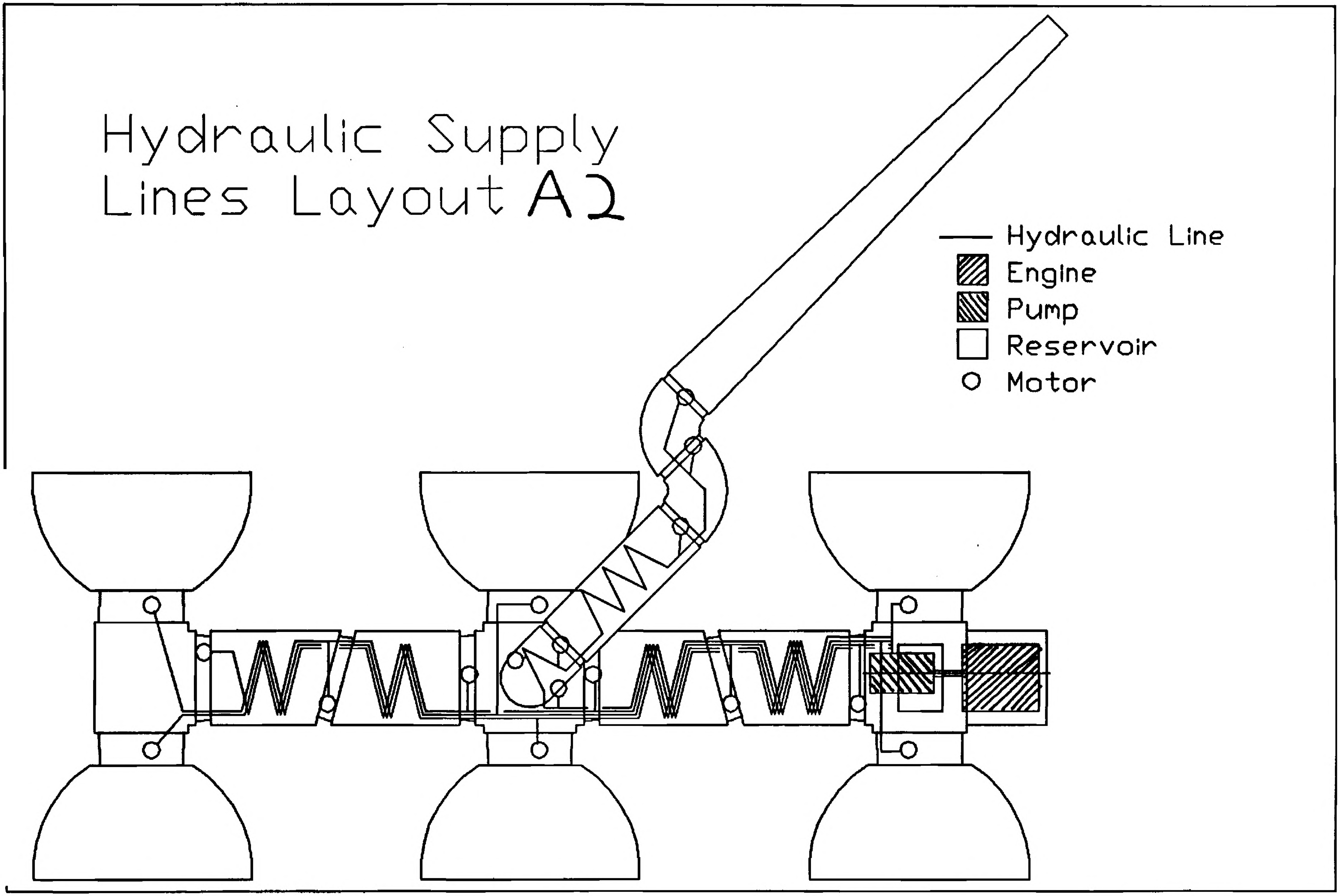
Flow Diagram A1



Power Supply – Group 5

ME4182d
Revision 4

Hydraulic Supply Lines Layout A2



APPENDIX B

Appendix B

Accumulator

If additional fluid flow above the capacity of the pump is required, then a hydraulic accumulator can be added to the system. The accumulator should be of the gas-piston type so that it can be mounted horizontally. Also, it should have a maximum working pressure above the pressure of the system.

In order to determine the additional volume of fluid that is required; the following equation can be used.

$$(GPM_{required} - GPM_{max}) \text{ X Time} = \text{Volume required} = Vw$$

After this volume is determined, the equation below can be used to calculate what size accumulator is needed.

$$V1 = \frac{vw(P3/P2)^{1/f}}{e(.95)[(P3/P2)^{1/n} - 1]}$$

where:

P3=maximum system pressure

P2=minimum system pressure

P1=gas precharge pressure

f=charge coefficient

n=discharge coefficient

e=gas charge ratio P1/P2

V1=accumulator size required

Vw=required volume of fluid

In order to save weight, a smaller accumulator can be used in conjunction with a small gas bottle. However, it must satisfy the following equations.

$$V_{accumulator} + V_{bottle} > V_1$$

$$V_{accumulator} > (1.2)(Vw)$$

This type of setup will allow the whole volume of the accumulator to be used.

PHE Hydraulics inc. supplies various size accumulators of the piston variety. These sizes range from 1/2 gallon to 5 gallons. Also, this company can supply a gas bottle system compatible with the accumulator. The prices of these accumulators range from $100 to $300.

APPENDIX C

Appendix C

Heat Exchanger

The hydraulic system generates heat as the fluid flows through it. If the heat causes the temperature of the oil to rise too high, then a heat exchanger is needed. According to the estimates we made, there should not be a need for a heat exchanger. However, these are only estimates; so when the system is actually built, the system should be tested and then it should be determined if in fact a heat exchanger is actually needed. The present system is readily adaptable to the addition of a heat exchanger.

HEAT EXCHANGER

Pump -> 15 % Loss => 5 hp X .15 = .75 hp
.75 hp = 1908 Btu/hr

motors -> very efficent but assume 10% Loss
= 1200 Btu/hr

Relief Valve -> 1.48 x psi x GPM = 1.48 x 2000 x 5.4
= 15984 x .1 = 1598 Btu/hr

Valves -> 1.48 x psi x GPM = 1.48 x 2000 x .4
= 1184 x 6 = 7104 Btu/hr

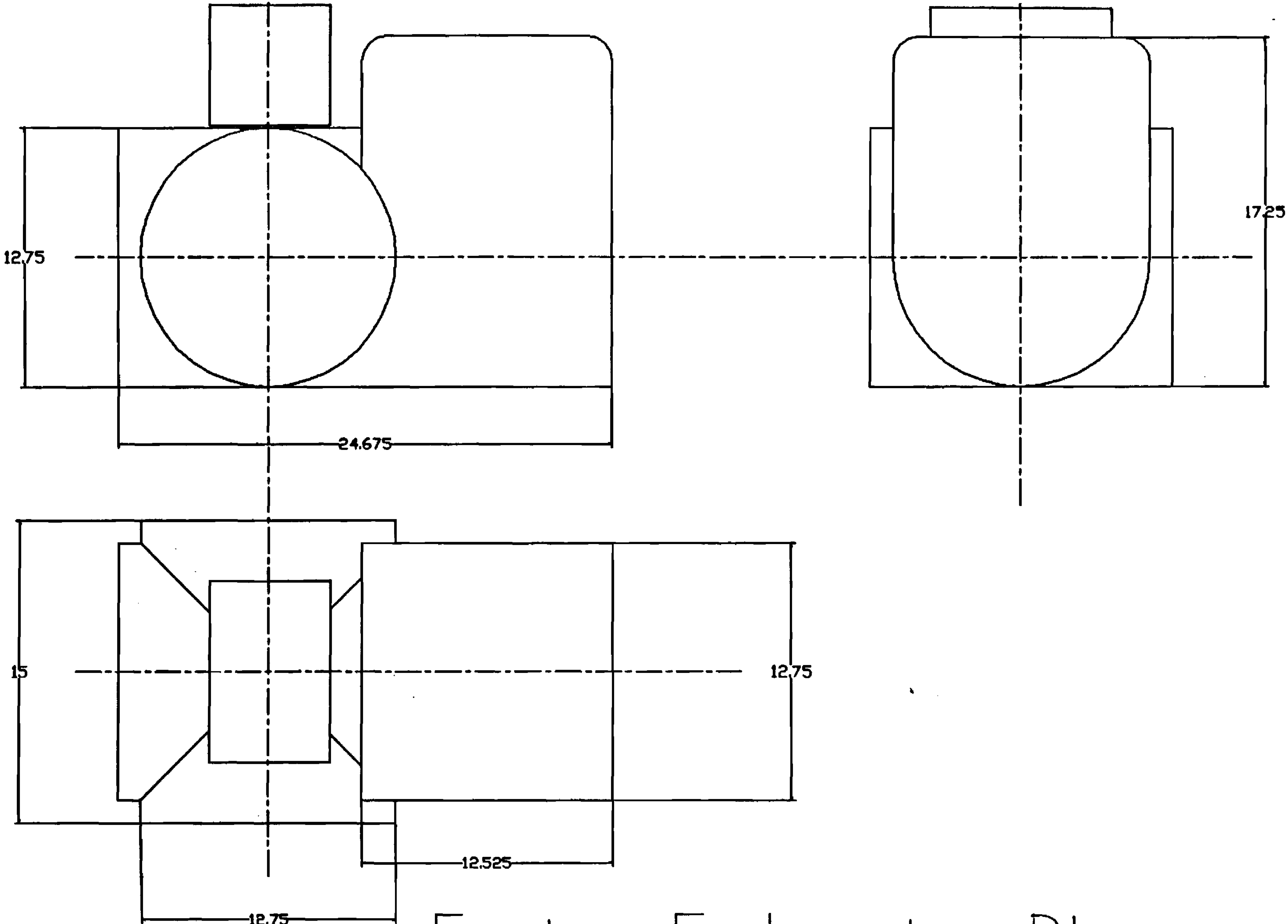
Lines 10% of hp = .5 hp = 1273 Btu/hr

Total 13,000 Btu/hr

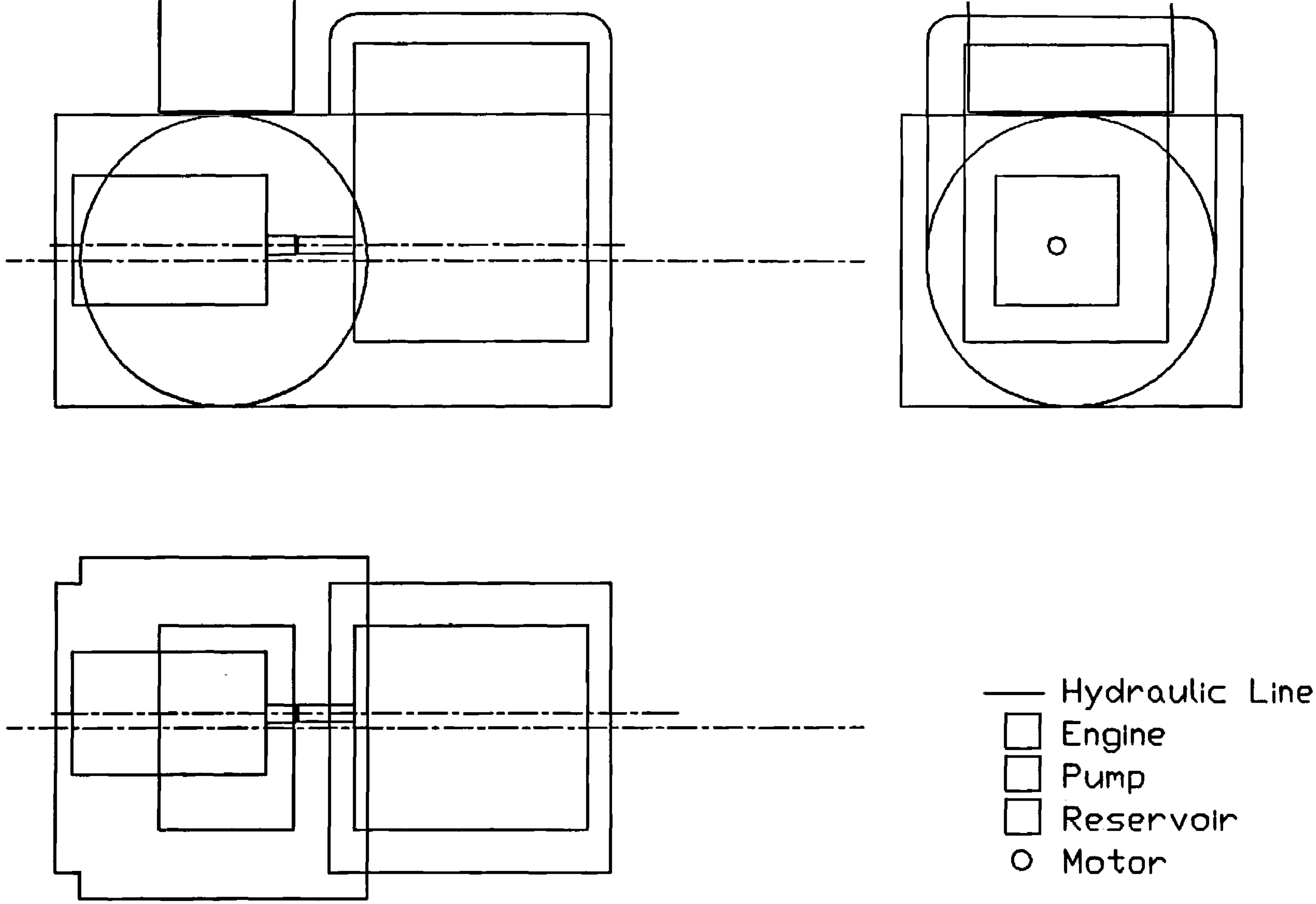
Books => > 15,000 Btu/hr then need a heat exchanger.
So NO Heat Exchanger Needed

However, these are only estimates. The System should be built and tested for heat output. If oil temp > 200°F then heat exchanger is needed

APPENDIX D

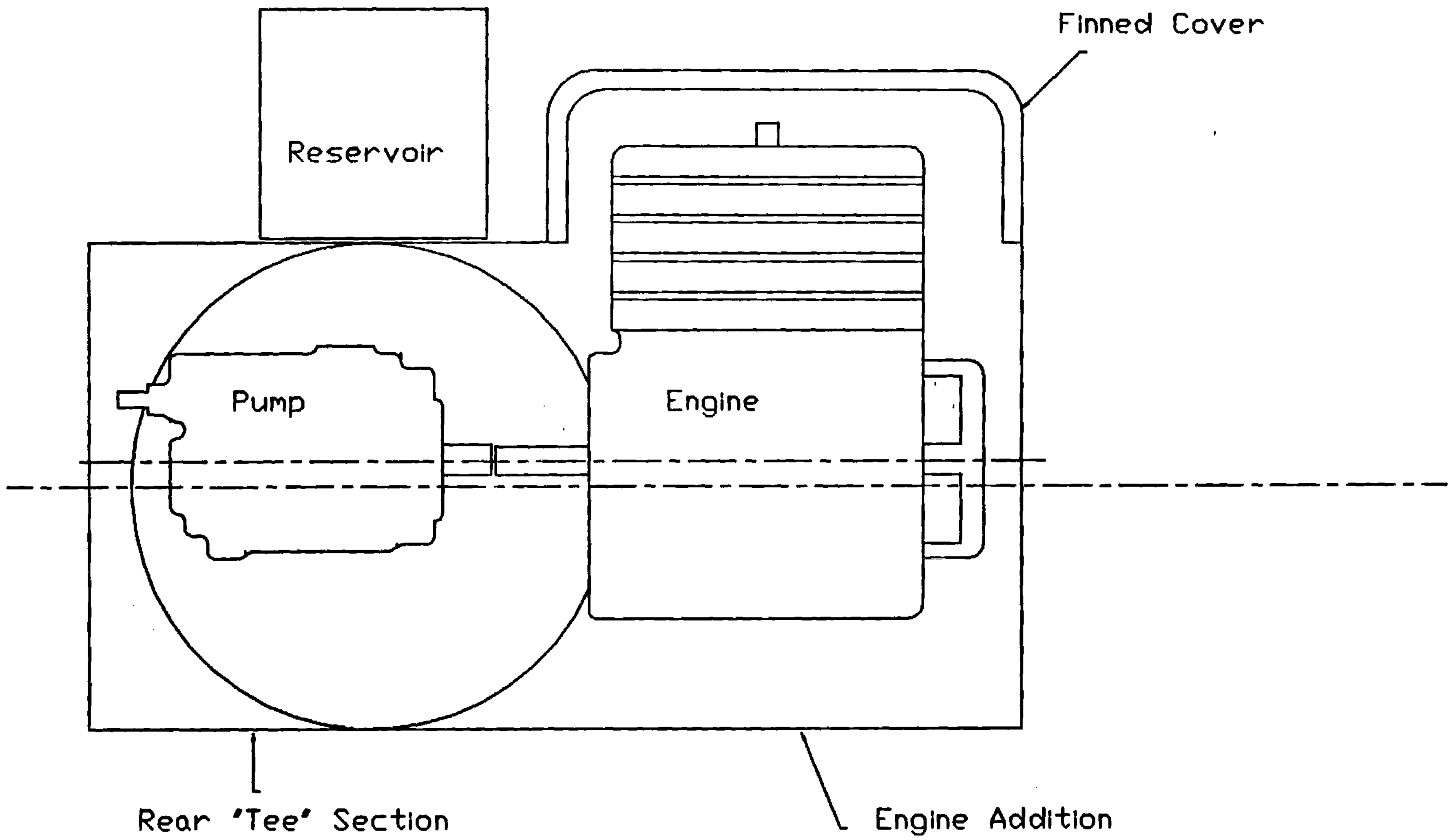


Engine Extension D1



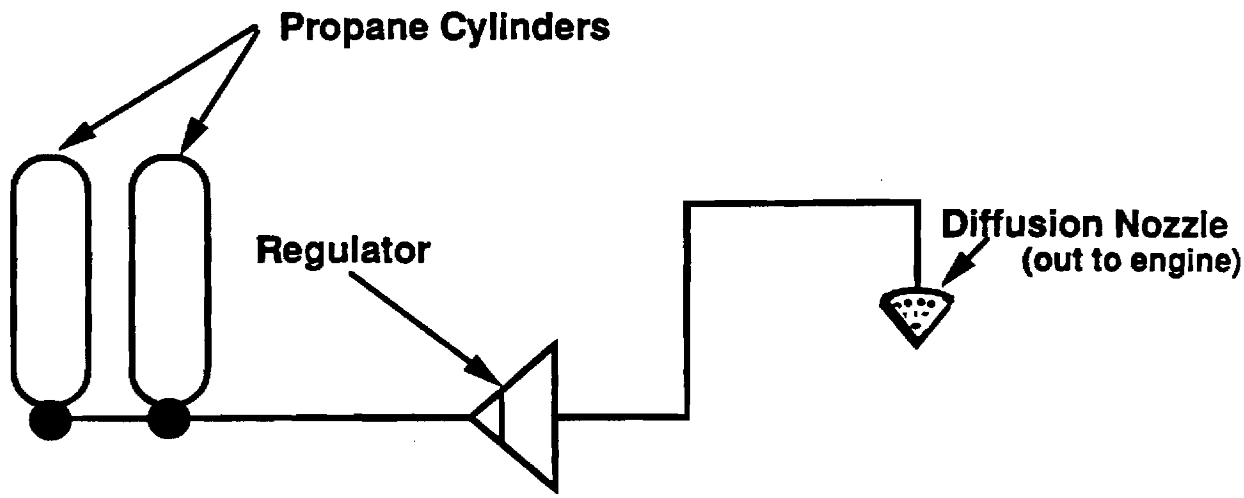
Engine Extension

Showing Engine, Pump and Reservoir D2

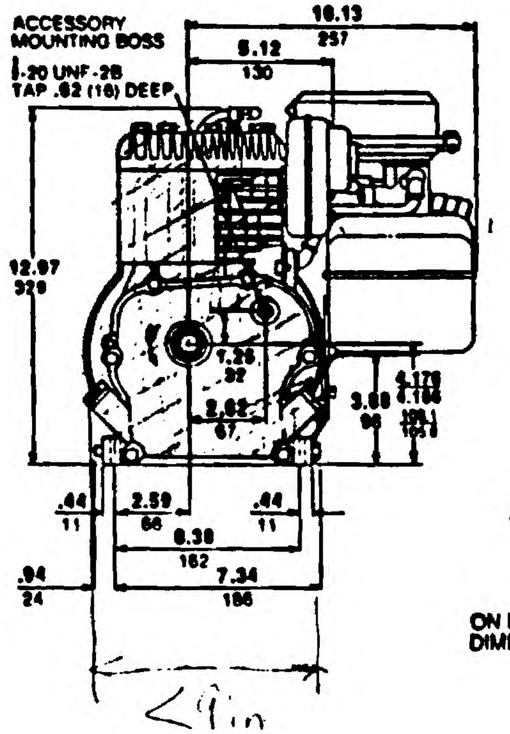


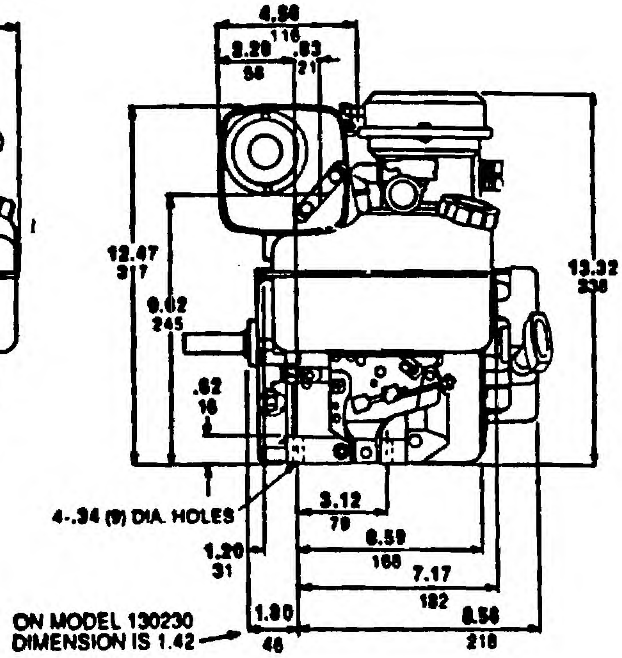
Cross Section D3

D4
PROPANE SYSTEM LAYOUT

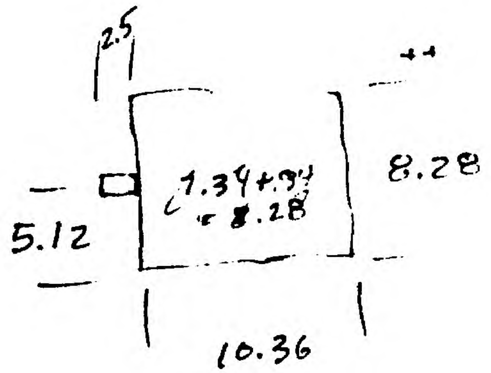


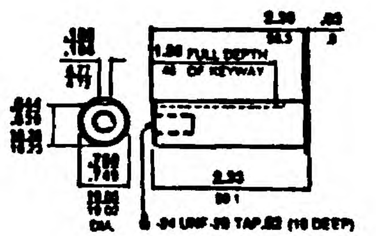
D5 Model Series 130200—5 HP



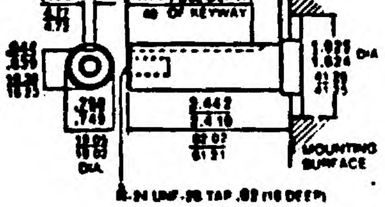


1.80
8.56
10.36

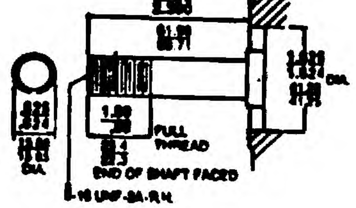




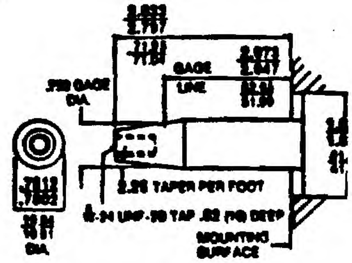
MODEL 130202 - C/S NO. 301079
3/4" DIA. .844" KEYWAY 1-7/8" LONG DBT 5/16-24



MODEL 130232 - C/S NO. 301730
3/4" DIA. .844" KEYWAY 1-7/8" LONG DBT 5/16-24

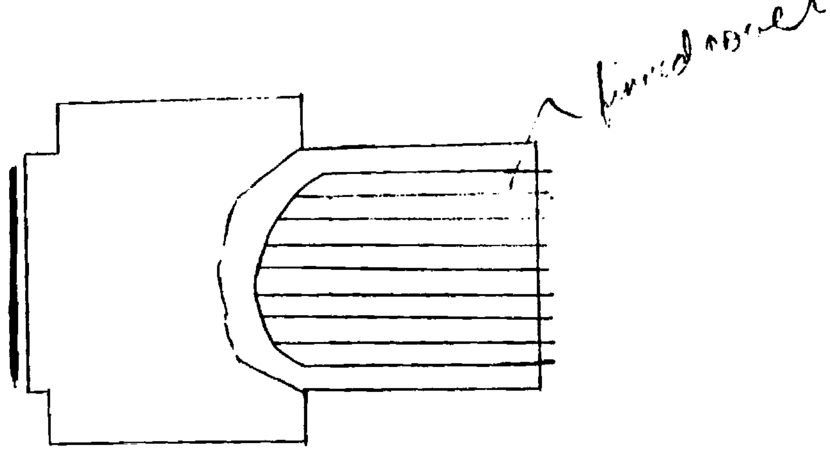


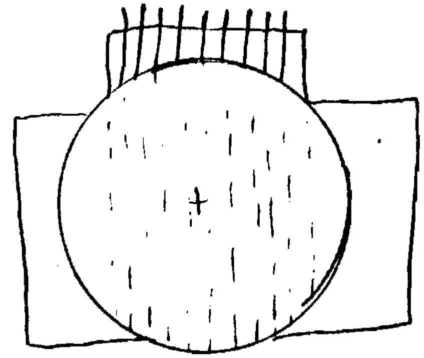
MODEL 130232 - C/S NO. 301732
5/8" DIA. 1" THREAD 5/8-18

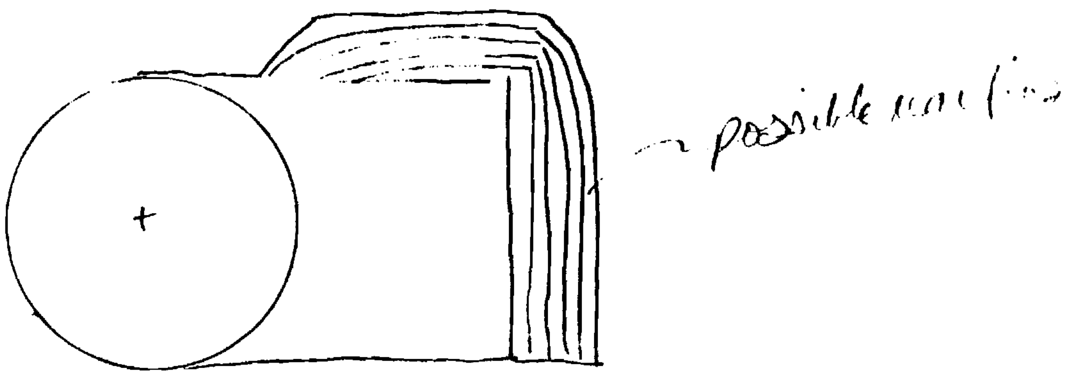


MODEL 130232 - C/S NO. 301733
3/4" DIA. TAPERED DBT 5/16-24

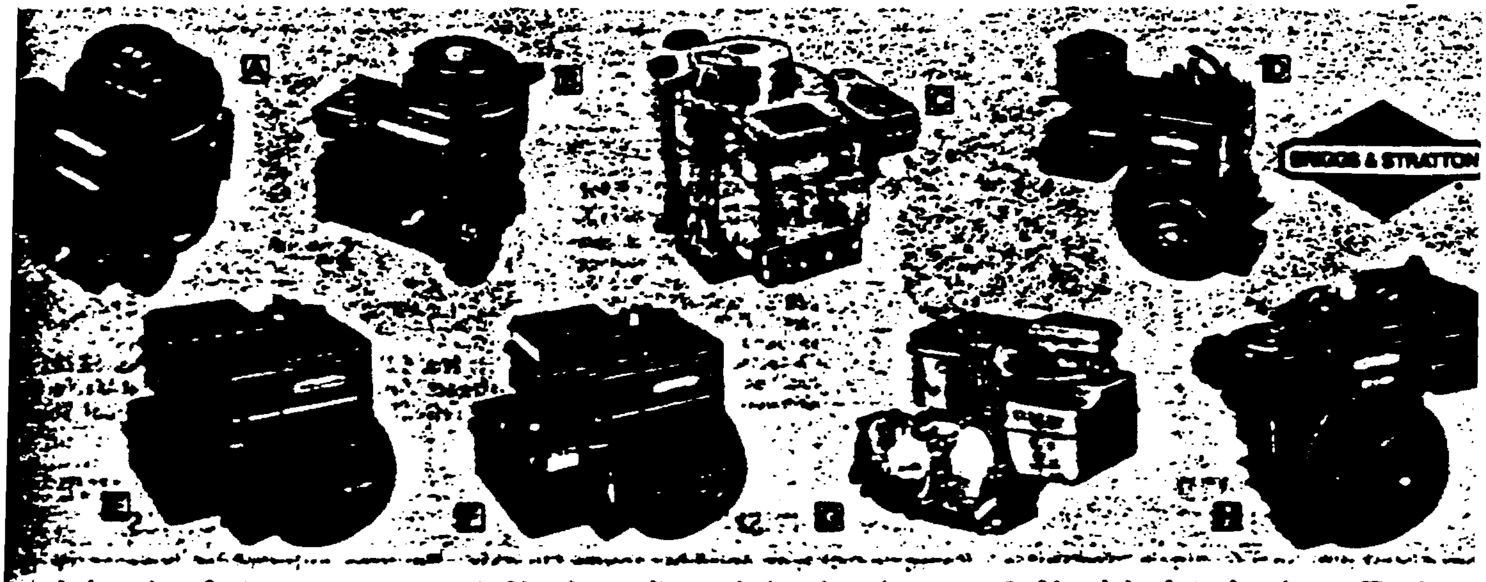
Engine Compartment Sketch







BRIGGS & STRATTON AIR-COOLED 4-CYCLE GASOLINE ENGINES



...cooled engines feature maintenance-free Magnetron® selective storage-type electronic ignition. No servicing needed, except spark plug

- ...h silicon-copper aluminum alloy cylinder and crankcase dissipate engine heat
- ...at treated ductile iron crankshaft with integral counterweights and ...shaft drive gear
- Aluminum alloy main bearings (except one ball bearing on power take-off side on Nos. 32904, 42684, 42690, 42689 & 42692)
- Counterclockwise (CCW) rotation when viewed from power take-off side
- Cold headed steel intake valve and austenitic forged steel exhaust valve
- Oil Foam™ or paper pleated cleaner exceeds automotive standards for filtration
- Aluminized steel, quiet muffler has longer life
- Positive-type recoil starter with dust sealed, self-lubricating anti-rollback starter clutch

BRIGGS & STRATTON LIMITED WARRANTY

Briggs & Stratton warrants vertical-shaft engines (... 42688, 42691, and 32906) for 2 years. All other engines for one year. Text of warranty available on request. "Manufacturers' Warranties" on page opposite inside back cover.

VERTICAL SHAFT ENGINES

...No. 42688 3.5 HP engine with 9.02 cu-in displacement. Pulsa-jet carburetor with single mixture adjustment and auto...

...No. 42691 3.0 HP engine with 12.57 cu-in displacement. Pulsa-jet carburetor with single mixture adjustment and sintered alloy insert valve seat.

...No. 32906 8.0 HP engine with 19.44 cu-in displacement. Float carburetor with ...and power mixture adjustment. Sintered alloy insert for intake valve and ...ess steel insert for exhaust valve. ...el tank.

HORIZONTAL SHAFT ENGINES

...No. 42687 2.0 HP engine with 6.65 cu-in displacement. Pulsa-jet carburetor with single mixture adjustment and sintered alloy insert valve seat.

...Nos. 42689 and 42690 3.0 HP engines with 7.75 cu-in displacment. Pulsa-jet carburetor with front mount control panel and sintered alloy insert valve seat.

... G Nos. 42683, 42684, 42692, and ...5.0 HP engines with 12.57 cu-in displacement. Pulsa-jet carburetor with ...mount control panel and sintered insert valve seat.

...No. 32904 8.0 HP engine with 19.44 cu-in displacement. Float carburetor with ...and power mixture adjustment. Sintered alloy insert for intake valve and ...less steel insert for exhaust valve.

BRIGGS & STRATTON ENGINE SPECIFICATIONS

Key	HP	Stock No.	Cylinder Bore	Cylinder Stroke	Cu. In. Displ.	Shaft Dia.	Shaft Configuration	Carburetor Type	Start
VERTICAL CRANKSHAFT ENGINES									
A	3.5	42688	[illegible]	[illegible]	9.02	[illegible]	Two Woodruff Keyway	Pulsa-jet	Recoil
B	3.0	42691	[illegible]	[illegible]	12.57	[illegible]	Two Woodruff Keyway	Pulsa-jet	Recoil
C	8.0	32906	3	[illegible]	19.44	1	Straight Keyway	Float	Recoil
HORIZONTAL CRANKSHAFT ENGINES									
D	2.0	42687	[illegible]	[illegible]	6.65	[illegible]	Straight Keyway	Vari-jet	Recoil
E	3.0	42689	[illegible]	[illegible]	7.75	[illegible]	Straight Threaded	Pulsa-jet	Recoil
	3.0	42690	[illegible]	[illegible]	7.75	[illegible]	Int. Thrd. w/Keyway	Pulsa-jet	Recoil
F	5.0	42683	[illegible]	[illegible]	12.57	[illegible]	Straight Keyway	Pulsa-jet	Recoil
	5.0	42692	[illegible]	[illegible]	12.57	[illegible]	Straight Threaded	Pulsa-jet	Recoil
	5.0	42684	[illegible]	[illegible]	12.57	[illegible]	Int. Thrd. w/Keyway	Pulsa-jet	Recoil
G	5.0	42686	[illegible]	[illegible]	12.57	[illegible]	PTO 6:1 Reduction	Pulsa-jet	Recoil
H	8.0	32904	3	[illegible]	19.44	1	Straight Keyway	Float	Recoil

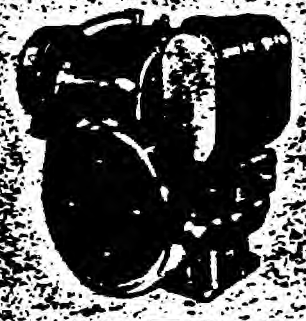
BRIGGS & STRATTON ENGINE ORDERING DATA

Key	HP	Applications	Overall Dimensions Length	Height	Depth	B&S Model	Stock No.	List	Each
VERTICAL CRANKSHAFT ENGINES									
A	3.5	Mower	[illegible]	[illegible]	[illegible]	92902-5015	42688	$180.80	$171.76
B	3.0	Mower	[illegible]	[illegible]	[illegible]	130902-1016	42691	275.00	261.82
C	8.0	Mower	[illegible]	[illegible]	[illegible]	191702-4015	32906	344.80	327.86
HORIZONTAL CRANKSHAFT ENGINES									
D	2.0	General	[illegible]	[illegible]	[illegible]	60102-4015	42687	203.40	193.23
E	3.0	Pump	[illegible]	[illegible]	[illegible]	80232-4030	42689	[illegible]	212.04
	3.0	Pump	[illegible]	[illegible]	[illegible]	80232-4035	42690	[illegible]	211.19
F	5.0	General	[illegible]	[illegible]	[illegible]	130202-4015	42683	257.30	244.24
	5.0	Pump	[illegible]	[illegible]	[illegible]	130232-4035	42692	283.40	269.23
	5.0	General	[illegible]	[illegible]	[illegible]	130232-4036	42684	283.80	269.61
G	5.0	Mixer	[illegible]	[illegible]	[illegible]	130252-4049	42686	340.10	323.10
H	8.0	General	[illegible]	[illegible]	[illegible]	190432-5535	32904	397.40	377.63

Included for Comparison (Kohler is larger & More $)

KOHLER 4-CYCLE GASOLINE ENGINES

KOHLER



No. 32793
10, 12, 14 and
16 HP Electric Start

No. 42356
8 HP Recoil Start

Replacement parts available. See page 1822 for listing.

No. 32971
23 HP Electric Start

...on construction of Kohler air-...4-cycle, 3600 RPM gas engines pro-...durability, dependability and long ...idely used in industry, agriculture ...nstruction for garden tractors, liq-...mps, compressors, etc. Gives excel-...ld weather starting, especially for ...

...positive starting is provided by ...tented automatic compression re-...(ACR) feature. It releases com-...n during cranking and automati-...restores full power when engine ...

...t, reliable gear drive starter on ...c start models eliminates belts and ...etely disengages to give more usea-...rsepower.

...learance baffling permits chaff, ...lippings and other airborne debris ...through and away from engine. This results in cool running engine and long engine life.

Float-feed carburetor provides for remote throttle and choke hookup. Air cleaner has replaceable dry-type paper element. Fuel tank capacity of all models is 1.5 gallons except No. 32971 which has no fuel tank.

Heat-treated ductile iron crankshaft has integral counterweights and includes 7/16"-20 x 1½" deep threaded hole in PTO end of shaft. No. 32971 has 5/8-18, 1½" deep threaded hole. Antifriction ball bearings on both ends of horizontal crankshaft, except No 32971 which also has a sleeve bearing on the flywheel end. CCW rotation when viewed from PTO end of crankshaft. Pistons are cam-ground, permanent mold aluminum alloy.

Other engine features include precision oil-bathed internal flyweight-type governor, Stellite* faced exhaust valves with rotators, Stellite* faced valve seats, oil filler tube and dipstick, and air intake precleaner element.

8 to 16 HP single cylinder, L-head engines feature magneto ignition with easily adjustable external breaker points. Stop button is mounted on breaker cover. High tension spark plug wire with boot facilitates all-weather starts. Electric start models include battery ignition, 15 amp flywheel alternator system and instrument panel.

23 HP horizontally opposed twin cylinder, L-head engine No. 32971 features cast-iron crankcase, full pressure lubrication with spin-on oil filters, and aluminized dual mufflers for quiet operation and rust resistance. Has control panel consisting of choke, throttle, oil pressure gauge, ammeter, start button and stop switch—all positioned for easy operation. Has battery ignition with resistor-type spark plugs. 12 volt electric start with 15 amp flywheel alternator system.

Stock No.	Cylinder Bore	Cylinder Stroke	Cu-in Displ.	Shaft Dia.	Shaft Configuration	Carburetor Type	Carburetor Start	Fuel Tank
42356	2.94"	2.75"	18.64	1"	Straight Keyway	Float Feed	Recoil	6 Qt.
32791	2.94	2.75	18.64	1	Straight Keyway	Float Feed	Electric	6
32793	3.25	2.88	23.85	1.125	Straight Keyway	Float Feed	Electric	6
32795	3.38	3.25	29.07	1.125	Straight Keyway	Float Feed	Electric	6
32969	3.50	3.25	31.27	1.125	Straight Keyway	Float Feed	Electric	6
32796	3.75	3.25	35.90	1.125	Straight Keyway	Float Feed	Electric	6
32971	3.00	3.00	57.73	1.437	Straight Keyway	Float Feed	Electric	None

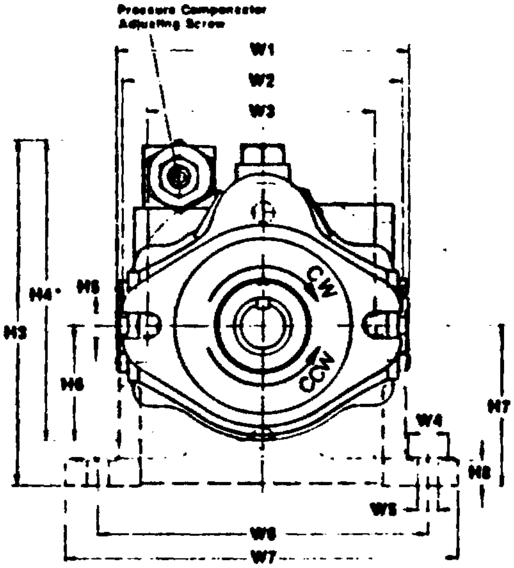
Applications	Overall Dimensions, in. Length	Height	Depth	Spec. No.	Kohler Model	Stock No.	List	Each	Shpg. Wt.
General	14.50	20.06	15.89	30766	K181T	42356	$710.45	$666.34	79.0
General	14.50	20.06	13.91	30767	K181S	32791	844.56	792.83	80.0
General	19.125	18.562	15.66	46810	K241S	32793	1054.26	985.32	150.0
General	19.125	18.562	15.66	47739	K301S	32795	1105.17	1034.77	144.0
General	19.125	18.562	15.66	60426	K321S	32969	1181.85	1119.00	145.0
General	20.97	18.562	15.66	71309	K341S	32796	1233.07	1153.67	146.0
General	20.5	22.06	18.94	36340	K582S	32971	2432.40	2283.89	202.0

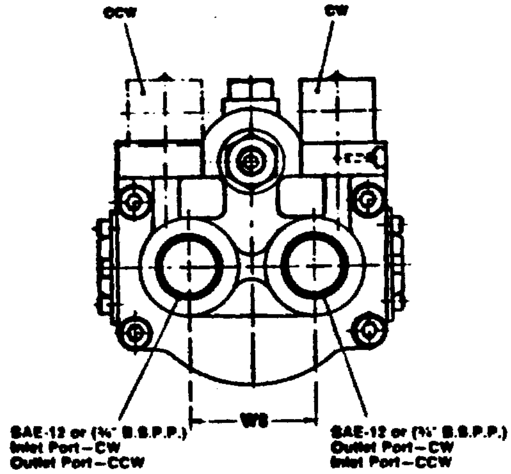
GAS ENGINE CHAIN SAWS AVAILABLE

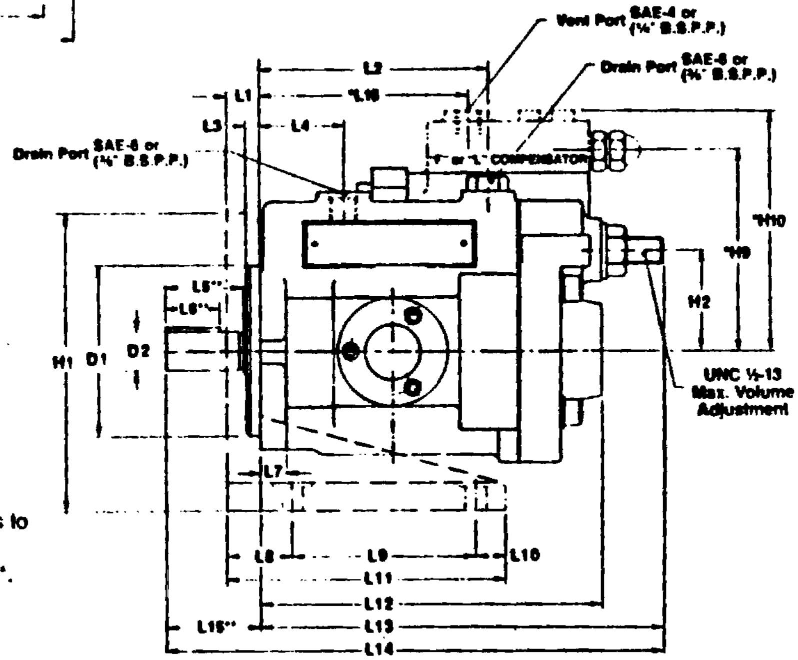
...ws listed in this catalog can handle nearly ...tting job. Medium-duty chain saws tackle the ...wner's tree trimming chores. Heavy-duty saws ...r cutting jobs of farmers and forestry crews, as well as being precision built for the heavy-duty demands of tree service operators who cut every working day. Complete product descriptions available.

See Index under Chain Saws.

APPENDIX E







Installation Drawing 23-9596
Assembly Drawing SD-01537
Optional Foot Bracket S14-00128

*"F" or "L"—Ventable Compensator
"T"—Power Limiter (See page 10)

Note: Two bolt mounting flange conforms to SAE "A" specifications, except for dimensions marked with asterisks.

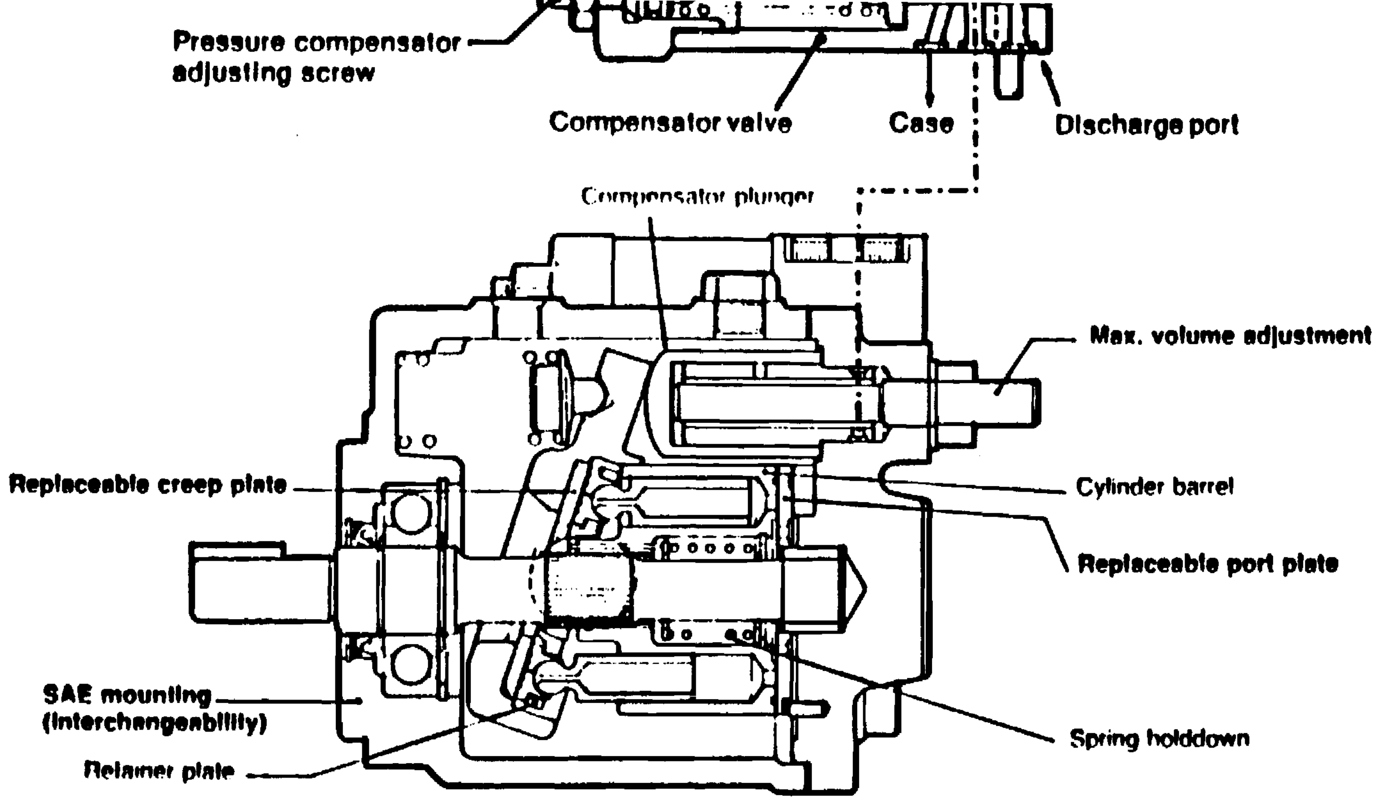
Installation Dimensions
PV6

	L1	L2	L3	L4	L5	L6	L7	L8	L9	L10	L11	L12	L13	L14	L15	L16	H1	H2	H3
Inch	0.59	4.27	0.25	1.57	See Below	See Below	0.49	1.19	3.50	0.56	5.25	6.43	8.1	9.89	See Below	4.11	5.64	1.89	6.48
mm	15.1	108.5	6.4	39.9	See Below	See Below	12.4	30.2	88.9	14.3	133.4	163.3	205.7	251.2	See Below	104.4	143.2	48	164.7

	H4	H5	H6	H7	H8	H9	H10	W1	W2	W3	W4	W5	W6	W7	W8	D1	D2
Inch	5.62	0.47	2.15	3.0	0.46	3.78	4.56	5.43	5.12	4.17	0.75	0.41	6.13	7.25	2.25	3.250 3.248	See Below
mm	142.7	11.9	54.6	76.2	11.7	96	115.8	138	130	106	19	10.3	155.6	184.2	57.2	82.55 82.50	See Below

Shaft Dimensions
PV6

						**Key Shaft Dimensions				**Spline Shaft Dimensions				
Shaft Code	Shaft Type	DIM.	L5	L6	L15	D2 Dia.	Sq. Key Section	Dim. Overkey	Key Lgth.	Major Dia. D2	No. Teeth	Pitch	Press. Angle	Spline Lgth. L6
1	Splined SAE-B	inch	1.30	X	1.63	X	X	X	X	.875/.853	13	16/32	30°	0.98
		mm	33.0		41.4					22.2/21.7				24.9
2	Keyed SAE-A	inch	1.46	1.00	1.79	.750/.749	.1884/.1874	.832/.827	1.00	X	X	X	X	X
		mm	37.1	25.4	44.5	19.05/19.02	4.785/4.760	21.13/21.00	25.4					



PRESSURE COMPENSATOR CONTROL WITH MAXIMUM VOLUME ADJUSTMENT

The C and F pressure compensator control allows the pump to deliver full volume from the outlet port until the pressure rises to the value set by the control. One turn clockwise of the pressure compensator adjusting screw represents a pressure increase of approximately 650 PSI (44.8 bar).

The control then reduces the pump volume to that required by the system while maintaining the preset pressure at the outlet port. The stroking piston is controlled by a 3 way valve which is shifted by discharge pressure.

The fast response (typically 50 ms off stroke and 120 ms on stroke) and high flow capacity of this valve holds pressure overshoot and undershoot to a minimum.

The minimum compensating pressure is 130 PSI (9 bar). An adjusting screw complete with locknut allows the pump volume to be set between maximum and zero.

Clockwise rotation pumps have the pressure compensator control located on the left side of the pump body, on counter clockwise rotation pumps the control is on the right side.

APPENDIX F

Line Configuration

1/4" cross sec. area = 4.9×10^{-6} in^2 Q = 530.14 in^3/min = 2.29 gal/min

5/16 = 7.67×10^{-6} in^2 Q = 828.35 in^3/min = 3.586 gal/min

1/2 return worst case 5.8 gal/min = 22.33 in^3/sec   velocity = 113.73 in/sec = 9.5 ft/sec

3/8   Q = 5.164 gal/min

configuration

wheels   assume T split occurs prior to rear wheel/axle centerline

2 × (233.7 in + 2ft = 40.95 ft of 1/2 in   add 4 ft

40.95 ft of 3/8 in   "   "

joints   246.5 + 2 ft = 22.54 ft   22.54 ft of 1/4   add 3 ft

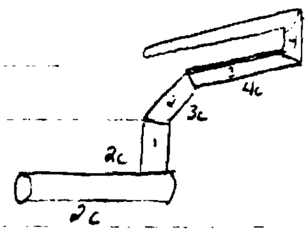
22.54 ft of 3/8   "   "

single supply line
with 2 turns between joints
except 1 turn for 1st joint

boom   284.1 in + 2 ft = 25.67 ft

25.67 ft of 1/4   add 2 ft

25.67 ft of 3/8   "   "



2	24
3	56
4	22

additions

supply   6 in 3/8 from pump to cross

6 in 3/8 from cross to T (wheel)

6 in 1/4 from cross to T (joints)

? 8 in 1/4 for each T con. to wheel (4 ft) ?

return   10 in 1/2 hose

valves   boom   4   3 pos. valves

joints   6   3 pos.

Pressure Loss Calculation

pressure loss

assume coils to be minor losses

wheels — one side total length of supply line 41 + 4 ft = 45 ft / 2 = 22.5 ft

max pressure loss at max gradation - 4 wheels at front end

flow: 1.4 gal/min

oil factor: .85

loss due to length

S: 2.1, water, [illegible] = 17 lbs/in²

17 × 22.5 ft = 3.825 lbs/in²

3.825 / .85 = 4.5 lbs/in²

[illegible] 16 ([illegible])

42.5 × [illegible] = [illegible]

40 × [illegible] = 4.35 lbs/in²

4.35 / .85 = 5.117 lbs/in²

4 × 5.117 + 2 × 4.5 29.468 lbs/in²

loss due to valves 16.6 lb/in² × 4 = 66.4

total wheel loss = 95.868

(over)

3 psi valve 16.6 lbs/in²

Q = 1.6 gal/min

joints total length = 23 + 3 ft = 26 ft

d+1
p drop = 20 lbs/in²

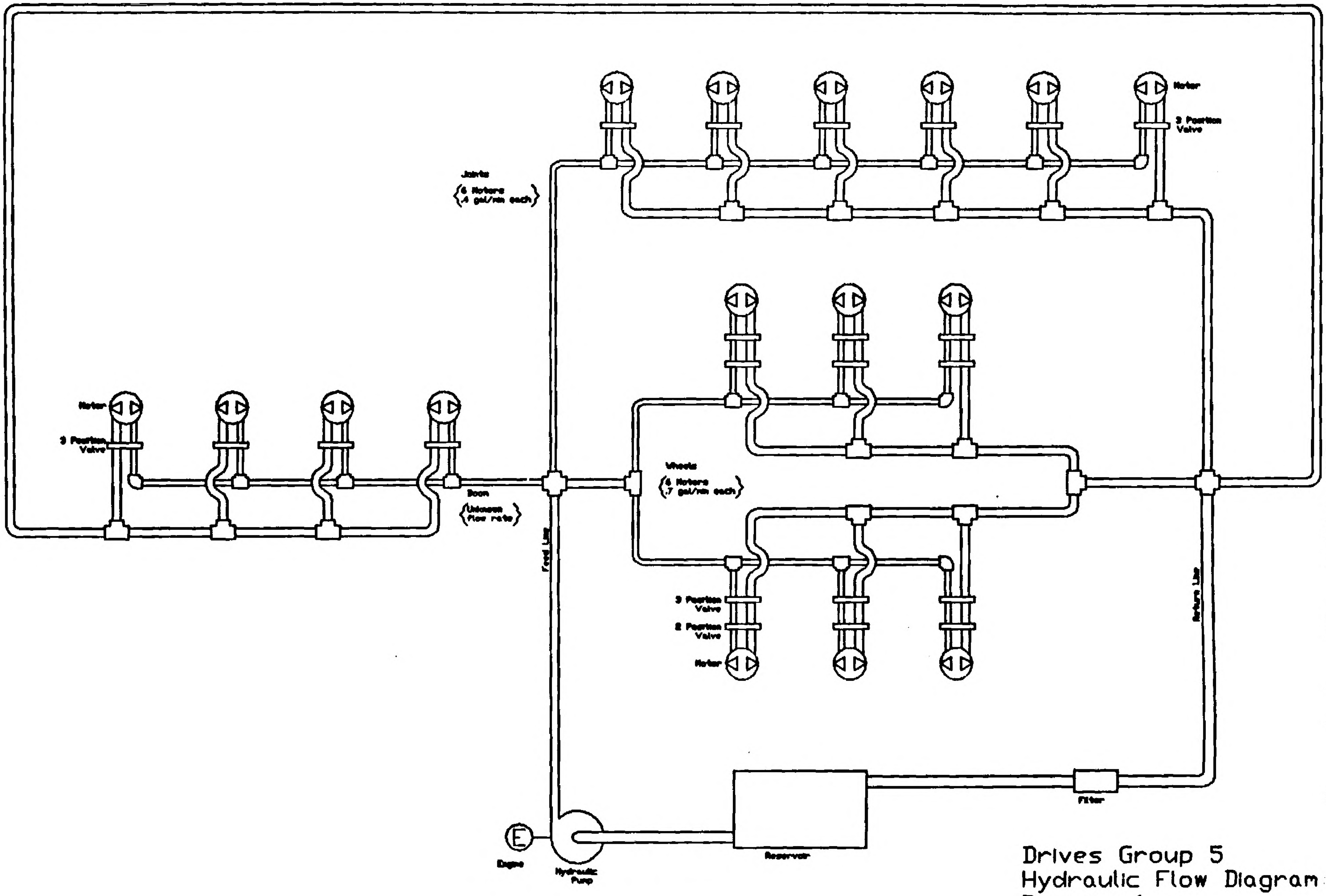
$20 \times {}^{26}/_{100} = 5.2$ lbs/in²

$5.2/.85 = 6.117$ lbs/in²

4 valve loss = 66.4

total = 72.517 psi

worst case senario for total loss = $\boxed{168.385 \text{ psi}}$



Drives Group 5
Hydraulic Flow Diagram
Revision 1
F1

Application Data

Hose Selection Chart

Hose Number	H166	H369	H747	H017	H243	H209	H213 H448	H324	H066	H069	H169
Usage	High Temp. Truck	Truck Freon 12	Freon 12	Gen. Purp. Hyd.	Gen. Purp. Hi-temp	Car Wash	High Temp. Truck	Power Steering	High Temp. Truck	Truck and Hyd.	Truck and Hyd.
Meets	DOT All	—	—	USCG[2] MSHA	—	—	DOT A1	—	DOT All	—	USCG[1,2] MSHA
SAE No.	J1402 Type All	J51 Type B2	J51 Type D	100R3	100R14 Type A	—	J1402 Type A1	J188 Type II	J1402 Type All	J1402 Type All 100R5	—
Temperature Range	−40°F +300°F	−20°F +250°F	−40°F +250°F	−40°F +212°F	−65°F +450°F	−40°F +200°F	−40°F +300°F	−40°F +250°F	−40°F +300°F	−40°F +300°F*	−40°F +212°F
Inner Tube	Nitrile	Nitrile	Rubber Nylon Rubber	Neoprene	Teflon®	Nylon 11	CPE	Neoprene	Nitrile	Nitrile	Neoprene
Reinforcement	1 Fiber & 1 S.S. Braid	1 Fiber & 1 Steel Braid	2 Fiber Spiral	2 Fiber Braids	1 S.S. Braid	1 Fiber Braid	1 Fiber & 1 Wire Braid	2 Fiber Braids	1 Fiber & 1 Steel Braid	1 Fiber & 1 Steel Braid	1 Fiber & 1 Steel Braid
Outer Cover	Fiber Braid	Fiber Braid Red	Butyl	Neoprene	Stainless Steel	Poly-urethane	Fiber Braid	Neoprene	Fiber Braid	Fiber Braid	Neoprene
Hose I.D.	MAXIMUM RECOMMENDED OPERATING PRESSURE — PSI										
3/16	1500	500					1500		2500	3000	3000
1/4	500			1250	3000	2250	1500		2250	3000	3000
5/16	500	500	500		2500	1750	1500		2000	2250	2250
3/8				1125	2000	1350		1400			
13/32	500	500	500				1250		2000	2000	2000
1/2	450	500	500	1000	1750	1000	1000		1750	1750	1750
5/8	450	500	500				750		1500	1500	1500
3/4				750	1000						
7/8	250	500					400		800	800	800
1				560	1000						
1 1/8		500								625	625
1 1/4				375							
1 3/8		350								500	500
1 13/16										350	350
2 3/8										350	
3										200	
Coil-O-Crimp®	Pages 68-70	Pages 68-70	Page 62	Pages 39-50	Pages 63-67	Pages 63-67		Pages 39-50	Pages 68-70	Pages 68-70	Pages 68-70
Reusable	Pages 94-99	Pages 94-99		Page 92			Page 100		Pages 94-99	Pages 94-99	Pages 94-99

Note: Teflon is a registered trademark of DuPont.
[1]Fuel and Lube Accepted [2]Fluid Power Accepted *Sizes -4 through -12, +250° Sizes -16 through -48.

Hose Selection Chart

Application Data

Hose Number	H104	H435	H436	H114	H145	H146	H300	H425	H195	H430	H439	H470
Usage	Hyd.	Hyd.	Hyd.	Ag. Hyd. & Hyd. Synthetic	Hyd.	Ag. Hyd. & Hyd. Synthetic	Hyd.	Hyd.	Synthetic Hyd.	High Pressure Hyd.	High Pressure Hyd.	High Pressure Hyd.
Meets	USCG[1,2] MSHA	—	—	MSHA	MSHA	—	MSHA	USCG[1,2] MSHA	MSHA	USCG[2] MSHA	USCG[2] MSHA	MSHA
SAE No.	100R1AT	100R7 Non-Cond.	100R7	Exceeds 100R1	Exceeds 100R1	Exceeds 100R1	—	100R2AT	—	100R12	100R12	—
Temperature Range	−40°F +212°F	−40°F +200°F	−40°F +200°F	−40°F +250°F	−40°F +250°F	−65°F +250°F	−40°F +212°F	−40°F +212°F	−40°F +200°F	−40°F +250°F	−40°F +250°F	−40°F +250°F
Inner Tube	Neoprene	Nylon 11	Nylon 11	Hytrel	Nitrile	Hytrel	Neoprene	Neoprene	EPDM	Neoprene	Neoprene	Neoprene
Reinforcement	1 Steel Braid	2 Fiber Braids	2 Fiber Braids	1 Steel Braid	1 Steel Braid	1 Steel Braid	2 Spiral 1 Wire Braid	2 Steel Braids	2 Steel Braids	Multi Spiral Steel	Multi Spiral Steel	Multi Spiral Steel
Outer Cover	Neoprene	Poly-urethane Orange	Poly-urethane Black	Neoprene	Neoprene	Polyester Braid	Neoprene	Neoprene	EPDM Green	EPDM or Neoprene	EPDM or Neoprene	EPDM
Hose I.D.	MAXIMUM RECOMMENDED OPERATING PRESSURE — PSI											
¼	2750	2750	2750	3000	3000	3000	3000	5000	5000			
5/16		2500	2500									
⅜	2250	2250	2250	3000	3000	3000	3000	4000	4000			
½	2000	2000	2000	3000		3000	3000	3500	3500	4000	4000	5000
⅝	1500						3000	2750				
¾	1250	1250	1250				3000	2250	2250	4000	4000	5000
1	1000	1000	1000					2000	2000	4000	4000	5000
1¼	625							1625	1625	3000	3000	5000
1½								1250	1250	2500	2500	5000
2								1125	1125	2500	2500	5000
Coll-O-Crimp®	Pages 39-50	Pages 63-67	Pages 63-67	Pages 39-50	Pages 39-50	Pages 39-50	Pages 39-50	Pages 39-50, 71-78	Pages 71-78	Pages 71-78	Pages 71-78	Pages 79-80
Reusable		Pages 104-105	Pages 104-105					Pages 106-109				
Hose Number	H104	H435	H436	H114	H145	H146	H300	H425	H095	H430	H439	H470

[1]Fuel and Lube Accepted
[2]Fluid Power Accepted

WEATHERHEAD®

Hose Selection Chart

CHART 1. Hose Flow Rate vs Pressure Drop

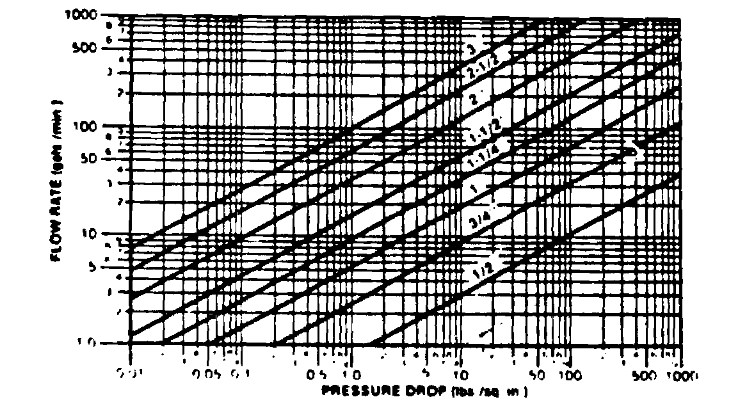
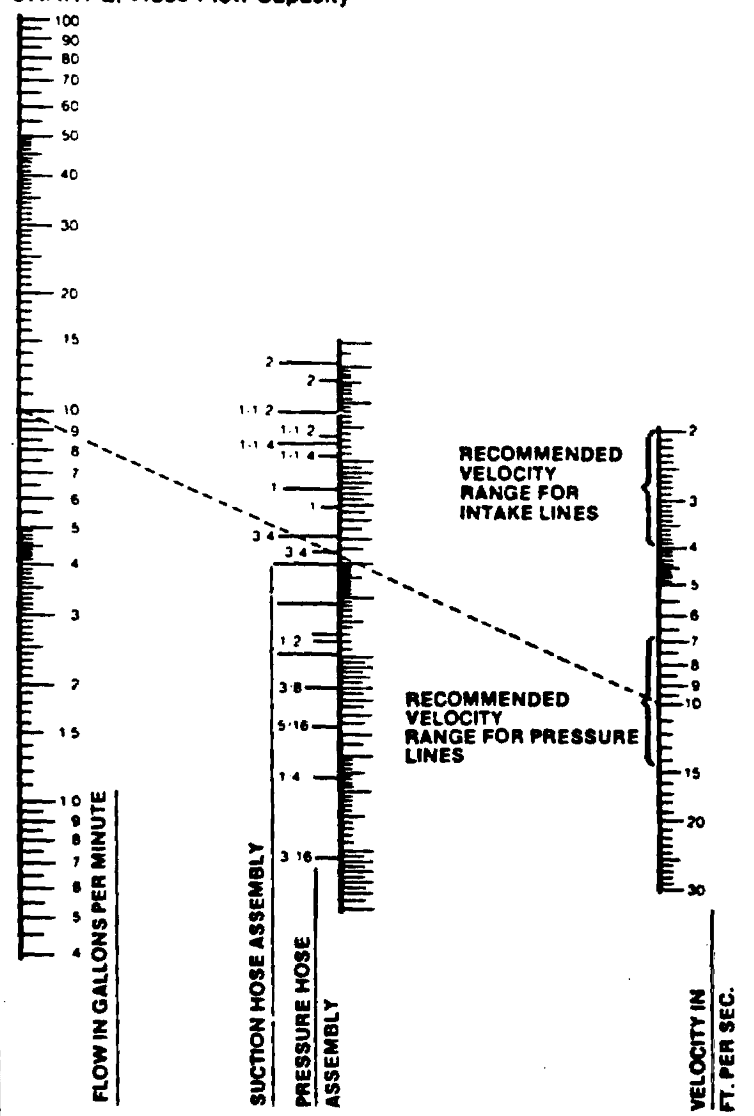


CHART 2. Hose Flow Capacity



There are several factors which enter into the selection of a hose sized such that it will provide the desired rate of flow at the required pressure; these are:

- **Hose size**
- **Hose length**
- **Hose ends & fluid conveyed**
- **Bends**
- **Static head pressure**

Hose Size

Undersized pressure lines produce excessive pressure drop with attendant energy loss and heating, and undersized suction lines cause cavitation at the pump inlet. Oversized hose assemblies, on the other hand, are excessively costly and generally too heavy.

In selecting hose for hydraulic systems, the following empirical values can be used to achieve minimum pressure drop consistent with reasonable hose size (see Chart 2):

Velocity of pressure lines ......7 to 15 ft/sec
Velocity of short pressure lines ...to 20 ft/sec
Velocity of suction lines ....2 to 5 ft/sec

To use Chart 2, lay a straight-edge across the chart as shown by the dotted line. Always use the next larger size hose shown if the line passes between sizes listed.

Hose Length

Chart 1. gives the pressure drop in different-sized hoses based on hoses of 100-foot length, and is based on water as the fluid conveyed. For hoses of a different length, these values must be corrected. For example, a 100-foot length of ½" hose causes a pressure drop of 100 lbs./in² at a flow rate of 10 gal./min. If the hose in question is 50 feet long, the pressure drop derived from Chart 1. must be corrected by multiplying the value by the ratio of the actual length to 100 feet, or $^{50}/_{100}$, or 0.5. Therefore, the actual pressure drop caused by a 50-foot length of ½" hose, at a flow rate of 10 gal./min., is 50 lbs./in² (0.5 x 100 = 50 lb./in²).

Hose Ends & Fluid Conveyed

In most cases, the end fitting openings are slightly smaller than the hose itself. However, this varies widely with hose end designs from "full-flow" ends which have the same ID as the hose, down to as much as ⅛-in. smaller ID than the hose bore. To allow for this, assume a 10-to 15% greater flow rate than actually measured in the system when determining pressure drop.

Chart 1. is based on water as the fluid conveyed, and for other fluids it is necessary to correct for the difference in specific gravity and viscosity. Chart 3. lists common fluids, their specific gravities, viscosities, and corresponding correction factors.

To determine the pressure drop for a specific fluid, first determine the pressure drop from

Chart 1. for the hose length then divide this by the correction factor found in Chart 3.

For example, the 50-foot length of ½" hose just described had a pressure drop of 50 lbs./in² at a flow of 10 gal./min. of water. To determine the pressure drop if #2 fuel oil is the fluid conveyed, divide by 0.752 (from Chart 3)... 50 ÷ 0.752 = 66.5 lbs./in² pressure drop. If, on the other hand, the fluid conveyed is Type #3 gasoline, the pressure drop would be 50 ÷ 1.19 = 42 lbs./in²

Chart 3. Fluid Flow Correction Factors

Liquid	Specific Gravity	Viscosity Centistokes (CS)	Viscosity Centipoises (CP)	Correction Factor R
Acetic Acid 100%	1.05		1.1	0.975
Acetic Acid 70%	1.07		2.7	0.843
Ammonia liquid 100%	0.66	0.30		1.290
Ammonia liquid 26%	0.907		1.3	0.943
*Asphalt @ 120 F	1.40		300	0.350
*Beer	1.01	1.15		0.990
Benzene (Benzol)	0.88	.743		1.08
Brine, Calcium Chloride 25%	1.23	3.80		0.78
Brine, Sodium Chloride 25%	1.19	2.07		0.88
Butyl Alcohol	0.81	3.64		0.783
*Castor Oil	0.96	900		0.27
*Crude Petroleum (typical)				
1. Pennsylvania Crude @ 100 F	0.80		3	0.78
2. California Crude @ 150 F	0.975		9	0.64
3. 33 API Crude @ 100 F	.86	7.2		0.685
4. Texas Crude @ 150 F	0.875		3	0.792
5. Mexican Crude @ 150 F	1.05		550	0.267
Decane - n	0.73	1.24		0.975
Ethyl Alcohol 100%	0.788		1.25	0.93
Ethyl Alcohol 80%	0.808		1.45	0.904
Ethyl Alcohol 40%	0.937		3.00	0.807
Ethyl Glycol	1.12		24.00	0.55
Formic Acid	1.22			0.94
*Fuel Oils				
No. 1 @ 100 F Sp. Gr. .82 to .95 Visc. 30 to 40 SSU	0.88	2.45		0.85
No. 2 @ 100 F Sp. Gr. .82 to .95 Visc. 35 to 50 SSU	0.88	4.50		0.752
No. 3 @ 100 F Sp. Gr. .82 to .95 Visc. 55 SSU max	0.88	8.6		0.66
No. 5 @ 100 F Sp. Gr. .82 to .95 Visc. 60 to 450 SSU	0.88	55.0		0.47
No. 6 @ 122 F Sp. Gr. .82 to .95 Visc. 430 to 2900 SSU	0.88	38.0		0.493
*Gasoline (representative)				
Type #1	.74	.88	-	1.04
Type #2	.72	.64	-	1.11
Type #3	.68	.46	-	1.19
Glycerine (Glycerol) 100% @ 150 F	1.26	-	75.0	0.45
Glycerine & Water 50%	1.13	-	6.5	0.717
Heptane - n	.684	0.60	-	1.16
Hexane - n	.66	0.49	-	1.21
Hydrochloric Acid 31.5%	1.16	-	1.92	0.92
Isobutyl Alcohol	0.817	-	3.90	0.745
Isopropyl Alcohol	0.785	-	2.70	0.828
Kerosene	0.80	2.23	-	0.892
Lubricating Oil (Machine Oil)	0.90		198	0.35
Lubricating Oil (Automotive)	0.893	-	110	0.39
Methyl Alcohol (Methanol) 100%	0.79	.74	0.60	1.072
Methyl Alcohol - 90%	0.824		0.77	1.03
Methyl Alcohol 40%	0.937		2.00	0.863
*Milk	1.03	1.15	-	0.99
Motor Oil	0.893		110	0.39
Naphthalene	1.15	0.9		1.04
Nitric Acid 95%	1.50		1.13	1.07
Nitric Acid 60%	1.37		2.35	0.913
Nonane - n	0.718	.97	-	1.02
Octane - n	0.70	.77		1.068
Olive Oil	0.91	93.0	-	0.41
Pentane - n	0.63	0.37	-	1.24
Propyl Alcohol	0.804	2.8	-	0.828
Rapeseed Oil	0.91	180	-	0.36
Sodium Hydroxide 50%	1.53	-	95.0	0.443
Soya Bean Oil	0.924	86	-	0.418
Sperm Oil	0.88	21	-	0.55
Sugar Solution 20%	1.08	1.9	-	0.895
Sugar Solution 40%	1.18	5.3	-	0.728
Sugar Solution 60%	1.29	44.0	-	0.475
Sulfuric Acid 100%	1.83	14.6	-	0.59
Sulfuric Acid 95%	1.83	14.5	-	0.593
Sulfuric Acid 60%	1.50	4.4	-	0.755
Toluene	0.866	-	0.6	1.092
Turpentine	0.86	1.83		0.90
Water (Fresh)	1.0	1.10	-	1.00
Water, Salt	1.03	1.10	-	1.00
Xylene (Xylol)	0.87	0.93	-	1.03

*These figures are approximate or averages of those values available

Bends

If a hose of a given length is bent, the pressure drop will increase by some definite amount... the sharper the bend and the smaller the radius of bend the greater the pressure drop. The effect of a bend may be neglected if it is slight or if there are but few bends in a long length of hose. This is because the additional pressure drop caused by these bends is not significant when compared to the total pressure drop.

However, a dock hose may have four sharp 90° bends in a 25-foot length, and if pressure drop is important, these bends must be considered because they constitute a significant portion of the overall pressure drop.

The curves in Chart 4. show the effects of resistance due to 90 degree bends. This data can also be used as a guide for smooth bends less or greater than 90 degrees. For example, a 45 degree bend has about 4/10 the resistance of a 90 degree bend.

Problem: Determine the equivalent length, in terms of hose inside diameters, of a 90 degree and a 180 degree bend whose relative radii are 12 inches.

Solution: Referring to the "total resistance curve," the equivalent length for a 90 degree bend is 34.5 hose diameters. The equivalent length of a 180 degree bend is 34.5 diameters for one 90 degree bend, 18.7 diameters for resistance due to length, and 15.8 ÷ 2 diameters for bend resistance. Adding these 34.5, 18.7 and *15.8 ÷ 2 equals 61.1 diameters for a 180 degree bend. Note that this loss is less than the sum of losses through two 90 degree bends separated by tangents.

Static Head Pressure

Static head is the difference in height between the inlet and outlet ends of a hose. Before using Chart 1., it is necessary to correct for static head pressure because the values in Chart 1. are pressure losses due to friction only.

To correct for static head pressure, the difference in height is determined and multiplied by 0.433 to convert the head to an equivalent pressure in PSI (one foot of water exerts 0.433 PSI pressure).

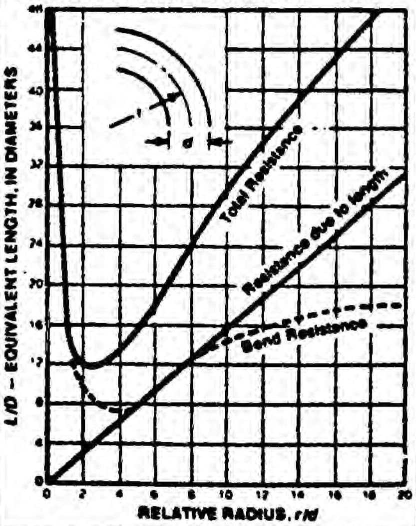
If the inlet is higher than the outlet, the pressure equivalent is added to the pump pressure. If the outlet is higher than the inlet, the pressure equivalent is subtracted from the pump pressure. In both cases, it is assumed that the pump pressure is the pressure available at the inlet end and that the pump is outside of the hose system.

Installation Design

Hose should not be twisted or put in torsion either during the installation or while in service. Sharp or excessive bends may cause the hose to kink or rupture.

Be sure to allow enough slack to provide for changes in length which will occur when pressure is applied. This

CHART 4. Resistance of 90° Bends



change in length can vary from +2% to −4%.

Design the installation so the hose assembly is accessible for inspection and easy removal.

Bend radius is important. A good working rule is that the minimum bend radius should be five or more times the OD dimension of the hose.

*In a continuous bend of 180 degrees the second 90 degree bend produces approximately one-half the resistance of the first bend.

Application Data

Chemical Compatibility Chart

These tables alphabetically list commonly used materials of various chemical composition. After each agent listing you will find the basic hose core and fitting materials rated according to their chemical resistance to each individual agent. The chart is intended to be used as a guide only.

⚠ **WARNING - Selection of Hose:** Selection of the proper hose for the application is essential to the proper operation and safe use of the hose and related equipment. Inadequate attention to selection of the hose for your application can result in serious bodily injury or property damage. In order to avoid serious bodily injury or property damage resulting from selection of the wrong hose, you should carefully review the information in this catalog.

⚠ **WARNING - Proper Selection of Hose Ends:** Selection of the proper end fittings for the hose and application is essential to the proper operation and safe use of the hose and related equipment. Inadequate attention to the selection of the end fittings for your application can result in serious bodily injury or property damage. In order to avoid serious bodily injury or property damage resulting from selection of the wrong end fitting, you should carefully review the information in this catalog.

Where unusual conditions exist, or where questions arise, consult WEATHERHEAD® Division of Dana Corporation for assistance on your hose application problems.

FLUID	HOSE MATERIAL									HOSE END FITTINGS		
	Nitrile	Neoprene	Teflon	Nylon	Polyurethane	EPDM	CPE	Hypalon	Hytrel	Brass	Steel	316 Stainless
Acetaldehyde Solvent	X	X	G	F	X	G	-	X	X	-	-	G
Acetic Acid (concentrated)	X	F	G	X	X	G	G	-	-	X	X	F
Acetic Acid (diluted)	X	G	G	F	X	G	G	G	G	X	X	G
Acetic Anhydride	F	G	G	X	X	G	F	G	X	X	F	F
Acetone	X	F	G	G	X	G	G	F	F	G	G	G
Air	G	G	G	G	G	G	G	-	-	G	G	G
Alcohols (Meth. & Ethanol)	G	G	G	G	X	G	G	G	G	G	F	G
Aluminum Chloride	G	G	G	X	G	G	X	F	-	X	X	F
Aluminum Floride	G	F	G	G	G	G	X	-	-	X	X	X
Aluminum Hydroxide	G	G	G	G	G	G	-	-	-	G	-	G
Aluminum Sulfate	G	G	G	-	G	G	X	G	-	X	X	G
Alums	G	G	G	-	G	G	X	-	-	X	X	F
Ammonium Chloride	G	G	G	X	-	G	G	G	G	X	G	F
Ammonium Hydroxide	F	F	F	F	X	G	F	G	X	X	G	G
Ammonium Phosphate	G	G	G	G	-	G	G	-	-	X	X	G
Ammonium Sulfate	G	G	G	G	-	G	G	G	G	X	F	F
Amyl Acetate	X	X	G	G	X	G	X	X	F	F	F	G
Amyl Alcohol	F	C	G	G	X	G	G	G	G	G	F	F
Aniline	X	F	G	X	X	G	X	X	X	F	G	F
Aniline Dyes	F	F	G	X	X	G	X	-	-	F	X	F
Animal Oils & Fats	G	X	G	G	X	F	F	F	G	-	-	G
Anti-Freeze (Glycol Base)	G	G	G	G	-	G	-	G	-	-	-	-
Asphalt	F	F	G	X	X	X	X	X	-	G	G	G
Barium Chloride	G	G	G	G	G	G	G	-	-	X	F	F
Barium Hydroxide	G	G	G	G	X	G	G	G	-	F	F	G
Barium Sulfide	G	G	G	F	G	X	G	G	X	F	X	G
Beet Sugar, Liquors	G	G	G	-	-	G	G	-	-	-	G	G
Benzene, Benzol	X	X	G	G	X	X	F	X	F	G	G	G
Benzaldehyde	X	X	F	G	X	G	X	X	-	-	-	G
Black Sulfate Liquor	G	G	G	F	X	G	F	F	-	X	G	G
Borax	F	F	G	G	G	G	G	-	G	G	G	G
Boric Acid	G	G	G	F	G	G	X	G	G	X	X	F
Brake Fluid (petroleum base)	G	G	-	G	-	X	-	X	-	-	-	-
Brine	G	G	G	-	-	G	G	-	-	X	-	F
Butane	G	G	G	G	X	X	-	G	G	G	G	G
Butyl Acetate	X	X	G	G	X	F	F	X	F	G	G	G
Butyl Alcohol, Butanol	G	G	G	F	X	G	G	-	-	G	G	G
Calcium Bisulfite	G	G	G	F	G	-	X	-	-	X	X	X
Calcium Chloride	G	C	G	X	G	F	G	-	G	X	F	F
Calcium Hydroxide	F	G	G	F	X	G	G	-	-	F	G	G
Calcium Hypochlorite	F	F	G	F	X	G	G	F	F	X	F	F
Caliche Liquors Na Nitrate	G	G	G	-	-	G	G	-	-	G	G	G
Cane Sugar Liquors	G	G	G	-	X	G	G	-	-	F	G	G
Carbolic Acid (Phenol)	X	X	G	X	X	X	X	X	X	F	X	F
Carbon Dioxide (Dry)	F	F	G	G	G	F	G	G	G	G	-	G
Carbon Dioxide (Wet)	F	F	G	G	G	F	-	-	-	F	-	G
Carbon Disulphide (Bisulfide)	X	X	G	F	G	X	-	-	-	G	G	G
Carbon Monoxide (Hot)	F	F	G	-	F	F	G	G	G	X	F	G
Carbon Tetrachloride	X	X	G	G	X	X	F	X	F	F	F	G
Carbonic Acid	G	G	G	G	F	G	X	-	-	X	X	F
Castor Oil	G	G	G	G	F	F	G	G	F	G	G	G
Cellosolve Acetate	X	X	G	F	X	G	X	-	-	X	X	G
Cellulube-90,150,220,300, 550,1000	X	X	G	G	X	G	-	-	-	G	G	G
Chlorinated Solvents	X	X	G	F	-	X	X	-	-	G	G	F
Chloracetic Acid	X	X	G	-	X	F	X	F	-	X	X	F
Chloroform	X	X	G	-	X	X	X	X	X	G	-	G
Chlorosulphonic Acid	X	X	G	-	X	X	X	X	X	X	F	X
Chromic Acid	X	X	G	X	X	X	X	F	X	X	X	F
Citric Acid	G	G	G	G	X	G	X	-	G	F	X	F
Coke Oven Gas	F	F	F	G	X	X	-	-	-	F	G	G
Copper Chloride	G	G	G	G	G	G	X	-	G	X	X	G
Copper Sulfate	G	G	G	G	G	G	X	-	G	X	X	G
Cottonseed Oil	G	G	G	G	G	F	G	G	G	G	G	G
Creosote	F	X	G	X	F	X	F	F	-	F	-	G
Cresol	X	X	G	-	X	X	X	X	-	-	-	G
Diesel Fuel	G	F	G	G	F	X	G	F	F	G	G	G
Downtherm A & E	X	X	G	-	-	X	X	-	-	X	F	G
Ethers (Ethyl Ether)	X	X	G	G	F	-	X	X	-	G	G	G
Ethyl Acetate	X	X	G	G	X	G	F	X	F	F	G	G
Ethyl Cellulose	F	F	G	G	F	F	G	-	-	F	G	F
Ethyl Chloride	F	X	G	G	F	G	X	X	X	F	F	G
Ethylene Dichloride	X	X	G	G	X	X	X	X	X	G	X	X
Ethylene Glycol	G	G	G	F	F	G	G	G	G	F	G	G
Ethylene Oxide	X	X	G	F	X	F	X	X	G	-	-	-
Fatty Acids	G	X	G	G	-	X	-	-	-	-	-	G
Ferric Chloride	G	G	G	F	F	G	X	G	-	X	X	X
Ferric Sulfate	G	G	G	G	-	-	X	-	G	X	X	F
Formaldehyde	F	F	G	X	X	G	X	G	F	F	F	G
Formic Acid	F	F	G	X	X	G	X	G	X	F	X	G
Freon 12*	G	F	F	G	X	X	F	G	X	G	F	G
Fuel Oil	G	F	G	G	F	X	G	-	-	F	G	G
Furfural	X	F	G	G	-	F	F	F	-	F	G	G
Gasoline (refined)	G	F	G	G	F	X	G	F	G	G	G	G
Gasoline (unleaded)	G	X	G	G	X	X	G	F	X	G	G	G
Gasoline (10% Ethanol)	G	X	G	G	X	X	-	-	X	X	G	G
Gasoline (10% Methanol)	F	X	X	-	X	X	-	-	X	X	G	G

*, Use approved Freon Hose.

CODES: G-Good resistance F-Fair resistance X-Incompatible -No data available

Chemical Compatibility Chart

Application Data

APPLICATION

FLUID	HOSE MATERIAL									HOSE END FITTINGS		
	Nitrile	Neoprene	Teflon	Nylon	Polyurethane	EPDM	CPE	Hypalon	Hytrel	Brass	Steel	316 Stainless
Glycerine, Glycerol	G	G	G	G	X	G	G	-	G	G	G	G
Greases	G	F	G	G	G	X	G	F	G	G	G	G
Green Sulfate Liquor	G	G	G	-	G	G	X	-	-	X	X	G
Heptane	G	F	G	G	F	X	G	F	G	G	G	G
N-Hexane	G	F	F	G	F	X	G	F	G	G	G	G
Houghto Safe 271 to 640	G	G	G	G	-	X	-	-	-	G	G	G
Houghto Safe 5046, 5046W	G	G	G	G	-	-	-	-	-	G	G	G
Houghto Safe 1010,1055 1115,1120,1130	X	X	G	G	X	G	G	-	-	G	G	G
Hydraulic Oil (Petroleum Base)	G	G	G	G	G	X	G	F	G	G	G	G
Hydrobromic Acid	X	X	G	X	X	F	X	G	X	X	X	X
Hydrochloric Acid	X	X	G	X	X	G	X	G	X	X	X	X
Hydrocyanic Acid	F	F	G	-	-	G	X	G	-	X	F	G
Hydrofluoric Acid	X	X	G	X	X	F	X	G	-	X	X	X
Hydrofluosilicic Acid	F	X	G	-	-	G	X	G	-	X	X	X
Hydrogen Peroxide	X	X	G	-	-	F	X	F	-	F	X	G
Hydrogen Sulfide	X	F	G	-	-	G	X	G	G	F	F	F
Hydralube H-2	G	F	G	G	-	-	-	-	-	G	G	G
Isopropyl Alcohol	F	G	G	G	X	G	G	G	G	G	F	G
Iso-Octane	G	F	G	G	X	X	G	G	G	-	F	-
Kerosene	G	F	G	G	G	X	G	F	F	G	G	G
Lacquer	X	X	G	G	X	X	F	-	-	G	X	G
Lacquer Solvents	X	X	G	G	X	X	F	X	F	G	X	G
Lactic Acid	G	G	G	G	X	G	X	G	X	F	X	F
Lindol	X	X	G	G	-	G	-	-	-	F	G	G
Linseed Oil	G	G	G	G	F	F	G	G	-	F	G	G
Lubricating Oils	G	G	G	G	F	X	G	F	G	G	G	G
Lime Sulfur	X	G	G	-	-	G	-	-	-	X	-	G
Magnesium Chloride	G	G	G	G	G	G	G	-	-	X	F	F
Magnesium Hydroxide	F	F	G	X	X	G	G	-	-	G	G	G
Magnesium Sulphate	G	G	G	-	-	G	G	-	-	F	G	G
Mercuric Chloride	G	G	G	X	-	G	X	-	-	X	F	X
Mercury	G	G	G	G	G	G	G	G	G	X	G	G
Methane	F	F	G	-	F	X	G	-	-	-	-	-
Methyl Chloride	X	X	G	G	X	X	F	-	-	G	G	G
Methyl Ethyl Ketone	X	X	G	G	X	G	G	X	G	F	F	F
Methyl Isobutyl Ketone	X	X	G	G	X	G	X	X	-	-		G
Methyl Isopropyl Ketone	X	X	G	G	X	G	X	X	-	F	F	F
Mineral Oil	G	G	G	G	G	X	G	G	G	G	G	G
Naphtha	G	F	G	G	F	X	G	X	G	F	G	G
Naphthalene	X	X	G	G	F	X	G	X	F	F	G	F
Nickel Acetate	X	F	G	-	-	G	-	-	-	-	-	-
Nickel Chloride	G	G	G	-	X	G	X	-	-	X	X	F
Nitric Acid (concentrated)	X	X	G	X	X	X	X	X	X	X	X	F
Nitric Acid 10%	X	F	G	X	X	G	X	G	X	X	X	G
Nitric Acid 70%	X	X	G	X	X	X	X	X	X	X	X	G
Nitrobenzene	X	X	G	G	X	X	F	X	X	X	G	F
Olaic Acid	F	F	G	G	F	X	X	F	G	F	F	G
Oxalic Acid	F	G	G	F	-	G	X	-	-	F	F	F
Oxygen	F	F	F	F	G	G	X	-	-	G	G	G
Palmitic Acid	G	F	G	-	X	F	F	X	X	X	F	F
Perchloroethylene	X	X	G	-	X	X	F	X	X	X	F	F
Petroleum Oils	G	F	G	G	G	X	G	-	G	G	G	G
Picric Acid (molton)	X	X	G	-	-	X	X	-	-	X	X	F
Picric Acid (solution)	X	X	G	-	F	G	X	F	-	X	X	F
Potassium Chloride	G	G	G	G	G	G	G	-	-	X	F	G
Phosphate Esters	X	X	X	G	X	G	G	-	-	-	-	-
Phosphate Esters (Alkyl)	X	X	G	G	X	G	G	X	G	G	G	G
Phosphate Esters (Anyl)	X	X	G	F	-	-	-	X	-	G	G	G
Phosphate Ester (Blend)	X	-	G	F	-	-	-	X	-	G	G	G
Potassium Cyanide	G	G	G	G	G	G	G	-	-	-	G	G
Potassium Dichromate	F	G	G	-	G	G	X	G	-	X	G	G
Potassium Hydroxide	G	G	G	G	X	G	X	G	[illegible]	F	X	G
Potassium Sulphate	G	G	G	F	G	G	G	-	-	F	F	G
Propane	Use H366 Hose											
Pydraul F-9, 150	X	X	G	G	X	G	G	-	-	G	G	G
Pyridine	X	X	X	-	-	F	X	X	-	-	-	F
Sea Water	G	G	G	F	X	G	-	G	G	F	X	G
Skydrol (All)	X	X	G	G	X	G	G	-	-	G	G	G
Soap Solution	G	F	G	G	G	G	G	G	G	G	G	G
Soda Ash, Sodium Carbonate	G	G	G	G	-	G	G	-	-	F	F	G
Sodium Bisulphate	G	G	G	G	-	G	G	-	-	X	F	G
Sodium Chloride	G	G	G	G	G	G	G	G	G	X	F	G
Sodium Cyanide	G	G	G	G	-	G	G	-	-	X	-	G
Sodium Hydroxide	F	F	G	G	X	G	F	-	-	F	X	G
Sodium Hypochlorite	F	F	G	-	X	F	F	G	G	X	X	F
Sodium Nitrate	G	G	G	G	-	G	G	-	-	F	F	F
Sodium Perborate	G	G	G	-	X	G	X	G	-	F	F	G
Sodium Peroxide	G	G	G	-	X	G	X	G	-	X	F	G
Sodium Phosphates	X	X	G	G	G	G	X	-	-	F	F	F
Sodium Silicate	G	G	G	G	-	G	G	-	-	F	F	G
Sodium Sulphate	G	G	G	G	G	G	G	-	-	F	F	G
Sodium Sulphide	G	G	G	G	G	G	G	-	-	F	X	G
Sodium Thiosulphate	G	G	G	G	G	G	G	-	-	F	X	G
Soybean Oil	G	G	G	G	G	X	G	G	G	G	G	G
Stannic Chloride	G	G	G	G	-	G	X	F	-	X	X	X
Steam 450°	X	X	G	X	X	X	X	X	X	F	F	G
Stearic Acid	F	F	G	G	G	F	G	F	G	F	F	G
Sulphur	F	G	G	G	-	G	G	-	-	X	X	G
Sulphur Chloride	F	G	G	-	-	X	G	X	X	X	X	X
Sulphur Dioxide	F	F	G	F	-	G	X	-	-	X	F	G
Sulphuric Acid (10% Cold)	G	G	G	F	X	G	X	G	F	X	X	G
Sulphuric Acid (10% Hot)	X	F	G	F	X	F	X	G	X	X	X	G
Sulphuric Acid (75% Cold)	X	F	G	X	X	F	X	-	X	X	X	X
Sulphuric Acid (75% Hot)	X	X	G	X	X	X	X	-	X	X	X	X
Sulphuric Acid (95% Cold)	X	X	G	X	X	X	X	F	-	X	X	X
Sulphuric Acid (95% Hot)	X	X	G	X	X	X	X	X	X	X	X	F
Sulphuric Acid (Fuming)	X	X	G	X	X	X	X	X	X	X	F	F
Tannic Acid	F	G	G	G	G	G	X	G	G	F	X	F
Tar	G	F	G	X	-	X	X	-	-	F	F	G
Tartaric Acid	F	F	G	-	G	F	X	G	-	F	X	F
Tetrachloroethane	X	X	G	G	X	X	-	-	-	-	-	G
Toluene	X	X	G	G	X	X	F	X	F	G	G	G
Transmission Oil	G	G	G	G	-	X	-	-	-	-	-	-
Trichloroethylene	X	X	G	F	X	X	F	X	X	G	G	G
Turpentine	F	X	G	G	X	X	G	X	-	F	F	G
Varnish	X	X	G	G	X	X	F	-	-	G	G	G
Water	G	G	G	F	G	G	G	G	G	F	F	G
Water (Glycol)	G	G	G	F	-	G	G	G	-	F	G	G
Water (Petroleum)	G	F	G	F	X	X	G	F	G	G	F	G
Xylene	X	X	G	G	X	X	F	X	F	G	G	G
Zinc Chloride	F	F	G	G	-	G	X	-	G	X	X	F
Zinc Sulfate	G	G	G	G	-	G	X	-	-	X	F	G

CODES: G-Good resistance F-Fair resistance X-Incompatible -No data available

NOTES: All data given herein is believed to be accurate and reliable but presented without guaranty, warranty, or responsibility of any kind, express or implied, on our part. Chemical resistance will vary with the wide diversity of possible mixtures and service conditions. It is not therefore possible to give any guarantee whatsoever in individual cases.

Hose | Medium Pressure

H069 General Purpose Truck Hose
SAE 100R5
SAE J1402 TYPE II†
DOT AII†

Typical Application: Medium pressure hydraulic, air, oil, fuel or grease lines on fork lifts, trucks & off highway vehicles: -16 thru -48 sizes are not DOT approved.
Inner Tube: Nitrile
Reinforcement: 1 Fiber Braid - 1 Steel Braid
Cover: Fiber Braid
Temp. Range: -40°F to +300°F (-40°C to +149°C) sizes -4 thru -16
-40°F to +250°F (-40°C to +121°C) sizes -20 thru -48
†sizes -4 thru -12 only

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H06904	3/16	33/64	3000	12000	3	30	16	50',250'	
H06905	1/4	37/64	3000	12000	3-3/8	30	20	50',250'	
H06906	5/16	43/64	2250	9000	4	30	23	50',250'	Coll-O-Crimp
H06908	13/32	49/64	2000	8000	4-5/8	30	27	50',250'	069 'E' Series
H06910	1/2	59/64	1750	7000	5-1/2	30	36	50',250'	Pages 68-70
H06912	5/8	1-5/64	1500	6000	6-1/2	30	41	50',250'	Reusable
H06916	7/8	1-15/64	800	3200	7-3/8	20	46	50',250'	069 'D',
H06920	1-1/8	1-1/2	625	2500	9	20	51	50'	069 'T' &
H06924	1-3/8	1-3/4	500	2000	10-1/2	15	59	50'	247 'N' Series
H06932	1-13/16	2-7/32	350	1400	13-1/4	11	90	50'	Pages 94-99
H06940	2-3/8	3-7/8	350	1400	24		143	50'	
H06948	3	3-9/16	200	800	33		209	50'	

H169 Hydraulic Hose

USCG Fluid Power and Fuel/Lube Systems Accepted
MSHA Accepted
Typical Application: Medium pressure air, fuel, grease, oil, truck and power steering lines.
Inner Tube: Neoprene
Reinforcement: 1 Fiber Braid - 1 Steel Braid
Cover: Neoprene Perforated
Temp. Range: -40°F to +212°F (-40°C to +100°C)

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H16904	3/16	33/64	3000	12000	3	30	16	50',250'	
H16905	1/4	37/64	3000	12000	3-3/8	30	20	50',250'	Coll-O-Crimp
H16906	5/16	43/64	2250	9000	4	30	23	50',250'	069 'E' Series
H16908	13/32	49/64	2000	8000	4-5/8	30	27	50',250'	Pages 68-70
H16910	1/2	59/64	1750	7000	5-1/2	30	36	50',250'	Reusable
H16912	5/8	1-5/64	1500	6000	6-1/2	30	41	50',250'	069 'D' &
H16916	7/8	1-15/64	800	3200	7-3/8	20	46	50',250'	247 'N' Pages 94-99
H16920	1-1/8	1-1/2	625	2500	9	20	51	50'	
H16924	1-3/8	1-3/4	500	2000	10-1/2	15	59	50'	
H16932	1-13/16	2-7/32	350	1400	13-1/4	11	90	50'	

H104 Hydraulic Hose
SAE 100R1 TYPE AT

USCG Fluid Power and Fuel/Lube Systems Accepted
MSHA Accepted
Typical Application: Most commonly used for medium pressure hydraulic lines. Especially popular for farm implement hydraulic lines.
Inner Tube: Neoprene
Reinforcement: 1 Steel Braid
Cover: Neoprene
Temp. Range: -40°F to +212°F (-40°C to +100°C)

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H10404	1/4	17/32	2750	11000	4	30	15	50',250' 500'	
H10406	3/8	11/16	2250	9000	5	30	22	50',250' 500'	
H10408	1/2	13/16	2000	8000	7	30	29	50',250' 500'	Coll-O-Crimp 'U' Series
H10410	5/8	15/16	1500	6000	8	30	34	50',250'	Pages 39-50
H10412	3/4	1-3/32	1250	5000	9-1/2	20	40	50',250'	
H10416	1	1-13/32	1000	4000	12	20	50	50',250'	
H10420	1-1/4	1-23/32	625	2500	16-1/2	20	78	50'	

Medium Pressure | Hose

H114 HYDRAULIC HOSE
Exceeds SAE 100R1

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H11404	1/4	1/2	3000	12000	4		11	50',250' 500'	Coll-O-Crimp "U" Series Pages [illegible]
H11406	3/8	41/64	3000	12000	5		18	50',250' 500'	
H11408	1/2	13/16	3000	12000	7		25	50',250' 500'	

MSHA Accepted
Typical Application: 3000 PSI continuous working pressure hose. Ideal for farm implement high temperature hydraulic lines. Will work with synthetic or petroleum/water based fluids.
Inner Tube: Hytrel
Reinforcement: 1 Steel Braid
Cover: Neoprene
Temp. Range: -40°F to +250°F (-40°C to +121°C)

H145 HYDRAULIC HOSE
Exceeds SAE 100R1

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H14504	1/4	1/2	3000	12000	2		13	50'250' 500'	Coll-O-Crimp "U" Series Pages [illegible]
H14506	3/8	41/64	3000	12000	2-1/2		18	50',250' 500'	

MSHA Accepted
Typical Application: Ideal for use in high pressure lines on off the road construction equipment, farm equipment, and other high pressure applications where a small bend radius is needed.
Inner Tube: Nitrile
Reinforcement: 1 Steel Braid
Cover: Neoprene
Temp. Range: -40°F to +250°F (-40°C to +121°C)

HOSE

Hose | Medium Pressure

H146 3000 PSI Hydraulic Hose
Exceeds SAE 100R1

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H14604	1/4	33/64	3000	12000	4		11	50',250'	Coll-O-Crimp 'U' Series Pages 38-50
H14606	3/8	11/16	3000	12000	5		18	50',250'	
H14608	1/2	51/64	3000	12000	7		25	50',250'	

Typical Application: 3000 PSI continuous working pressure hose. Ideal for farm implement high temperature hydraulic lines. Will work with synthetic or petroleum/water based fluids.
Inner Tube: Hytrel
Reinforcement: 1 Steel Braid
Cover: Abrasion Resistant Fiber Braid
Temp. Range: -65°F to +250°F (-54°C to +121°C)

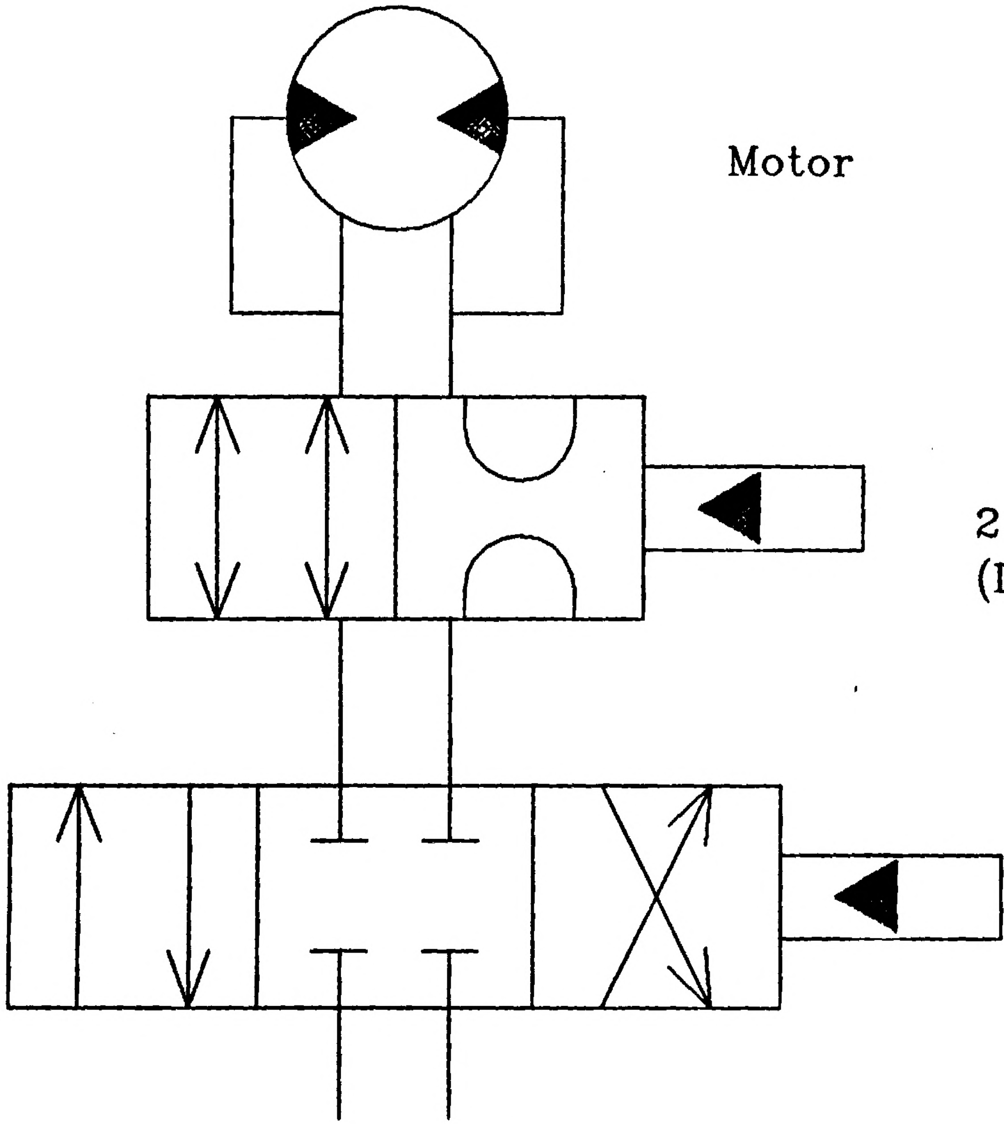
H300 3000 PSI Hydraulic Hose

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H30004	1/4	9/16	3000	12000	4		15	50',250'	
H30006	3/8	23/32	3000	12000	5		25	50',250'	Coll-O-Crimp 'U' Series Pages 38-50
H30008	1/2	29/32	3000	12000	7		45	50',250'	
H30010	5/8	1-1/32	3000	12000	8		55	50',250'	
H30012	3/4	1-3/16	3000	12000	9-1/2		65	50',250'	

MSHA Accepted
Typical Application: 3000 PSI constant working pressure hose. Ideal for farm implement construction and industrial equipment where high presssure hydraulic needs exist.
Inner Tube: Neoprene
Reinforcement: 4-6 size 1 steel braid; 8 size 2 steel braids; 10 and 12 size 2 spiral and 1 steel braid.
Cover: Neoprene
Temp. Range: -40°F to +212°F (-40°C to +100°C)

APPENDIX G

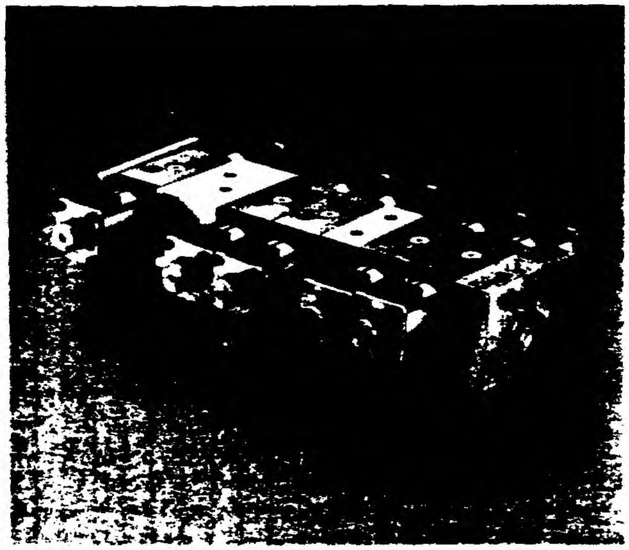
Valve Detail



G1 Power Supply – Group 5

ME4182c
Revision 1

ʼressure Compensated, :lectrohydraulic, Proportional)irectional Control Valves



Applicatio

Series six MCV valve systems are ideal for applications that require precise control of load, speed and direction. They provide precise and variable speed control without lurching during start–up. Series six valves provide smooth acceleration and deceleration, thus providing precise and predictable metering throughout the flow range.

Operation

The Series 6, MCV–ISO valves are controlled by proportional, push–type oil immersed solenoids, which provide precise metering characteristics, regardless of load variations.

Features

- Low Cost
- Pressure Compensation per Spool
- Precise low flow metering
- Compact and lightweight – saves space
- Adaptable to larger HPI proportional valves
- No external pilot or drain lines
- No null adjustment
- Adjustable flow control for each cylinder port
- Spade or Hirschmann connectors

Installation Data

Temp. Range	–40°F to 180°F (–40°C to 82°C)
Fluid Viscosity	35 to 1750 SSU recommended
Filtration	25 micron return line
Fluids	Petroleum base (consult factory for other fluids)
Mounting	Valve will function in any position; however, it is recommended valve be mounted horizontally for optimum performance.

Operating Specifications

Flow Ratings	.2 GPM (.76 LPM) min.— 7 GPM (26.5 LPM) max.
Max. Pressure	3000 PSI (204 Bar) maximum
Flow Settings For other input currents, please specify when ordering	Available in increments of .10 GPM *(.2 – 7 GPM) * "A" spool is limited to 5 GPM output flow. Output flow is based on a solenoid current of: 2100 ma.–12 VDC (A & C spool) 1310 ma.–24 VDC (A & C spool)
Seals	Nitrile (Standard) Viton (Optional)

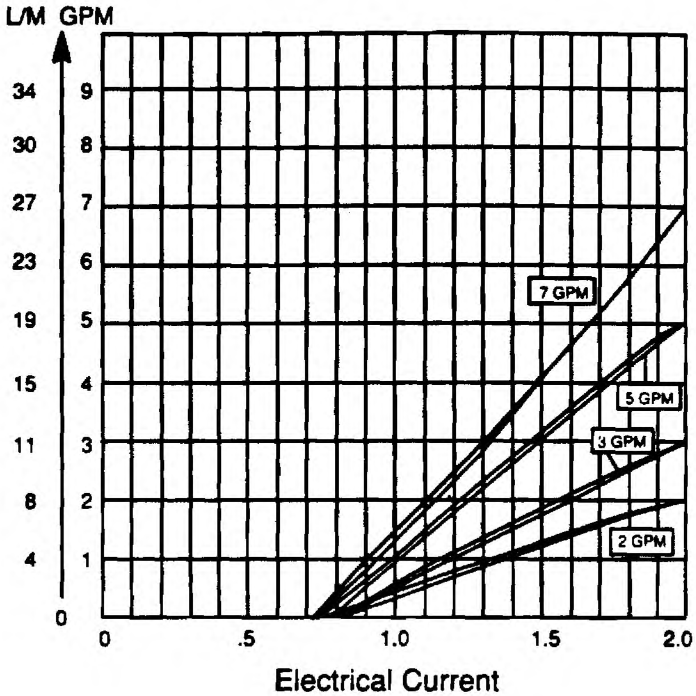
Electrical Specifications

Voltages	12 VDC or 24 VDC
PWM Control Required	To obtain precise, proportional control, pulse width modulation (PWM) is required: 40 Hz (P–Q Controllers) 65 Hz (OEM Controllers)
Current Range	Cracking to full flow: 12 VDC 500–2100 ma. (C spool) 800–2100 ma. (A spool) 24 VDC 310–1310 ma. (C spool) 500–1310 ma. (A spool) Output flow is based on a solenoid current of: 2100 ma.–12 VDC (A & C spool) 1310 ma.– 24 VDC(A & C spool)
Coil Resistance	At 68°F (20°C): 12 VDC 3.9 ohms 24 VDC 13.7 ohms At 140°F (60°C): 12 VDC 4.5 ohms 24 VDC 13.8 ohms
Duty	Continuous duty rated
Grounding	No grounding is required. No ground terminal is available
Connections	Standard: 1/2" male spade terminals
Optional	DIN 43650 appliance type plug with gasket for 6 mm to 8 mm cable diameter with one mounting screw. To order optional appliance plug separately, use P/N 15000–14

Performance Data

Hysterisis Curve (12 VDC)

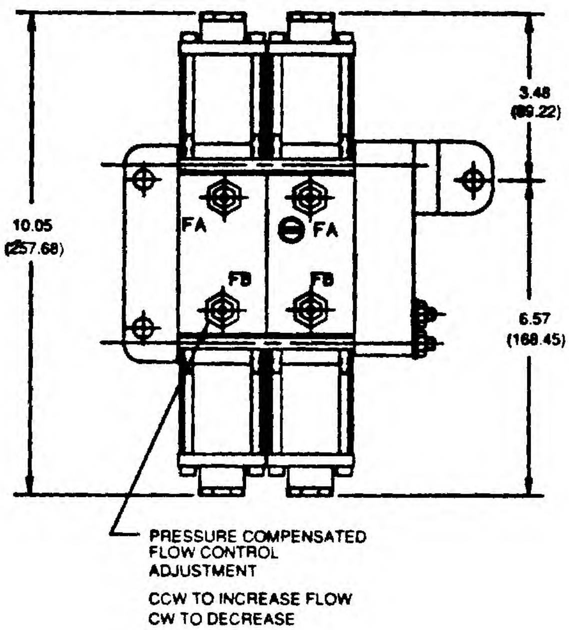
Various Flow Settings

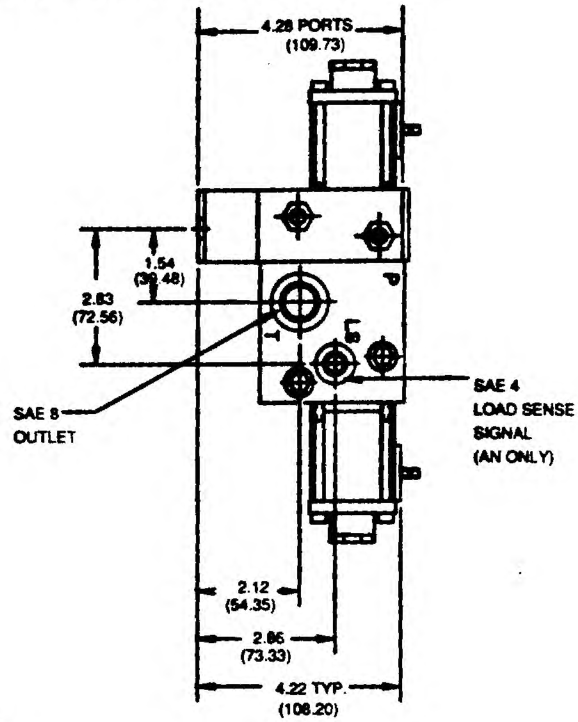


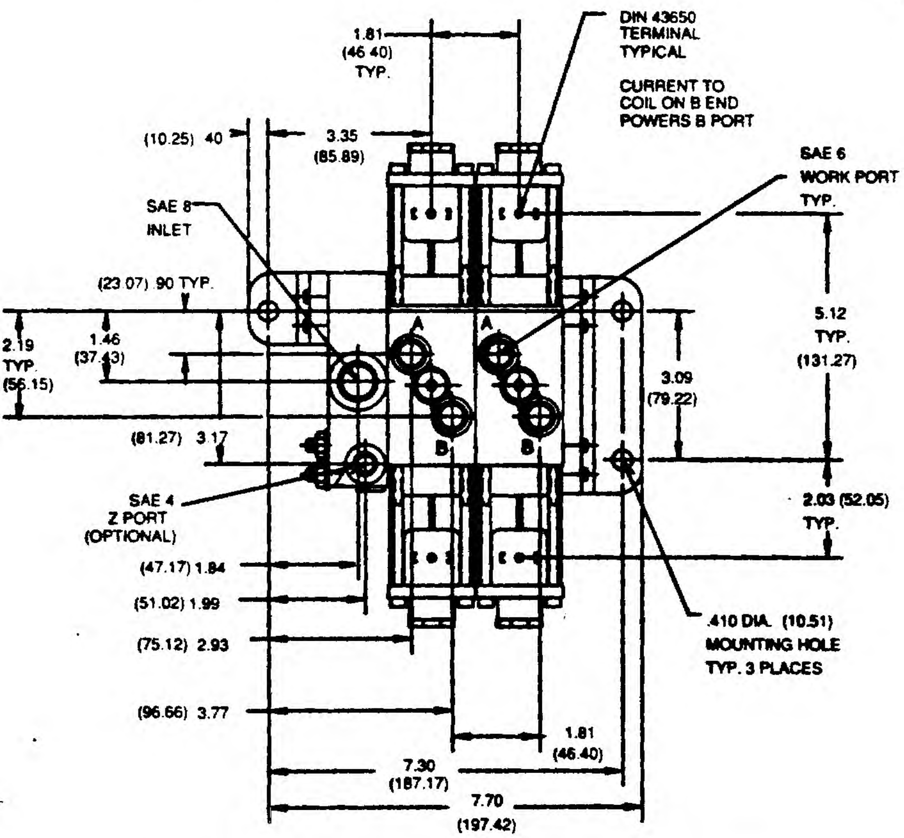
mensions

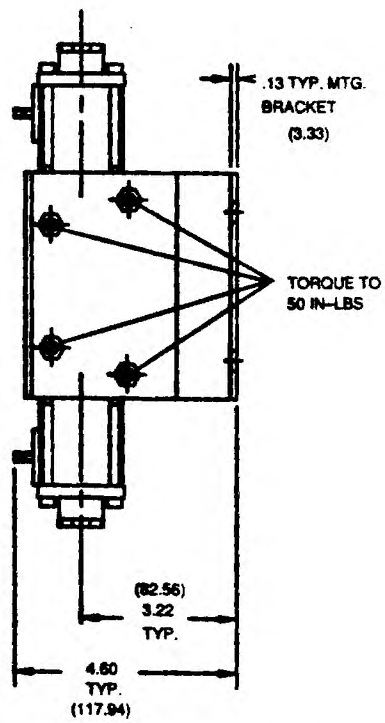
ilimeter equivalents for inch dimensions are shown in (**)

ese drawings show a typical two function electrohydraulic proportional valve assembly which is used with load nse pumps and auxiliary shuttle option. Mounting brackets can be reversed 180°









Engineering Performance Data

D1V Series Pressure Drop vs. Flow

The chart to the right provides the flow vs. pressure drop curve reference for D1V Series valves by spool type.

Example:
Find the pressure drop at 6 GPM or a D1V with a number 1 spool.

Using the top chart, locate the numeral 1 in the spool number column. To the right of the numeral 1, locate the numeral 4 in the P–A column, and the numeral 1 in the B–T column.

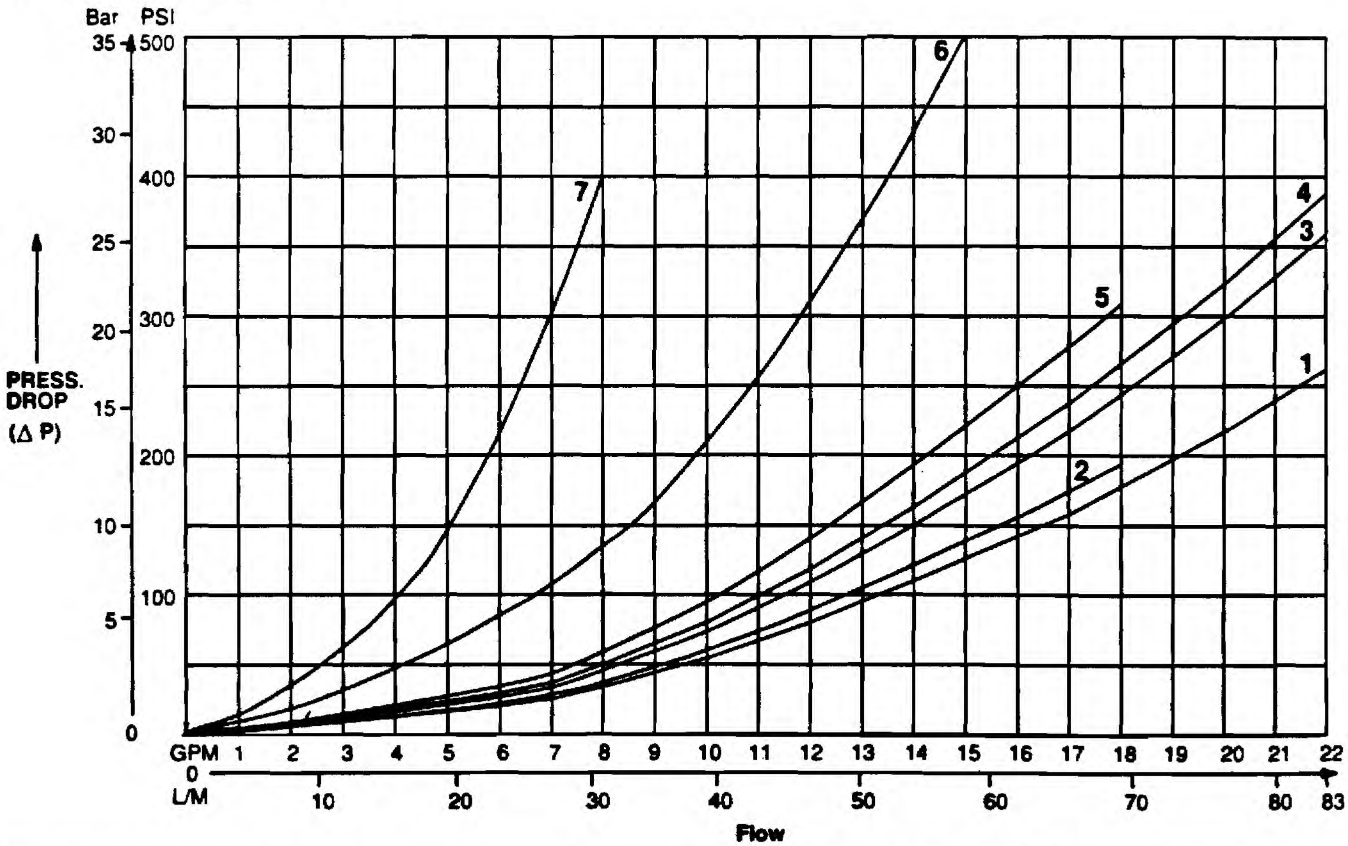
Using the bottom graph, locate curve 4; the pressure drop P–A is 28 PSI at 6 GPM. Then using curve 1, the pressure drop B–T at 6 GPM is 22 PSI. Total pressure drop through the valve is then 28 + 22 = 50 PSI.

Note: Pressure drops should be checked for all flow paths, especially when using non–symmetrical spools (spools 3, 5, 7, 14, 15, 16, 21 & 22) and unbalanced actuators.

D1V Pressure Drop Reference Chart

Spool No.	Curve Number						
	P–A	P–B	P–T	A–T	B–T	B–A	A–B
1	4	4	–	1	1	–	–
2	3	3	4	1	1	–	4
3	4	4	–	1	1	–	–
4	4	4	–	1	1	–	–
5	3	4	–	1	1	–	–
6	3	3	–	1	1	–	–
7	4	3	6	1	1	–	6
8	2	2	6	4	4	–	6
9	2	2	6	4	4	–	6
10	4	4	–	–	–	–	–
11	4	4	–	1	1	–	–
14	3	4	–	1	1	–	–
15	4	4	–	1	1	–	–
16	4	3	–	1	1	–	–
20	5	5	–	2	2	–	–
21	4	7	–	1	–	4	–
22	7	4	–	–	1	–	4
30	5	5	–	2	2	–	–

Pressure Drop Chart

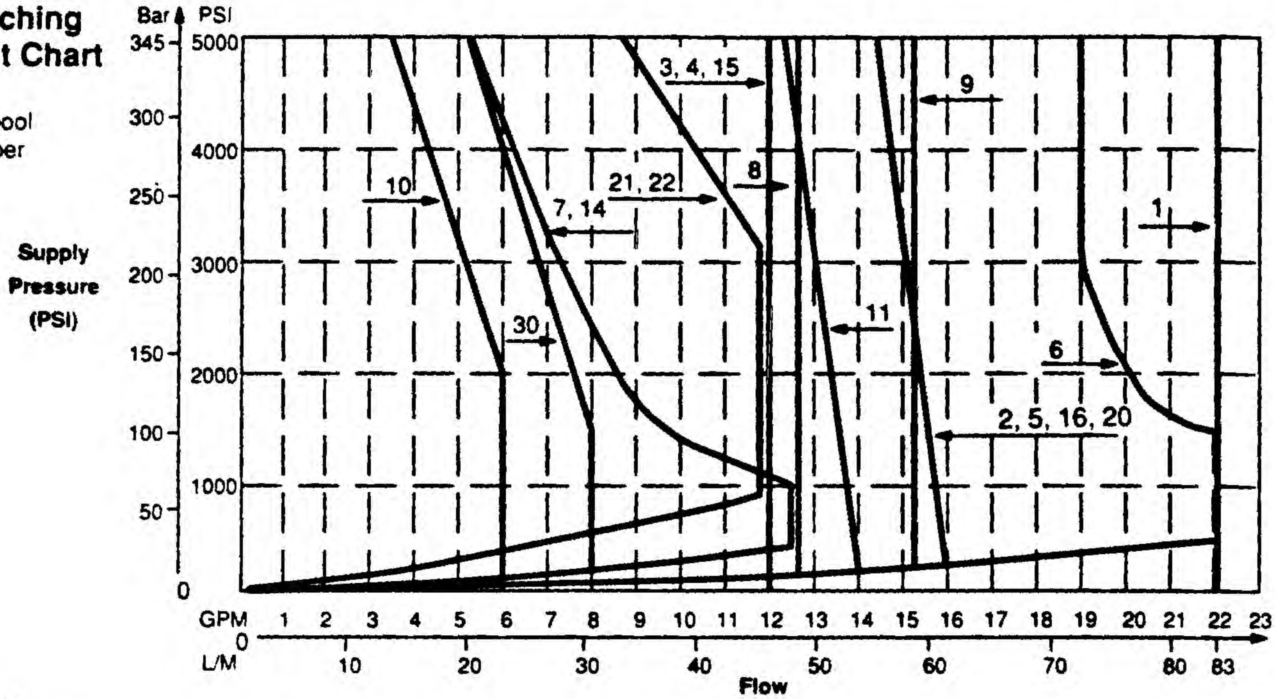


Curves were generated using 100 SSU hydraulic oil. For any other viscosity, pressure drop will change as per chart.	VISCOSITY CORRECTION FACTOR							
	Viscosity (SSU)	75	150	200	250	300	350	400
	% of ΔP (Approx.)	93	111	119	126	132	137	141

For additional information — call your local Parker Fluidpower Distributor.

Switching Limit Chart

by Spool Number



Example:

Determine the maximum allowable flow of a D1V Series Valve (#30 spool) at 1200 PSI (83 Bar) supply pressure. Locate the curve marked "30". At 1200 PSI (83 Bar) supply pressure, the maximum flow is 8 GPM (30 L/M). At 2000 PSI (138 Bar) the flow is 7.5 GPM (28 L/M).

Notes:
1. For F & M style valves, reduce flow to 70% of that shown
2. For AC low–watt coils, reduce flow to 40% of that shown, 3000 PSI max.
3. For DC low–watt coils, reduced flow to 85% of that shown.

For Maximum Valve Reliability, Adhere to the Following Installation Information

Recommended Mounting Surface

Surface must be flat within .0004 inch T.I.R. and smooth within 32 micro–inch. Torque bolts to 50 in.–lbs. (5.6 N.m)

Mounting Position

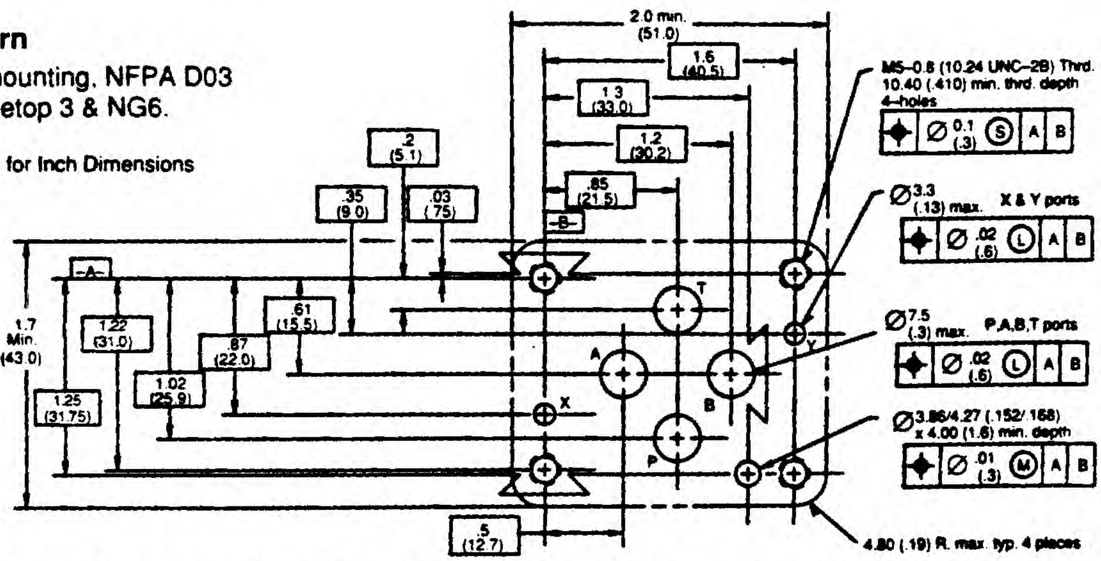
Detent – Unrestricted (Lever)
Detent – Horizontal (Solenoid, Pilot Operated)
Spring Offset – Unrestricted
Spring Centered – Unrestricted

E

Mounting Pattern

D1V – Subplate mounting. NFPA D03 (Formerly D01), Cetop 3 & NG6.

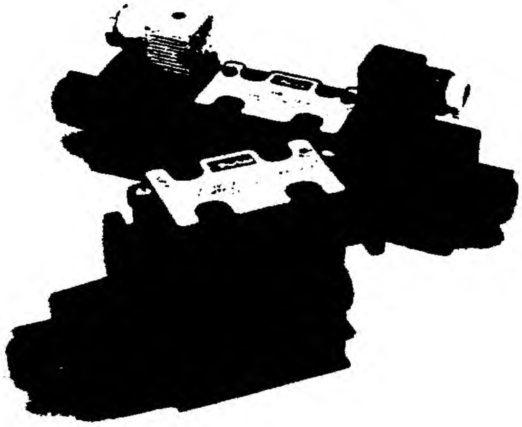
"Millimeter Equivalents for Inch Dimensions are shown in (**)"



Mounting Surface D1V Directional Control Valve Manifold M'TD. (NFPA D03, Formerly D01); Cetop 3 & NG6
Note: X and Y Ports are required for subplate piloted D1VP valves only

Engineering Performance Data

Mounting Pattern	NFPA D03, (Formerly D01); CETOP 3; NG 6
Maximum Pressure	Operating 5000 PSI (345 Bar) C.S.A. rating 4000 PSI Tank Line: 1500 PSI (103 Bar) – Standard 3000 PSI (207 Bar) – Optional 200 PSI (13.8 Bar) with monitor switch (10 option)
Nominal Flow	8.5 GPM (32 Liters/Min.)
Maximum Flow	See chart on page 3



Solenoid Electrical Characteristics *

Solenoid Code	Nominal Volts/Hz	In Rush Amps	Holding Amps	Watts
Y	120/60 110/50	2.00 2.10	0.49 0.58	25 27
YF**	120/60 110/50	0.90 0.94	0.18 0.20	10 9.5
T	240/60 220/50	1.00 1.05	0.26 0.31	25 27
R	24/60	10.50	2.70	27
RF**	24/60	5.50	0.85	10
L	6 VDC	—	5.00	30
LF+	6 VDC	—	4.00	24
K	12 VDC	—	2.50	30
KF+	12 VDC	—	2.00	24
J	24 VDC	—	1.25	30
JF+	24 VDC	—	1.00	24
D	120 VDC	—	0.25	30
DF+	120 VDC	—	0.20	24
Z	250 VDC	—	0.12	30
ZF+	250 VDC	—	0.10	24

*Based on nominal voltage @ 72°F.

**3000 PSI, 40% rated flow (low watt A.C.)

+ 3000 PSI, 85% rated flow

C.S.A. file LR60407

Response Time

Nominal response time (milliseconds) at 5000 PSI, 8.5 GPM

Solenoid Type	Pull–In	Drop–Out
AC	13	20
DC	32	40

Solenoid Ratings

Insulation	Class F
Allowable Deviation from rated voltage	–10% +15%
Wet Armature Type	

Explosion Proof Solenoid Ratings

U.L. (P07)	Class I, Div. 1 & 2, Groups C & D. Class II, Div. 1 & 2, Groups E, F & G. As defined by the N.E.C.
M.S.H.A. (P06)	Complies with 30 CFR, Part 18.

Solenoid Electrical Characteristics * P06 and P07

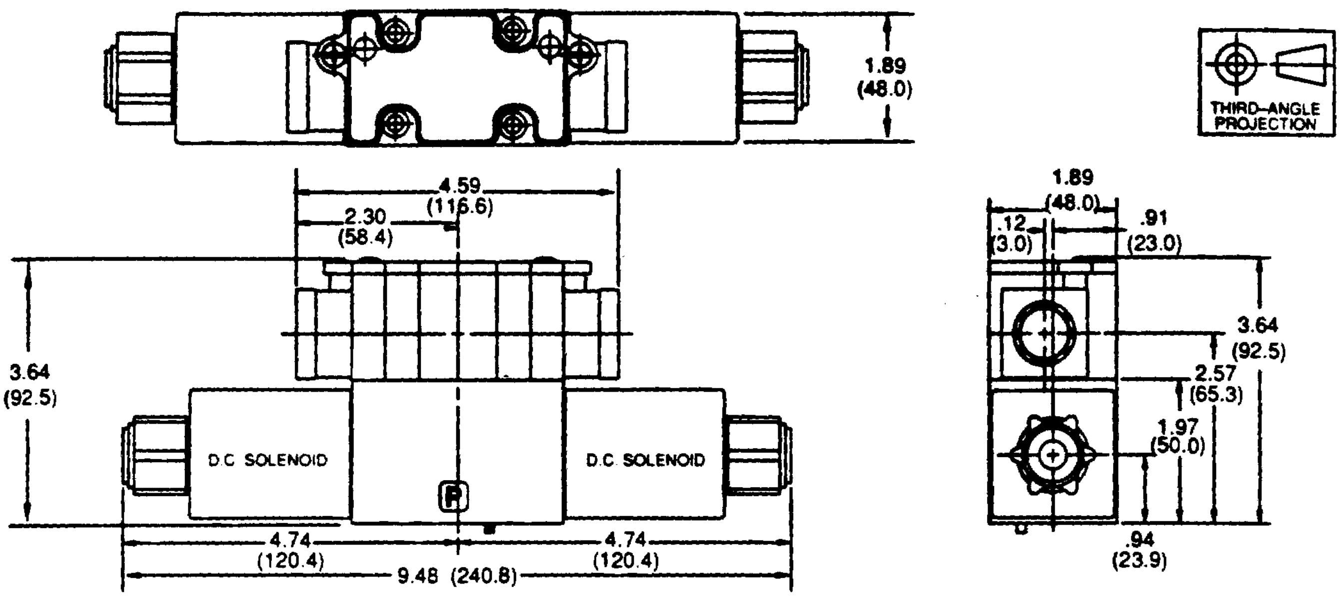
Solenoid Code	Nominal Volts/Hz	In Rush Amps	Holding Amps	Watts
Q	100/60	2.60	0.70	27
Y	120/60	2.20	0.58	27
T	240/60	1.10	0.29	27
R	24/60	11.10	2.90	27
L	6 VDC	—	5.50	33
K	12 VDC	—	2.75	33
J	24 VDC	—	1.38	33
D	120 VDC	—	0.28	33
Z	250 VDC	—	0.13	33

E

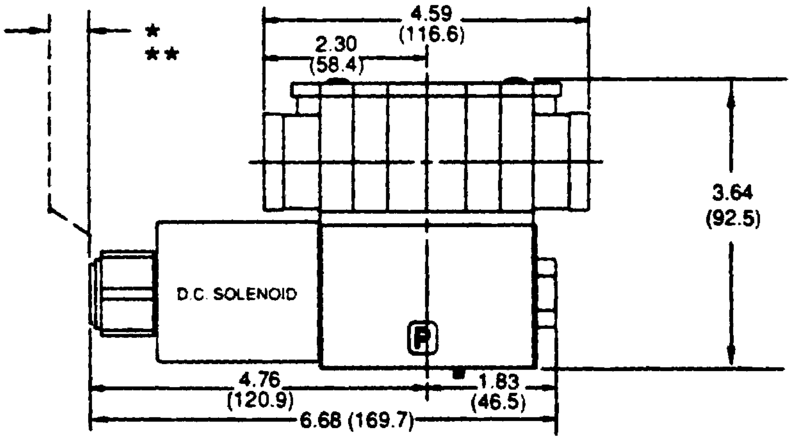
Dimensions

"Millimeter equivalents for inch dimensions are shown in (**)"

Conduit Box, D.C. Solenoid, Double



Conduit Box, D.C. Solenoid, Single

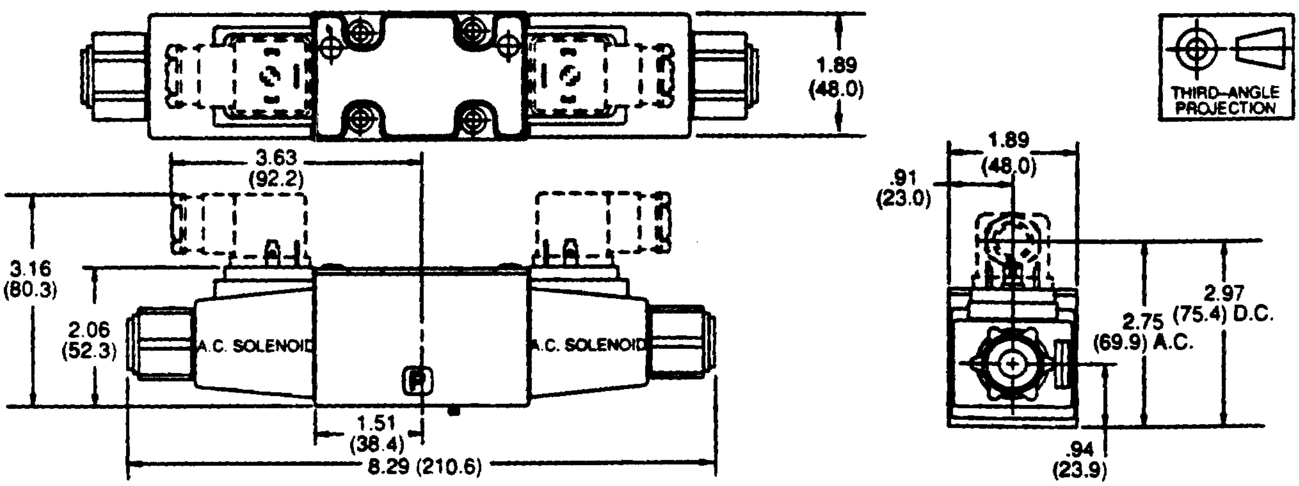


* Minimum clearance for coil removal
— D.C. 2.25 (57.2)

** Minimum clearance for solenoid removal
— D.C. 1.00 (25.4)

E

Hirschmann, A.C. Solenoid, Double



D	1V	W							
DIRECTIONAL CONTROL VALVE	BASIC VALVE	ACTUATOR	SPOOL	STYLE	ATTACHMENTS OR VARIATIONS	SEAL COMPOUND	SOLENOID	SOLENOID MODIF.	DESIGN SERIES

Basic valve: NFPA D03, (formerly D01)

Actuator: Wet armature solenoid

Design series — Note: not required when ordering

Spool

Code	Symbol	Code	Symbol
1		10	
2		11	
3		14	
4		15	
5		16	
6		20*	
7		21	
*8, 9**		22	
		30**	

* 8 & 20 spool have closed crossover
** 9 & 30 spool have open crossover

Seal Compound

Code	Type
Omit	Nitrile
V	Viton

Attachments or Variations

Code	Description
Omit	Standard Valve No variations
4	CSA approval
5	Signal lights
6	Manaplug plug–in
10 ⊙	Monitor switch
56	Lights & manaplug
630	5 pin plug–in manaplug w/single solenoid valve
P06 ⊙	Exp. proof M.S.H.A.
P07	Exp. proof U.L.
P10 ⊙	Spade lug conn.
P14	Extended override & boot
P16 ⊙	High pressure solenoid tube

⊙ Not C.S.A. Approved

Solenoid Modif.

Code	Description
F	Low amp coil

Note: Low amp A.C. coils are not CSA approved.

Solenoid

Code	Description
Y	120V / 60 Hz 110V / 50 Hz
YW	Standard Hirschmann #
YY	Hirschman with plug
T	240V / 60 Hz 220V / 50 Hz
TW	Standard Hirschmann #
TT	Hirschmann with plug
R	24V / 60 Hz
L	6 VDC
K	12 VDC
KW	Standard Hirschmann #
KK	Hirschmann with plug
J	24 VDC
JW	Standard Hirschmann #
JJ	Hirschmann with plug
D	120 VDC
DW	Standard Hirschmann #
DD	Hirschmann with plug
Z	250 VDC

\# Mating connectors must be ordered separately.

Style

Code	Description	Symbol
B†	Sgl. solenoid, 2 position, spring offset. P to A and B to T in offset position.	
C	Dbl. solenoid, 3 position, spring centered	
D†	Dbl. solenoid, 2 position, detent	
E	Sgl. solenoid, 2 position, spring offset to center. P to B and A to T when energized.	
F	Sgl. solenoid, 2 position. Spring offset, energized to center. Position spool spacer on a side. P to A and B to T in spring offset position.	
H†	Sgl. solenoid, 2 position, spring offset. P to B and A to T in offset postion.	
K	Sgl. solenoid, 2 position. Spring offset to center. A side. P to A and B to T when energized..	
M	Sgl. solenoid, 2 position, spring offset, energized to center position. Spool spacer on B side. P to B and A to T in spring offset position.	

Subplate Note: See **"Installation Information, Directional Control Valves"** section of this catalog for subplate drawing and model numbers.

This condition varies with spool code

† Only spools 20 & 30

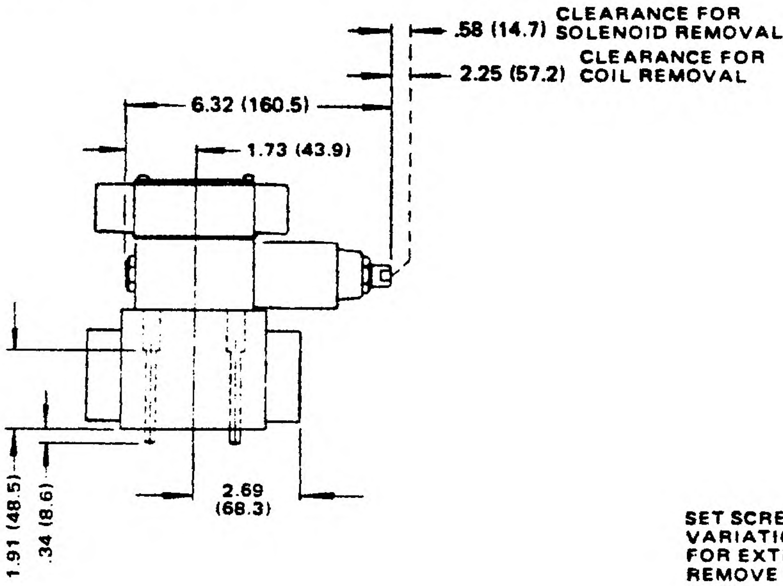
Unit Weight:
Single Solenoid 3.0 lbs. (1.36 kg)
Double Solenoid 3.5 lbs. (1.6 kg)

IENSIONS

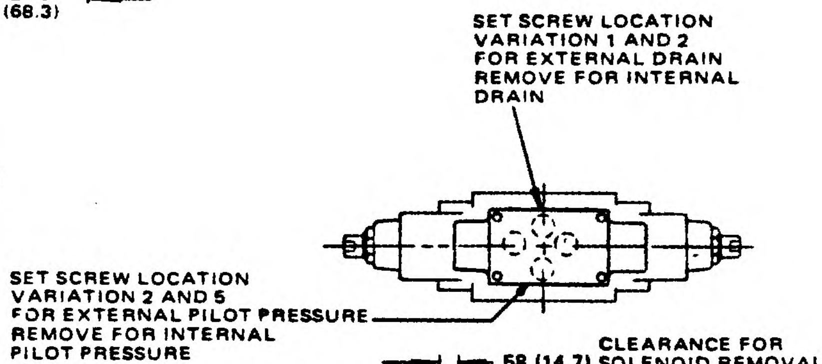
METER EQUIVALENTS FOR INCH DIMENSIONS ARE SHOWN IN (**)"

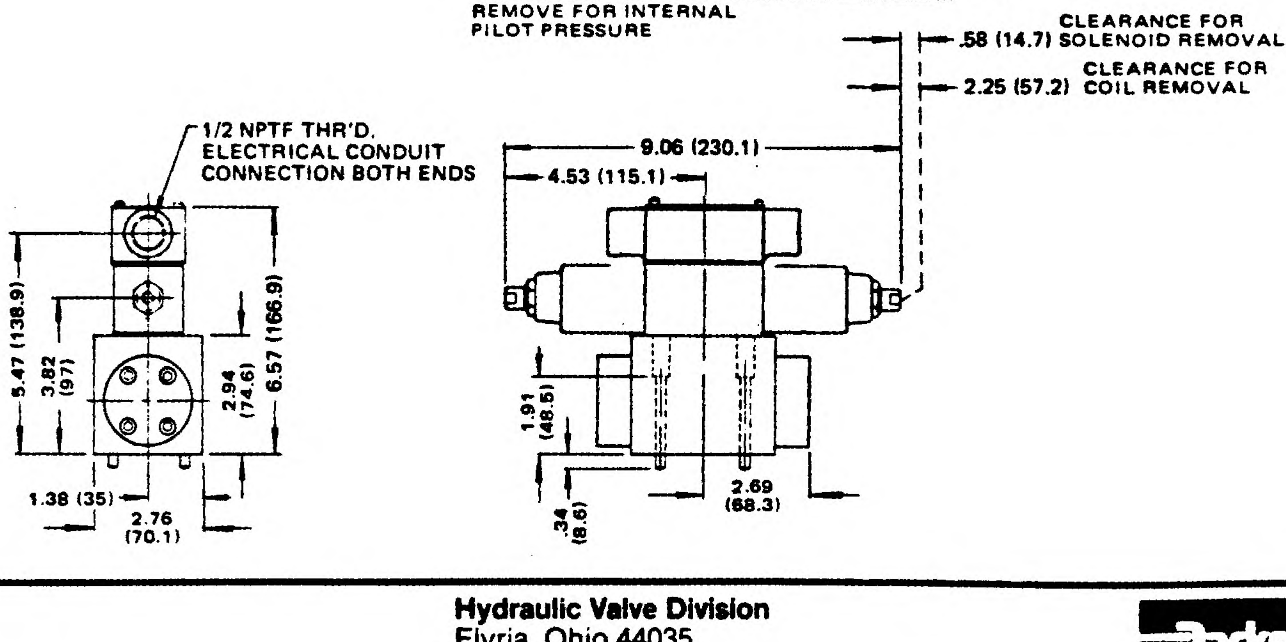
SOLENOID

;LE SOLENOID, SPRING OFFSET MODELS
VW*B*, D31VW*E*, D31VW*F*, D31VW*H*



ENTERED AND DETENTED MODELS
31VW*C*, D31VW*D*

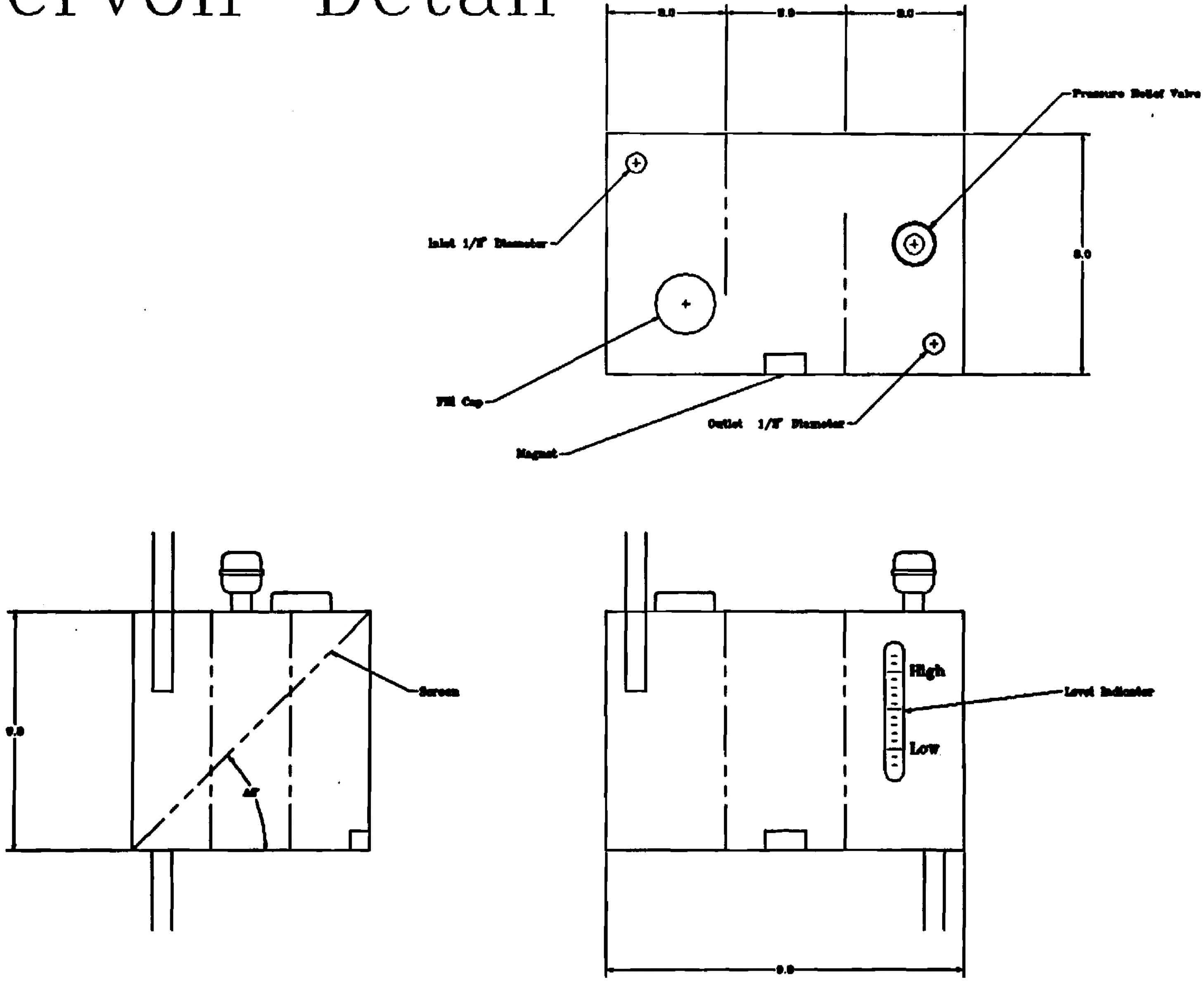




E

APPENDIX H

Reservoir Detail



H1

Power Supply – Group 5

1/4" = 1"
ME4182a1
Revision 1

Appendix H 2

Reservoir Description

The body of the reservoir is to be made from steel which is about 40 thousandths of an inch thick. This is so that the sides can be welded to the top and the bottom. The screen is 100 mesh steel screen. It should be placed at a 30° to 45° angle to the bottom. This will provide optimum performance of removing the air from the fluid. The baffles are to be made from the same steel as the body. Two baffels were used to increase the lenght the fluid has to flow without increasing the velocity of the fluid.

The level indicator is available from Grainger and is made by Dayton. The model to be used includes a temperature guage and is model number No. 1A760 Reservoir Fluid Level Guage. The cost is less than $14. The filler cap is a standard container cap available from several vendors. The pressure relief valve is a standard relief valve and should be set for about 2 psi. The magnet is a permant type and should be mounted on the bottom of the reservoir where the velocity of the fluid is the least. The inlet and outlet ports will require fittings to connect them with the hydraulic lines.

ME 4192 FINAL PROJECT
SPRING 1993

THE ENABLER:
CHASSIS JOINTS AND DRIVES

STEVEN INTRONE
BRIAN JAY
DAVID SHELDON

EXECUTIVE SUMMARY

The purpose of the actuators and motors group is to select appropriate motors and drive systems for the chassis that will enable the wheels and the joints to move. This report finalizes the drive systems by specifying the components based on the torque and power requirements. The chassis needs six motors for the wheels and six motors for the joints. Char-Lynn model #101-1012 hydraulic motors with four-bolt flanges were chosen for the wheels, and Char-Lynn model #101-1061 hydraulic motors with four-bolt flanges were chosen for the articulation joints. The drive system for the wheels includes the motor connected to the wheel shaft by way of a Lovejoy coupler. The drive system for the articulation joints include a Diamond double-strand chain run by Morse sprockets, one on the motor and the other on the joint connector, creating a 3.43:1 gear ratio.

DESIGN OVERVIEW

Wheel Drive

The wheel drive involves five main parts: the Char-Lynn model #101-1012 hydraulic motor, a Lovejoy coupler, a drive shaft, a motor bracket, and a wheel plate which connects the drive system to the wheel hub. Only the motor bracket and the wheel plate will need custom machining. The motor will be bolted to the bracket by four flat-head, 3/8"-16 UNC, countersunk screws (max. thread engagement is 0.6"). The bracket will be screwed in place. The coupler is keyed to the motor shaft and consists of two hub halves (model #L099 x 1" hub) and one bronze insert (model #L099). This assembly allows for misalignment and is the point where the wheel may be detached from the vehicle. The wheel plate is to be welded in place, and no bearings are needed to support the shaft. Refer to the appendix for appropriate drawings.

Articulation Joint Drive

Driving the articulation joint requires considerably more power and is more complex than the wheel drive. Refer to the drawings in the appendix for more clarity. The main components are the Char-Lynn model #101-1061 hydraulic motor, a Diamond double-strand 3/8" pitch drive chain (58 links per joint), two Morse sprockets (48T and 14T), a motor bracket, a 4" diameter bronze journal bearing, a custom joint coupler, and a joint plate. The bracket is to be custom made from 3/8" 6066 aluminum plate. It will be screwed into place on the face of the upper bearing sleeve. The motor is bolted to the bracket in the same manner as the wheel motors. The bearing is anchored in the bracket using two Industrial Retaining Rings (model #3100-400). The 48T sprocket (cat. #D35B48) is to be bored out to an inner diameter of 4" (so as to fit on the bearing; don't forget to use lots of grease). This sprocket also needs two 5/16" dowel pin holes set 180^{o} apart at a diameter of 4.182". The 14T sprocket (cat. #D35B14) is to be broached to fit the splined motor shaft. The dowel pins will be fixed in the joint coupler so that they mate exactly with the holes in the 48T sprocket. This facilitates easy disassembly of the joint. The joint coupler is to be fixed to the joint plate by four 9/32" bolts, equally spaced, on a 4.895" bolt circle, and then the plate is fixed to the face of the lower bearing sleeve.

SELECTIONS FOR THE WHEEL DRIVE

Motor

The selection of the motor is the most important part of our project. We have chosen the Char-Lynn H-series #101-1012 hydraulic motor to drive the wheels. The motor has a 1" straight shaft with woodruff key and 7/8-14 o-ring ports, with a four-bolt flange for connection to the motor bracket. With the 0.6 GPM flow rate given, each motor will be able to obtain a RPM of 33.8 RPM. The rated torque of each motor is 2824 in·lbs and it can run at a pressure of 2250 psi, which is well within the given 2000 psi. The motor will have a direct connection to the wheel shafts by couplers. The motor is optimum because of its' abilities and its' size. This motor is relatively small and is easy to mount. The motor is easily accessible and relatively inexpensive, $168 per motor. We will use six of these motors, one for each wheel.

Motor Bracket

The motor bracket was designed to keep the motor stable during operation. It is a piece of ANSI 6066 Aluminum, 4" wide and 0.375" thick. The bracket will be mounted onto the upper sleeve with six ANSI UNC Type 8, 0.164 diameter SAE Grade 1 steel machine screws. Each screw will be 1" in length. In the middle, the motor is connected by the use of the four-bolt flange and four 3/8-16 UNC machine screws, each 1" in length. The dimensions were selected to obtain the maximum support for the motor while using size, weight, and ease of manufacture as the constraints.

Coupler

The purpose of the coupler is to allow for misalignment. We have chosen the Lovejoy coupler which consists of two hub halves (model #L099 x 1") and one bronze insert (model #L099). The coupler is keyed to the motor shaft by a 1.0625"x1/4"x1/8" keyway while a 1" drive shaft is keyed to the other side of the coupler. This provides a point of easy detachment within the motor drive system.

Drive Shaft

The drive shaft of the wheel drive connects the power from the motor to the wheels. We will be using a stock 1" diameter AISI 1040 steel shaft which is available from Bearings and Drives Incorporated of Atlanta. The shaft will be connected to the wheel plate on one end and to the coupler on the other. The shaft is 3.98" long and will have two key ways cut into to it to hold the AISI 1040 steel keys. The section cut into the shaft that connects onto the coupler has dimensions of 1.0625"x1/4"x1/8". The section cut into the shaft that holds the shaft onto the wheel plate has dimensions 2.25"x1/4"x1/8". The shaft will be press fitted into the wheel plate.

Wheel Plate

The wheel plate was designed so that the wheels could be connected to the drive shaft. The wheel plate is connected onto the hub by eight equally spaced 8-SAE grade 1 machine screws. The wheel plate will be made out of 1/2" 6066 Aluminum plate. The drive shaft will connect onto the wheel plate by way of a 1040 steel key. A groove will be cut into the wheel plate with dimensions 2.25"x1/4"x1/8". The drive shaft will be press fitted into the wheel plate.

SELECTIONS FOR ARTICULATION JOINT DRIVE

Motor

The motor selection was based on the articulation joint constraints. The motor selected is the Char-Lynn H-series #101-1061 hydraulic motor. The motor has a 1" SAE 6B splined shaft and 7/8-14 o-ring ports. It is mounted on a bracket by a four-bolt flange and it will be able to obtain an RPM of 7.5 RPM. The rated torque of each motor is 3151 in·lbs, and with the gear ratio of 3.43:1, the final available torque on the joint will be 10807 in·lbs, which is sufficient for this application. The motor will run at a pressure of 2000 psi. which is well below the max pressure of 2150 psi. The motor will provide power to the joints by using sprockets and a double-strand drive chain.

Drive System

The articulation joints will be driven by a sprocket and a double-strand drive chain. A Morse 14 tooth double steel sprocket (model #D35B14) will be connected onto the splined motor shaft and kept there by a cap and endscrew. A Morse 48 tooth double steel sprocket (model #D35B48) will slide onto a bronze bearing sleeve. A double-strand 3/8" pitch drive chain will run between the two sprockets. The resulting gear ratio is 3.43:1. The 48 tooth sprocket will be connected onto a coupler by two 0.3125 diameter dowel pins. The coupler is then connected onto the joint plate, which in turn, moves the joints.

Motor Mounting Bracket

The mounting bracket was designed to incorporate the gear system. The material is 6066 Aluminum and the mounting face is 3/8" thick, with slotting holes for accurate motor positioning on the top. As seen in the appendix, the center of the bracket will have a 4" diameter hole cut into it to allow for the bearing sleeve to fit in. A 48 tooth sprocket will fit onto the bearing sleeve and the two side pieces on the bracket will allow for the connection of "Z" brackets which hold the sprocket in place during rotation. The mounting bracket will connect onto the upper bearing flange inside the joint by using six ANSI UNC-8 bolts on a 5.8125" diameter bolt circle. The purpose of the hole in the middle is to allow space for different hoses and cables to run through.

Bearing Sleeve

The bearing sleeve was selected to conserve space radially around the center of the articulation joint, while supporting the loads that might be imposed on it. The sleeve is a 4" stock bronze bushing with an inner diameter of 3.5" to allow for the passage of hydraulic line and cables. It is anchored in the motor bracket by two industrial retaining rings to facilitate easy replacement.

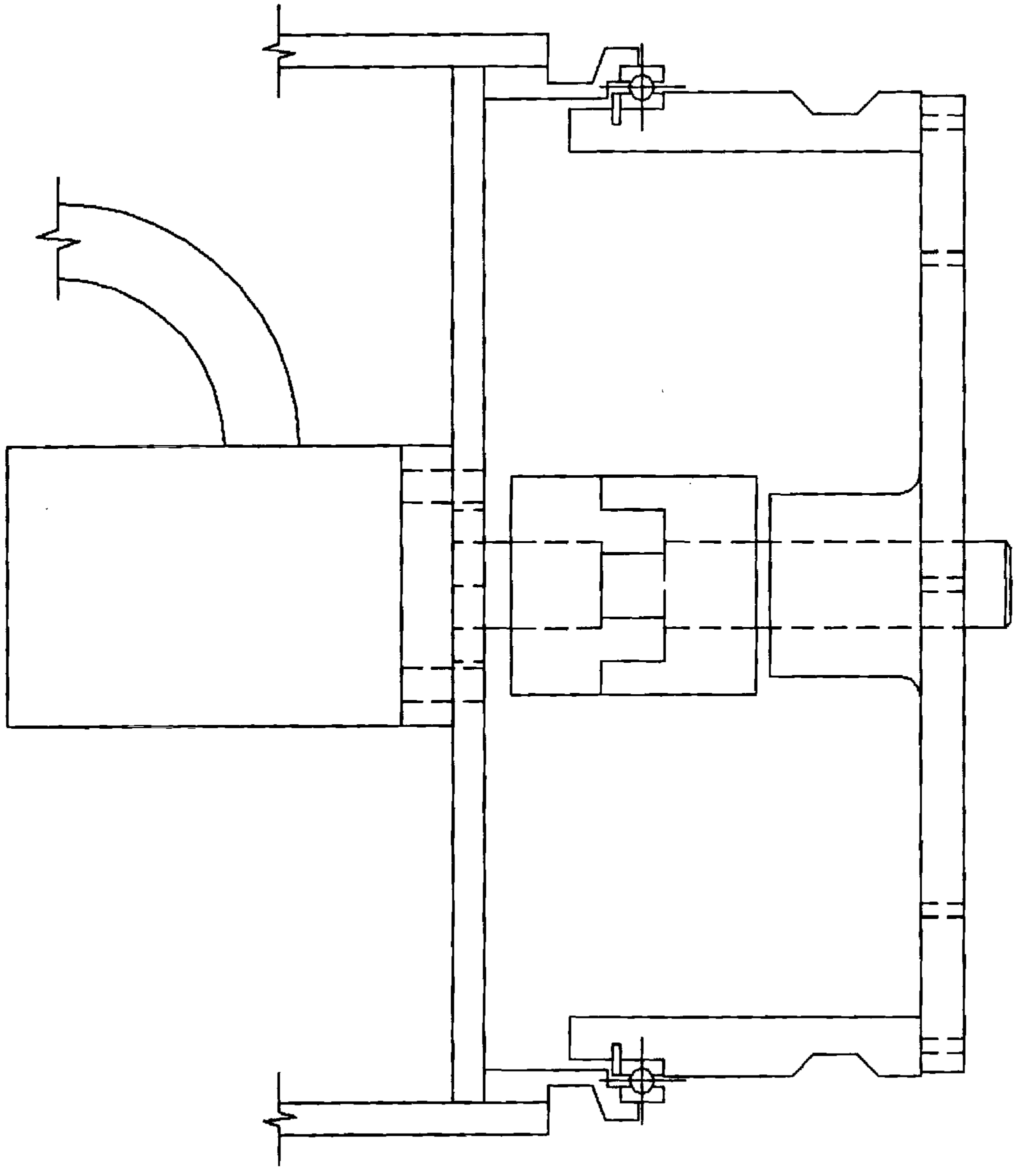
Joint Coupler

The 6066 Aluminum joint coupler connects the power from the 48 tooth sprocket to the joint plate. The coupler consists of two rings, one with four holes and one with two holes (where the dowel pins are anchored), welded to a short tube. Two 5/16″ dowel pins which are anchored in the coupler transmit torque to the coupler from the sprocket. The inner diameter of the coupler is 3.5″ which allows the appropriate hoses and cables to run through. The coupler is then connected to the joint plate with four 5/16″ bolts.

Joint Plate

The joint plate connects the coupler to the lower bearing sleeve. It will be machined out of a 3/8″ 6066 Aluminum plate. The plate is connected onto the coupler by four 5/16″ bolts that are located on a 4.895″ bolt circle in the center of it. There is a 3.5″ diameter cutout in the center to allow for the hoses and cables. The joint plate is attached to the lower bearing sleeve using six SAE Grade 1 steel machine screws on an 11.625″ circle.

Appendix



ME4182 - Enabler
Group 3
Wheel Assembly
Drawn By: DSS
Date: 6/2/93
Scale: none

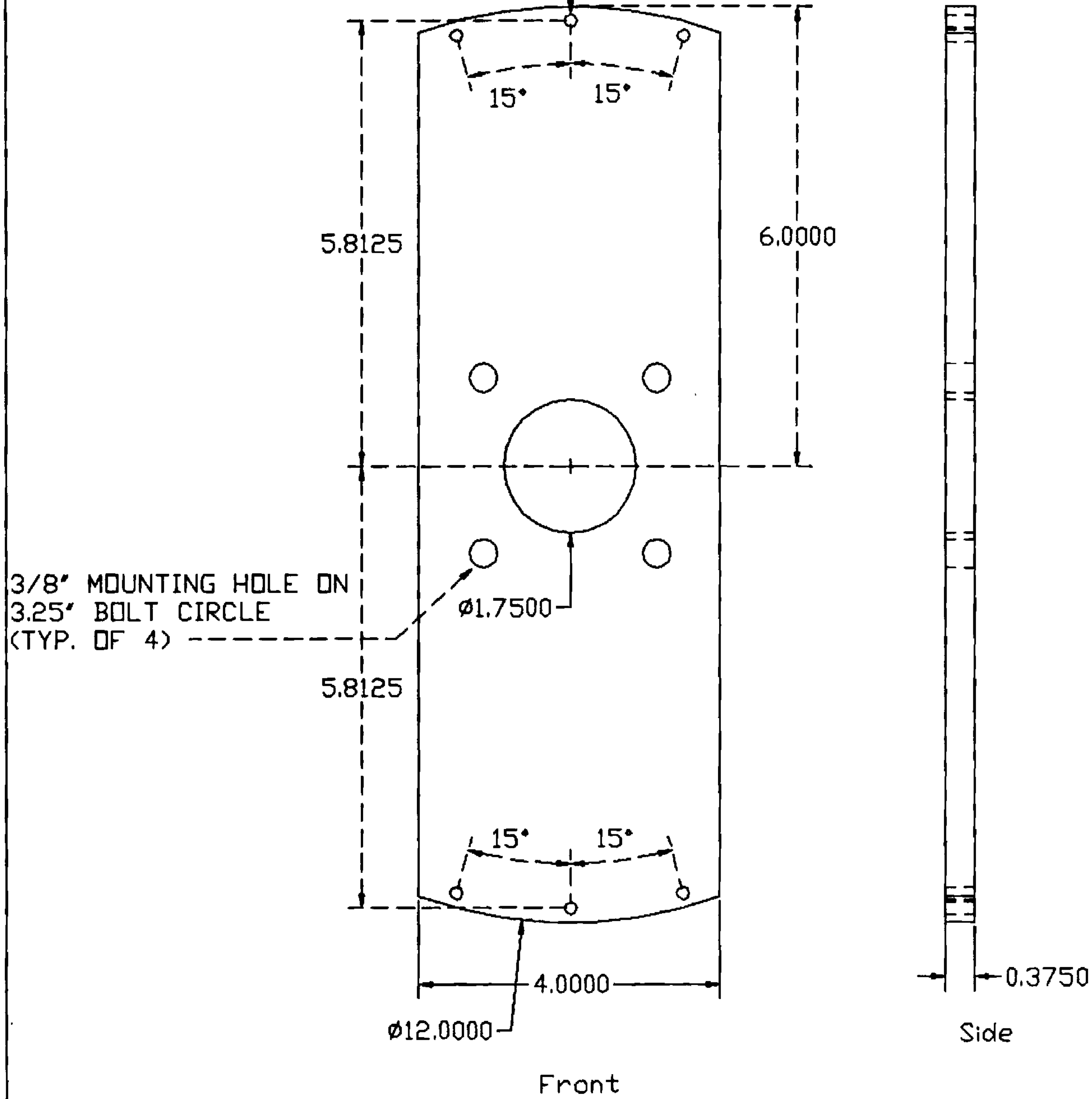
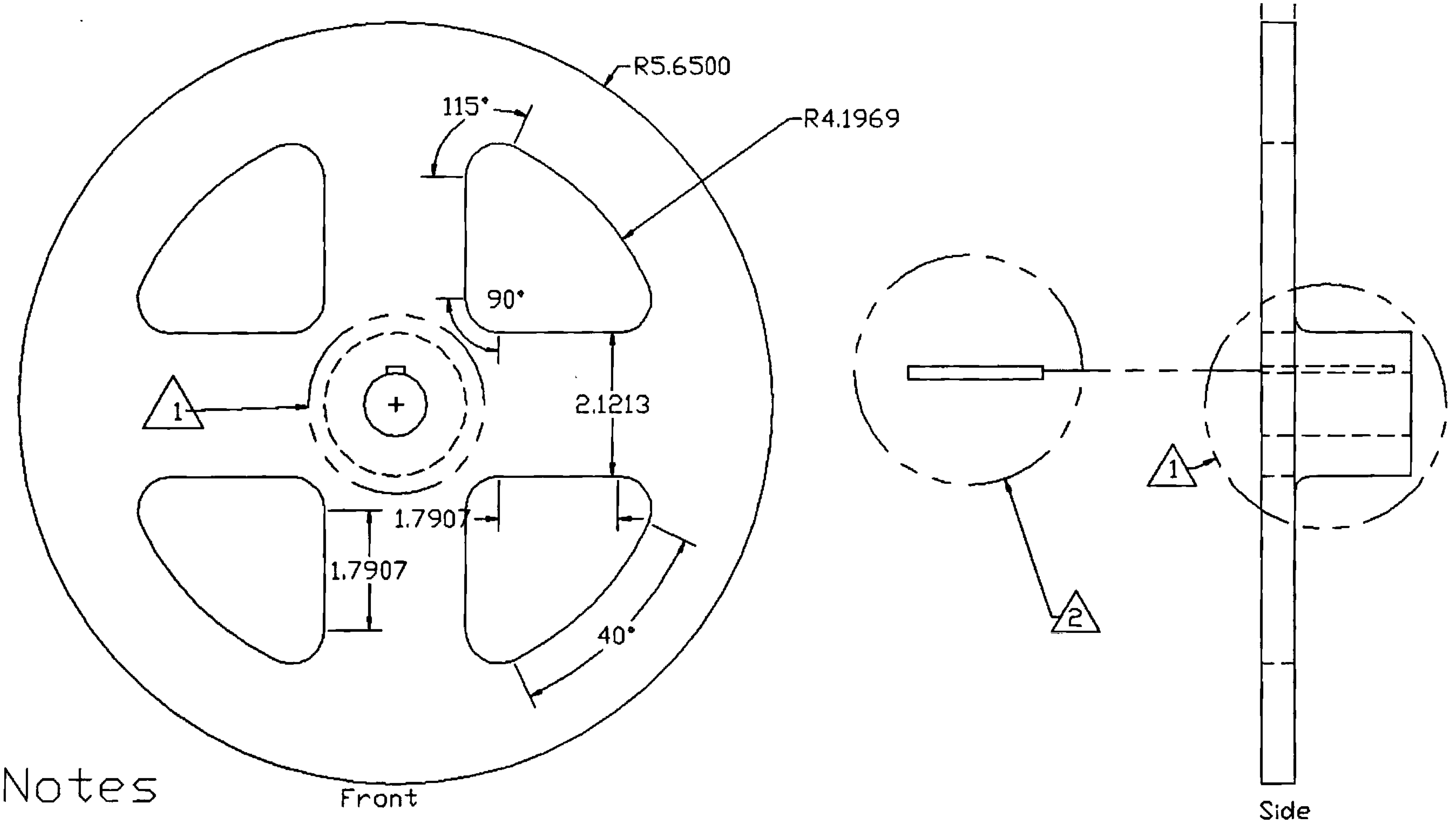


Figure 2

NOTES:

1. Material - ANSI 6066 Aluminum
2. ANSI UNC-8 Bolts on 11.625″ diameter bolt circle

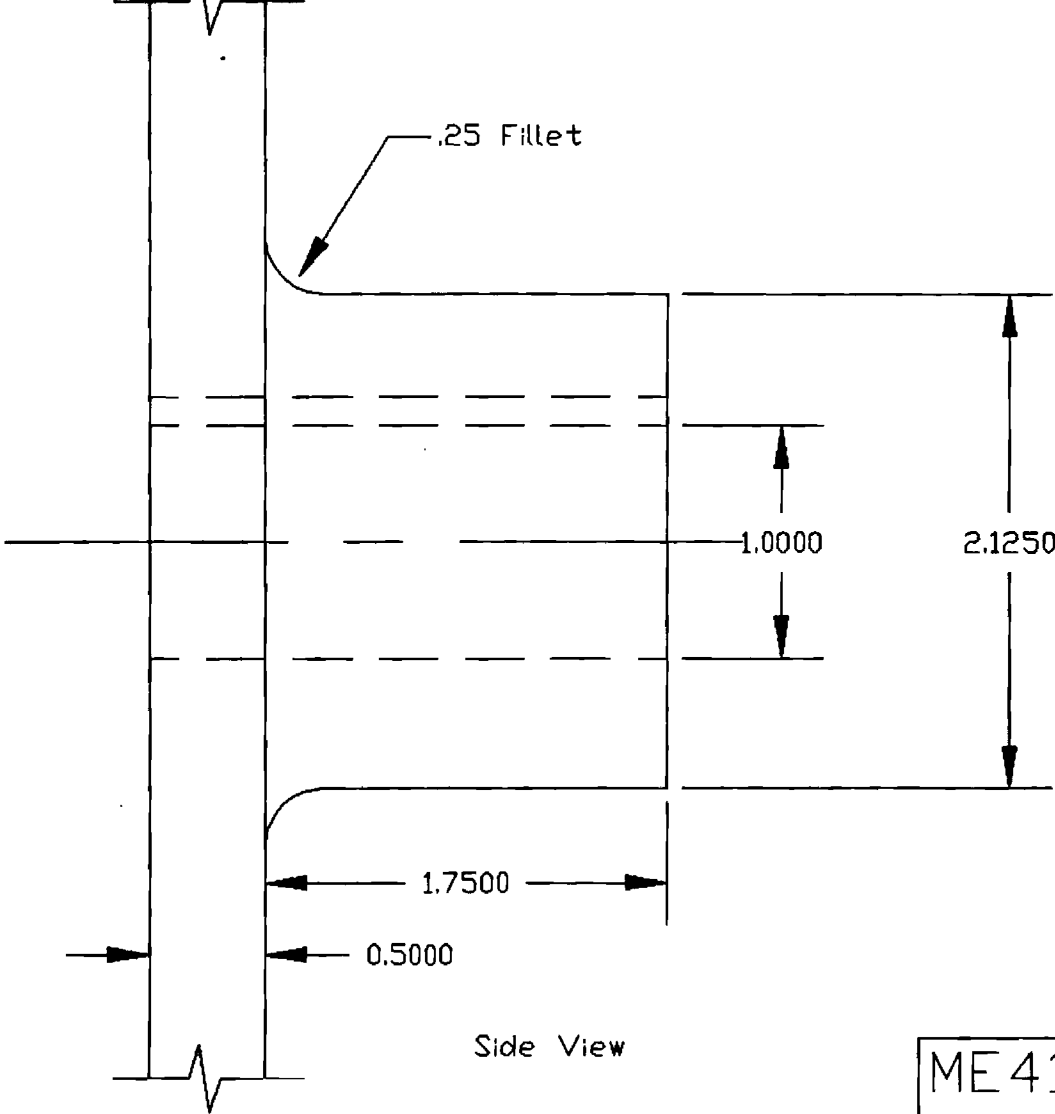
ME4192 - Enabler
Group 3 - Wheels
Name: Bracket
Drawn By: DSS
Date: 6/2/93
Scale: none



Notes

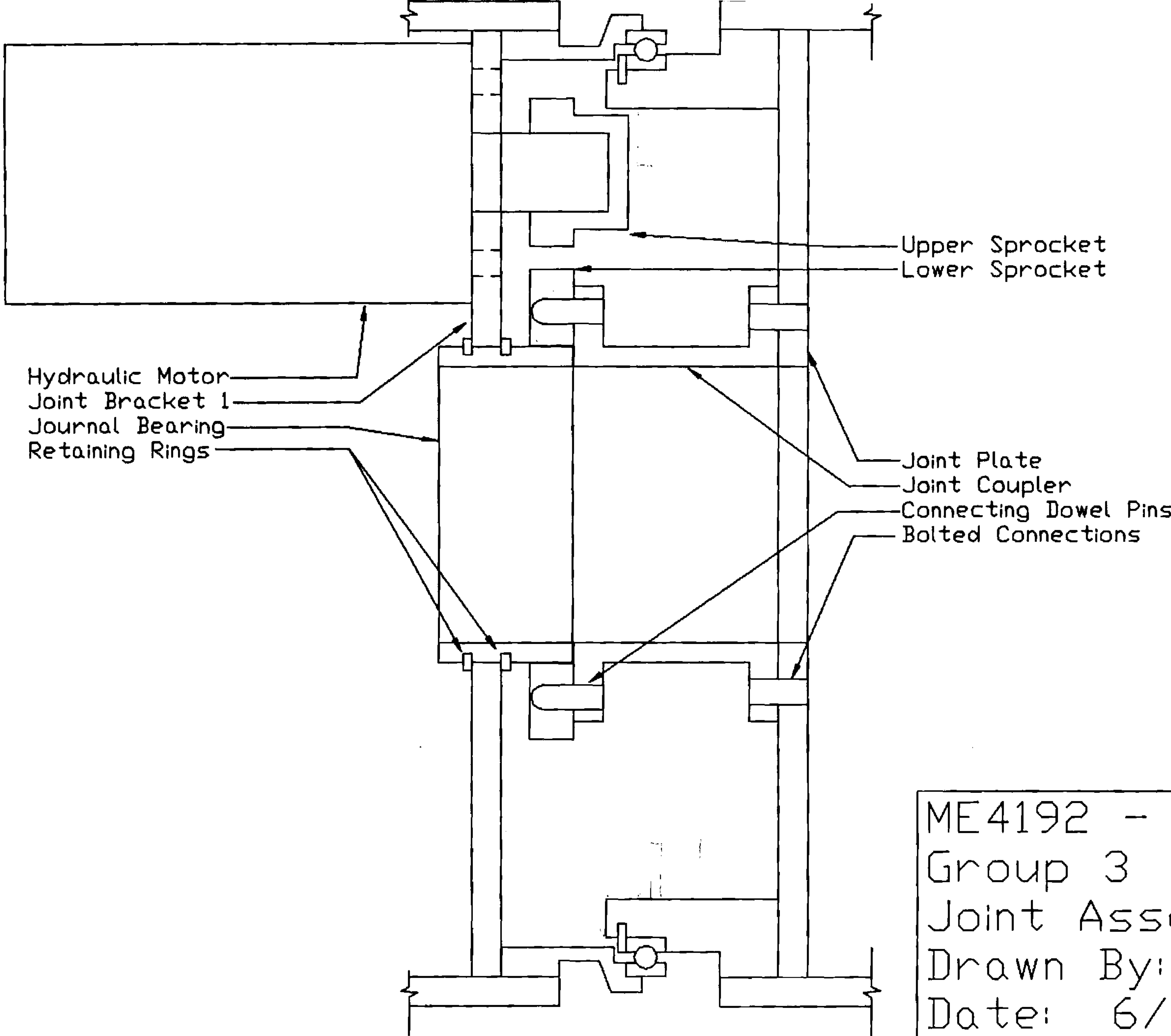
1. See Detail Drawing 1
2. 2-1/4" L x 1/4" H x 1/4" W AISI 1040 Steel Key
3. Made From 6066 Aluminum

ME4192 - Enabler
Group 3
Name: Wheel Plate
Drawn By: JRS
Date: 6/2/93
Scale: none

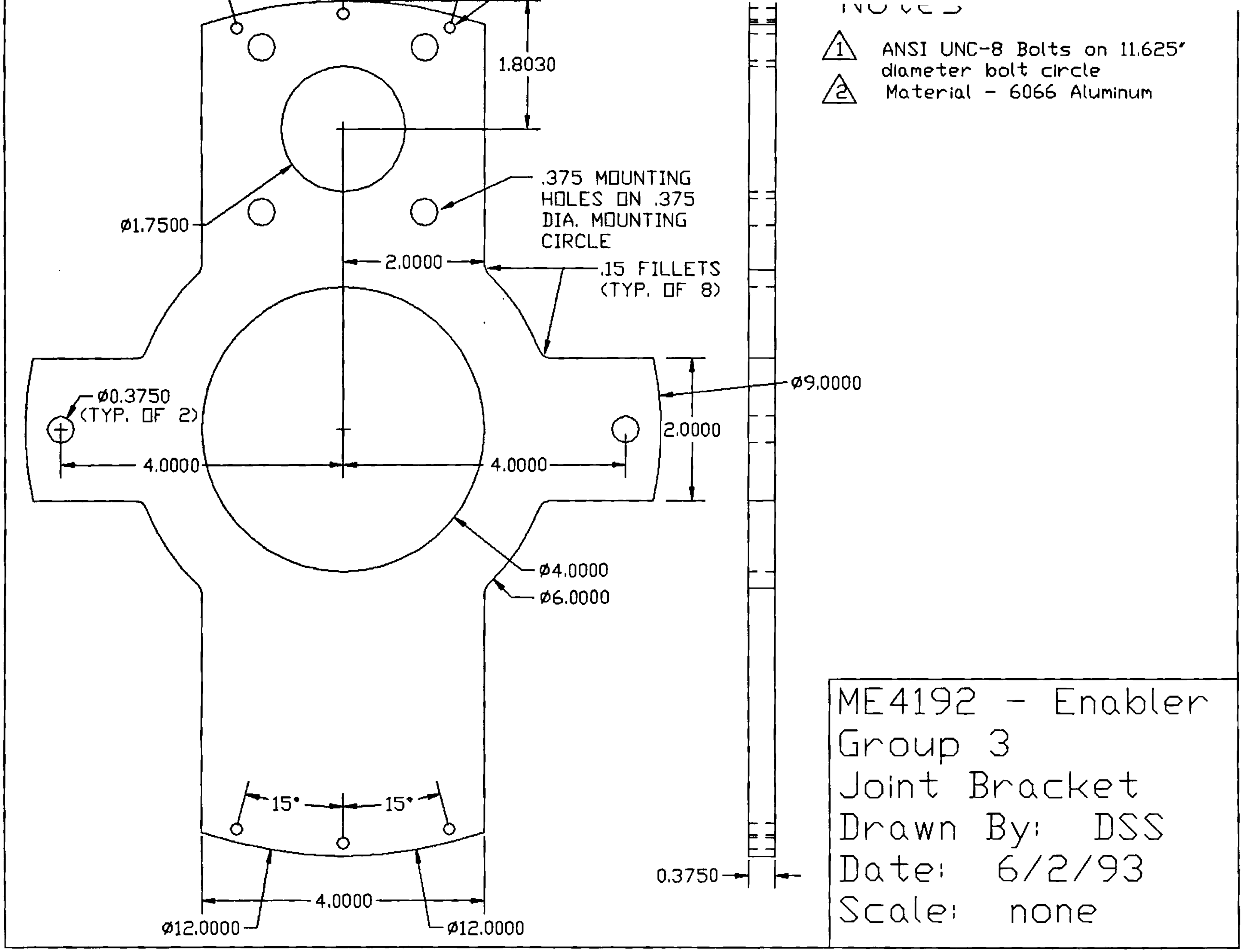


Detail Drawing 1

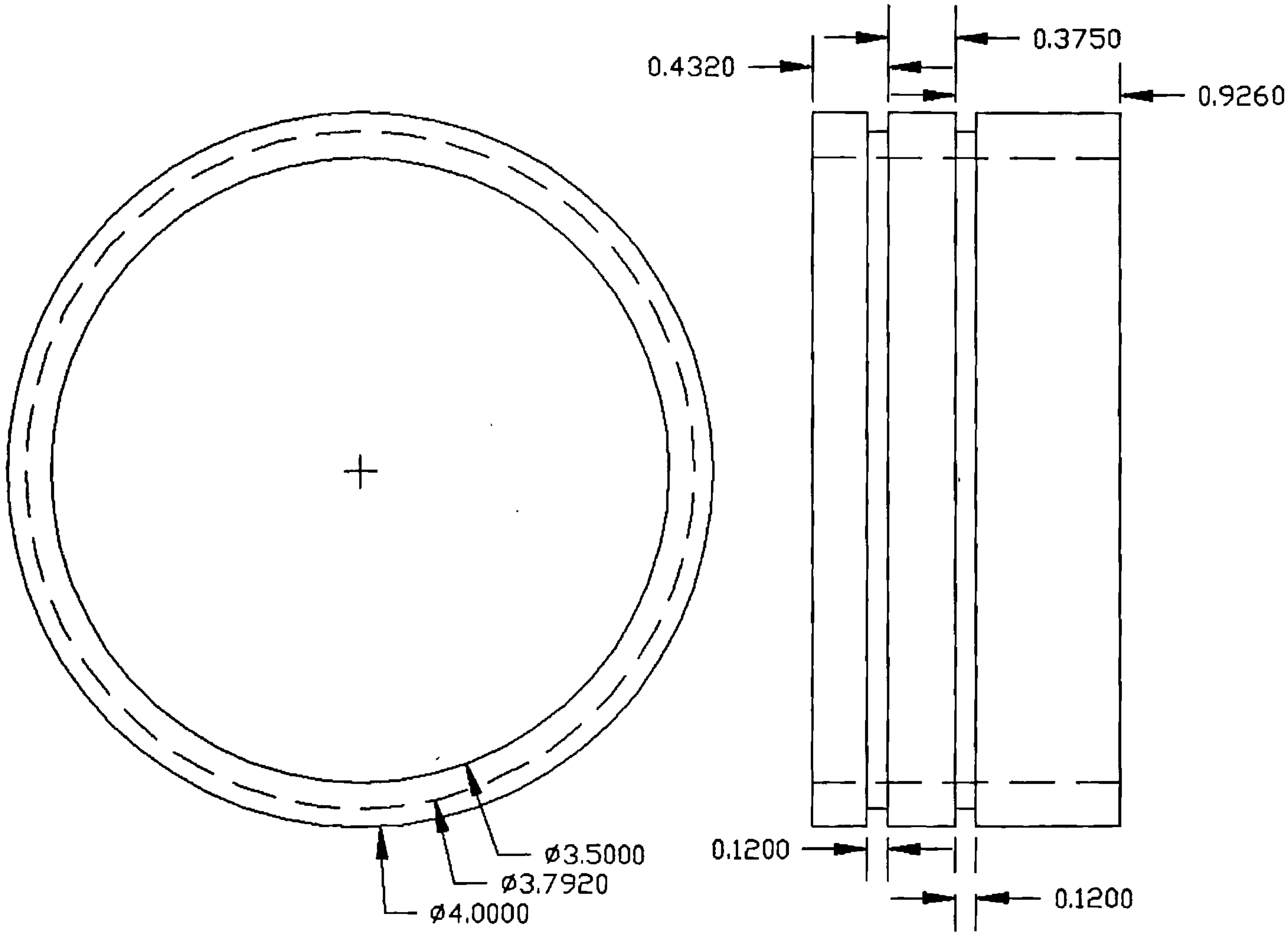
ME4192 - Enabler
Group 3
Name: Detail 1
Drawn By: DSS
Date: 6/2/93
Scale: none



ME4192 - Enabler
Group 3
Joint Assembly
Drawn By: DSS
Date: 6/2/93
Scale: none



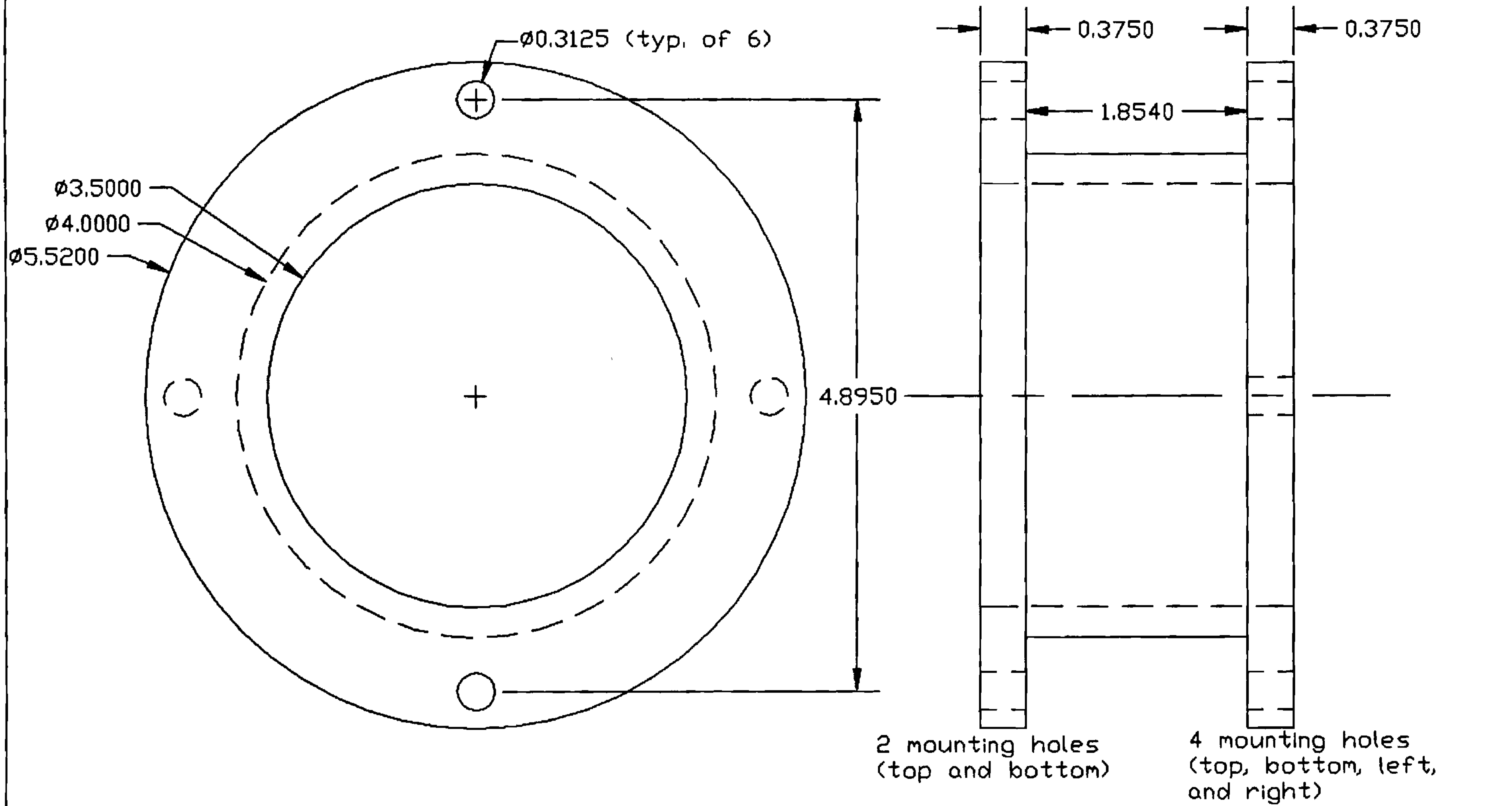
1.8030
Ø1.7500
.375 MOUNTING HOLES ON .375 DIA. MOUNTING CIRCLE
2.0000
.15 FILLETS (TYP. OF 8)
Ø0.3750 (TYP. OF 2)
4.0000
4.0000
Ø9.0000
2.0000
Ø4.0000
Ø6.0000
15°
15°
4.0000
Ø12.0000
Ø12.0000
0.3750
1 ANSI UNC-8 Bolts on 11.625" diameter bolt circle
2 Material - 6066 Aluminum
ME4192 - Enabler
Group 3
Joint Bracket
Drawn By: DSS
Date: 6/2/93
Scale: none



Notes

△1 Pre-Purchased Bronze Bushing Custom Cut for the Enabler

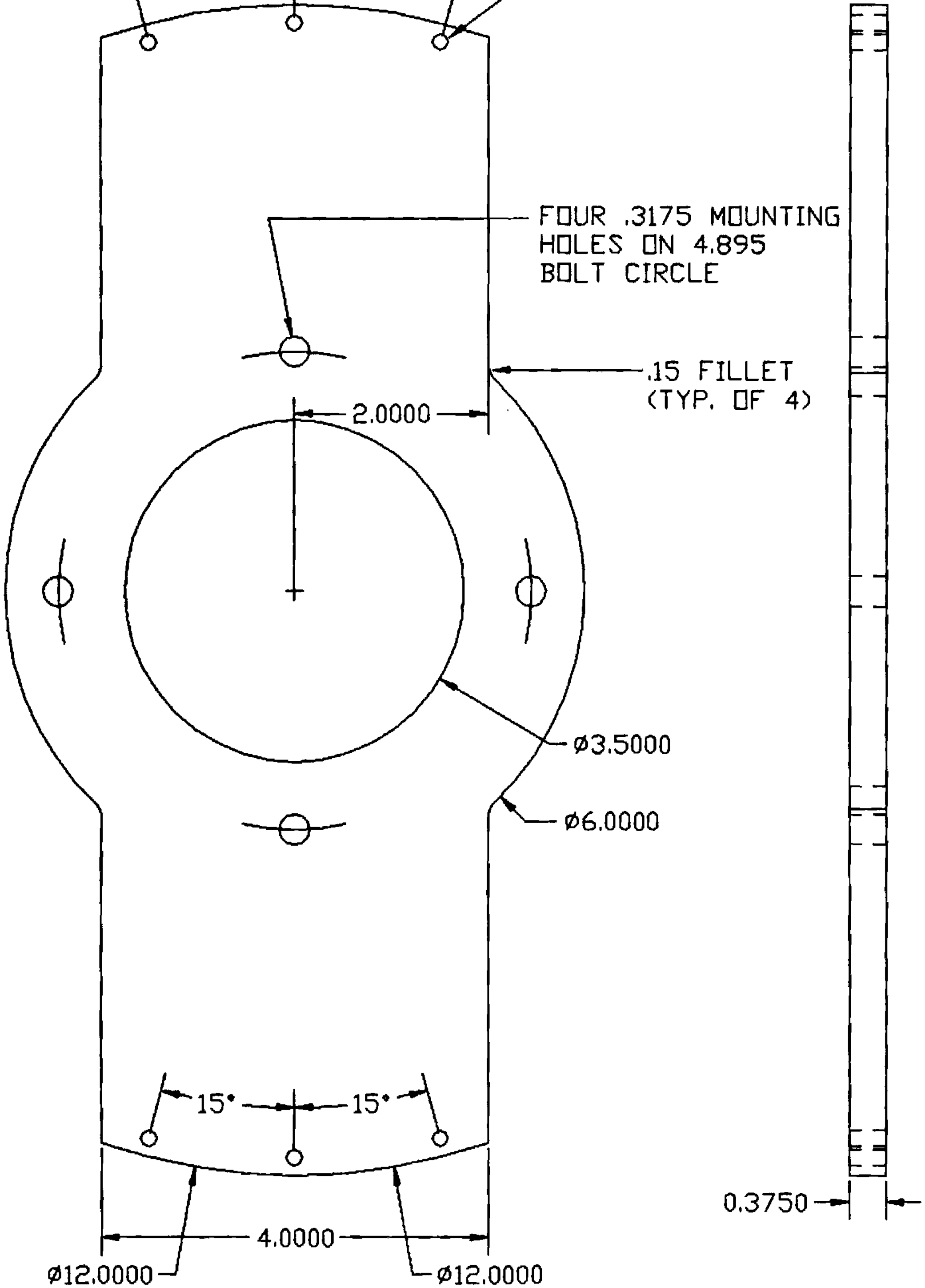
ME4192 - Enabler
Group 3
Bearing Sleeve
Drawn By: DSS
Date: 6/2/93
Scale: none



Notes

1 Material - 6066 Aluminum

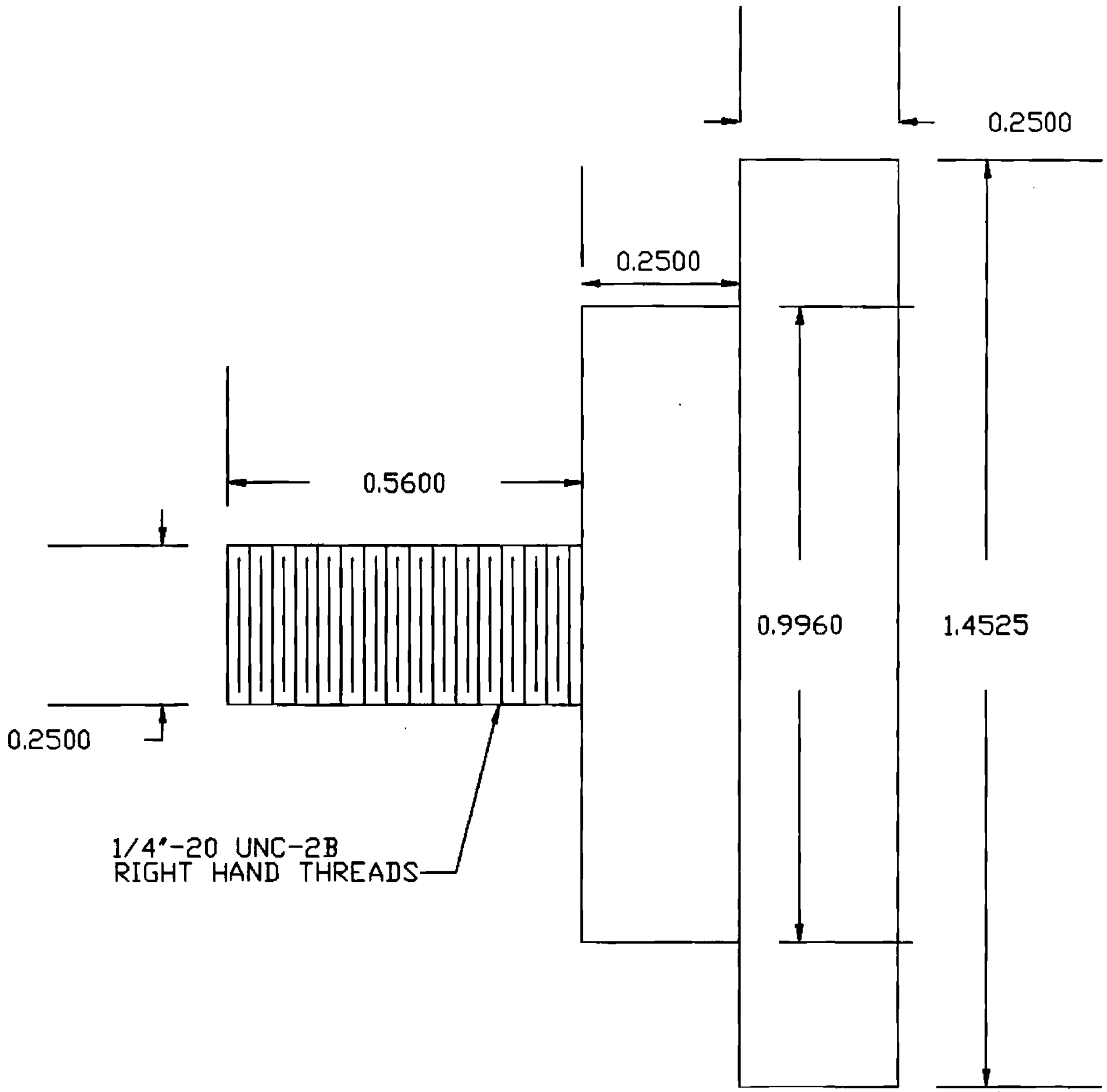
ME4192 - Enabler
Group 3
Name: Jnt Coupler
Drawn By: DSS
Date: 6/2/93
Scale: none



Notes

1. ANSI UNC-8 Bolts on 11.625" diameter bolt circle
2. Material - 6066 Aluminum

ME4192 - Enabler
Group 3
Joint Plate
Drawn By: DSS
Date: 6/2/93
Scale: none



ME4192 - Enabler
Group 3
Sprocket Plug
Drawn By: DSS
Date: 6/2/93
Scale: none

ME 4182
First Lunar Shelter Student Contest Final Report

Professor J.W. Brazell
George W. Woodruff
School of Mechanical Engineering

Design Group One

Derek Garland
Christopher Hill
Clint Kervin
John Leffler
Bob Swarner

1.0 Executive Summary

This report presents a final design for the ME4182 design competition. The purpose of this project is to design, construct, and test a machine capable of off-loading a lunar habitat from a lander, and placing it accurately within the contest area. In addition to this, the machine design must provide radiation shielding for the habitat.

The design which is presented in this report utilizes an aluminum H frame, which consists of 1" x 1/8" square aluminum tubing. Four aluminum legs, which are attached to this frame, provided mounting locations for wheels. These wheels are driven by ladder chain and sprockets which are housed within the aluminum frame. Foreword, reverse, and angular motion is achieved by operating two drive motors in various combinations of speed and direction.

The design also features a loading mechanism which consists of two "forks" which are attached to two arms. These arms are driven by a worm gear which is connected to a permanent magnet DC motor through a series of gear reductions. This loader system not only facilitates the placement of the habitat, but it also provides a means for placing radiation shielding over the vehicle.

The radiation shielding utilized by this design consists of a pre-fabricated aluminum shell which will be placed over the habitat, once it is in position.

The vehicle is powered by three variable voltage divider circuits, which enable independent continuous speed control of all vehicle functions.

This vehicle is valued at $402.29, with a cost for scoring purposes of $370.98. The maximum power which should be available to the vehicle is 25.2 Watts. The actual power used by the vehicle at any given time is only 5.82 Watts, a number which should be used for scoring purposes.

In summary, this design provides a relatively low cost, low weight concept which will allow the efficient transportation of a lunar habitat, as well as radiation shielding for it.

Table of Contents

List of Figures

2.0 Introduction

This design project is based on a contest that is sponsored by the American Society of Civil Engineers. The purpose of the contest is to design, construct, test, and demonstrate a machine that will satisfactorily unload a lunar habitat from a lander, and position it accurately on another portion of the contest area. In addition to these functions, radiation shielding for the habitat must also be provided.

The machine built to accomplish these functions is subject to several rigid constraints which are presented in the contest rules. These constraints will also be discussed in relevant sections of this report. In addition to meeting the constraints, the final design must also address the following issues:

* Minimize cost.
* Minimize weight.
* Minimize execution time.
* Maximize dust containment.
* Maximize system autonomy.
* Maximize simplicity in mechanical design and operation.
* Minimize power requirements.
* Maximize shielding effectiveness.

In order to accomplish this task, while effectively utilizing the knowledge and previous experience, the project is divided into five systems, with a different person responsible for each. These systems are defined as follows:

* Chassis system - basic framework for machine.
* Transportation system - drive mechanism of machine.
* Loading system - mechanism for off loading and placing habitat.
* Control system - responsible for providing adequate control of other systems.
* Radiation shielding system - providing adequate radiation shielding for habitat.

The following sections of this report deal with the individual issues related to each of these systems. The report then concludes with a financial analysis of the project, and a statement of future work.

3.0 Chassis System

3.1 Problem Description

Abstract

The purpose of this project is to design, construct, and test a machine capable of off-loading a lunar habitat from a lander, and placing it accurately within the contest area. In addition to this, the machine design must provide radiation shielding for the habitat.

The design which is presented in this report utilizes an aluminum H frame, which consists of 1" x 1/8" square aluminum tubing. Four aluminum legs, which are attached to this frame, provided mounting locations for wheels. These wheels are driven by ladder chain and sprockets which are housed within the aluminum frame. Foreword, reverse, and angular motion is achieved by operating two drive motors in various combinations of speed and direction.

The design also features a loading mechanism which consists of two "forks" which are attached to two arms. These arms are driven by a worm gear which is connected to a permanent magnet DC motor through a series of gear reductions. This loader system not only facilitates the placement of the habitat, but it alsoprovides a means for placing radiation shielding over the vehicle.

The radiation shielding utilized by this design consists of a pre- fabricated aluminum shell which will be placed over the habitat, once it is in position.

The vehicle is powered by three variable voltage divider circuits, which enable independent continuous speed control of all vehicle functions.

In summary, this design provides a relatively low cost, low weight concept which will allow the efficient transportation of a lunar habitat, as well as radiation shielding for it.

June 2, 1993

Bob Swarner
Box 32820, Ga Tech Station
Atlanta, GA 30332

Mr. James W. Brazell
School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, GA 30332

Dear Mr. Brazell:
Please accept our entry into the spring quarter ME4182 design competition. Our transporter design is based on a simple two motor drive mechanism. In addition to this drive mechanism, our vehicle utilizes an innovative loading mechanism that will effectively locate the habitat, as well as the radiation shielding. Thank you for all of your guidance and advice throughout this project. It has proved invaluable to the success of our team and design. We also welcome any comments that you may have after reading the final report and watching the demonstration.

Sincerely,

Bob Swarner
Design Team I

The need for a single structure to connect the transport and unloading systems was determined from initial studies of the design problem. A single frame was developed to mount all transport and unload systems. Design goals were determined to be:

1) provide a rigid and low mass structure to connect all other sub-systems into one unit.
2) provide a structure that would allow all components to be mounted inside the required space as determined by the contest rules.

Constraints:

1) All components must fit within the design "box" space allowed. Primary limit on frame size was initially given as the cube space of the habitat. This was modified during the design process to the cube of the lander's horizontal dimension, per instructions from the instructor.

2) Frame must not deflect under normal operating loads.

3) Must effectively interface with all other systems.

3.2 Problem Solution:

The entire frame is fabricated from square cross section aluminum of grade 6061-T6. The square cross-section was chosen for its resistance to bending and torque loading along the tube axis. One inch outside width tubing was chosen for all frame components. Consistent size was an advantage in designing joints because most of the fasteners in the model are of the same style. The one inch tubing size is readily available and is relatively low in cost. The basic concept for the final chassis design is an " H" frame. Two side rails are joined at the front and rear of the chassis by cross members. Overhangs at the ends of the side rails provide mounting for the unload sub-system and transport drive. Uprights are connected to the frame side rails at cross member mounting locations. These uprights serve to provide both ride height for ground clearance and constant elevation for removing the habitat from the lander. All frame joints are bolted using bridge plates at each joint. Original design called for welded joints. However; Due to concern over heat deformation in the welding process, the bolt on bridge plates were developed.

The H Frame design was developed using the following concepts:

1) The perimeter frame design allows for a large number of mounting points to be developed around the chassis. This concept also allows for some protection to vital components from collision damage by requiring that all components be placed inside the frame. The frame serves a limit on the overall size of the device by limiting the design "box". The "H" frame design allows for maximum weight spread during load/unload sequences. By providing a wide mounting base tip-over problems were reduced. The load imposed by the load/unload sub-system is transmitted to two points, by using each side rail as a separate beam, the loads on the chassis were simplified (see calculation 3.1)

2) The use of square cross section tubing allows for the development of a very rigid frame (see calculations 3.2). An additional feature of this tubing choice is that it allows the chain drive lines of the transport sub-system to be mounted inside the frame rails and uprights . This prevents dust from damaging these components. The square cross section of the tubing allows for a number of easily indexed and prepared mounting surfaces. This greatly simplified the process of aligning mounting the various sub-systems.

3.3 Feasibility of Solution for Lunar Application:

The limiting factor in the feasibility of this design for real world applications is tip-over. Given the mass of the habitat compared to an initial estimate of real world device mass (see calculation 3.4), it appears that some ballast might be necessary. The following points need to be evaluated:

1) Power sources, either battery or electro-chemical could require substantial mass. This mass could be used to balance the transporter.

2) Much closer tolerances could be developed on a real world device to reduce the moment arm imposed by the unloader/loader device.

3) Contact with the lander could be used to brace the transporter during unloading. Further bracing could be developed to assist in the lowering to the surface phase of the unload.

The major liberty taken in the development of the model is the size of components in the frame. While the overall dimensions of the frame are to scale, items such as bushings, shaft sizes, and the frame tube size are not, due to the need to manufacture the model. Ease of manufacture was deemed a key element of the final model design.

Considerations for future development:

In a real world lunar transporter we would power each wheel independently and provide a control system for all wheel drive. Traction control and steering would be incorporated in this system.

One concept that would greatly improve the design is variable ride height. This would allow for reduced tip over, greater maneuverability over difficult terrain, and a more compact shipping configuration.

4.0 Transportation Subsystem

4.1 PROBLEM STATEMENT:

One of the primary tasks in the competition is to place the habitat in a position quite distant and differing in orientation from its original position on the lunar lander. Because the habitat must be moved approximately seven feet, and rotated about a vertical axis 90 degrees, our group has chosen to load the habitat on a wheeled vehicle, carry it to the new site, orient the habitat, then place the habitat on the surface. To traverse the simulated lunar surface, the vehicle must not strand itself on obstacles such as rocks and pits.

The vehicle should have enough power to climb some rocks, and to get out of holes. The orientation of the vehicle should be completely controllable, preferably without lateral translation if pure rotation is desired by the operator. The relative weight of gear motors is high compared to other vehicle components. It is desired, therefore, to minimize the number of guarantors used. The volume taken up by the transportation subsystem's components should be minimized and positioned to the best advantage of other subsystems.

The goals to be satisficed are adequate ground clearance, sufficient power, good maneuverability, minimized weight, and a high degree of integration with other subsystems. The first goal, ground clearance is constrained by two boundaries: maximum vehicle volume and requirements of the loader subsystem. The requirements of the loader fall with in the maximum allowable volume, therefore the loader requirements are the primary restriction on ground clearance. Five inches of clearance between wheel to surface contact is required by the loading subsystem, therefore this was the bound on ground clearance.

4.2 PROBLEM SOLUTION:

Sufficient power to drive over rocks and out of pits is needed. The availability of several high torque gear motors satisfied this goal. At startle speed, they have a torque of 80-100 oz-inches, which is adequate for our needs.

The limitations of space around the lander and at the habitat site require that our vehicle be easily maneuvered. Steerable wheels provide for rotation and translation of the vehicle, but require translation if rotation is to occur, and the center of rotation would be the non steerable wheels. Our group has chosen to allow the right and left side wheels to be driven at different rates, allowing the operator choices between pure rotation to pure translation, with the center of the vehicle as the rotational center. The maximum fore/aft speed should be at least four inches per second, which will allow the distance to be traversed in approximately twenty seconds. The minimum speed should allow the operator adequate positioning control. A speed of approximately .5 inches per second should suffice. The motors selected provide a speed range which, when the shaft speed is reduced by a factor of two, is perfect in meeting these goals.

To minimize weight, our group decided to use to drive motors. Using chain and sprockets allowed us to use one motor per side, and place the motor at the five inch chassis height where the motor will not be subject to ground clearance problems. See Figure 4.1 for the side view layout of this subsystem, and Figure 4.3 for the spur and pinion layout. The chain and sprockets were chosen because of their size and performance characteristics. All shafting is 3/16 inches in diameter. This allowed uniformity of gear and sprocket choice, and facilitated use of some components which were readily available. This diameter shafting will be of adequate strength to prevent critical deformation.

The use of two motors with chain drive allowed a variety of component placement options. Our group has chosen to run the chains inside the chassis, and place the motors aft for balance considerations. The chassis section of this report contains balance calculations and results.

When the drive motors were mounted, we discovered a redundancy in the design. Please refer to Figure 4.1 for the detailed design, and Figure 4.2 for the revised design. The original specifications provided for three standoffs for the motor mounts. When mounted with two standoffs on the long chassis axis, the mount would not deflect under applied test loads. The mount was redesigned and machined so as to have the two mounting holes in the described locations, and excess material was removed. The mount required minimal adjustment to bring the mounted parts within operating tolerances.

The assembled system operates smoothly, with a minimal amount of chain vibration. The load presented to the motor under free-wheeling conditions is approximately 40 oz-inches. The motor provides adequate torque to this system.

4.3 FEASIBILITY OF SOLUTION FOR ACTUAL LUNAR APPLICATION

The mechanical principles involved in our drive subsystem are viable on the moon. The wheeled drive has been used on former lunar missions. The transmission of power of power to the wheels would not be a direct scale-up of the chain drive which we utilized. Our chain drive was convenient for the scale at which we were modeling. In all likelihood, the full scale lander would use one or two motors with shaft drive and some suspension flexibility. The chain type transmission could feasibly be used, but due to vibrational and weight limitations, shafts would be more suitable.

Components such as bearings and shafts could be chosen to prevent dust penetration, and a channel or tubular design such as we have built could be used for the body of a full scale device. This provides many options for later additions. Motors so mounted could later be removed and used for other tasks. The chassis members and drive components could be designed so that the rover could later be converted to a crane or similar mechanism.

5.0 Loader System

5.1

PROBLEM STATEMENT:

The loader subsystem is required to lift the habitat off of the lander, place the habitat on the transport vehicle, and unload it at the specified site. The loader subsystem must also be capable of moving the radiation shielding into position if necessary.

CONSTRAINTS: There are several constraints which affect the design of the habitat loader. They include the following:

- the orientation of the habitat on the lander and the position of the lander treads
- the vertical distance to be traversed by the habitat
- the weight, dimensions and center of gravity position of the habitat -
- the weight, dimensions and center of gravity position of the vehicle and radiation shielding

5.2 GOALS: Several goals are specified in the contest (and grade) scoring. These include:

- minimize cost and weight
- resistance to environmental fouling
- operational and mechanical simplicity
- minimum execution time

5.3 PROBLEM SOLUTION:

DESCRIPTION OF SYSTEM: The system we have chosen is effectively a four-bar linkage which utilizes two parallel main beams, two parallel 'forks' attached to the beam ends, and a chain linkage to keep the forks parallel to the ground. The parallel main beams are driven simultaneously by a gear motor-driven shaft operated by a worm gear.The worm gear is used to provide some self-locking capability and to reduce the torque requirements of the motor. The forks engage 4 pins that are on the sides of the habitat. The pins are located at the fore and aft limits of the habitat's longitudinal center of gravity placement.

EXPLANATION OF DESIGN DECISIONS: We have evaluated numerous designs for this subsystem. Below is a brief description of these systems and the reasons for discarding some of the designs.

1) The first design assumed an adjustable chassis height as part of the accommodation for the vertical distance to be traveled. Approximately half the height to be traversed (total height = 11.65 inches) was taken up by the ride height travel, and the other half with a mechanism based on a standard forklift. The design incorporated a motor on the forklift saddle to flip the fork over. This would be used to flip the habitat onto its top and into a saddle on the vehicle. This was discarded due to the mechanical complexity of the forklift and flip systems, and because we thought the next system would work better.

2) The second design incorporated two main parallel beams, much as we are going to use. These beams would incorporate a u-shaped saddle on the end that would hook onto a pair of pins located at the habitat center of gravity. This wouldn't necessarily have required the variable ride height, although we still had planned on using it at that point. This design was discarded due to a re-reading of the contest rules which specified that the center of gravity position of the habitat was not exactly known.

3) The third design used a standard application of a four-bar linkage. The linkage was used to keep a fork parallel to the ground, and this fork contacted 4 pins on the sides of the habitat just as we are now using. The reason for discarding this design was that we could

3) The third design used a standard application of a four-bar linkage. The linkage was used to keep a fork parallel to the ground, and this fork contacted 4 pins on the sides of the habitat just as we are now using. The reason for discarding this design was that we could only achieve a linkage transmission angle of twelve degrees at the limits of travel. Forty degrees is usually the preferred minimum. At this point, we had decided to discard the variable ride height system due to its mechanical complexity. Using it may have improved our transmission angles slightly, but it wouldn't have made a significant difference.

4) Our fourth design is the one we have decided on. The chain linkage allows the benefits of the four-bar linkage without the transmission angle worries. The drive system is such that the loads can be easily tied into major structure. The fork system allows for varied position of the habitat center of gravity, and should be relatively easy to align with the habitat when unloading. The benefits of the worm drive have been mentioned, and the components should be relatively easy to make. The design uses tapered, 0.188" thick 6061-T6 aluminum main beams which have an I-section. See Drawing 5.2 in the Section 5 appendix. The stress analysis for these beams resulted in a safety factor of approximately 37 for simple bending about the major (X) axis. See the calculations on pages 1-4 in the appendix. A tapered design was used in order to efficiently distribute the loading stresses throughout the beam length. A stress analysis was also performed for the worst-case loading that would be seen about the beam's minor (Y) axis. The results of this indicate a safety factor of approximately 6. The calculations for these values can also be found on pages 1-4 of the Section 5 appendix. The beams are driven at the larger end by a 1/4" shaft driven by a gear motor and 20:1 worm reduction. The shaft ends are supported in double shear by plastic bushings. The loading fork is attached to the small end of the beam by a 3/16" shaft which is pinned to the fork and passes through a ball bearing pressed into the end of the beam. The #21 pitch ladder chain and chain sprocket is attached to this shaft, and by fastening the chassis ends of the chain to the chassis, the fork is kept parallel to the ground. Each fork is made from 0.125 6061-T6 aluminum. See Drawing 5-3 in the Section 5 appendix. The forks pick up on the pins mounted on the side of the habitat; the rear pins are mounted in such a way as to provide lateral restraint. The stress calculations for the forks resulted in safety factors of 50 and 7 in major (x) and

minor (y) axis bending, respectively. See Section 5 appendix page 5 for the calculations of these factors. Buckling failure is not a factor due to the design geometry. The calculated maximum mechanical power required for this subsystem is 1.17E-6 horsepower; the calculations are in the Section 5 appendix, page 6.

5.4 FEASIBILITY OF SOLUTION FOR LUNAR APPLICATION

5.4.1 PHYSICAL FEASIBILITY: The design we have chosen should function in the same manner whether on the moon or on the earth. The design is resistant to the environment, does not require an atmosphere, and is somewhat fail-safe in the event of power loss. It requires no setup to speak of, little maintenance, and
should be able to be operated by a space-suited person.

5.4.2 ACTUAL SIZE REQUIREMENTS: Wherever possible, standard structural shapes have been used; this should allow the easy scaling-up of structure dimensions. The full-size motor required for this application would probably be smaller proportionally than the one we are using. Similarly, the chain linkage may be larger than it would need to be if scaled up.

ASSUMPTIONS MADE FOR MODELING: In this design, we are assuming that the loading and unloading of the habitat and radiation shielding will be done with the vehicle stationary. Additionally, we have chosen motors which will operate the system slowly, so dynamic complications should be minimal. We are assuming that the lander and vehicle will be on a fairly flat surface when in operation

6.0 Radiation Shielding Subsystem
6.1 Radiation Shielding Subsystem Goals

The radiation shielding must protect habitat and occupants from cosmic radiation and mild solar flares. Additionally, the shielding must be of minimum cost and be able to be easily transported and/or set up by the vehicle.

6.2 Radiation Shielding Subsystem Constraints

The radiation shielding method will follow one of two possible scenarios. It must either be transported to the moon by a separate lunar lander or it must arrive on the same lander used by the vehicle. If the latter case is chosen, then it must not adversely affect the operation of the vehicle or obstruct the handling of the habitat. In addition to this it must also conform to the following:

The shielding must fit on a habitat-sized lunar lander.

It must provide radiation protection equal to or greater than 15 g/cm^2 of water.

The shielding must completely cover the habitat.

The shielding must allow access to the habitat for repairs and scheduled maintenance.

6.3 Description of System

The system used to provide radiation shielding for the habitat consists of a half-cylindrical structure that is pre-fabricated on earth and then transported to the moon (figure 6.1). This structure allows a minimum of 2 inches on any side of the habitat for maintenance to the habitat. The structure is made entirely of aluminum which is used as the shielding agent as well as the structural support. The ends of the shield are folded and epoxyed so as to provide total shielding from radiation. The dimensions and specifications of the shield are as follows:

Height:	9.625 in	(refer to figure 6.2 for dimension placement)
Width:	11.625 in	
Length:	20.124 in	
Radius of top:	5.8125 in	
Thickness:	0.087 in	
Weight:	5.33 lb	(refer to calc. 6.1 for weight calc.)

These dimensions easily fit within the lunar lander constraints for transplant to the moon. The shielding is assumed to have been delivered to the moon on either the same lander as the vehicle or a similar lander and has been off loaded close to the site at which the habitat is to be located. The vehicle would off load the habitat and deliver it to the proposed site and then get the radiation shielding and cover the habitat.

6.4 Explanation of Design Decisions

To construct a radiation shield, the contents of the radiation must be known. Cosmic radiation is primarily (85%-90%) made up of protons, the remaining particles being alpha particles and higher atomic number particles. However, the primary concern in cosmic radiation is not the particles but the gamma rays. Alpha particles cannot penetrate human skin and protons take less material to be attenuated than the gamma radiation, as shown later. For these reasons the cosmic radiation is modelled as gamma radiation.

The requirement is that the shielding has to be as effective as 15 g/cm^2 of water, so this is used as the basis for determining the thickness of the shielding. To simplify the selection of the material and thickness of that material a simple radiation intensity transmissibility equation is used. This way the thickness of a material can be found by comparing the effectiveness of the material at a given energy level using that material's linear attenuation coefficient. Threshold energy of 10 Mev was chosen as the constant energy because this is where the maximum thickness of material is required as seen in figure 6.3.

Aluminum was chosen as the radiation shielding agent due to its previous use in radiation shielding and its good structural qualities. It is an inexpensive and readily available material that is proven to hold up well in space due to the many missions it has already been used in. Using these conditions and the transmissibility ratio as the comparison between water and aluminum, the needed thickness of the aluminum is 0.087 inches (calculations 6.2). The thickness of material for attenuation of gamma rays being greater than that of protons can now be shown in calculation 6.3 in the appendix.

6.5 Feasibility of Solution for Actual Lunar Application

This structure is physically feasible due to the use of physical verification of all radiation shielding requirements. No structural problems are foreseen in using aluminum as the structural material due to its common usage in building structures on earth. Its sufficient thickness and the reduced gravity on the moon work to reenforce these assumptions. Aluminum is currently and has in the past been used in radiation shielding both on earth and in space. This backs up the decision to use this material for our radiation shielding. Aluminum is readily available and easy to work with, therefore the prefabrication of the structure should be of minimal effort. In modelling the shielding structure, ribbed aluminum sheet is used as a modelling liberty because of its cost and ease of construction.

Its transportation to the moon should be as easy as transporting the vehicle due to the shielding easily fitting within the maximum allowable dimensions of payload on the lunar lander. The shield even allows for additional payload (ie. habitat) to be transported to the moon with it. The shield can easily be lifted over the habitat and into place with the vehicle due to its simple shape and rigid structure. The shielding's prefabrication on earth allows for minimal setup time on the moon so that a habitat can be installed into place in the shortest time possible.

7.0 Control System

7.1 Problem Statement

The primary purpose of the control system is to provide the power necessary to adequately position all parts of the machine. In addition to this primary requirement, the control system must also meet constraints which are set forth in the contest rules. The constraints applicable to the control system are listed as follows:

- All vehicle systems must be tele-robotically controlled.
- No human intervention is allowed inside the contest box.
- Power and communications cables may not mechanically enhance the construction process.
- Vehicle must operate from 115 V single phase power and require no more than 20 amps.

The design may ignore the final constraint, provided the design team furnishes an alternative power source. In addition to the constraints posed by the contest rules, the control system should ideally consider all of the following goals:

- Minimize cost.
- Require minimal operator training.
- Be easily portable.
- Require minimal maintenance.
- Be as simple as possible to construct.
- Use standard readily available components.

7.2 Problem Solution

7.2.1 Design Decisions

Conceptual Design

The first step in solving the design problem presented above was to decide between open loop and closed loop control. It was decided that the added complexity of a closed loop control system would be necessary to allow accurate vehicle positioning on all types of terrain. A block diagram of the desired control function is included in the appendix as figure 7.1.

Once closed loop control was chosen, the degree of system automation had to be decided on. Since the ideal control system is a totally autonomous one, this concept was explored thoroughly. After careful consideration, it was determined that a fully autonomous system is not practical for this application. The main reason for this is that detailed data concerning the contest area terrain was not available during the design stages. This lack of data would have required the vehicle to incorporate many sensors in order to perform its duties efficiently and to prevent catastrophic failure. This factor, coupled with financial constraints and the fact that this vehicle will only be used one time, made a fully autonomous system impractical for this application.

Once it was decided that human intervention would be required, the question of radio control vs. hard wiring arose. It was decided that since direct wiring was permitted by the contest rules it would be more suited for this application. Reasons for this include the threat of radio interference, and the lower costs associated with the components required for a hard wired system.

Another conceptual design decision was whether to use a commercially available motor speed controller, or custom built. Due to financial considerations, a custom built controller was opted for.

Several concepts were explored for the actual control circuit. The first of these, which is the simplest, involves wiring a rheostat in series with each motor. This concept, although it is the simplest and would produce satisfactory results, was not chosen due to the relatively high cost of the required hardware.

The second concept for the motor control circuit utilizes a variable transistorized voltage divider circuit. This circuit provides a reliable means of motor speed control at a relatively low cost. The main disadvantage to this concept is the complexity involved in its design and construction.

Detail Design

The main decisions involved in the control system's detail design involve the selection of components. A detailed analysis of this circuit is included in the appendix. The actual optimum values of the components were determined experimentally, using the calculated values as a starting point. Once the circuit design was completed, an enclosure was selected and the circuit boards were laid out. Although the circuit board layout is not optimum, it is acceptable because it is not a circuit designed for mass production, and the time required to optimize the layout could not be justified.

A much more important layout decision involved the control panel. At first it was desired to have the drive motor speed regulating knobs mounted on horizontal shafts. This was desired to make the vehicle drive control more suited to human operation. This issue was finally abandoned because the knob diameter required to avoid interference problems between the enclosure and the potentiometer was so large, that it would have required an unacceptably large human motion for a small control action. A drawing of the final control box layout is included in the appendix as figure 7.2.

7.3 SYSTEM DESCRIPTION

The final design of the control system involves three separate voltage divider circuits; each one controlling a separate permanent magnet DC motor. A schematic of these circuits is included in the appendix as figure 7.3. These circuits control the individual motor speeds by regulating the voltage across their terminals. All motors used on the vehicle have the same armature windings and are rated for a nominal 12 V DC.

The regulation of the motor voltage is accomplished by limiting the amount of current allowed to flow into the base of the transistors (N1, N2, and N3). This is accomplished by adjusting the respective potentiometer (R2, R5, and R8). As the resistance of the potentiometer decrease, more current is allowed to flow into the transistors' bases. These base currents are amplified and allowed to flow into the collectors, which are connected to the motors. The fixed resistors in the circuit (R1, R4, and R7) serve to limit the maximum amount of current available to the transistors' bases. The remaining potentiometer (R3, R6, and R9) allow an adjustment of the control knobs' sensitivity by varying the voltage drop across the base emitter junction of the transistors.

Directional control of the motors is achieved by three double pull - double throw toggle switches; one for each circuit. These switches serve to reverse the polarity of the voltage sent to the motors, which will reverse the direction of the motor travel.

7.4 SYSTEM OPERATION

The control system is operated by setting the appropriate toggle switch to one of three positions. These positions are foreword (switch up), reverse (switch down), and off (switch centered). Once the toggle switch is set in the correct position, the control knob may be rotated to regulate the motor speed. A drawing of the control panel is included in the appendix as figure 7.4.

Directional control of the vehicle is achieved by varying the relative speeds and/or directions of the two drive motors. Depending on the relative speeds of the two motors, The vehicle may travel straight, turn gradually, or turn very sharply. In addition to the position of the vehicle, the control system also regulates the position of the loader arms. Care must be exercised by the operator during this maneuver because the loader arms are not equipped with limit switches. Allowing the loader arms to travel too far in the reverse direction may result in damage to the vehicle. Given proper operator care, this phenomenon should not occur because the loader arms travel very slowly, so ample time is always available to de-energize the system before damage occurs.

7.5 FEASIBILITY OF SOLUTION FOR ACTUAL LUNAR APPLICATIONS

The control system for an actual lunar application could operate on the same principle as the one designed for this model. However, this system would have to be capable of handling much higher power levels. This increased load would result in much more complex circuitry. Due to the increased circuit complexity, the control unit is not built to a scale of 1:24. In addition to this, a lunar vehicle using this form of control must have video cameras on board so that a remotely located operator could monitor vehicle motion, as well as the vehicle's surroundings.

8.0 COST ANALYSIS

This section describes the total cost breakdown for the project. These figures are based on actual costs to the team, as well as fair market values of all used components. Metal costs are based on the actual amount of material used in the vehicle. This figure does not include extra material which was included in standard stock lengths. However, the metal cost figure includes all waste generated by the fabrication process. A detailed breakdown of the project cost is included in the appendix.
Documentation to support these claims is included in the final project notebook.

The project cost is divided into the following categories:

Drive components	\$355.24
Chassis components	\$ 14.74
Shielding components	\$ 1.00
Electronic components	\$ 31.51

This produces a total cost of \$402.29. For scoring purposes, this cost is reduced to \$370.98. This reduction is necessary because the contest rules state that anything not in the contest box does not count in weight or cost computations.

9.0 Power Requirements

The power requirements for the final design can be divided into three categories: maximum rated power, minimum available power, and actual maximum power usage.

The first category, maximum rated power, is defined as the sum of the rated input voltage times the rated stall current for each motor. This vehicle utilizes three permanent magnet DC motors, each with identical armature windings. These windings are rated for an input voltage of 12V and a maximum stall current of 700 mA. These ratings combine to yield a maximum rated power of 25.2 Watts. Since inefficiency in the control unit is negligible compared to the power ratings of the motors, the power required to operate the control circuitry is not included in this figure.

The second category, minimum available power, is defined as the actual amount of power that the vehicle requires to sustain operation of all systems under full load. This is an experimentally determined number which is obtained from the following data. The drive system requires a maximum of 490 mA to sustain a turn at full operating speed. In addition to the drive system, the loader system requires 240 mA, when under full load. When all systems are operating simultaneously, under maximum load, the current required is 720 mA. Since all of these readings were taken with an input voltage of 12 VDC, the minimum required power is equal to 8.64 Watts. Since all measurements were taken at the input to the control system, this number represents the minimum amount of power which must be available to the vehicle at all times.

The third category, actual maximum power usage, is defined as the maximum amount of power that the vehicle requires under normal operating conditions. This value can be obtained from the measurements cited above. Since the drive system and the loader system are not designed to operate simultaneously, the system requiring the most power will represent the actual power requirements. Since the drive system requires 490 mA under full load, the actual maximum power usage is 5.88 Watts.

The actual maximum power usage of 5.88 Watts should be used for scoring purposes, because it represents the maximum power that the vehicle requires when operated as intended. Even though the previous number should be used for scoring, the actual power available to the vehicle should be a minimum of 25.2 Watts. This amount should be available to account for any unexpected circumstances which may arise.

10.0 Conclusion

The design team has succeeded in producing a low weight, relatively low cost, model for the competition. This design, although functional, could be improved if time and funding were available.

A major improvement which could be made in the existing design is further weight reduction. For this model 1/8" thick aluminum box beam is utilized. The primary reason for this is to allow holes to be tapped directly into the beam. Given enough time and money, the beams could be machined to an appropriate thickness, in most places, while still maintaining adequate thickness for threaded holes, where necessary.

APPENDIX

REFERENCES

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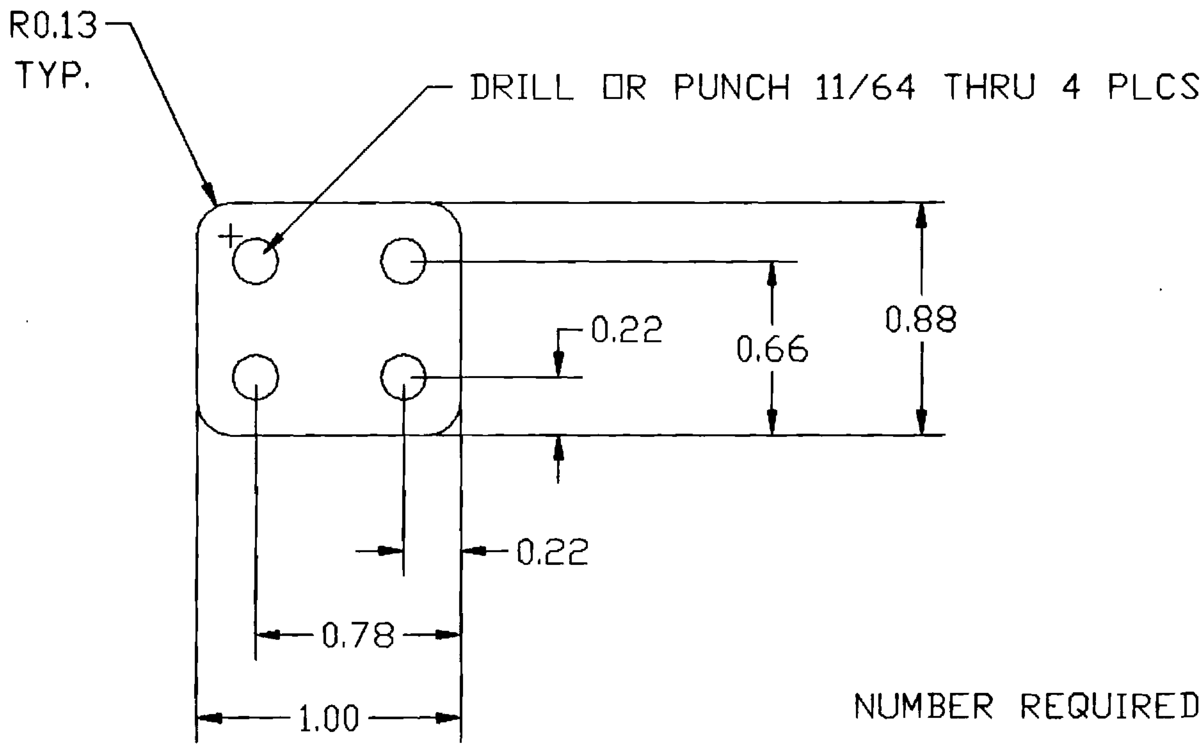
5. *Radiation Protection at Diablo Canyon Power Plant , General Employee Training*, 1992, Internally Published

6. J.R. Boutwell, Engineer - Nuclear Group, Reed National Air Products Group, Holland, Ohio

7. Dr. Carrom, Professor of Nuclear Engineering, Georgia Institute of Technology, Atlanta, Georgia

8. Dr. Paul Simony, Professor of Physics, Jacksonville University, Jacksonville, Florida

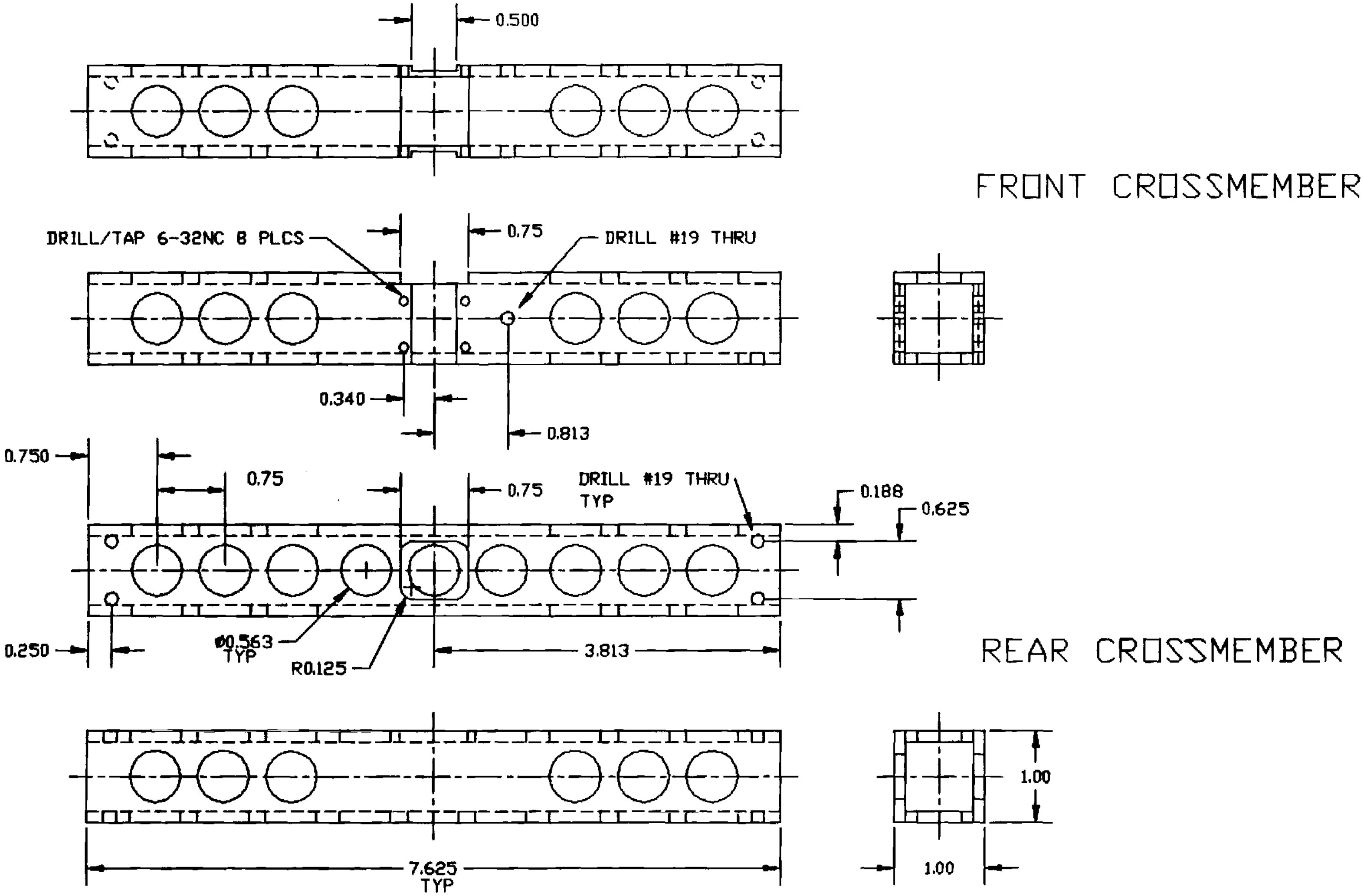


NUMBER REQUIRED: 16

Tolerances			
±1/64	.xx .01 .xxx .003	±1/2°	
Material			
ALUM 6061/2024			

Date	5/18/93
Draft	J. LEFFLER
Checked	
Appvd	

TUBE CONNECTOR
SANDWICH PLATE



PARTS SYMMETRIC
SIDE TO SIDE

Tolerances				
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Material			Checked	
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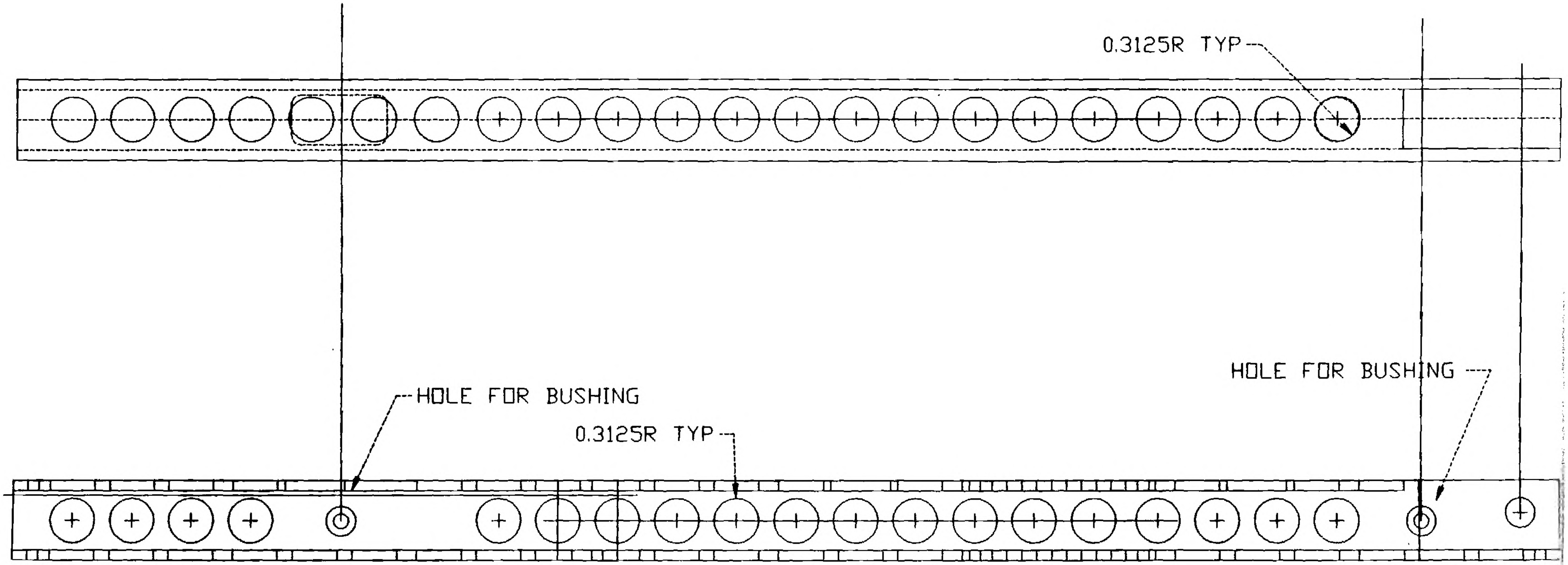
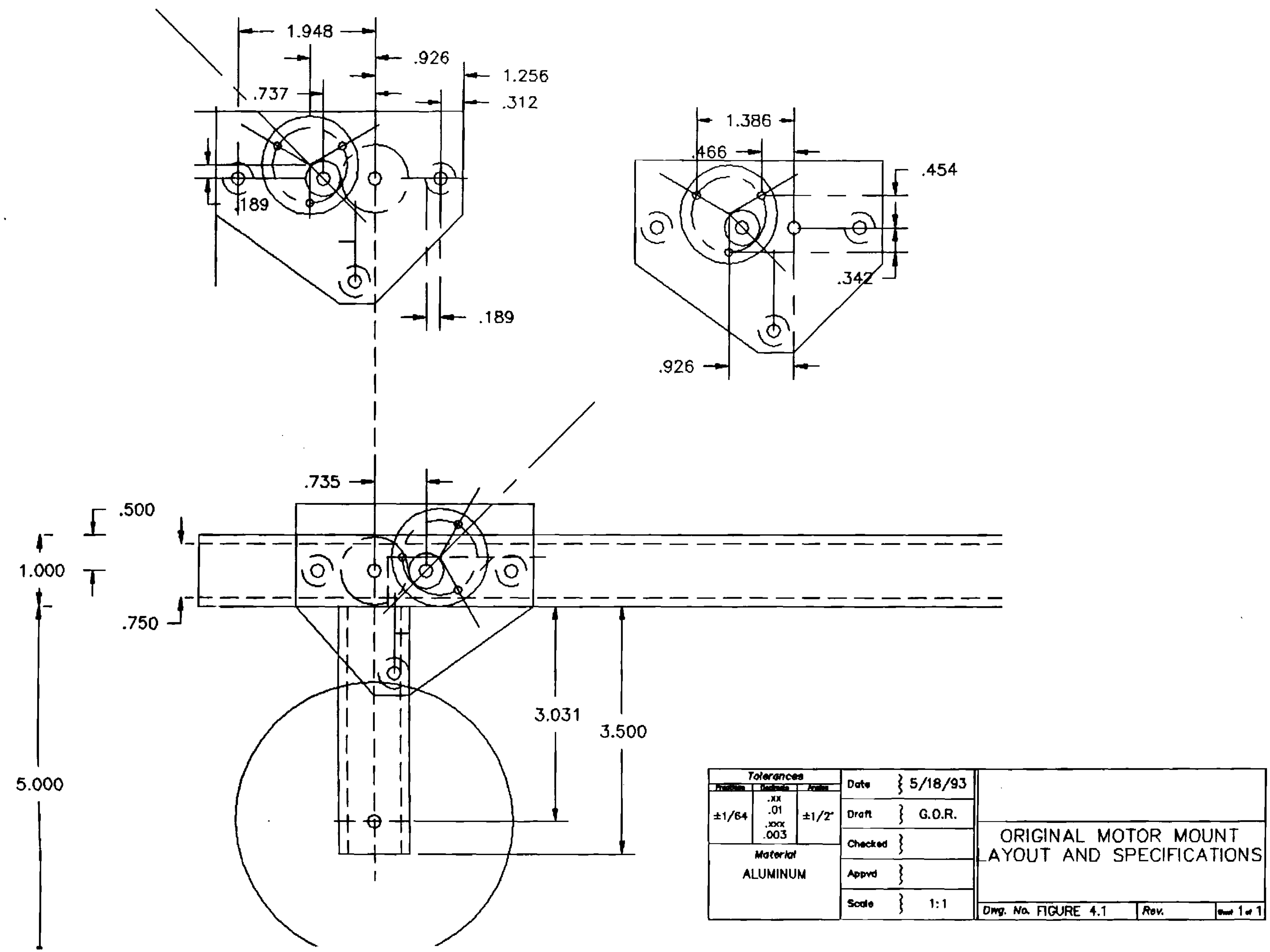
			Scale	FULL

FRONT AND REAR
CROSSMEMBER TUBES
CHASSIS SUBSYSTEM



Tolerances			Date	5/22/93	
Fractions	Decimals	Angles	Draft	CHRISTOPHER HILL	
±1/64	.xx .01 .xxx .003	±1/2°	Checked		CHASSIS LONGERON TUBE
Material			Appvd	awH	
ALUMINUM			Scale	1:1	
					Dwg. No. FIGURE 3.3 — Rev. 2 — Sheet 1 of 1


1.948
.926
1.256
.737
.312
.189
.189
1.386
.466
.454
.342
.926
.735
.500
1.000
.750
3.031
3.500
5.000
Tolerances
Fractions
Decimals
Angles
±1/64
.xx
.01
.xxx
.003
±1/2°
Material
ALUMINUM
Date 5/18/93
Draft G.O.R.
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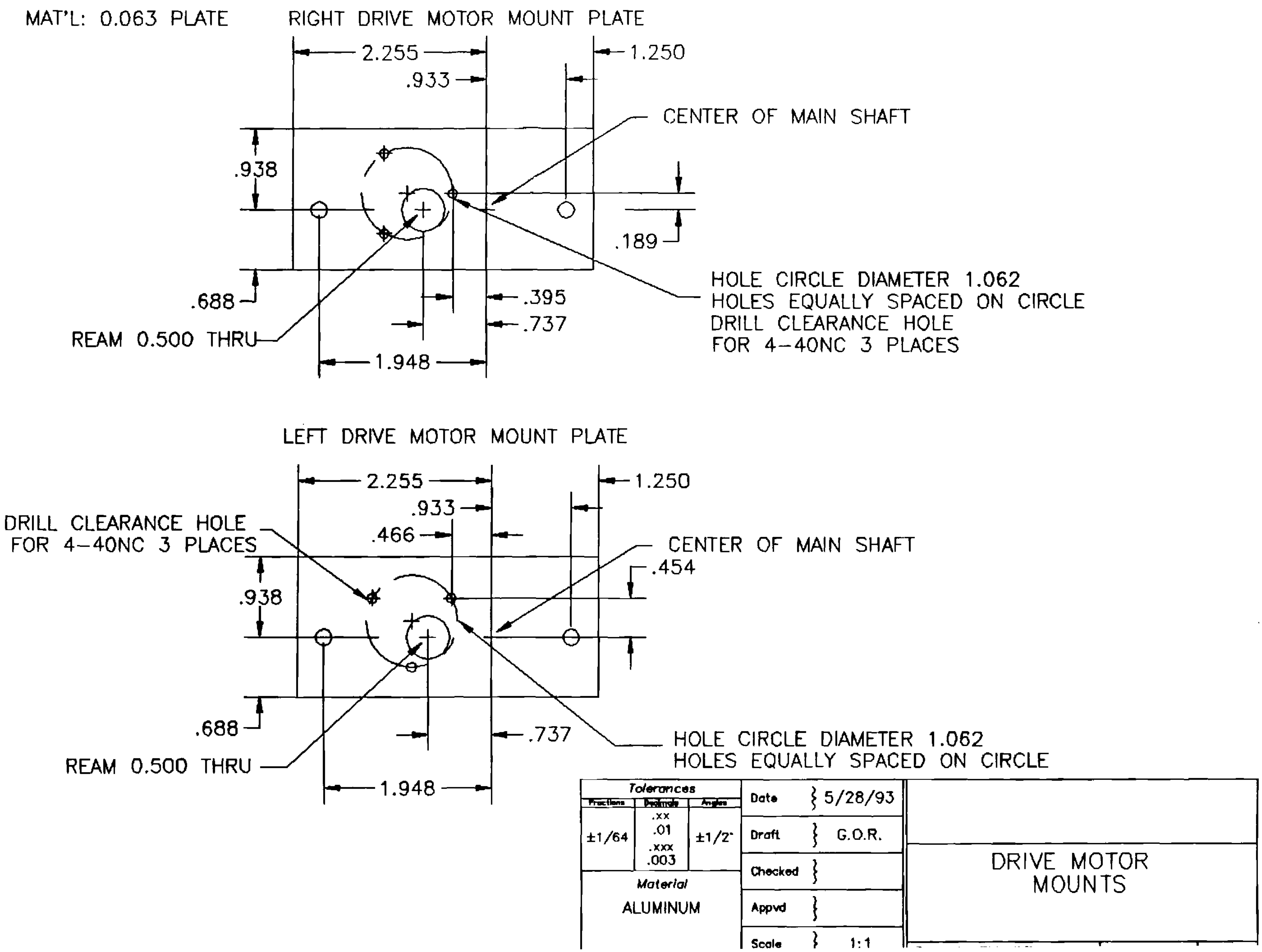
Appvd
Scale 1:1
ORIGINAL MOTOR MOUNT LAYOUT AND SPECIFICATIONS
Dwg. No. FIGURE 4.1
Rev.
Sheet 1 of 1


MAT'L: 0.063 PLATE
RIGHT DRIVE MOTOR MOUNT PLATE
2.255
.933
1.250
CENTER OF MAIN SHAFT
.938
.189
HOLE CIRCLE DIAMETER 1.062
HOLES EQUALLY SPACED ON CIRCLE
DRILL CLEARANCE HOLE
FOR 4-40NC 3 PLACES
.688
.395
.737
REAM 0.500 THRU
1.948
LEFT DRIVE MOTOR MOUNT PLATE
2.255
1.250
.933
DRILL CLEARANCE HOLE
FOR 4-40NC 3 PLACES
.466
CENTER OF MAIN SHAFT
.454
.938
.688
.737
REAM 0.500 THRU
HOLE CIRCLE DIAMETER 1.062
HOLES EQUALLY SPACED ON CIRCLE
1.948
Tolerances
Fractions
Decimals
Angles
±1/64
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.xxx .003
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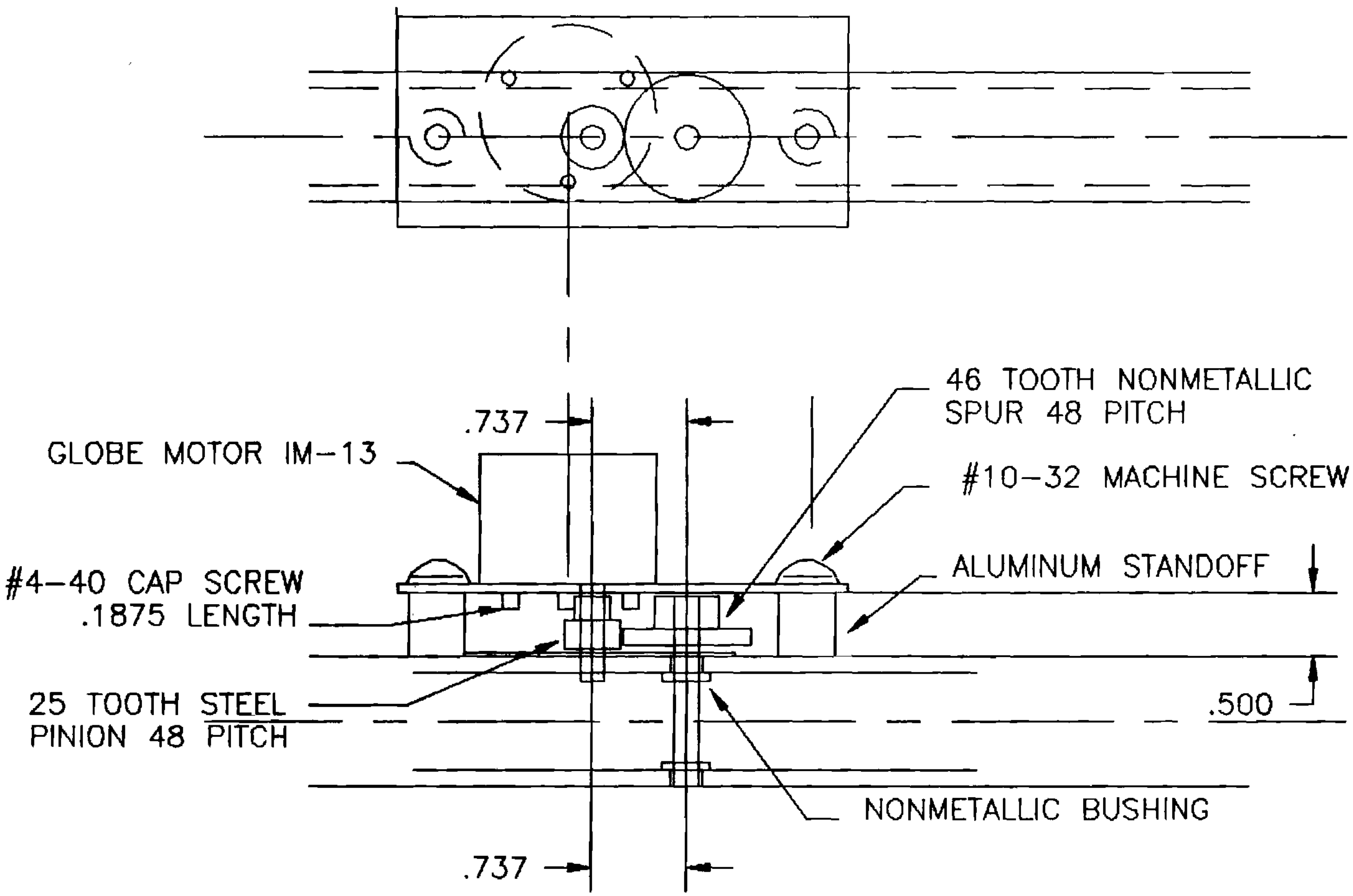
DRIVE MOTOR
MOUNTS



Tolerances					
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Material			Checked		
ALUMINUM			Appvd		
			Scale	1:1	

APPENDIX

SECTION 5

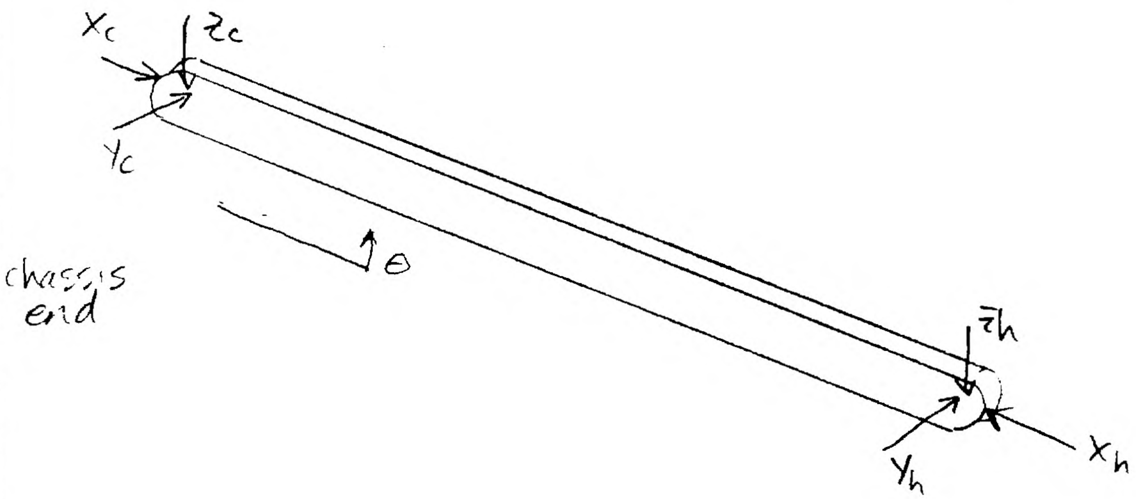
LOADER SUBSYSTEM

CALCULATIONS

DESIGN OF LOADER BEAMS + FORKS

- maximum cargo weight = 2 lbf, rounding up from radiation shield weight of 1.7 lbf.
- dynamic considerations consist of minimum safety factor of 5 for static calculations
- SEE DWG 5-1 FOR ASSEMBLY

BEAM → Tapered I-section, triangular in some form



Case 1: $\theta = 0$, horizontal - max tip deflection, bending + shear stress

→ from fork geometry, moment arm from radshield contact point to beam-pivot joint = 3.3 in

Fork mass estimate = .1 lbf max, so max moment

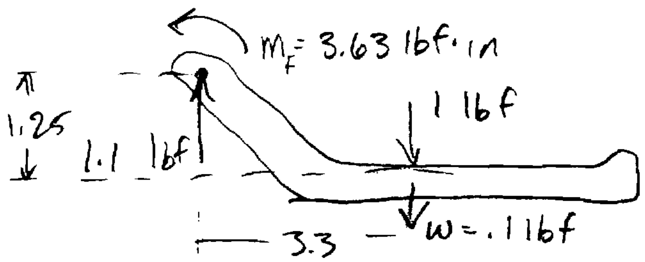
$= 3.3\,in\left(.1\,lbf + 2\,lbf/_{2\,forks}\right) = 3.63\ lbf \cdot in$

radius of chain sprocket = .33 in, so load on chain

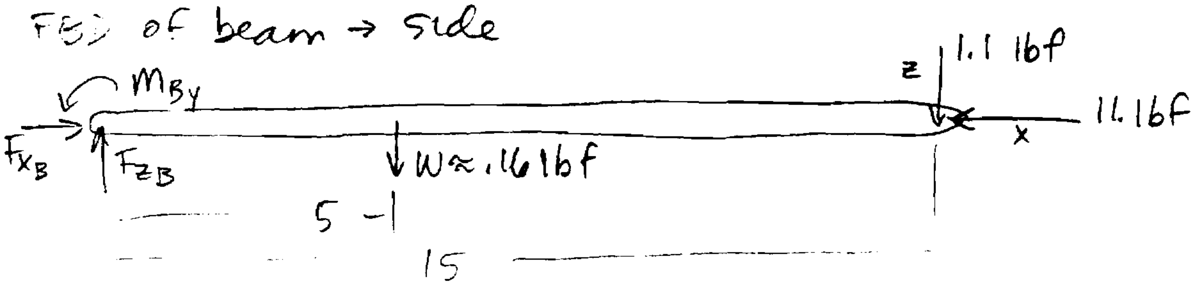
$= \dfrac{3.63\ lbf \cdot in}{.33\ in} = 11\ lbf$ → because of linkage, this is countered by 11 lbf compressive stress in X direction; always present at any angle.

JOHN LEFFLER

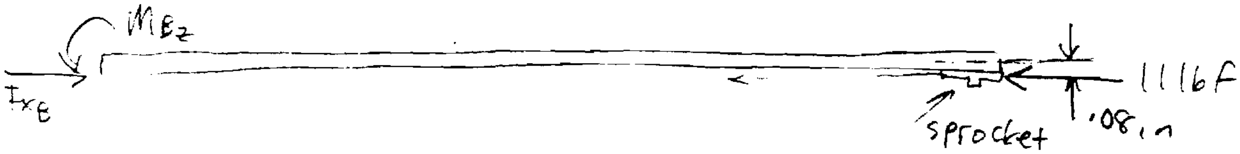
FBD of fork + shaft



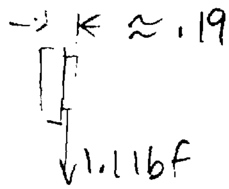
FBD of beam → side



FBD of beam - top



FBD of beam: end



⇒ moment from offset mounting of fork

so $M_{Bx} = (1.1 \text{ lbf})(.19 \text{ in}) = .21 \text{ in}\cdot\text{lbf}$

→ ignored

$\Sigma F_x = 0 = 11 \text{ lbf} - F_{x_B}$ $\quad F_{x_B} = 11 \text{ lbf}$

$\Sigma F_y = 0$ $\quad \Sigma F_z = 0 = 1.1 \text{ lbf} + .16 \text{ lbf} - F_{z_B}$

$F_{z_B} = 1.26 \text{ lbf}$

$\Sigma M_y \overset{+}{=} 0 = (1.1 \text{ lbf})(15 \text{ in}) + (.16 \text{ lbf})(5 \text{ in}) - M_{By} = 0$

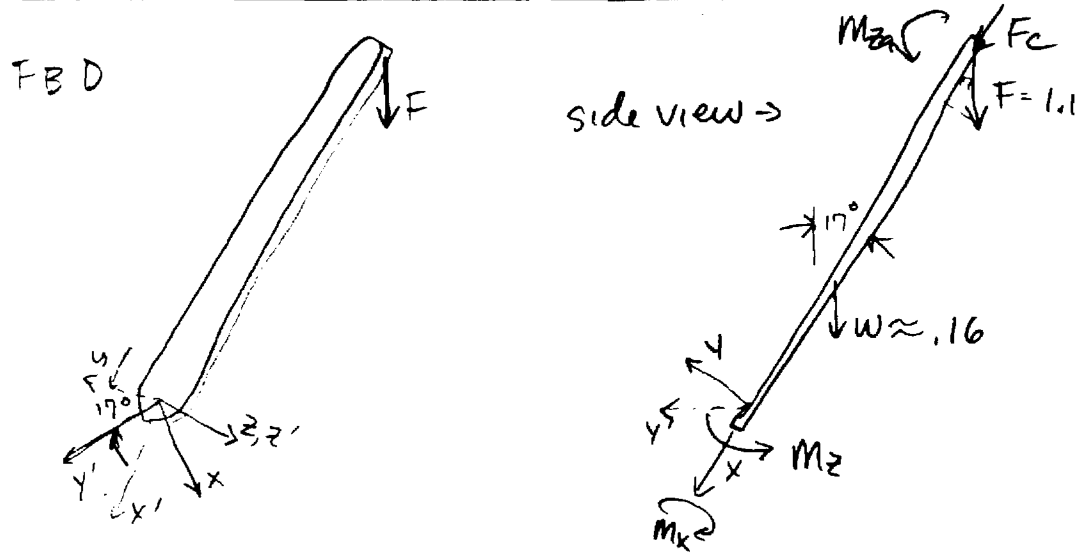
$M_{By} = 17.3 \text{ lbf}\cdot\text{in}$

$\Sigma M_z \overset{+}{=} 0 = (11 \text{ lbf})(.08 \text{ in}) - M_{Bz} = 0$ $\quad M_{Bz} = .88 \text{ in}\cdot\text{lbf}$

→ ignored

Case 2: $\Theta = 90$. Maximum buckling likelihood. However, by intuition, the worst case failure about the y-axis (which would be the buckling axis) would be with arms and radshield raised to full height and at a roll angle of 17°, caused by a 10 cm rock. This sizeable angle applies less to a buckling equation + more to a bending equation.

FBD



$$\Sigma F_x = 0 = (1.1\ \text{lbf} + .16\ \text{lbf})\cos 17 + 11\ \text{lbf} + F_x$$

$$F_x = -12.2\ \text{lbf}$$

$$\Sigma F_y = 0 = (1.1\ \text{lbf} + .16\ \text{lbf})\sin 17 - F_y$$

$$F_y = .37\ \text{lbf}$$

$$\Sigma F_z = 0$$

$$\Sigma M_x = 0 = 3.3\ \text{in}\,(1.1\ \text{lbf}) - M_x \qquad M_x = 3.63\ \text{in}\cdot\text{lbf}$$

$$\Sigma M_y = 0 = 0$$

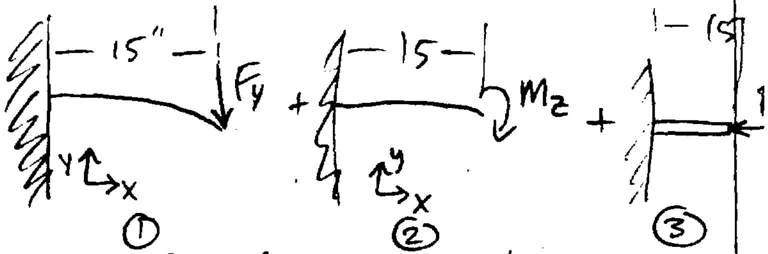
$$\Sigma M_z = 0 = (.16\ \text{lbf}\,(5\ \text{in}) + 1.1\ \text{lbf}\,(15\ \text{in}))\sin 17$$

$$-1.25''(1.1\ \text{lbf}) - M_z \qquad M_z = 3.68\ \text{lbf}\cdot\text{in}$$

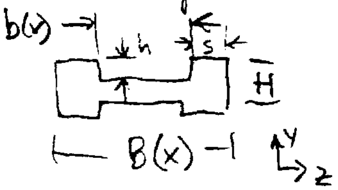
SO: $F_x = -12.2$ lbf $\quad F_y = .37$ lbf

$F_z = 0 \quad M_x = 3.63$ in·lbf $\quad M_z = 3.68$ in·lbf

Modeling beam as →



→ will calculate max stresses in beam at 1 inch intervals, calculate principle stresses + compare with yield stress. Using flexure formula $\sigma_y = \frac{M_y}{I_y}$



$$I_y = \frac{2sH^3 + b(H-2h)^3}{12} \qquad s = \frac{B-b}{2}$$

$$\text{so } \sigma_y = \frac{(H/2)(12)}{(B-b)H^3 + b(H-2h)^3}\left(F_y(15) + M_z\right) + \frac{F_x}{A(x)}$$

⇒ yield max compressive stresses

---

* SEE FIG. MCAD1 FOR SHEAR STRESSES

→ shear stresses ignored → all 20 lbf/in² or less

* TOTAL ARM TWIST $= \frac{TL}{GI_p}$ → small end used for section, $G = 3.8E6$ psi, $I_p = I_x + I_y = .00593$ in⁴

$$\text{total twist} = \frac{M_x L}{G I_p} = \frac{3.63 \text{ in·lbf}(15 \text{ in})}{3.8E6 \text{ lbf/in}^2(.00593 \text{ in}^4)} = .0024 \text{ rad}$$

→ ignored

* YIELD STRESS FOR 6061-T6 = 40,000 psi

1) Safety factor for X-axis bending failure (Case 1)

* SEE FIG. MCAD2 → max $\sigma_x = 1.056E3$ psi

$n_x = 40E3/1.056E3 = 37.8$ or 37

2) Safety factor for Y-axis bending failure (Case 2)

* SEE FIG. MCAD3 → max $\sigma_y = 5.854E3$ psi

$n_y = 40E3/5.854E3 = 6.83$ or 6

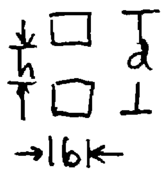
FORK CALCULATIONS

→ SEE DWG 5-3 FOR DIMENSIONS

STRESS EVALUATED ACROSS FIRST LIGHTENING HOLE

1) X-AXIS BENDING:

$\sigma_x = \dfrac{M_F}{Z_x}$ $Z_x \Rightarrow$



$Z_x = \dfrac{b(d^3 - h^3)}{6d}$ in^3

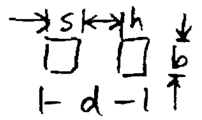
from Machys HandBook, 24th Ed

$Z_x = \dfrac{.125''((.5'')^3 - (.25'')^3)}{6(.5'')} = .00684\ in^3$ $M_F = 3.63$ lbf-in from calculation pg 2

$\therefore\ \sigma_x = \dfrac{3.63\ in \cdot lbf}{.00456\ in^3} = 796\ lbf/in^2$

2) Y-AXIS BENDING

$\sigma_y = \dfrac{M_x}{Z_y}$ $Z_y \Rightarrow$



$Z_y = \dfrac{sb^2}{3}$ in^3

$Z_y = \dfrac{(.125'')(.125'')^2}{3} = 6.5E-4\ in^3$

$M_x = 3.63$ lbf·in from calculation pg 3

$\therefore\ \sigma_y = \dfrac{3.63\ in \cdot lbf}{.00065\ in^3} = 5576\ lbf/in^2$

---

* SAFETY FACTORS: ALUM 6061-T6 YIELD STRESS > 40E3 psi

X-AXIS BENDING: $n_x = \dfrac{40E3\ psi}{796\ psi} = 50$

Y-AXIS BENDING: $n_y = \dfrac{40E3\ psi}{5576\ psi} = 7$

* BUCKLING NOT LIKELY WITH DESIGN USED

* SHEAR STRESSES: $MAX = \dfrac{V}{A} = \dfrac{1.1\ lbf}{2(.125'')^2} = 35.2\ psi$ → ignored

ESTIMATED POWER + TORQUE REQUIREMENTS

Using Stock Drive Products 32 pitch, $14^{1/2°}$ pressure angle, single thread, lead angle 4° 5', lead = .0982 20:1 ratio. Efficiency value from Boston Gear Catalog:

$$E = \frac{\tan\gamma(1 - f\tan\gamma)}{f + \tan\gamma}$$

with γ = lead angle, f = coefficient of friction. Using their upper coefficient of friction value (.05)

$$E = \frac{\tan 4°5'(1 - .05\tan 4°5')}{.05 + \tan 4°5'} = 58.6\%$$

TORQUE ON SHAFT: From calculations pages 1 and 2, the estimated max torque on the shaft is M_{By}, Case 1, so T = 17.3 in·lbf or 277 in·oz.

With the 20:1 ratio the motor torque needed for perfect efficiency is 13.84 in·oz = T_s

With the worm drive efficiency, $T_s = \frac{13.84 \text{ in·oz}}{58.6\%} = 23.6$ in-oz

With the sizeable drag of the 5 shaft bushings and the fork chains, we reduce the efficiency by 2/3.

so $T_s = \frac{23.6 \text{ in·oz}}{.333} = 70.9$ in·oz → optimistic perhaps

MECHANICAL POWER: For the desired arm rotation speed of 1 rpm, 20 rpm are needed from the motor

so $P = T_s\omega = (70.9 \text{ in·oz})(1 \text{ rpm})(2\pi \text{ rad/rev})(1 \text{ m}/60\text{s})$

$= 7.42 \text{ in·oz}\left(\frac{1\text{ft}}{12\text{in}}\right)\left(\frac{1\text{lbf}}{16\text{oz}}\right)\left(\frac{\pi \text{ rad}}{30\text{s}}\right) = .00405 \frac{\text{ft·lbf·rad}}{\text{s}}$

= 1.17E-6 HP or 8.7E-4 W

FIGURE MCAD 1

CALCULATION OF SHEAR STRESSES FOR TAPERED I-BEAM ABOUT X-AXIS AND Y-AXIS, MEASURED AT INTERVALS OF 1 INCH (x) FROM THE SMALL END.

X-AXIS:

VARIABLES ARE b=width of each section cutout, B=overall width, P=tip load, Hb=lg end height, flange thickness=.125, stresses are in lbf/in^2

$x := 0..15 \qquad B := .1875 \qquad b := .0625 \qquad P := 1.26 \qquad Hb := .9375$

$$Ax_x := \left[B\cdot\left[.5+\frac{x}{15}\cdot(Hb-.5)\right]-2\cdot b\cdot\left[.25+\frac{x}{15}\cdot(Hb-.5)\right]\right] \qquad \sigma x_x := \frac{P}{Ax_x}$$

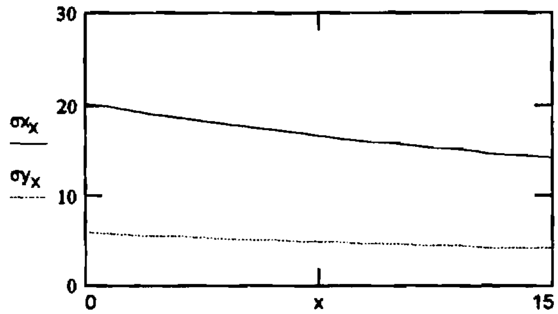
Y-AXIS:

VARIABLES ARE: H=overall section height, h=section cutout heights, Fy=force on tip, Bb=I-Beam large end diameter

$H := .1875 \qquad h := .0625 \qquad Bb := .9375 \qquad Fy := .37$

$$Ay_x := \left[H\cdot\left[.5+\frac{x}{15}\cdot(Bb-.5)\right]-2\cdot h\cdot\left[.25+\frac{x}{15}\cdot(Bb-.5)\right]\right] \qquad \sigma y_x := \frac{Fy}{Ay_x}$$

σx_x	σy_x
20.16	5.92
19.589	5.752
19.049	5.594
18.538	5.444
18.054	5.301
17.594	5.167
17.157	5.038
16.742	4.916
16.346	4.8
15.968	4.689
15.608	4.583
15.263	4.482
14.933	4.385
14.618	4.292
14.315	4.204
14.024	4.118

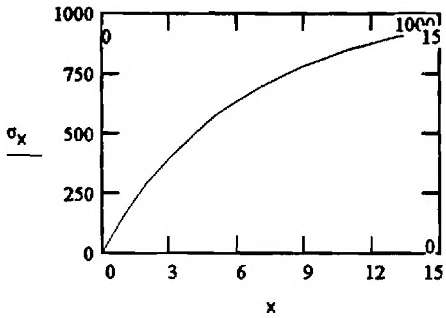


CALCULATION OF NORMAL STRESSES FOR TAPERED I-BEAM, SMALL END = .5" TALL, MEASURED AT INTERVALS OF 1 INCH (x) FROM THE SMALL END. X-AXIS ROTATIONS. VARIABLES ARE b=width of each section cutout, B=overall width, P=tip load, Hb=lg end height, flange thickness=.125, stresses are in lbf/in^2

MCAD 2

$x := 0..15 \qquad B := .1875 \qquad b := .0625 \qquad P := 1.26 \qquad Hb := .9375$

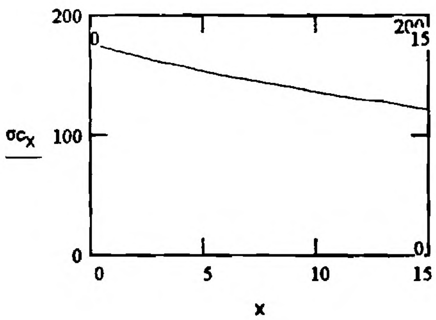
$$\sigma_x := \frac{P\cdot x\cdot\left[\frac{\left[.5+\frac{x}{15}\cdot(Hb-.5)\right]}{2}\right]}{\frac{1}{12}\cdot\left[B\cdot\left[.5+\frac{x}{15}\cdot(Hb-.5)\right]^3-2\cdot b\cdot\left[.25+\frac{x}{15}\cdot(Hb-.5)\right]^3\right]}$$



$$\sigma = \begin{pmatrix} 0 \\ 159.615 \\ 291.397 \\ 401.151 \\ 493.234 \\ 570.977 \\ 636.955 \\ 693.192 \\ 741.295 \\ 782.553 \\ 818.01 \\ 848.524 \\ 874.797 \\ 897.417 \\ 916.873 \\ 933.578 \end{pmatrix}$$

CALCULATION OF COMPRESSIVE STRESS AT 1-INCH SECTIONS ALONG BEAM DUE TO 11 lbf COMPRESSIVE FORCE.

$$\sigma c_x := \frac{11}{\left[B\cdot\left[.5+\frac{x}{15}\cdot(Hb-.5)\right]-2\cdot b\cdot\left[.25+\frac{x}{15}\cdot(Hb-.5)\right]\right]}$$



$$\sigma c = \begin{pmatrix} 176 \\ 171.012 \\ 166.299 \\ 161.839 \\ 157.612 \\ 153.6 \\ 149.787 \\ 146.159 \\ 142.703 \\ 139.406 \\ 136.258 \\ 133.249 \\ 130.37 \\ 127.613 \\ 124.97 \\ 122.435 \end{pmatrix} \qquad \sigma + \sigma c = \begin{pmatrix} 176 \\ 330.627 \\ 457.697 \\ 562.99 \\ 650.846 \\ 724.577 \\ 786.742 \\ 839.351 \\ 883.998 \\ 921.958 \\ 954.268 \\ 981.773 \\ 1.005\cdot 10^3 \\ 1.025\cdot 10^3 \\ 1.042\cdot 10^3 \\ 1.056\cdot 10^3 \end{pmatrix}$$

FIGUR
MCAD :

CALCULATION OF COMPRESSIVE STRESSES FOR TAPERED I-BEAM, SMALL END = .5" TALL, FOR CONDITION OF VERTICAL BEAM POSITION WITH RADSHIELD ATTACHED, 17 DEG. ROLL; MEASURED AT INTERVALS OF 1 INCH (x) FROM THE SMALL END. Y-AXIS ROTATIONS. VARIABLES ARE: H=overall section height, h=section cutout heights, Mz=moment around Z axis, Fy=force on tip, Bb=I-Beam large end diameter, Fx=chain force (compressive), s=flange thickness

$x := 0..15 \quad H := .1875 \; h := .0625 \quad Mz := 3.68 \quad Fy := .37 \quad Bb := .9375 \quad Fx := 12.2 \quad s := .125$

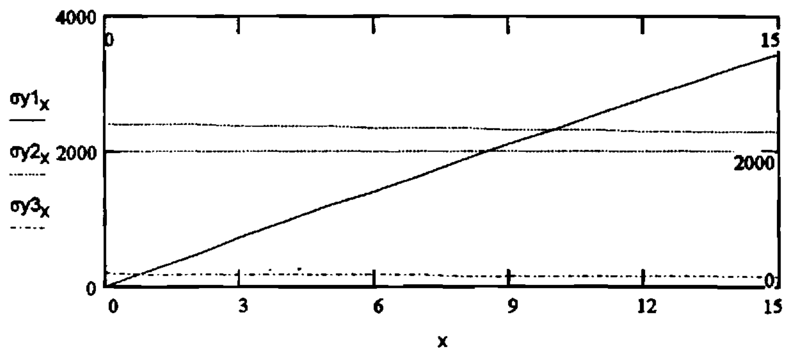
$$A_x := \left[H\cdot\left[.5 + \frac{x}{15}\cdot(Bb - .5)\right] - 2\cdot h\cdot\left[.25 + \frac{x}{15}\cdot(Bb - .5)\right]\right]$$

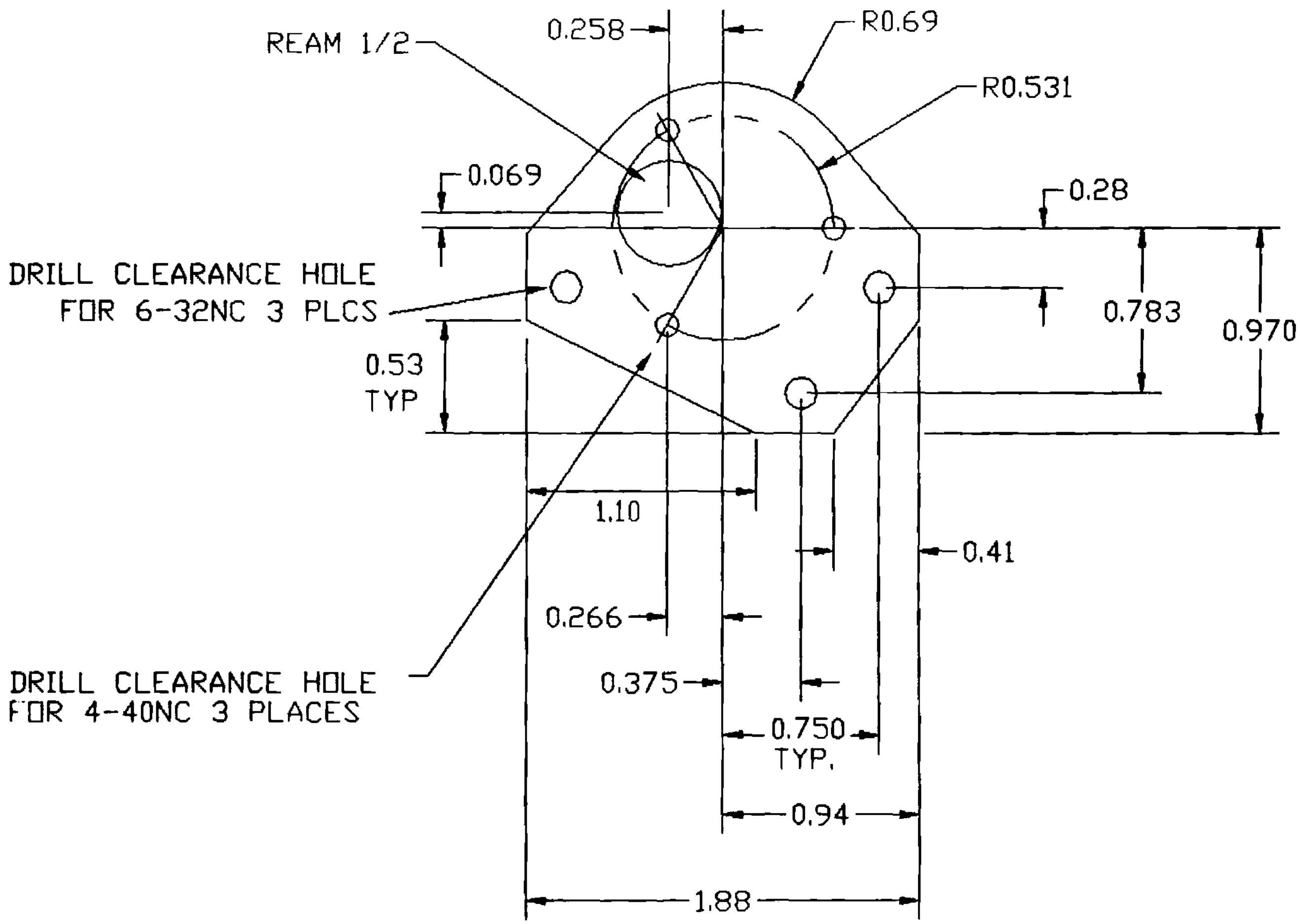
$$\sigma y1_x := \frac{H\cdot 6}{2\cdot s\cdot H^3 + \left[.25 + \frac{x}{15}\cdot(Bb - .5)\right]\cdot(H - 2\cdot h)^3}\cdot(Fy\cdot x)$$

$$\sigma y2_x := \frac{H\cdot 6}{2\cdot s\cdot H^3 + \left[.25 + \frac{x}{15}\cdot(Bb - .5)\right]\cdot(H - 2\cdot h)^3}\cdot Mz$$

$$\sigma y3_x := \frac{Fx}{A_x}$$

$$\sigma y1 = \begin{bmatrix} 0 \\ 242.555 \\ 483.106 \\ 721.676 \\ 958.291 \\ 1.193\cdot 10^3 \\ 1.426\cdot 10^3 \\ 1.657\cdot 10^3 \\ 1.886\cdot 10^3 \\ 2.113\cdot 10^3 \\ 2.338\cdot 10^3 \\ 2.562\cdot 10^3 \\ 2.784\cdot 10^3 \\ 3.004\cdot 10^3 \\ 3.222\cdot 10^3 \\ 3.439\cdot 10^3 \end{bmatrix} \quad \sigma y2 = \begin{bmatrix} 2.422\cdot 10^3 \\ 2.412\cdot 10^3 \\ 2.402\cdot 10^3 \\ 2.393\cdot 10^3 \\ 2.383\cdot 10^3 \\ 2.373\cdot 10^3 \\ 2.363\cdot 10^3 \\ 2.354\cdot 10^3 \\ 2.344\cdot 10^3 \\ 2.335\cdot 10^3 \\ 2.326\cdot 10^3 \\ 2.316\cdot 10^3 \\ 2.307\cdot 10^3 \\ 2.298\cdot 10^3 \\ 2.289\cdot 10^3 \\ 2.28\cdot 10^3 \end{bmatrix} \quad \sigma y3 = \begin{bmatrix} 195.2 \\ 189.668 \\ 184.441 \\ 179.494 \\ 174.806 \\ 170.356 \\ 166.128 \\ 162.104 \\ 158.27 \\ 154.614 \\ 151.123 \\ 147.785 \\ 144.593 \\ 141.535 \\ 138.604 \\ 135.791 \end{bmatrix} \quad \sigma y1 + \sigma y2 + \sigma y3 = \begin{bmatrix} 2.618\cdot 10^3 \\ 2.845\cdot 10^3 \\ 3.07\cdot 10^3 \\ 3.294\cdot 10^3 \\ 3.516\cdot 10^3 \\ 3.736\cdot 10^3 \\ 3.955\cdot 10^3 \\ 4.173\cdot 10^3 \\ 4.388\cdot 10^3 \\ 4.602\cdot 10^3 \\ 4.815\cdot 10^3 \\ 5.026\cdot 10^3 \\ 5.235\cdot 10^3 \\ 5.443\cdot 10^3 \\ 5.65\cdot 10^3 \\ 5.854\cdot 10^3 \end{bmatrix}$$





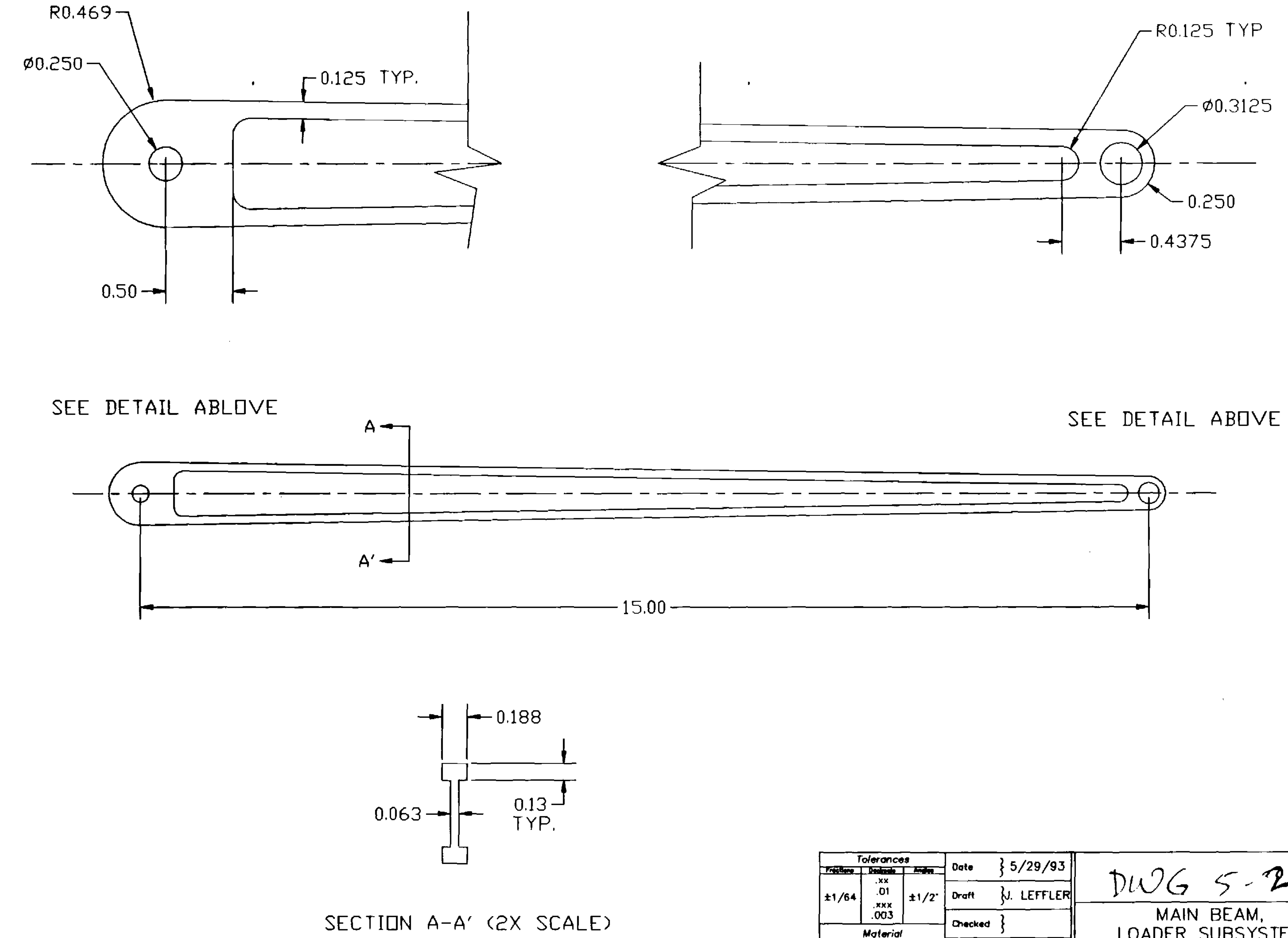
Tolerances		
Fractions	Decimals	Angles
±1/64	.xx .01 .xxx .003	±1/2°

Material
ALUM 6061T6
OR EQUIV

Date	5/15/93
Draft	J. LEFFLER
Checked	
Appvd	

DWG 5-1

MOUNTING PLATE FOR
GLOBE MOTORS
GEARMOTOR IM-13



Tolerances					
Fractions	Decimals	Angles	Date	5/29/93	DWG 5-2
±1/64	.xx .01 .xxx .003	±1/2°	Draft	J. LEFFLER	MAIN BEAM, LOADER SUBSYSTEM
Material			Checked		
ALUM 6061-T6			Appvd		
			Scale		

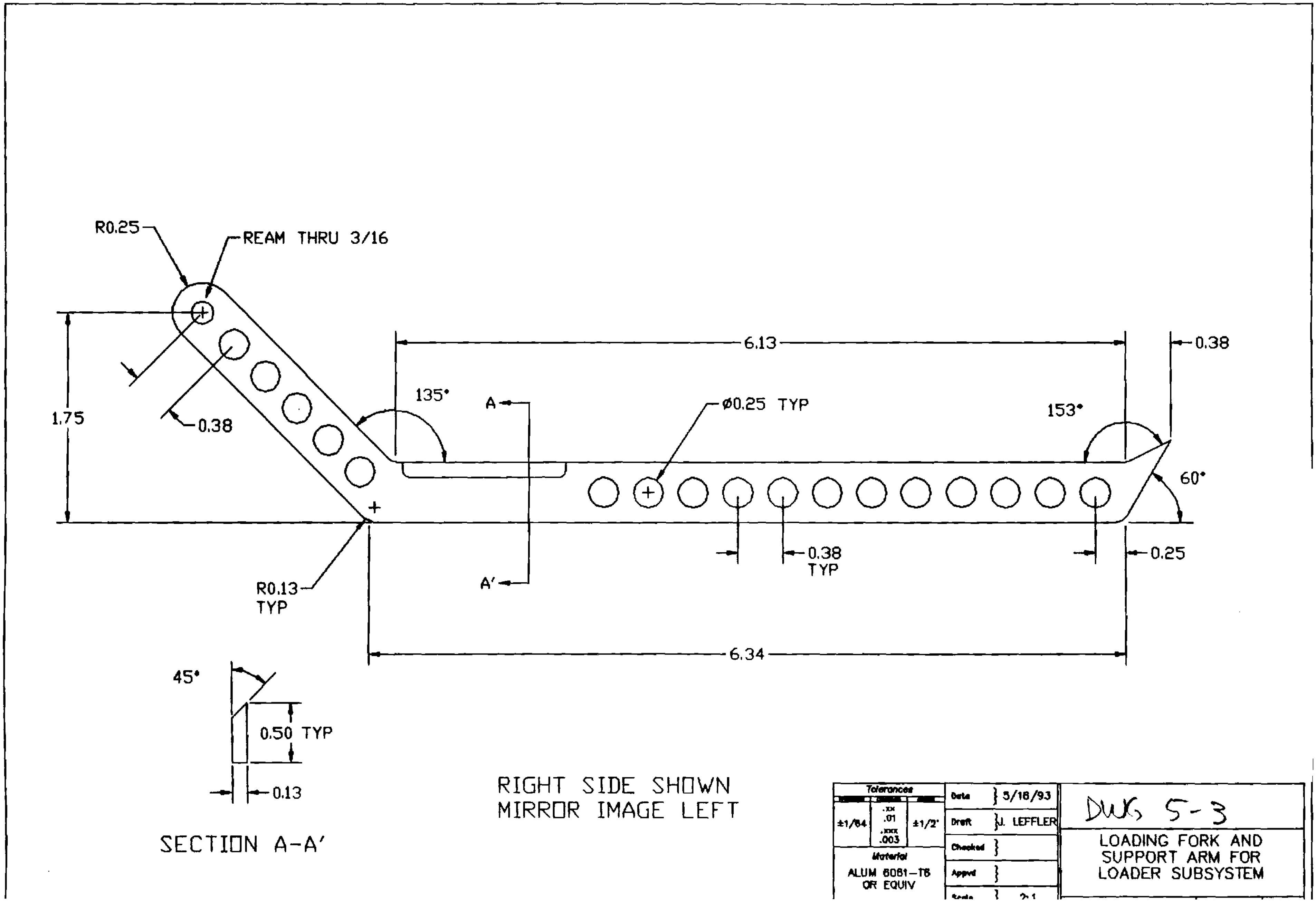

R0.25
REAM THRU 3/16
1.75
0.38
135°
A
A'
6.13
0.38
Ø0.25 TYP
153°
60°
0.38
TYP
0.25
R0.13
TYP
6.34
45°
0.50 TYP
0.13
SECTION A-A'
RIGHT SIDE SHOWN
MIRROR IMAGE LEFT
Tolerances
±1/64
.xx
.01
.xxx
.003
±1/2°
Material
ALUM 6061-T6
OR EQUIV
Date
5/18/93
Draft
J. LEFFLER
Checked
Appvd
DWG 5-3
LOADING FORK AND
SUPPORT ARM FOR
LOADER SUBSYSTEM

FIGURE 6.1

RADIATION SHIELD

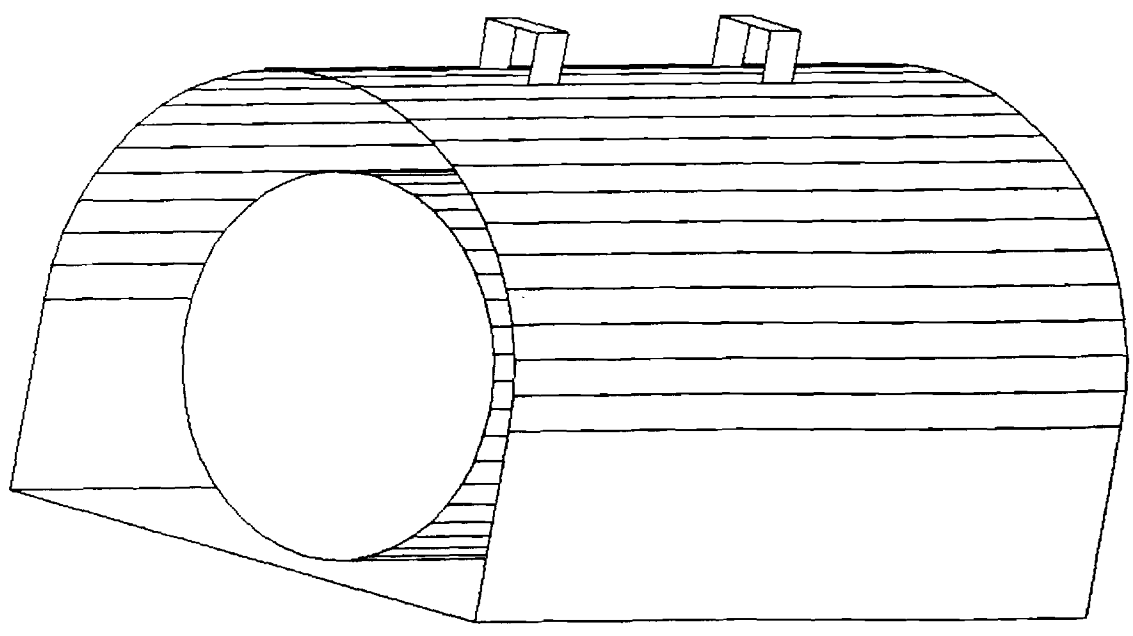
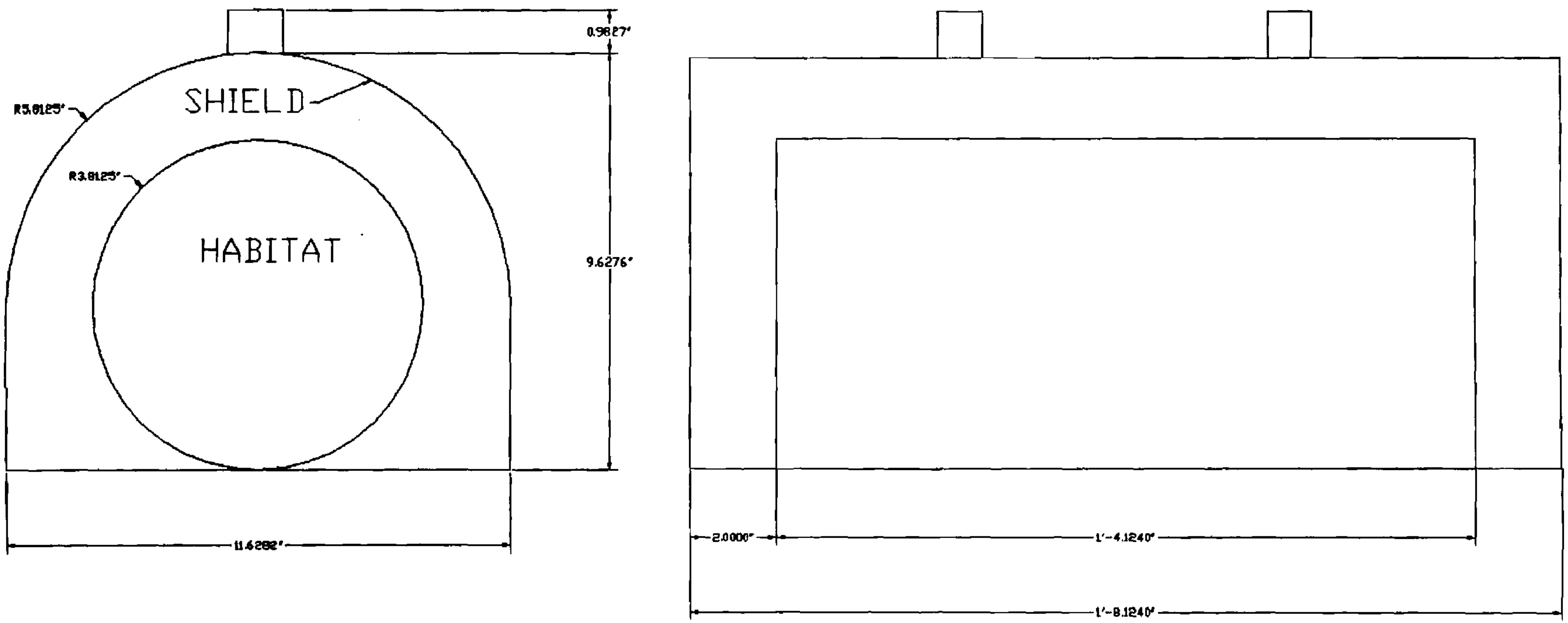


FIGURE 6.2

RADIATION SHIELDING



CALCULATIONS 6.1

For Al thickness = 0.087"

length = 20.124"

circumference = $\pi \cdot$ radius + 2 sides

$= \pi(5.8125") + 2(3.8125)$

$= 25.89"$

volume of shielding top = (25.89")(20.124")(0.087")

$= 45.32"^3$

volume of shielding side = $[\frac{1}{2}\pi \cdot \text{radius}^2 + \text{height} \cdot \text{length}](.087)$

$= [\frac{1}{2} \cdot \pi(5.8125) + (3.8125)(11.625)](.087)$

$= [53.45](.087)$

$= 4.65"^3$

total volume = $45.32"^3 + (4.65)2$

$= 54.62"^3$

density of Al = 2.699 g/cm³

$= 2.699\ g/cm^3 \left(\frac{2.54\ cm}{1\ in}\right)^3 \left(\frac{1\ lb}{453.6\ g}\right)$

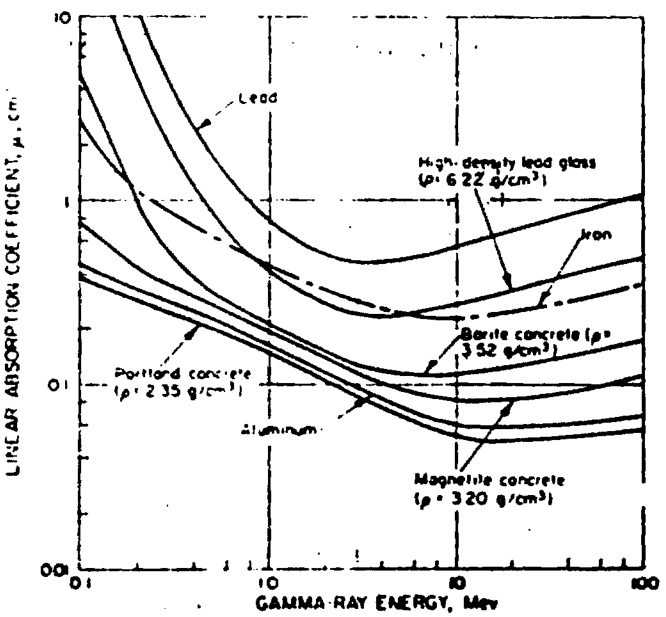
$= 0.0975\ \frac{lb}{in^3}$

total weight = total volume · density

$= (54.62\ in^3)(0.0975\ lb/in^3)$

$= 5.33$ lb

FIGURE 6.3



Lead
High-density lead glass
(ρ= 6.22 g/cm³)
Iron
Barite concrete (ρ= 3.52 g/cm³)
Portland concrete
(ρ= 2.35 g/cm³)
Aluminum
Magnetite concrete
(ρ= 3.20 g/cm³)
LINEAR ABSORPTION COEFFICIENT, μ, cm⁻¹
GAMMA-RAY ENERGY, Mev

CALCULATIONS 6.2

$I = I_0 e^{-\mu t}$

where I = resulting intensity
I_0 = original intensity
μ = linear attenuation coeffecient
t = thickness

at 10 Mev

$\mu_{Aluminum} = 0.062\ cm^{-1}$

$\mu_{H_2O} = 0.022\ cm^{-1}$

transmissibility ratio for H_2O

$$\frac{I}{I_0} = e^{-\mu t}$$

$$= e^{-(0.022\ cm^{-1})(cm)}$$

$$= 0.7189$$

transmissibility ratio for Al has to equal transmissibility ratio for H_2O

$$\frac{\ln \frac{I}{I_0}}{\mu_{Al}} = t$$

$$\frac{\ln(.7189)}{-(.062)} = t$$

$$5.32\ cm = t$$

CALCULATIONS 6.3

It is known that approximately 10 g/cm² of carbon
nielding is sufficient to shield against inner-belt
rotons.

$$\frac{I}{I_o} = e^{-(0.044)\left(\frac{10\ g/cm^2}{2.25\ g/cm^3}\right)}$$

$$= e^{-(.044)(4.44)}$$

$$\frac{I}{I_o} = 0.822$$

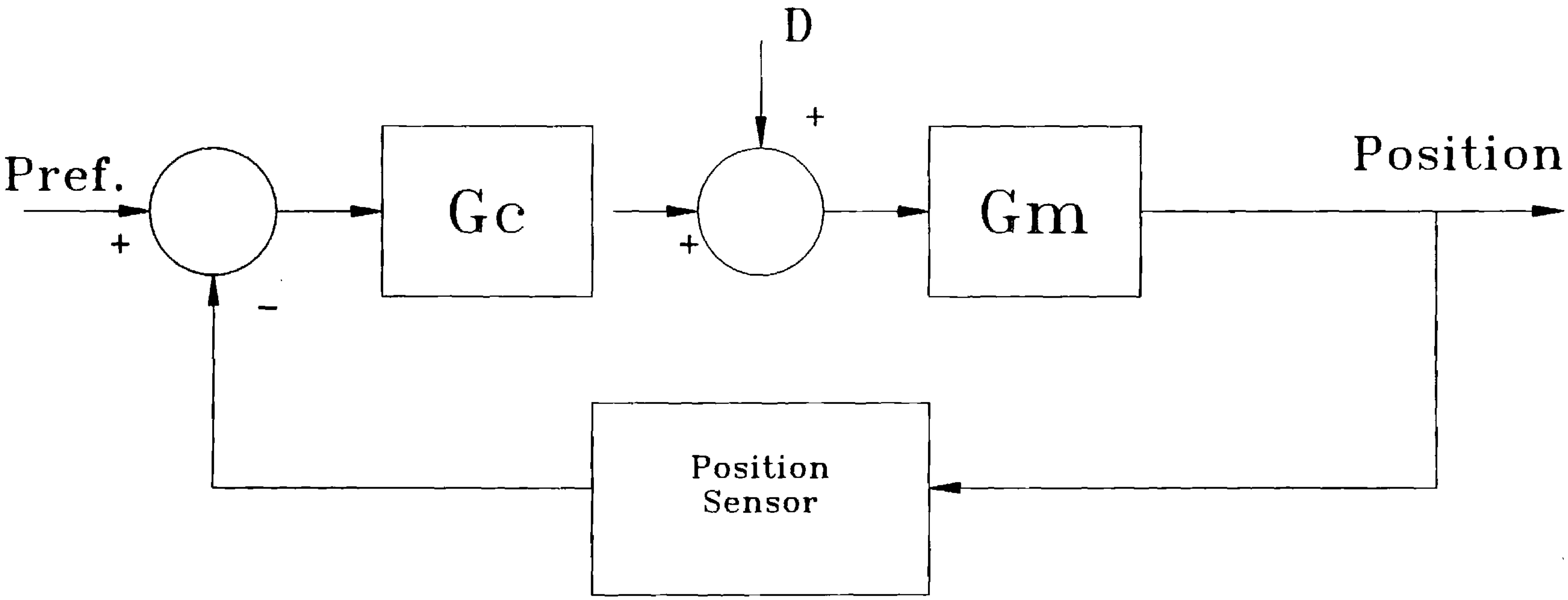
$$t = \frac{\ln \frac{I}{I_o}}{\mu_{Al}}$$

$$= \frac{\ln(0.822)}{-(.062)}$$

$$t = 3.15\ cm$$

his is what is needed in aluminum to attenuate protons.

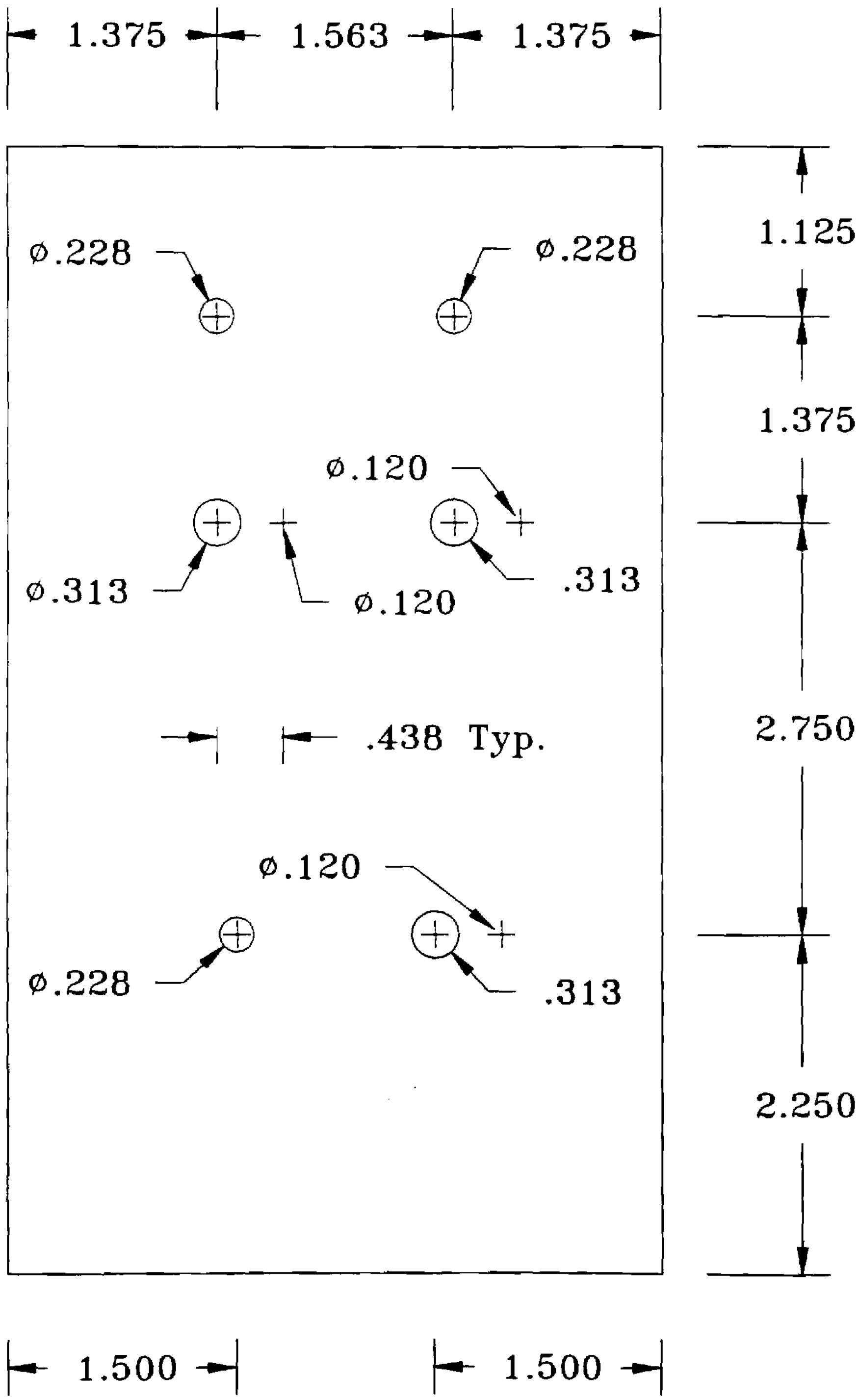
Control Block Diagram
Figure 7.1



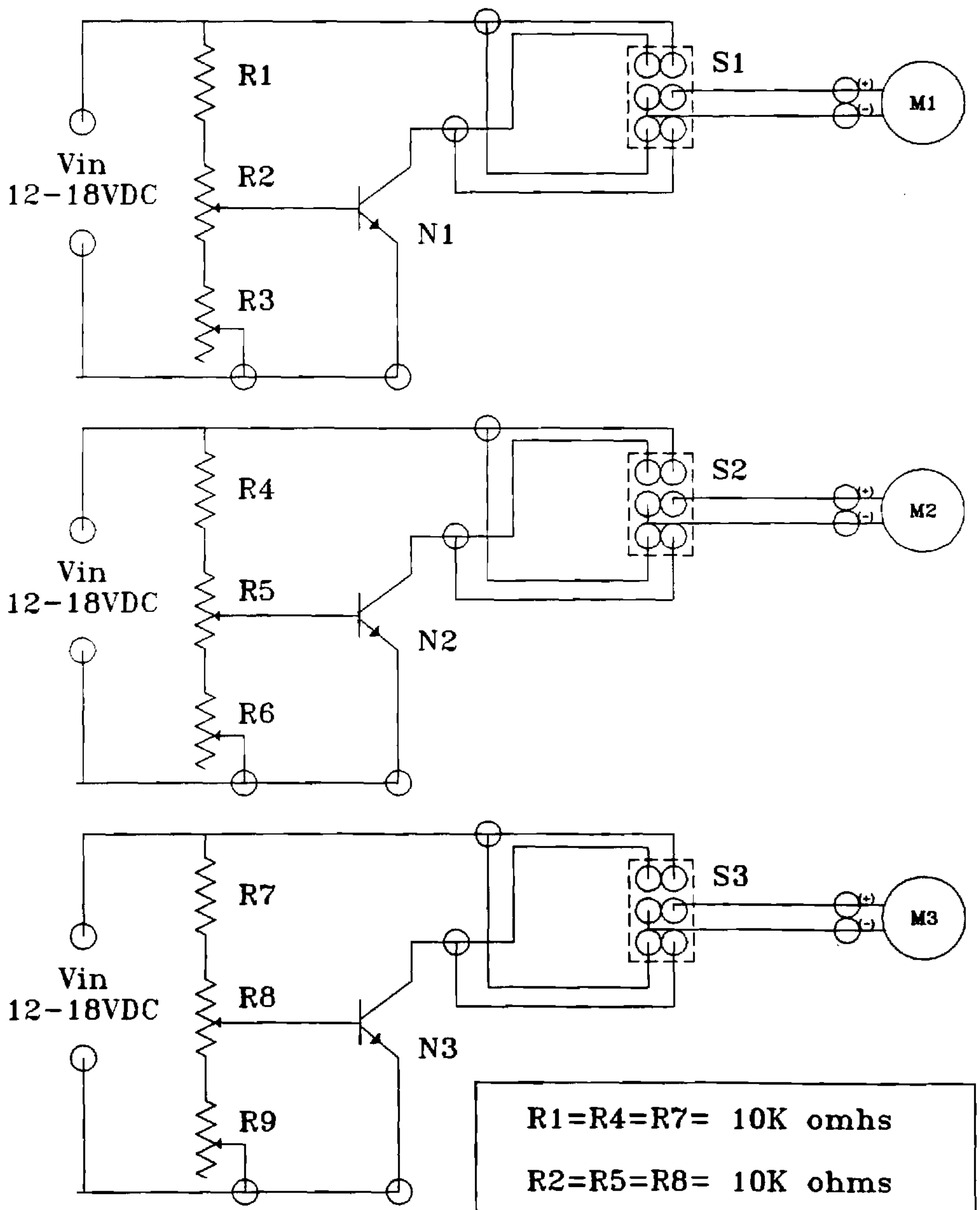
Note: The Pref. and position sensor functions are humanly generated for this application.

Control Panel Layout

Figure 7.2



Control Circuit
Figure 7.3



R1=R4=R7= 10K omhs

R2=R5=R8= 10K ohms

R3=R6=R9= 1K ohms

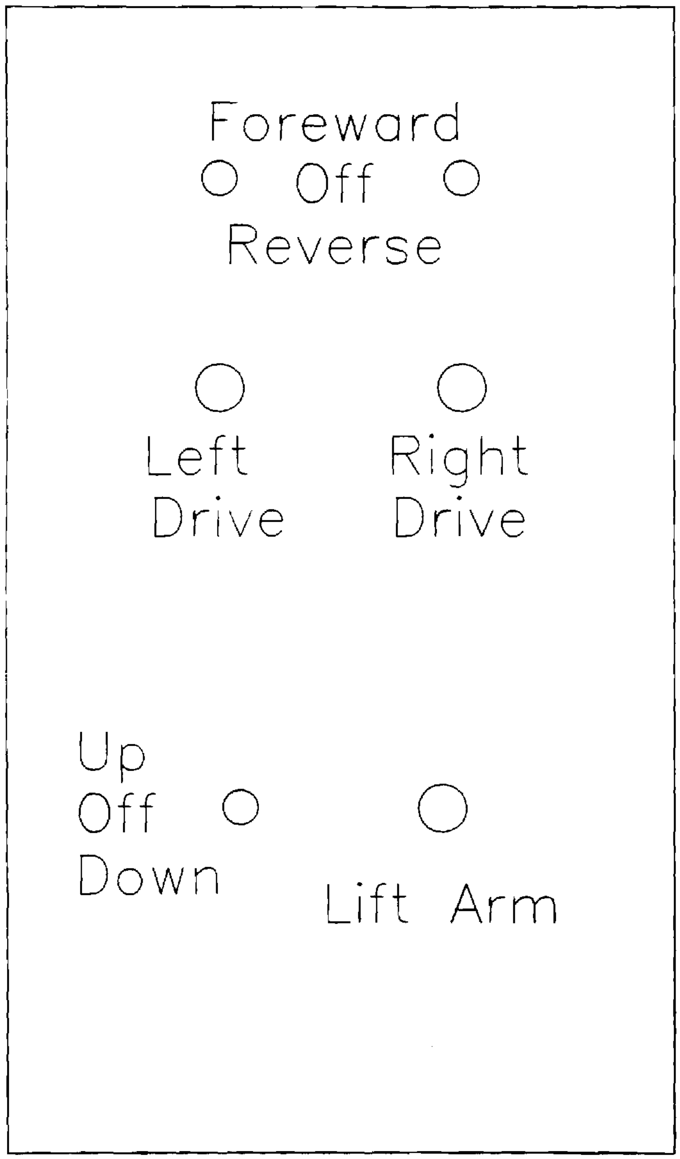
N1=N2=N3= TIP3015

S1=S2=S3= DPDT-CO (3A)

M1=M2= GLOBE 415A160.
M3= GLOBE 415A159.

Control Panel

Figure 7.4



Control Circuit Analysis

This analysis assumes the following:

- The voltage drop across the transistor base to emitter is .7V.
- The potentiometer is modeled as two constrained resistors (R2a and R2b).

Input Circuit

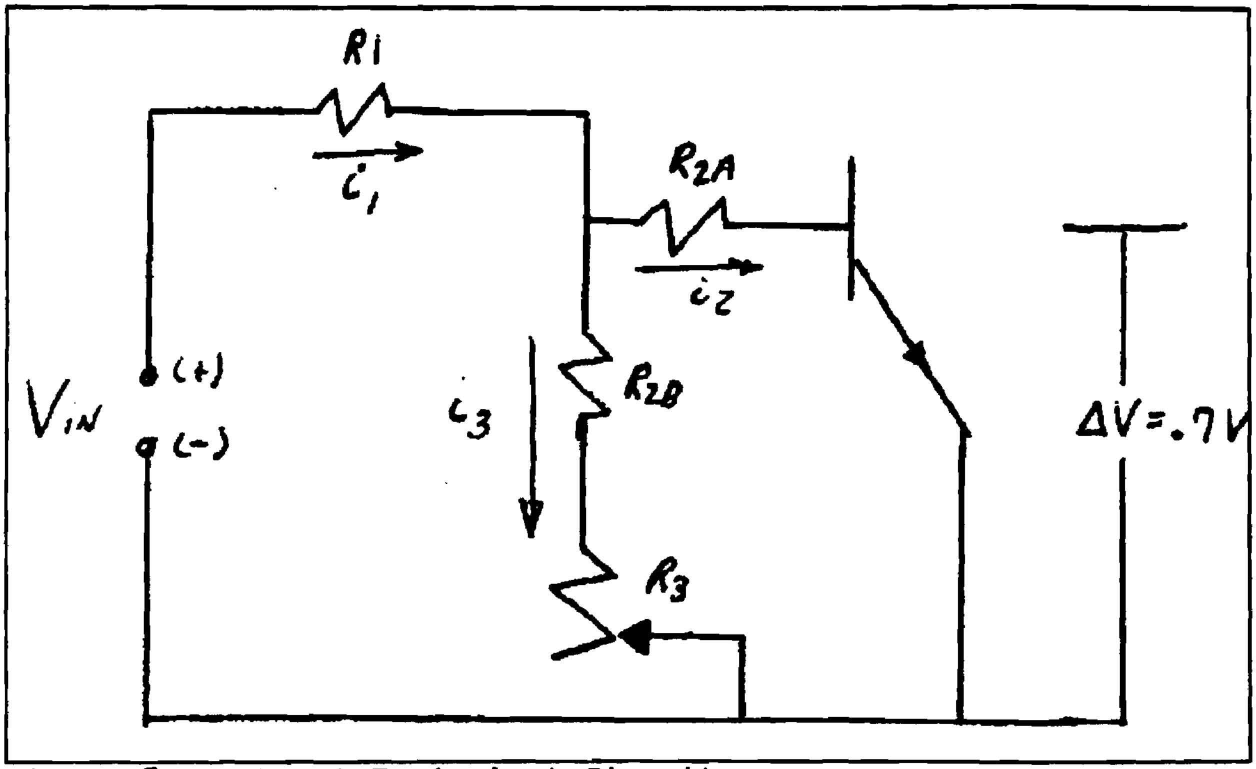


Figure 7-5: Input Equivalent Circuit

The following equations apply to this system. These equations were derived using Kirchoff's voltage and current laws.

$$V_i = R_1 i_1 + R_{2b} i_3 + R_3 i_3 \tag{1}$$

$$V_i = R_1 i_1 + R_{2a} i_2 + .7 \tag{2}$$

$$i_1 = i_2 + i_3 \tag{3}$$

Combining these equations and solving for i_2 gives:

$$i_2=\frac{(V_i-.7)\,(R_1+R_{2b}+R_3)}{(R_1+R_{2a})\,(R_1+R_{2b}+R_3)-R_1^2} \tag{4}$$

The resistors R2a and R2b must always sum to be the value of the potentiometer, R2.

Output Circuit

A model of the output circuit is shown in Figure 5.6.

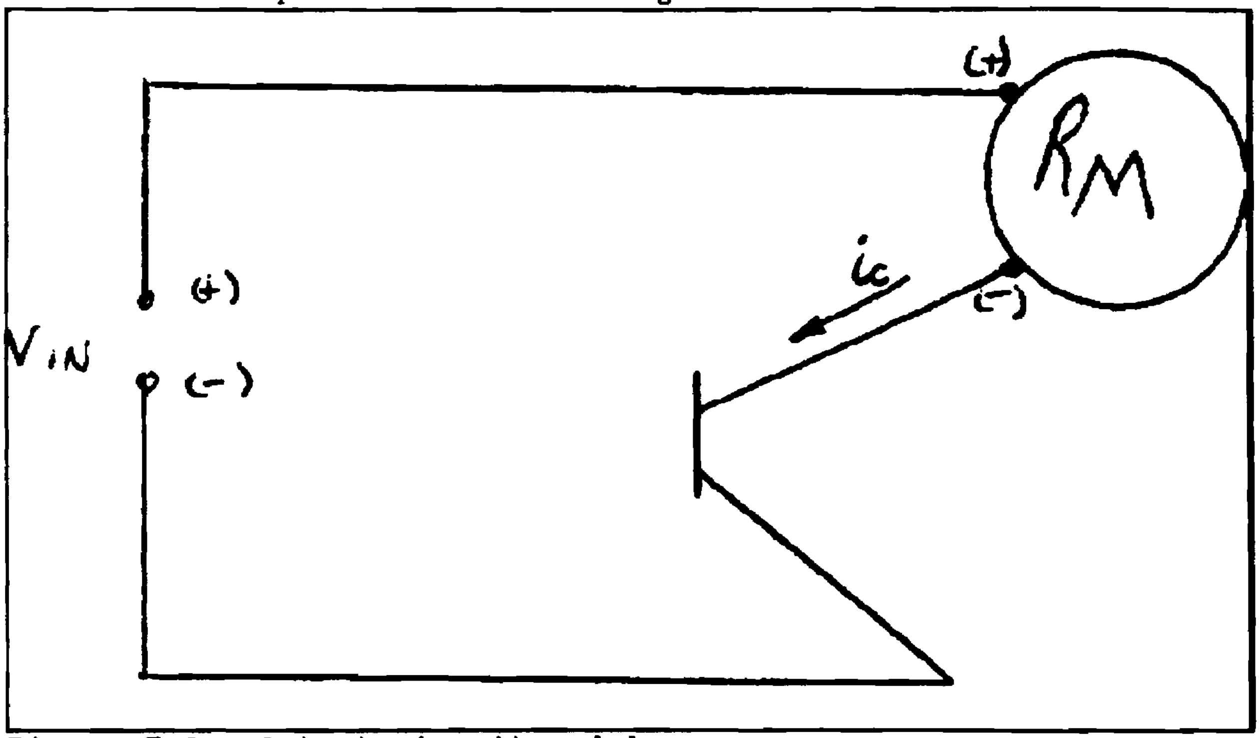


Figure 5-6: Output circuit model.

The resistance of the motor armature windings is Rm. This value is available in motor catalogs. For the particular motors used in this design, the value of Rm is 7.8 ohms. Assuming that the transistor gain is given as Beta, then the voltage across the motor terminals is given as :

$$V_m=\beta i_2 R_m \tag{5}$$

See the design notebook for speed vs torque and current curves for the motors used in this project.

Summary of Project Cost

Description	Amount	See Note#
Chain, sprockets, shafts, bushings, gears	$204.24	1
Wheels	$ 20.00	2
Shaft collars and wheel hubs	$ 20.00	3
Motors	$111.00	4
1" Square x 1/8" thick aluminum tubing	$ 8.32	5
Misc. aluminum plate	$ 4.00	6
Bolts	$ 2.42	7
Corrugated aluminum sheet	$ 1.00	8
Transistors, resistors, switches, enclosure, circuit boards	$ 26.51	9
Wire	$ 5.00	10
Total:	$402.29	

Notes:

1. Classified as drive components. Cost is actual cost of order from Stock Drive Products. See project notebook for invoice.

2. Classified as drive components. Since wheels were used, value is based on similar wheels available at local hobby stores.

3. Classified as drive components. Since these items were used, their value is based on similar components using Stock Drive Products as a reference.

4. Classified as drive components. Since all motors were used, their value is based on a quote of $37.00 each from South Atlantic Component Sales, Norcross, GA.

5. Classified as chassis components. Cost is actual cost for material used. See project notebook for invoice from Georgia Steel.

6. Classified as chassis components. Since all of this material was scrap, its value is based on new stock prices of approximately $4.00 per pound. This estimate was reached after a discussion with a sales representative from Georgia Steel.

7. Classified as chassis components. See invoice from B&C True Value Hardware included in project notebook.

8. Classified as shielding components. Since this material was scrap, its value is based on the new material cost of $.1625 per square foot.

9. Classified as electronic components. See invoices from Radio Shack included in the project notebook.

10. Classified as electronic components. Since all wire in the system was previously owned, its value is based on current prices using a R&S Electronics catalog as a reference.

Power Unit and Distribution System for the Lunar Enabler

ME 4182 Group 5

John Chamlee

Rick Canfield

Michael Johnson

Doug Kanipe

James Melchiors

Teresa Powell

March 10, 1993

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Letter of Transmittal

March 10, 1993

Mr. J. W. Brazell
School of Mechanical Engineering
Georgia Institute of Technology
Atlanta, GA 30332

Dear Mr. Brazell:

With this letter we are transferring the final report on our ME 4182 project to you. The report gives our recommendations for the power unit and power distribution system for the Enabler. We would like to thank you, Jeff Donnell, and Scott Pierce for your encouragement and assistance.

Sincerely,

John Chamlee, Group Leader

Rick Canfield

Michael Johnson

Doug Kanipe

James Melchiors

Teresa Powell

Executive Summary

The purpose of this paper is to describe the method of supplying power to the Enabler recommended by Group 5, **Power Unit and Distribution System**. The goal of Group 5 was to design a hydraulic system to provide desired pressure and flow to sufficiently power the enabler.

The power unit consists of a five horsepower Briggs and Stratton model 130200 internal combustion engine and a Hagglunds Denison PV6 Variable Piston Pump. This engine would be converted from gasoline to propane to reduce offensive emissions. The power unit is capable of supplying 2000 psi at a constant pressure and a maximum flow rate of 5.372 gallons per minute.

The power distribution system consists of a network of 1/4", 3/8" and 1/2" Weatherhead H104 wire braided Neoprene hydraulic hoses that supply hydraulic pressure to the wheels, the articulating joints, and the boom. Each boom or joint motor requires a Parker Fluidpower D1VW solenoid operated, three-position, directional control valve to control the flow direction. Each wheel motor needs a Parker Fluidpower MEV6 proportional directional control valve for smooth operation in forward and reverse and a Parker D31VW free-wheel / connect valve to control the free wheel and lock positions.

The auxiliary equipment required for the hydraulic system is a reservoir, a filter, and a relief valve. The reservoir is a custom fabricated 1.4 gallon box with internal baffles, a magnet, and a screen. The filter used is a 10 micron Parker 4Z618 and it is inline before the reservoir. The relief valve is a Parker RA 12005. It is located after the pump and it is set at 2300 psi for safety pressure relief.

The 2/3 scale Enabler model needs modifications so that the hydraulic pump and the engine can be located on the vehicle instead of having them external and connected by an umbilical cord. This modification consists of an extension on the body of the vehicle effectively changing one of the t-shaped body sections to a cross-shape.

Table of Contents

List of Figures

Introduction

The ME 4182 class meeting Winter Quarter, 1993 continued the work of other classes to design the Enabler, a lunar work vehicle. The focus was on designing a 2/3 scale working model to be built next quarter by the ME 4192 classes for NASA.

This report describes the specifications for the power unit and power distribution system as determined by Group 5, **Power Unit and Distribution System**. Nine components were examined for the power system. They are:

- Engine
- Pump
- Lines
- Valves
- Reservoir
- Heat Exchanger
- Filter
- Relief Valve
- Accumulator

Each component is individually analyzed to see what the operating requirements are and the equipment is selected to meet these requirements. This report details the hydraulic system required to power the Enabler.

The heart of the system is a Hagglunds Denison variable displacement, axial piston pump powered by a five horsepower Briggs and Stratton internal combustion engine converted to propane fuel. The hydraulic power is distributed to the six wheel motors, the six joint motors and the four boom motors and is controlled by two and three position servo valves. The hydraulic fluid, after flowing through a ten micron filter, returns to a reservoir where air and contaminants are removed.

Engine

The engine is a key component in any hydraulic system. The engine will provide the necessary input power to the hydraulic system and all of its members. The engine which will best meet the requirements of the enabler is a five horsepower, propane powered, Briggs and Stratton Model 130200 internal combustion engine. This engine will optimally satisfy the various constraints imposed by the hydraulic power system of the enabler. The engine will provide adequate power to supply all of the system components. This engine also will be the optimum design and size to be mounted on the body of the enabler. Finally this engine will meet the requirements of the indoor demonstration.

$$hp = P(Q) / 1714$$

The Briggs and Stratton engine meets the power constraints imposed on it by the hydraulic system. The power system of the enabler consists of sixteen individual hydraulic engines each of which require a certain flow rate and pressure to optimally perform their designated tasks. Using the flow rates for the sixteen engines and the above formula it can be shown that the enabler will require at least a 4.5 horsepower engine - a constraint well within the range of the five horsepower Briggs and Stratton. This horsepower figure is generated using the maximum pressure (P) in pounds per inch squared and flow (Q) in gallons per minute under the most adverse conditions (worst case scenario) in order to insure desired performance at all times.

The enabler will be constructed from a twelve inch inside diameter aluminum tube and moving over various, sometimes steep, terrain. The enabler will require its engine to be versatile enough to adapt to the constraints of this environment. Since the Briggs and Stratton is an internal combustion engine, it requires no bulky external power connections or potentially dangerous voltage (Aluminum being an excellent conductor). The size of the

Briggs and Stratton (see Appendix D) makes it easy to place in a engine compartment located at the rear of the craft (see Appendices A-5 to A-8). During operation, the enabler will be required to traverse various objects in its path. While traversing these objects the body of the enabler will be angled, requiring the engine, specifically the oil sump, to operate in this angled position. The Briggs and Stratton engine can be effectively operated at a constant angle of 25 degrees, reaching angles of up to 35 degrees intermittently.

A gasoline powered engine gives off many unpleasant exhaust gasses that can be decreased by conversion to propane power. The Briggs and Stratton can easily be converted from gasoline to propane operation, making it ideal for the indoor demonstration. The propane power system will be obtained at no charge from Combustion Labs Inc. in Riverdale, Ga. The propane power system will consist of a propane cylinder, a series of valves, a regulator, and finally an output port into the intake of the engine (Appendix A-9). For more detailed specifications, drawings and decision information see Appendix D.

Pump

The pump for the enabler is a Hagglunds Denison PV6 Variable Piston Pump. This pump supplies 2000 psi at a chosen variable flow rate of 5.4 gal/min. The Displacement is 0.88 inches^3/rev. The dimensions of the pump (see Appendix E) are small enough so that the pump fits inside the enabler chassis. The weight of the pump is 24 lbs. Additional features include Nine-piston rotating group, fast compensator response and quiet operation.

The requirements of the system on the pump include three different sub systems: the wheels, the articulating joints, and the boom. The calculations for the required flowrate are based on a "worst case" or the maximum power requirement of the combined sub systems. This worst case is defined as four wheels moving along with 4 articulating joints. The wheel motors need a flowrate of 0.7 gal/min, and the articulating joints need a flowrate

0.643 gal/min. If all eight of these motors are run at full power for five minutes the flowrate required would be 5.372 gal/min. Note that the boom will operate only when the enabler is stopped. Therefore, the worst case for the wheel-joint systems is assumed to be a larger drain on the hydraulic system than the boom sub system.

The PV6 pump has two specific features which make it a good choice for this hydraulic system. These features are the Adjustable Compensator and the Standard Maximum Volume Adjustment. The pump actually is able to supply up to 3000 psi however the Adjustable Compensator allows a preset pressure in this case 2000 psi to be maintained. Then a relief valve will be set for a pressure which is approximately 15 -20 percent higher than the 2000 psi operating pressure. The Compensator actually controls the swashplate angle. When the swashplate is perpendicular to the pistons, there is no stroke and therefore no displacement. However when the pump detects a lag in the pressure the Compensator moves the swashplate angle allowing the pistons to reciprocate ,and the pump generates more fluid output, resulting in a pressure increase.

The Standard Maximum Volume Adjustment allows the control of the flowrate leaving the pump. Using the equation:

$$Q = \frac{(D)(RPM)}{231}$$

[Q is flowrate in gal/min, D is displacement in inches^3/rev]

the maximum flowrate produced by the pump may be calculated. The speed supplied by the motor is 3600 rpm. Therefore the maximum flowrate possible is 13.7 gal/min. However, the needed flowrate is 5.372 gal/min for the worst case. So the pump maximum volume adjustment may be set for 5.372 gal/min. The adjustment basically reduces the displacement of the pump to correspond to the speed of the motor and the needed flowrate. This Standard Maximum Volume Adjustment also eliminates the need for an Accumulator.

Hydraulic Lines

The type hose that will be selected for the supply line is a 1/4 inch I.D. Weatherhead H104 hydraulic hose (SAE 100R1 Type AT) with an operating temperature range from -40°F to +212°F (-40°C to +100°C). The hose has a working pressure of 2750 psi and a minimum bursting pressure of 11000 psi. It has single steel wire braid reinforcement, a bending radius of 4 inches (17/32 inch O.D.), and the inner and outer hose material is made of Neoprene - a rubber material compatable with hydraulic oil. The return lines will also be selected as a Weatherhead H104 but one will have a 3/8 inch I.D. (11/16 inch O.D.) and the other will have a 1/2 inch I.D. (13/16 inch O.D.). The working pressure of the 3/8 inch return line is 2250 psi and a minimum bursting pressure of 9000 psi. The working pressure of the 1/2 inch return line is 2000 psi and a minimum bursting pressure of 8000 psi.

The Power Group will use hydraulics as a means of powering the Enabler. The hydraulic lines will serve as a way to transport fluid from the pump to the various motors located throughout the body of the Enabler. There, fluid flow is converted to mechanical rotation and the means by which the Enabler can move is established. Hydraulic horsepower is equal to the flow rate (gallons per minute) times the pressure (pounds per square inch) divided by a conversion factor of 1714. In order for the hydraulic system to be efficient and reliable, certain requirements and a safe configuration of the lines had to be met when selecting the type of hose.

The required working pressure of the Enabler will be 2000 psi and the hydraulic lines will have to meet this requirement. A single steel wire braid reinforcement will be sufficient for our pressure requirement. A maximum flow rate of 2.3 gal/min will also be required from any one supply line. Our calculations with a 1/4 inch supply line operating at

15 ft/sec would provide 2.3 gal/min and will be sufficient for our required flow rate. Higher flow velocities are certainly possible without adverse effects to the system. Hydraulic oil (petroleum base) will be the type of fluid used, so the inner surface of the hose will have to be compatible with the oil. A Neoprene inner and outer coating will be the best option for our application.

The cost of the hydraulic lines is another selection criteria. The lowest cost depends on the vendor and the most economical line configuration possible without affecting the performance of the Enabler. Due to space constraints, the hose will have to have a small bending radius in order to allow line connections in small areas, but not too small as it will incur excessive pressure losses. To have a cost effective line configuraion, two supply lines and two return lines both branching from T's and running the length of the Enabler will be used to supply power to the wheel motors. A single 1/4 inch line will supply power to the joint motors and a single 1/4 inch line will supply power to the boom motors. Both will have there own 3/8 inch return line.

Another problem is the possible torsion of hydraulic lines due to vehicle appendage rotation. The solution imposed is to actually coil the hydraulic lines through the body cavity of the Enabler to allow for rotation by winding the lines and unwinding them. A minimum number of coils is set at 2 coils per each section or 2 coils between each motor.

For additional information and detail drawings see Appendix F.

Servo-Valves

The servo-valves for the enabler are Parker Fluidpower directional control valves. For the wheels, two different types of valves will be needed: proportional directional control valves (Series MEV6) and free-wheel / connect valves (Series D31VW). For the articulating joints and the boom joints a solenoid operated, three-position, directional control valve (Series D1VW) will be used.

The two valves selected to control the direction of the fluid to the wheels will be solenoid operated, proportional, directional control valves and two position free-wheel / connect valves. The proportional control valve will have three positions. The center position will be stop, which will not let any flow through. The left position will be the forward position, which will drive the wheels in the forward direction. The right position will be the reverse position, which will reverse the flow of the fluid and drive the wheels in the opposite direction. The Series 6 valves provide precise and variable speed control without lurching when the wheels start moving. They are controlled by proportional solenoids, which provide an output flow that is proportional to the input signal. Because of the proportional signal, these valve will provide precise metering from minimum to full speed. The maximum flow for each wheel will be 0.7 GPM and this will correspond to a 12 VDC control signal. For the valve to be halfway open, a control signal of 6 VDC is required. The two position free-wheel / connect valve will be placed between the proportional valve and the wheel motor. This valve has two positions. The first position will be the free-wheel position, which will allow the wheel to move without flow being supplied to the motor. The connect position allows the wheel to operate as it normally would. The enabler will require one proportional control valve and one free-wheel / connect valve for every wheel.

The valves for the articulating joints will be a three position, solenoid operated, directional control valve. The three positions will be the same as the proportional valve for the wheels. The center position will be stop, the left position will be forward, and the right position will be reverse. These valves supply a sudden burst of fluid to the motors and not a gradual build up to the maximum flow as the proportional valves do. This sudden flow rate will cause the joints to jerk as they try to start turning. It is preferred to use the proportional control valves to eliminate the jerking of the joints and to control the different flow rates required by the motors, but because the proportional valves cost considerably more than the regular control valves, the above valves are recommended. A total of six valves are needed for the articulating joints.

The valves for the boom joints will be the same as the valves for the articulating joints. These valves will control the direction of the fluid supplied to the motors for the joints. Again it is preferred to use the proportional control valves used for the wheels to prevent the boom from jerking as it moves, but because of monetary restrictions we will recommend the normal directional control valve. A total of four valves will be required for the boom joints.

For more detailed specifications and details on the valves see Appendix G.

Reservoir

Due to the constraints of weight and space as well as the uniqueness of the system, it has been decided that the hydraulic reservoir should be custom made at the machine shop at Georgia Tech. This will allow for the optimum design for the system as well as a competitive cost for the reservoir. The total capacity is to be 1.4 gallons and the overall dimensions are approximately: Length- 9 in. Height- 6in. Width- 6 in. The complete drawings and dimensions as well as description can be found in Appendix H.

Typically the volume of a reservoir is 2 to 3 times the mean flow rate. This would correspond to a 13 gallon reservoir, and several vendors supply reservoirs in the 10 to 15 gallon range. However, the system does not require the reservoir to act as a heat exchanger so this volume can be reduced. Since weight is a main consideration, the 1.4 gallon reservoir will be sufficient.

The hydraulic system is part of a mobile vehicle so the reservoir must be pressurized in order to keep the outlet to the pump flooded. Most vendors do not supply this type of reservoir. This is another reason why the decision to custom make the reservoir was made.

Filter

The filter used is a hydraulic return line spin-on filter. The filter has a 10 micron cellulose medium capable of up to 20 GPM. The maximum pressure rating for the filter is 150 psi and the maximum temperature rating is 225°F. The filter is manufactured by Parker and is sold through Grainger. The filter has 3/4 in ports that will require couplings to accommodate the size difference of the return lines.

The main advantage to this filter is the fact that it is inexpensive and readily available. This filter will meet all the requirements of the system and is easily maintained.

Relief Valve

The relief vâlve for the system is a Parker RA1200S direct-operated pressure relief valve. Its maximum flowrate is 20 gal/min, its maximum operating pressure of 3000 psi. the relief valve will be manually set for 2300 psi. The position of the relief valve will be directly after the pump. This pressure relief will protect the system from harmful pressure increase due to a pump malfunction.

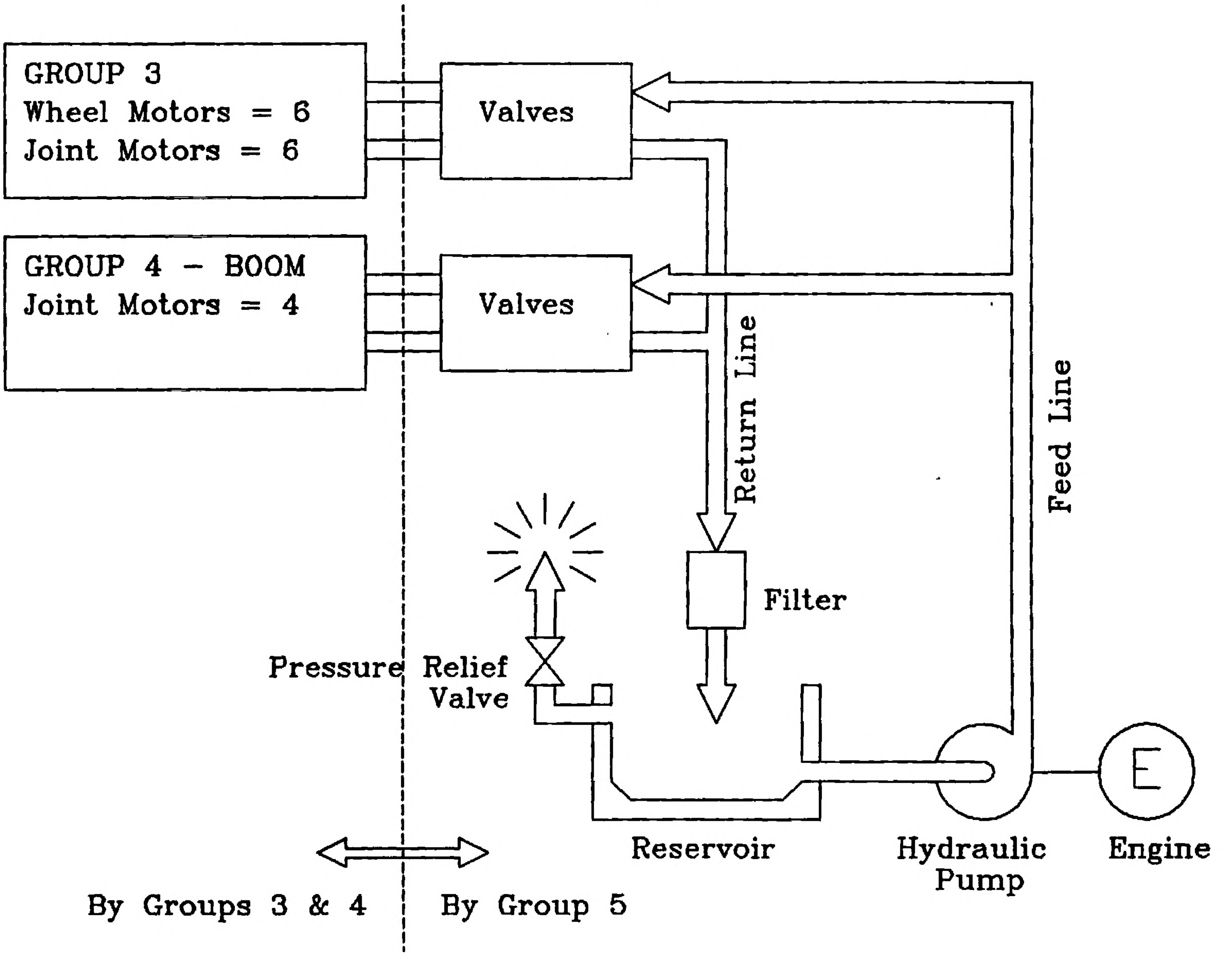
Bill of Materials

Component	#	Vendor	Model No.	Per Cost	Total Cost
Engine	1	Briggs & Stratton	130200	$253.50	$253.50
Pump	1	Hagglunds Denison	PV6	$714.00	$714.00
Proportional 3-way valves	6	Parker Fluidpower	MEV6	$700.00	$4,200.00
2-way valves	6	Parker Fluidpower	D31VW	$100.00	$600.00
3-way valve	10	Parker Fluidpower	D1VW	$150.00	$1,500.00
Relief valve	1	Parker	RA1200S	$100.00	$100.00
Filter	1	Parker	4Z618	$15.96	$15.96
Reservoir	1	Fabricated	N/A	$75.00	$75.00
1/4" Hydraulic hose	99 ft	Wheatherhead	H10404	$1.67 /ft	$158.65
3/8" Hydraulic hose	99 ft	Wheatherhead	H10406	$1.80 /ft	$171.00
1/2" Hydraulic hose	1 ft	Wheatherhead	H10408	$2.00 /ft	$2.00
3/8" cross fitting	2	Aeroquip	20806-6	$11.16	$22.32
1/4" female swival end	140	Aeroquip		$3.94	$551.60
3/8" female swival end	55	Aeroquip		$4.12	$226.60
1/2" female swival end	6	Aeroquip		$4.50	$27.00
1/4"-1/4" adapter	115	Aeroquip	2081 4-4	$1.00	$115.00
3/8"-3/8" adapter	30	Aeroquip	2081 6-6	$1.00	$30.00
1/2"-1/2" adapter	5	Aeroquip	2081 8-8	$1.00	$5.00
3/8"-1/4" adapter	19	Aeroquip	2081 6-4	$1.00	$19.00
1/2"-3/8" adapter	1	Aeroquip	2081 8-6	$1.00	$1.00
7/8"-1/4" adapter	24	Aeroquip	202702 10-4S	$3.50	$84.00
1/4" "T" fitting	13	Aeroquip	2090 4-4	$7.25	$94.25
3/8" "T" fitting	12	Aeroquip	2090 6-6	$7.78	$93.36
3/8" cross fitting	2	Aeroquip	2080 6-6	$11.16	$22.32
1/4" flow control valve	6	Aeroquip	2090 4-4	$3.60	$21.60

Appendices

APPENDIX A

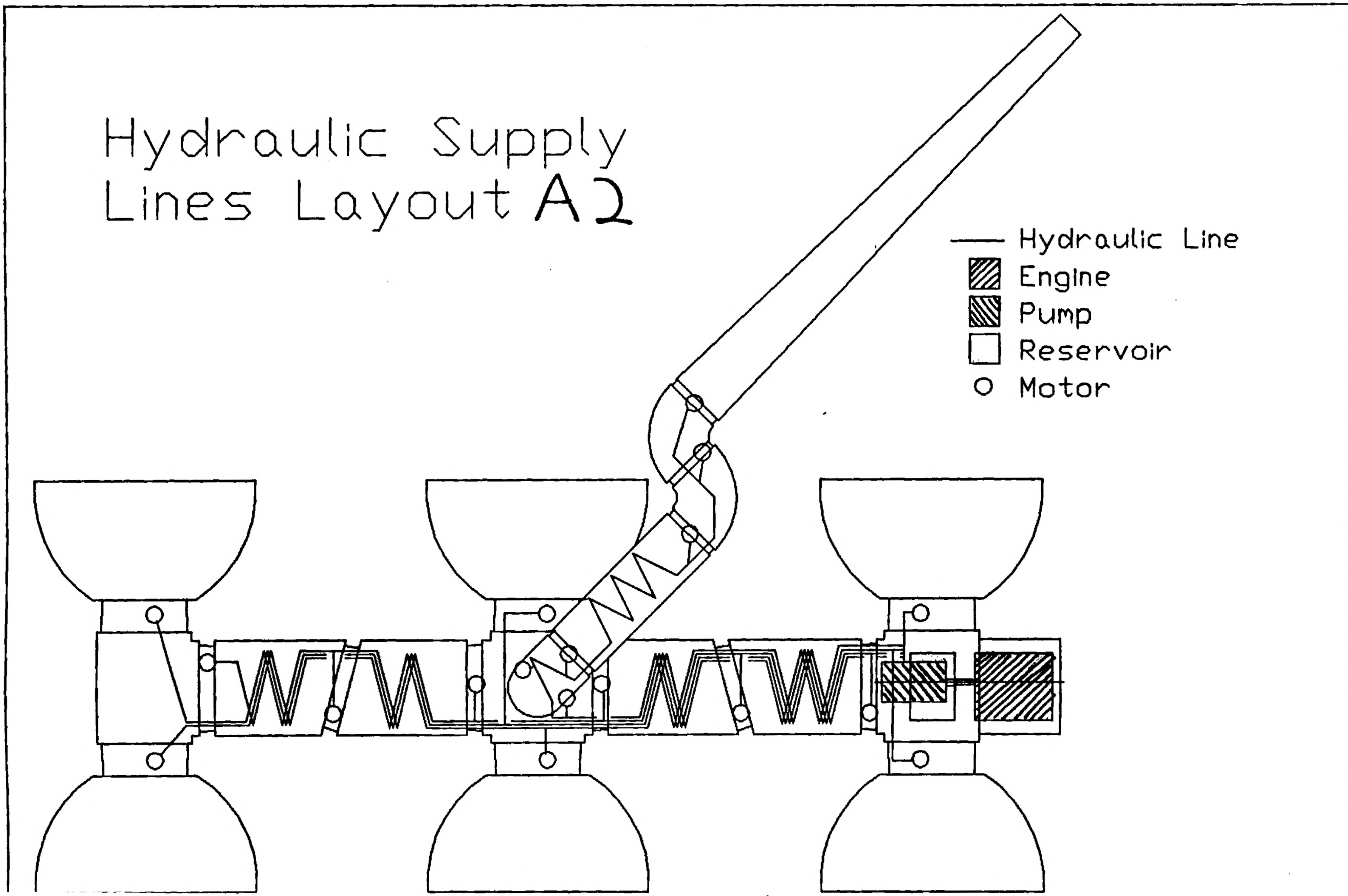
Flow Diagram A1



Power Supply – Group 5

ME4182d
Revision 4

Hydraulic Supply Lines Layout A2



APPENDIX B

Appendix B

Accumulator

If additional fluid flow above the capacity of the pump is required, then a hydraulic accumulator can be added to the system. The accumulator should be of the gas-piston type so that it can be mounted horizontally. Also, it should have a maximum working pressure above the pressure of the system.

In order to determine the additional volume of fluid that is required; the following equation can be used.

$$(GPM_{required}-GPM_{max}) \text{ X Time} = \text{Volume required} = Vw$$

After this volume is determined, the equation below can be used to calculate what size accumulator is needed.

$$V1 = \frac{vw(P3/P2)^{1/f}}{e(.95)[(P3/P2)^{1/n}-1]}$$

where:

P3=maximum system pressure

P2=minimum system pressure

P1=gas precharge pressure

f=charge coefficient

n=discharge coefficient

e=gas charge ratio P1/P2

V1=accumulator size required

Vw=required volume of fluid

In order to save weight, a smaller accumulator can be used in conjunction with a small gas bottle. However, it must satisfy the following equations.

$$V_{accumulator} + V_{bottle} > V_1$$

$$V_{accumulator} > (1.2)(Vw)$$

This type of setup will allow the whole volume of the accumulator to be used.

PHE Hydraulics inc. supplies various size accumulators of the piston variety. These sizes range from 1/2 gallon to 5 gallons. Also, this company can supply a gas bottle system compatible with the accumulator. The prices of these accumulators range from $100 to $300.

APPENDIX C

Appendix C

Heat Exchanger

The hydraulic system generates heat as the fluid flows through it. If the heat causes the temperature of the oil to rise too high, then a heat exchanger is needed. According to the estimates we made, there should not be a need for a heat exchanger. However, these are only estimates; so when the system is actually built, the system should be tested and then it should be determined if in fact a heat exchanger is actually needed. The present system is readily adaptable to the addition of a heat exchanger.

HEAT EXCHANGER

Pump -> 15 % Loss => 5 hp x .15 = .75 hp
.75 hp = 1908 Btu/hr

motors -> very efficent but assume 10% Loss
= 1200 Btu/hr

Relief Valve -> 1.48 x psi x GPM = 1.48 x 2000 x 5.4
= 15984 x .1 = 1598 Btu/hr

Valves -> 1.48 x psi x GPM = 1.48 x 2000 x .4
= 1184 x 6 = 7104 Btu/hr

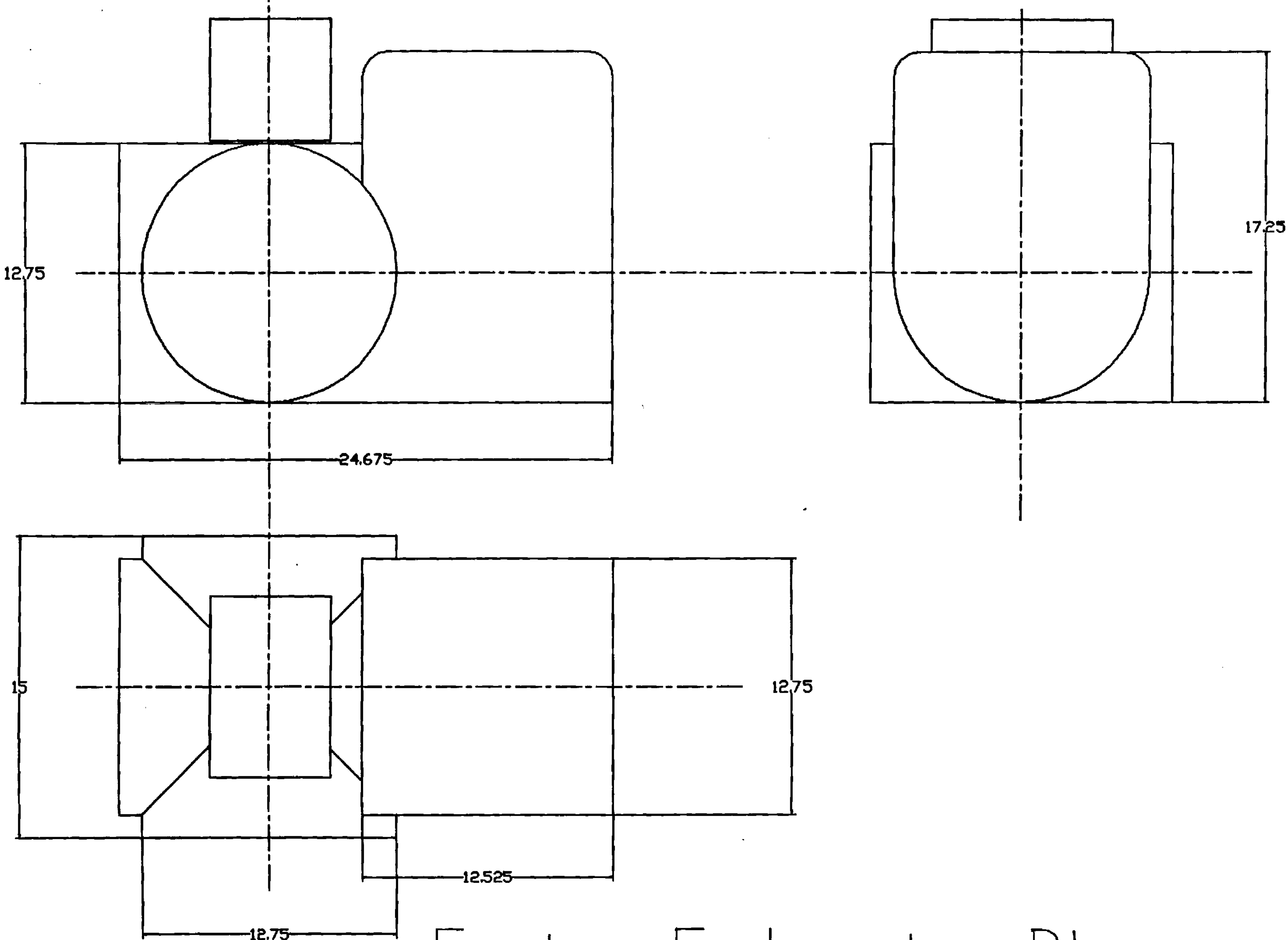
Lines 10% of hp = .5 hp = 1273 Btu/hr

Total 13,000 Btu/hr

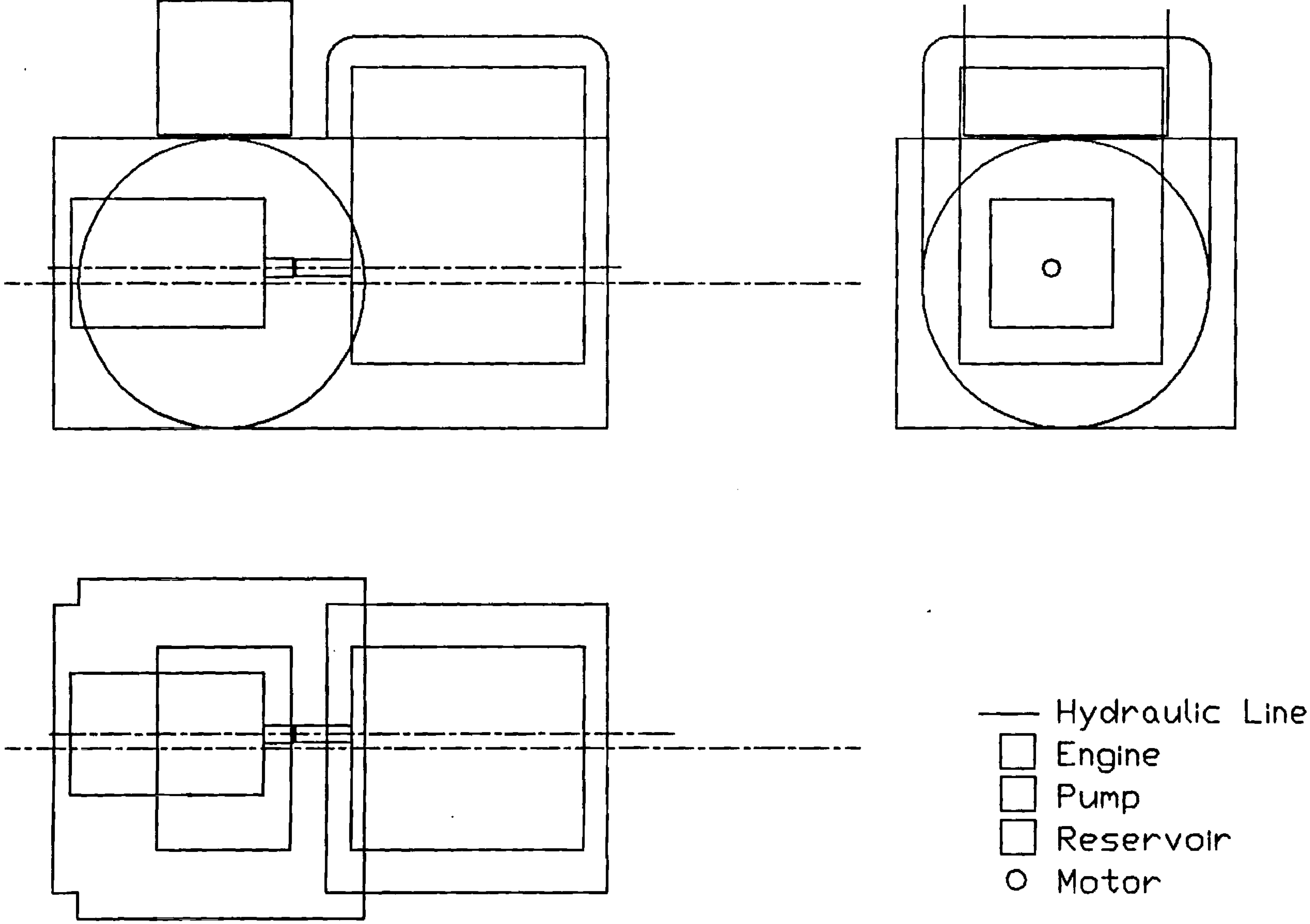
Books => > 15,000 Btu/hr then need a heat exchanger.
So NO Heat Exchanger Needed

However, these are only estimates. The System should be built and tested for heat output. If oil temp > 200°F then heat exchanger is needed

APPENDIX D

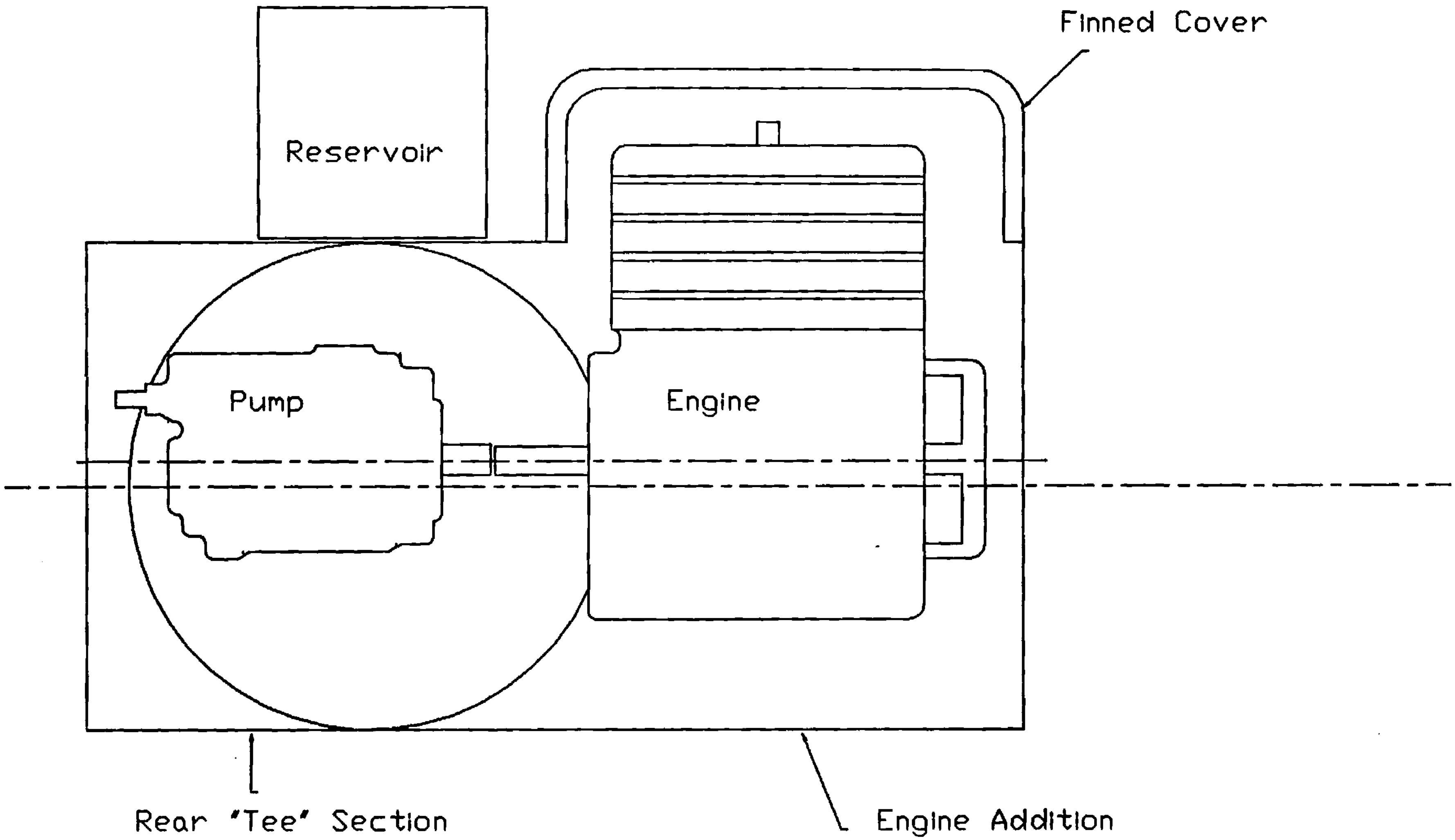


Engine Extension D1



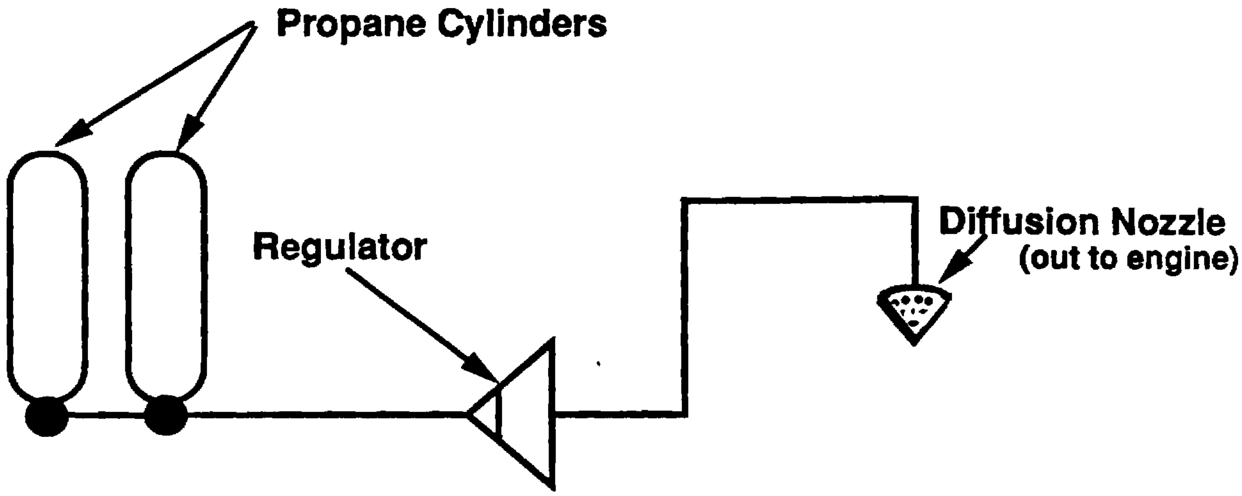
Engine Extension

Showing Engine, Pump and Reservoir D2



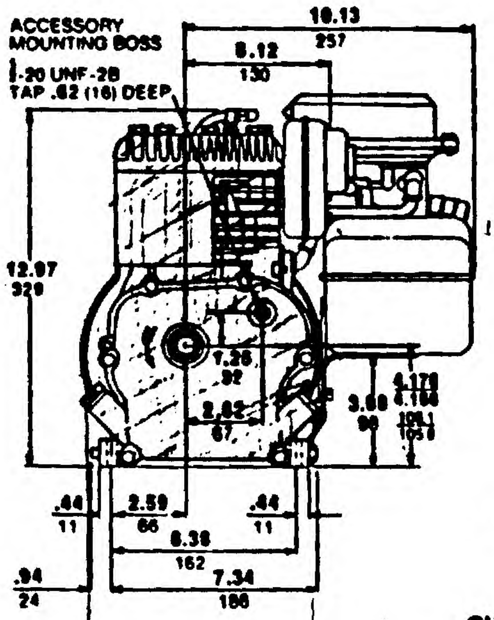
Cross Section D3

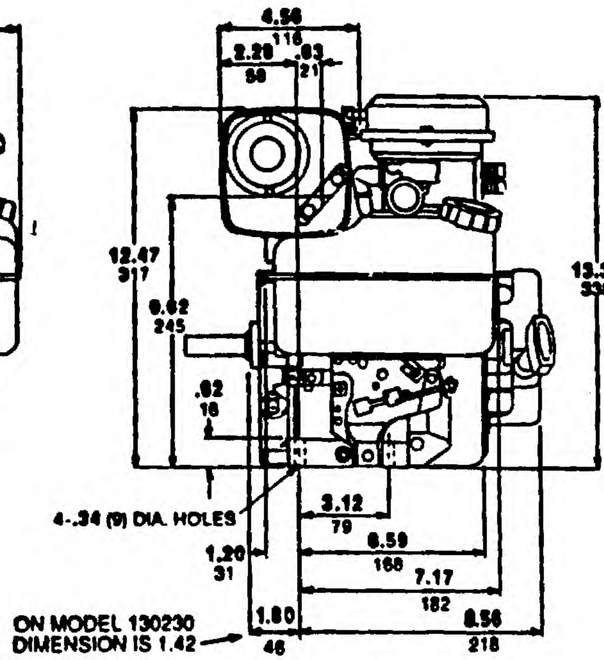
D4
PROPANE SYSTEM LAYOUT



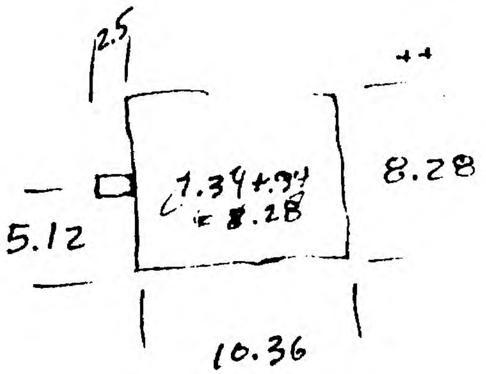
BRIGGS & STRATTON

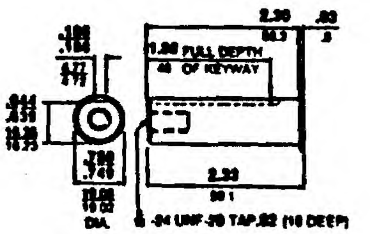
D5 Model Series 130200—5 HP



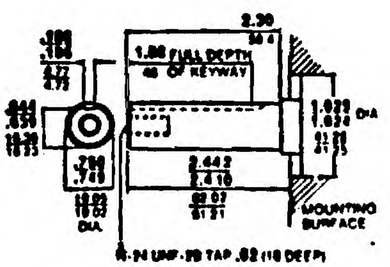


1.80
8.56
10.36

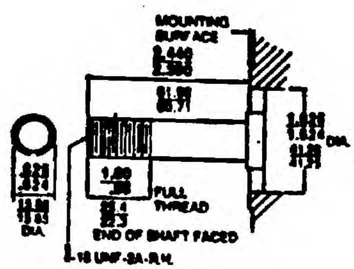




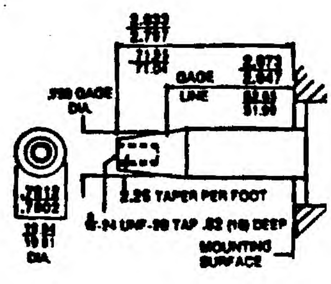
MODEL 130202 - C/S NO. 281679
3/4" DIA. .644" KEYWAY 1-7/8" LONG D&T 5/16-24



MODEL 130232 - C/S NO. 281730
3/4" DIA. .644" KEYWAY 1-7/8" LONG D&T 5/16-24

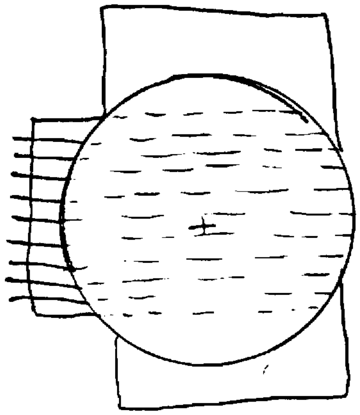


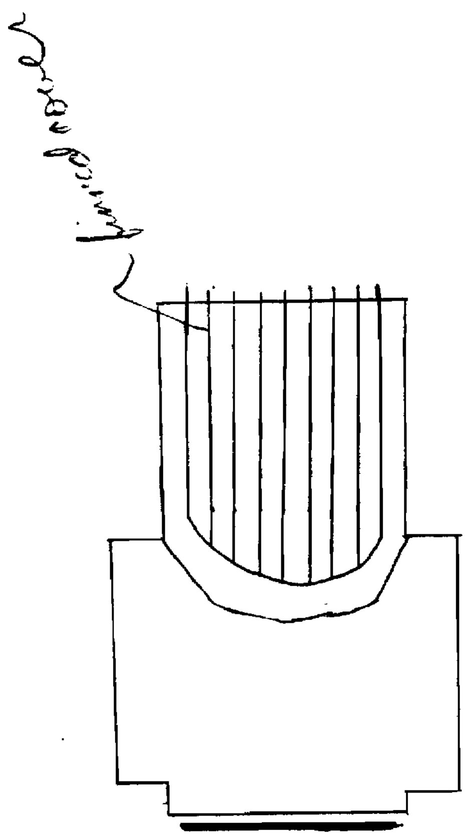
MODEL 130232 - C/S NO. 281732
5/8" DIA. 1" THREAD 5/8-18

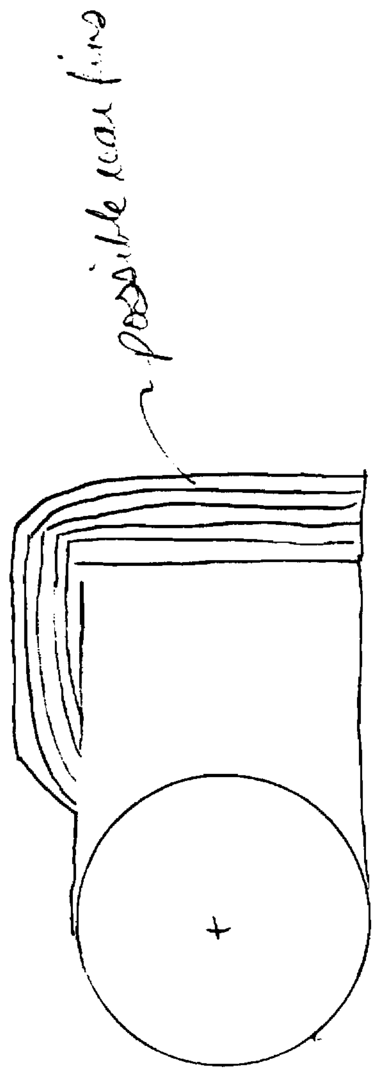


MODEL 130232 - C/S NO. 281733
3/4" DIA. TAPERED D&T 5/16-24

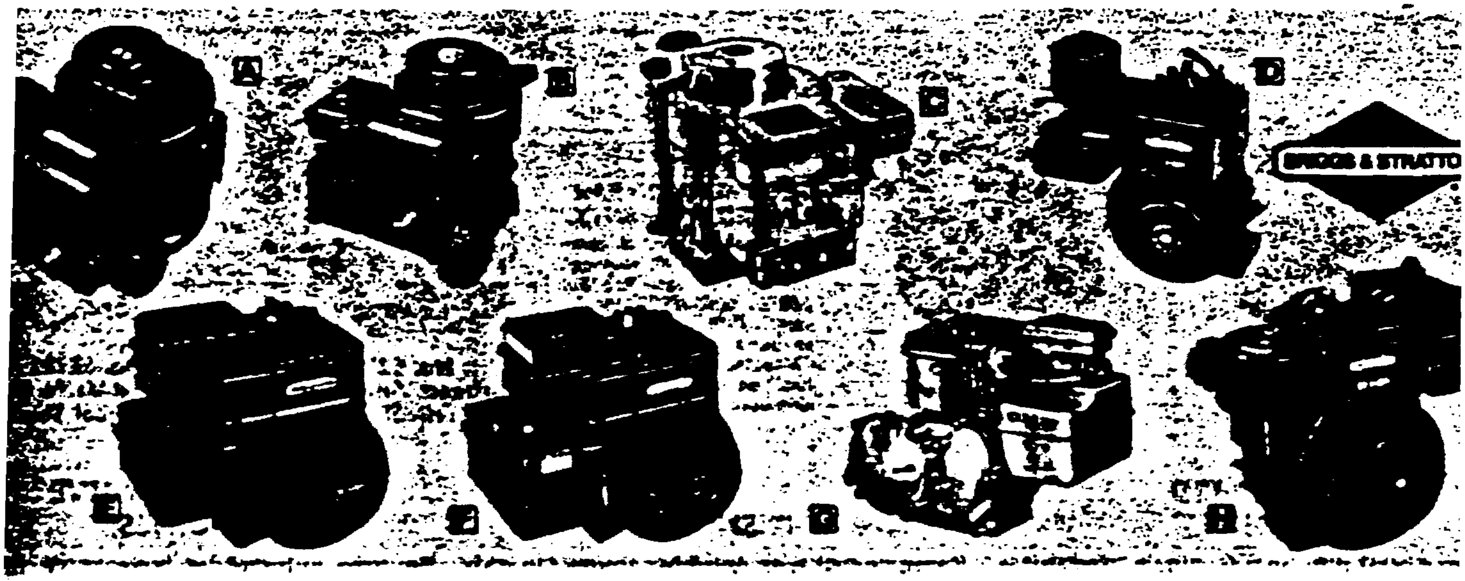
Engine Compartment Sketch







BRIGGS & STRATTON AIR-COOLED 4-CYCLE GASOLINE ENGINES



-cooled engines feature
maintenance-free Magnetron®
active storage-type electronic
ition. No servicing needed, except
rk plug

h silicon-copper aluminum alloy
inder and crankcase dissipate
gine heat

t treated ductile iron crankshaft
h integral counterweights and
minum drive gear

- Aluminum alloy main bearings (except one ball bearing on power take-off side on Nos. 32904, 42684, 42690, 42689 & 42692)
- Counterclockwise (CCW) rotation when viewed from power take-off side
- Cold headed steel intake valve and austenitic forged steel exhaust valve
- Oil Foam™ or paper pleated cleaner exceeds automotive standards for filtration
- Aluminized steel, quiet muffler has longer life
- Positive-type recoil starter with dust sealed, self-lubricating anti-rollback starter clutch

BRIGGS & STRATTON LIMITED WARRANTY

Briggs & Stratton warrants vertical-shaft engines (42688, 42881, and 32906) for 2 years. All other en for one year. Text of warranty available on request "Manufacturers' Warranties" on page opposite i back cover.

VERTICAL SHAFT ENGINES

. 42688 3.5 HP engine with 9.02 cu-in acement. Pulsa-jet carburetor with e mixture adjustment and auto

. 42691 5.0 HP engine with 12.57 displacement. Pulsa-jet carburetor single mixture adjustment and sin- alloy insert valve seat.

. 32906 8.0 HP engine with 19.44 displacement. Float carburetor with and power mixture adjustment. Sin- alloy insert for intake valve and less steel insert for exhaust valve. l tank.

HORIZONTAL SHAFT ENGINES

. 42687 2.0 HP engine with 6.65 cu-in acement. Pulsa-jet carburetor with mixture adjustment and sintered insert valve seat.

. 42689 and 42690 3.0 HP engines 7.75 cu-in displacment. Pulsa-jet car- r with front mount control panel sintered alloy insert valve seat.

d F Nos. 42683, 42684, 42692, and 5.0 HP engines with 12.57 cu-in dis- ment. Pulsa-jet carburetor with mount control panel and sintered insert valve seat.

. 32904 8.0 HP engine with 19.44 displacement. Float carburetor with and power mixture adjustment. Sin- alloy insert for intake valve and less steel insert for exhaust valve.

BRIGGS & STRATTON ENGINE SPECIFICATIONS

Key	HP	Stock No.	Cylinder Bore	Cylinder Stroke	Cu. In. Displ.	Shaft Dia.	Shaft Configuration	Carburetor Type	Start
VERTICAL CRANKSHAFT ENGINES									
A	3.5	42688	[illegible]	[illegible]	9.02	[illegible]	Two Woodruff Keyway	Pulsa-jet	Recoil
B	5.0	42881	[illegible]	[illegible]	12.57	[illegible]	Two Woodruff Keyway	Pulsa-jet	Recoil
C	8.0	32906	3	[illegible]	19.44	1	Straight Keyway	Float	Recoil
HORIZONTAL CRANKSHAFT ENGINES									
D	2.0	42687	[illegible]	[illegible]	6.65	[illegible]	Straight Keyway	Vacu-jet	Recoil
E	3.0	42689	[illegible]	[illegible]	7.75	[illegible]	Straight Threaded	Pulsa-jet	Recoil
E	3.0	42690	[illegible]	[illegible]	7.75	[illegible]	Int. Thrd. w/Keyway	Pulsa-jet	Recoil
F	5.0	42683	[illegible]	[illegible]	12.57	[illegible]	Straight Keyway	Pulsa-jet	Recoil
F	5.0	42682	[illegible]	[illegible]	12.57	[illegible]	Straight Threaded	Pulsa-jet	Recoil
F	5.0	42684	[illegible]	[illegible]	12.57	[illegible]	Int. Thrd. w/Keyway	Pulsa-jet	Recoil
G	5.0	42686	[illegible]	[illegible]	12.57	[illegible]	PTO 6:1 Reduction	Pulsa-jet	Recoil
H	8.0	32904	3	[illegible]	19.44	1	Straight Keyway	Float	Recoil

BRIGGS & STRATTON ENGINE ORDERING DATA

Key	HP	Applications	Overall Dimensions Length	Overall Dimensions Height	Overall Dimensions Depth	B&S Model	Stock No.	List	Each
VERTICAL CRANKSHAFT ENGINES									
A	3.5	Mower	10 [illegible]	9 [illegible]	12 [illegible]	92902-5015	42688	$[illegible]	$171.76
B	5.0	Mower	13 [illegible]	10 [illegible]	13 [illegible]	130902-1016	42881	275.00	261.82
C	8.0	Mower	14 [illegible]	11 [illegible]	13 [illegible]	194702-4015	32906	344.80	327.86
HORIZONTAL CRANKSHAFT ENGINES									
D	2.0	General	11 [illegible]	11 [illegible]	9 [illegible]	60102-4015	42687	203.40	193.23
E	3.0	Pump	11 [illegible]	11 [illegible]	9 [illegible]	80232-4036	42689	222.20	212.84
E	3.0	Pump	11 [illegible]	11 [illegible]	9 [illegible]	80232-4035	42690	222.30	211.19
F	5.0	General	11 [illegible]	12 [illegible]	9 [illegible]	130202-4015	42683	257.30	244.24
F	5.0	Pump	11 [illegible]	12 [illegible]	9 [illegible]	130232-4035	42682	263.40	250.23
F	5.0	General	11 [illegible]	12 [illegible]	9 [illegible]	130232-4036	42684	263.80	250.61
G	5.0	Mixer	11 [illegible]	12 [illegible]	13 [illegible]	130252-4049	42686	340.10	323.10
H	8.0	General	17 [illegible]	15 [illegible]	11 [illegible]	190432-5535	32904	397.40	377.53

Included for Comparison (Kohler is larger & More

KOHLER 4-CYCLE GASOLINE ENGINES

...iron construction of Kohler air-cooled, 4-cycle, 3600 RPM gas engines pro... ...ded durability, dependability and long ... widely used in industry, agriculture ... construction for garden tractors, liq... ...pumps, compressors, etc. Gives excel... ...cold weather starting, especially for ...

...y, positive starting is provided by ... patented automatic compression re... (ACR) feature. It releases com... ...sion during cranking and automati... ...ly restores full power when engine ...

...mpact, reliable gear drive starter on ...ectric start models eliminates belts and ...mpletely disengages to give more usea... ...horsepower.

...clearance baffling permits chaff, ... clippings and other airborne debris ...ow through and away from engine. This results in cool running engine and long engine life.

Float-feed carburetor provides for remote throttle and choke hookup. Air cleaner has replaceable dry-type paper element. Fuel tank capacity of all models is 1.5 gallons except No. 3Z971 which has no fuel tank.

Heat-treated ductile iron crankshaft has integral counterweights and includes 7/16"-20 x 1½" deep threaded hole in PTO end of shaft. No. 3Z971 has 5/8-18, 1½" deep threaded hole. Antifriction ball bearings on both ends of horizontal crankshaft, except No 3Z971 which also has a sleeve bearing on the flywheel end. CCW rotation when viewed from PTO end of crankshaft. Pistons are cam-ground, permanent mold aluminum alloy.

Other engine features include precision oil-bathed internal flyweight-type governor, Stellite® faced exhaust valves with rotators, Stellite® faced valve seats, oil filler tube and dipstick, and air int... precleaner element.

8 to 16 HP single cylinder, L-head engi... feature magneto ignition with easily ...justable external breaker points. S... button is mounted on breaker cover. H... tension spark plug wire with boot fa... tates all-weather starts. Electric s... models include battery ignition, 15 ... flywheel alternator system and ins... ment panel.

23 HP horizontally opposed twin cylin... L-head engine No 3Z971 features c... iron crankcase, full pressure lubrica... with spin-on oil filters, and alumin... dual mufflers for quiet operation and ... resistance. Has control panel consis... of choke, throttle, oil pressure gauge, ... meter, start button and stop switch—positioned for easy operation. Has batt... ignition with resistor-type spark plugs ... volt electric start with 15 amp flywh... alternator system.

... Stock No.	Cylinder Bore	Cylinder Stroke	Cu-In Displ.	Shaft Dia.	Shaft Configuration	Carburetor Type	Start	...
4Z356	2.94"	2.75"	18.64	1"	Straight Keyway	Float Feed	Recoil	6
3Z791	2.94	2.75	18.64	1	Straight Keyway	Float Feed	Electric	6
3Z793	3.25	2.88	23.85	1.125	Straight Keyway	Float Feed	Electric	6
3Z795	3.38	3.25	29.07	1.125	Straight Keyway	Float Feed	Electric	6
3Z969	3.50	3.25	31.27	1.125	Straight Keyway	Float Feed	Electric	6
3Z796	3.75	3.25	35.90	1.125	Straight Keyway	Float Feed	Electric	6
3Z971	3.00	3.00	57.73	1.437	Straight Keyway	Float Feed	Electric	N

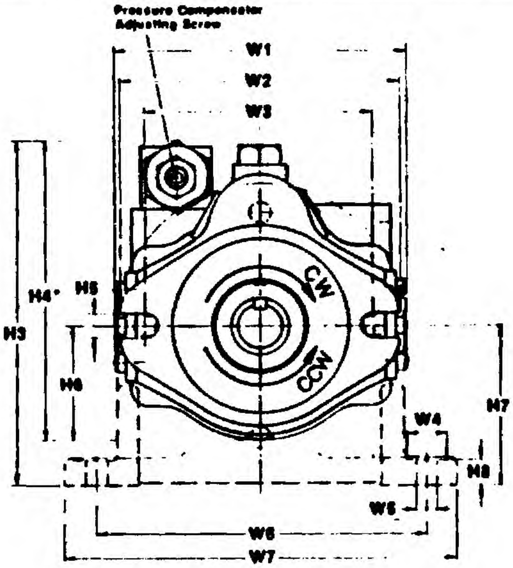
... Applications	Overall Dimensions, In. Length	Height	Depth	Spec. No.	Kohler Model	Stock No.	List	Each	...
General	14.50	20.06	15.89	30766	K181T	4Z356	$710.45	$666.34	
General	14.50	20.06	13.91	30767	K181S	3Z791	844.56	792.83	
General	19.125	18.562	15.66	46810	K241S	3Z793	1054.26	985.32	1
General	19.125	18.562	15.66	47739	K301S	3Z795	1105.17	1034.77	1
General	19.125	18.562	15.66	60426	K321S	3Z969	1181.95	1119.00	1
General	20.97	18.562	15.66	71309	K341S	3Z796	1233.07	1153.67	1
General	20.5	22.06	18.94	36340	K582S	3Z971	2432.40	2283.89	2

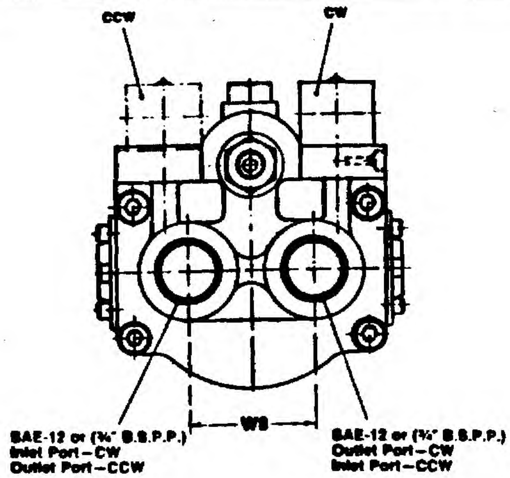
GAS ENGINE CHAIN SAWS AVAILABLE

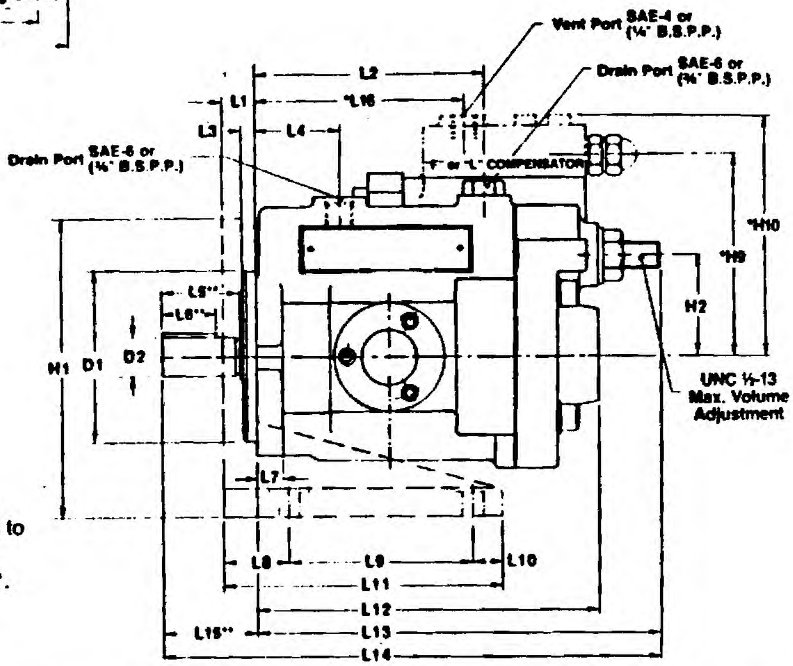
...ws listed in this catalog can handle nearly ...tting job. Medium-duty chain saws tackle the ...owner's tree trimming chores. Heavy-duty saws ...cutting jobs of farmers and forestry crews, as well as being precision built for the heavy-duty ...mands of tree service operators who cut every work day. Complete product descriptions available.

See Index under Chain Saws.

APPENDIX E







Installation Drawing 23-9596
Assembly Drawing SD-01537
Optional Foot Bracket S14-00128

*"F" or "L"—Ventable Compensator
"T"—Power Limiter (See page 10)

Note: Two bolt mounting flange conforms to SAE "A" specifications, except for dimensions marked with asterisks.

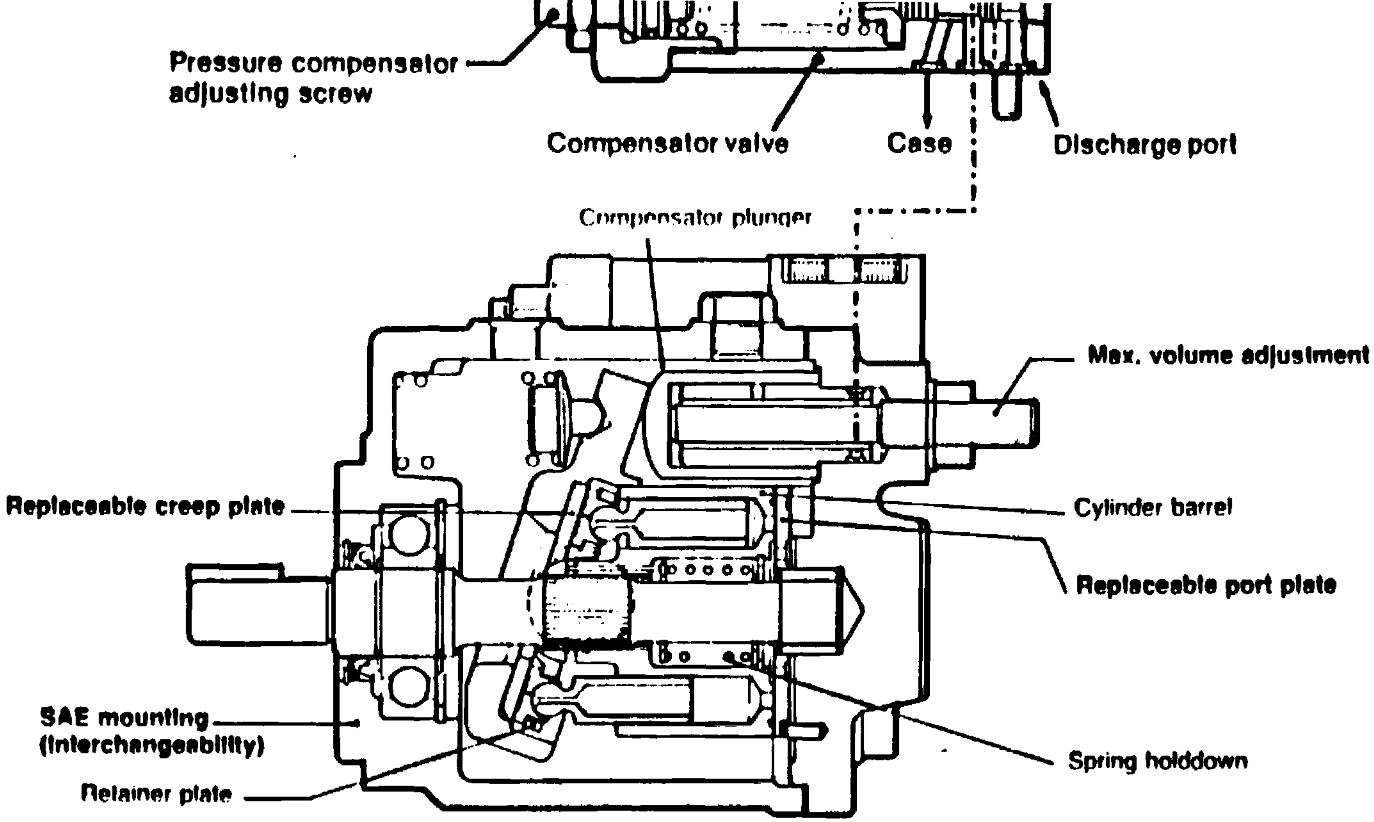
Installation Dimensions
PV6

	L1	L2	L3	L4	L5	L6	L7	L8	L9	L10	L11	L12	L13	L14	L15	L16	H1	H2	H3
Inch	0.59	4.27	0.25	1.57	See Below	See Below	0.49	1.19	3.50	0.56	5.25	6.43	8.1	9.89	See Below	4.11	5.64	1.89	6.48
mm	15.1	108.5	6.4	39.9	See Below	See Below	12.4	30.2	88.9	14.3	133.4	163.3	205.7	251.2	See Below	104.4	143.2	48	164.7

	H4	H5	H6	H7	H8	H9	H10	W1	W2	W3	W4	W5	W6	W7	W8	D1	D2
Inch	5.62	0.47	2.15	3.0	0.46	3.78	4.56	5.43	5.12	4.17	0.75	0.41	6.13	7.25	2.25	3.250 3.248	See Below
mm	142.7	11.9	54.6	76.2	11.7	96	115.8	138	130	106	19	10.3	155.6	184.2	57.2	82.55 82.50	See Below

Shaft Dimensions
PV6

Shaft Code	Shaft Type	DIM.	L5	L6	L15	**Key Shaft Dimensions: D2 Dia.	Sq. Key Section	Dim. Overkey	Key Lgth.	**Spline Shaft Dimensions: Major Dia. D2	No. Teeth	Pitch	Press. Angle	Spline Lgth. L6
1	Splined SAE-B	inch	1.30	X	1.63	X	X	X	X	.875/.853	13	16/32	30°	0.98
		mm	33.0		41.4					22.2/21.7				24.9
2	Keyed SAE-A	inch	1.46	1.00	1.79	.750/.749	.1884/.1874	.832/.827	1.00	X	X	X	X	X
		mm	37.1	25.4	44.5	19.05/19.02	4.785/4.760	21.13/21.00	25.4					



PRESSURE COMPENSATOR CONTROL WITH MAXIMUM VOLUME ADJUSTMENT

The C and F pressure compensator control allows the pump to deliver full volume from the outlet port until the pressure rises to the value set by the control. One turn clockwise of the pressure compensator adjusting screw represents a pressure increase of approximately 650 PSI (44.8 bar).

The control then reduces the pump volume to that required by the system while maintaining the preset pressure at the outlet port. The stroking piston is controlled by a 3 way valve which is shifted by discharge pressure.

The fast response (typically 50 ms off stroke and 120 ms on stroke) and high flow capacity of this valve holds pressure overshoot and undershoot to a minimum.

The minimum compensating pressure is 130 PSI (9 bar). An adjusting screw complete with locknut allows the pump volume to be set between maximum and zero.

Clockwise rotation pumps have the pressure compensator control located on the left side of the pump body, on counter clockwise rotation pumps the control is on the right side.

APPENDIX F

Line Configuration

1/4" cross sec. area = 4.9×10^{-2} in^2 Q = 530.14 in^3/min = 2.29 gal/min

5/16" = 7.67×10^{-2} in^2 Q = 828.35 in^3/min = 3.586 gal/min

1/2" return worst case 5.8 gal/min = 22.33 in^3/sec velocity = 113.73 in/sec = 9.5 ft/sec

3/8" Q = 5.164 gal/min

configuration

wheels assume T split occurs prior to rear wheel base centerline

2 × (233.732 in) + 2ft = 40.95 ft of 1/4 in add 4 ft

40.95 ft of 3/8 in " "

joints 246.482 + 2 ft = 22.54 ft 22.54 ft of 1/4 add 3 ft

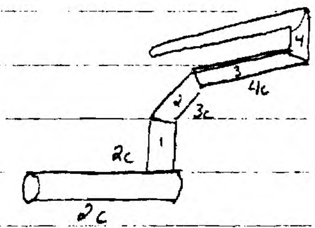
single supply line
with 2 turns between joints
except 1 turn for 1st joint

22.54 ft of 3/8 " "

boom 284.116 + 2 ft = 25.67 ft

25.67 ft of 1/4 add 2 ft

25.67 ft of 3/8 " "



lengths	
1	7"
2	2
3	5
4	30

additions

supply 6 in 3/8 from pump to cross.

6 in 3/8 from cross to T (wheel)

6 in 1/4 from cross to T (joint)

? 8 in 1/4 for each T conn. to wheel (4 ft) ?

return 12 in 1/2 hose

valves boom 4 3 pos. valves

joints 6 3 pos.

wheels 6 3 prop. pos. + 6 2 position

Pressure Loss Calculation

pressure loss

assume coils to be minor losses

wheels one side total length of supply line 41 + 4 ft = 45 ft / 2 = 22.5 ft

max pressure loss at max operation - 4 wheels at front end

flow = 1.4 gal/min

oil factor = .85

loss due to length

Q = 2.1, water, 1/4, 100 ft = 17 lbs/in²

17 × 22.5/100 = 3.825 lbs/in²

3.825/.85 = 4.5 lbs/in²

relative radius = 16 (4 inch bend)

43.5 × .25 = 10.875 in

40 × 10.875/100 = 4.35 lbs/in²

4.35/.85 = 5.117 lbs/in²

4 × 5.117 + 2 × 4.5 = 29.468 lbs/in²

loss due to valves 16.6 lb/in² × 4 = 66.4

total wheel loss = 95.868

(over)

$Q = 1.6\ gal/min$

joints total length = 23 + 3ft = 26 ft Δt p drop = 20 lbs/in²

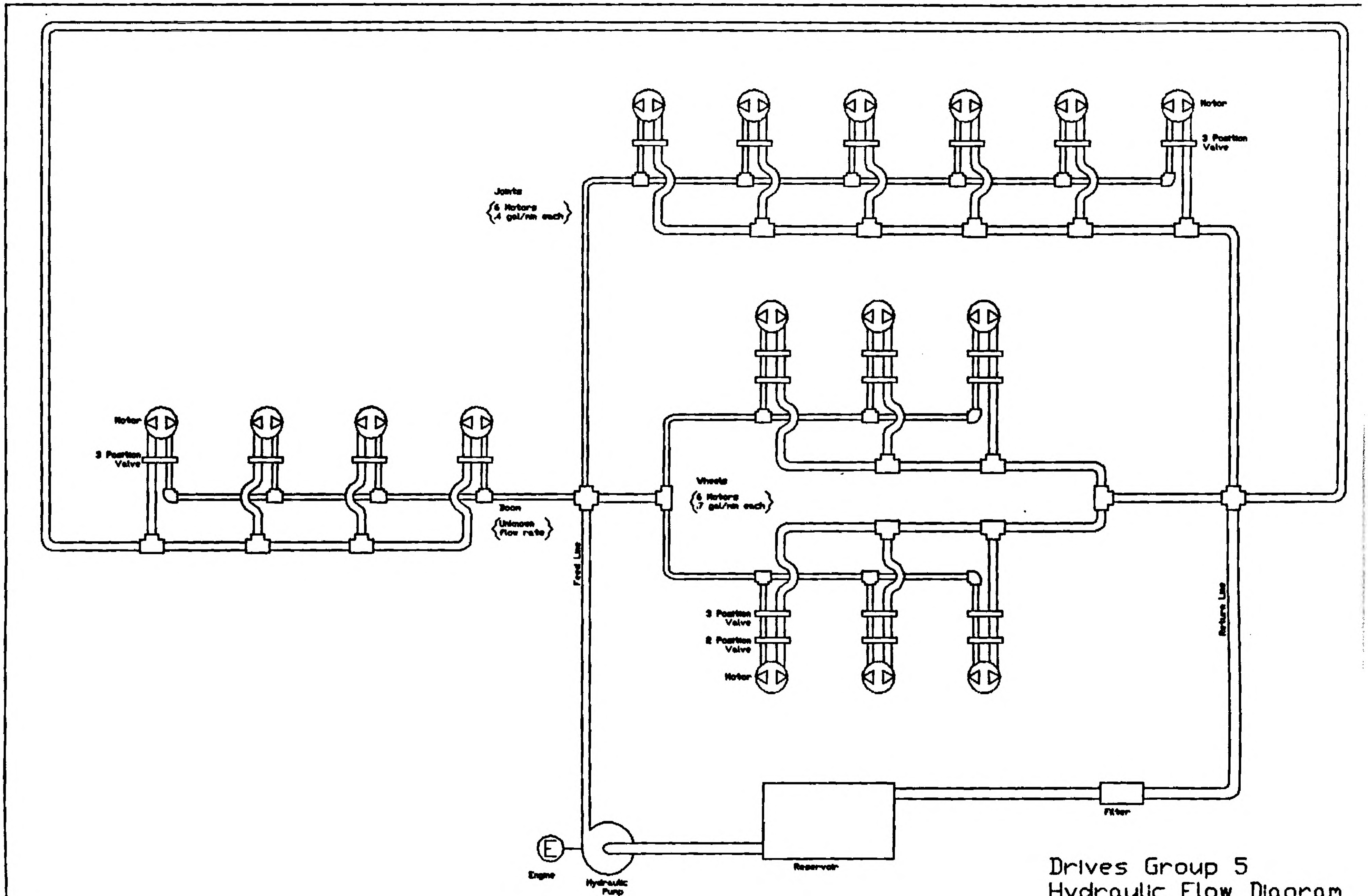
$20 \times 26/100 = 5.2\ lbs/in^2$

$5.2/.85 = 6.117\ lbs/in^2$

4 valve loss = 66.4

total = 72.517 psi

worst case senario for total loss = $\boxed{168.385\ psi}$



Drives Group 5
Hydraulic Flow Diagram
Revision 1
F1

Application Data

Hose Selection Chart

Hose Number	H166	H369	H747	H017	H243	H209	H213 H448	H324	H066	H069	H169
Usage	High Temp. Truck	Truck Freon 12	Freon 12	Gen. Purp. Hyd.	Gen. Purp. Hi-temp	Car Wash	High Temp. Truck	Power Steering	High Temp. Truck	Truck and Hyd.	Truck and Hyd.
Meets	DOT All	—	—	USCG[2] MSHA	—	—	DOT A1	—	DOT All	—	USCG[1,2] MSHA
SAE No.	J1402 Type All	J51 Type B2	J51 Type D	100R3	100R14 Type A	—	J1402 Type A1	J188 Type II	J1402 Type All	J1402 Type All 100R5	—
Temperature Range	−40°F +300°F	−20°F +250°F	−40°F +250°F	−40°F +212°F	−65°F +450°F	−40°F +200°F	−40°F +300°F	−40°F +250°F	−40°F +300°F	−40°F +300°F*	−40°F +212°F
Inner Tube	Nitrile	Nitrile	Rubber Nylon Rubber	Neoprene	Teflon®	Nylon 11	CPE	Neoprene	Nitrile	Nitrile	Neoprene
Reinforcement	1 Fiber & 1 S.S. Braid	1 Fiber & 1 Steel Braid	2 Fiber Spiral	2 Fiber Braids	1 S.S. Braid	1 Fiber Braid	1 Fiber & 1 Wire Braid	2 Fiber Braids	1 Fiber & 1 Steel Braid	1 Fiber & 1 Steel Braid	1 Fiber & 1 Steel Braid
Outer Cover	Fiber Braid	Fiber Braid Red	Butyl	Neoprene	Stainless Steel	Poly-urethane	Fiber Braid	Neoprene	Fiber Braid	Fiber Braid	Neoprene
Hose I.D.	MAXIMUM RECOMMENDED OPERATING PRESSURE — PSI										
3/16	1500	500					1500		2500	3000	3000
1/4	500			1250	3000	2250	1500		2250	3000	3000
5/16	500	500	500		2500	1750	1500		2000	2250	2250
3/8				1125	2000	1350		1400			
13/32	500	500	500				1250		2000	2000	2000
1/2	450	500	500	1000	1750	1000	1000		1750	1750	1750
5/8	450	500	500				750		1500	1500	1500
3/4				750	1000						
7/8	250	500					400		800	800	800
1				560	1000						
1 1/8		500								625	625
1 1/4				375							
1 3/8		350								500	500
1 13/16										350	350
2 3/8										350	
3										200	
Coil-O-Crimp®	Pages 68-70	Pages 68-70	Page 62	Pages 39-50	Pages 63-67	Pages 63-67		Pages 39-50	Pages 68-70	Pages 68-70	Pages 68-70
Reusable	Pages 94-99	Pages 94-99		Page 92			Page 100		Pages 94-99	Pages 94-99	Pages 94-99

Note: Teflon is a registered trademark of DuPont.
[1]Fuel and Lube Accepted [2]Fluid Power Accepted *Sizes -4 through -12, +250° Sizes -16 through -48.

Hose Selection Chart

Application Data

APPLICATION

Hose Number	H104	H435	H436	H114	H145	H146	H300	H425	H095	H430	H439	H470
Usage	Hyd.	Hyd.	Hyd.	Ag. Hyd. & Hyd. Synthetic	Hyd.	Ag. Hyd. & Hyd. Synthetic	Hyd.	Hyd.	Synthetic Hyd.	High Pressure Hyd.	High Pressure Hyd.	High Pressure Hyd.
Meets	USCG[1,2] MSHA	—	—	MSHA	MSHA	—	MSHA	USCG[1,2] MSHA	MSHA	USCG[2] MSHA	USCG[2] MSHA	MSHA
SAE No.	100R1AT	100R7 Non-Cond.	100R7	Exceeds 100R1	Exceeds 100R1	Exceeds 100R1	—	100R2AT	—	100R12	100R12	—
Temperature Range	−40°F +212°F	−40°F +200°F	−40°F +200°F	−40°F +250°F	−40°F +250°F	−65°F +250°F	−40°F +212°F	−40°F +212°F	−40°F +200°F	−40°F +250°F	−40°F +250°F	−40°F +250°F
Inner Tube	Neoprene	Nylon 11	Nylon 11	Hytrel	Nitrile	Hytrel	Neoprene	Neoprene	EPDM	Neoprene	Neoprene	Neoprene
Reinforcement	1 Steel Braid	2 Fiber Braids	2 Fiber Braids	1 Steel Braid	1 Steel Braid	1 Steel Braid	2 Spiral 1 Wire Braid	2 Steel Braids	2 Steel Braids	Multi Spiral Steel	Multi Spiral Steel	Multi Spiral Steel
Outer Cover	Neoprene	Poly-urethane Orange	Poly-urethane Black	Neoprene	Neoprene	Polyester Braid	Neoprene	Neoprene	EPDM Green	EPDM or Neoprene	EPDM or Neoprene	EPDM
Hose I.D.	MAXIMUM RECOMMENDED OPERATING PRESSURE — PSI											
¼	2750	2750	2750	3000	3000	3000	3000	5000	5000			
5/16		2500	2500									
⅜	2250	2250	2250	3000	3000	3000	3000	4000	4000			
½	2000	2000	2000	3000		3000	3000	3500	3500	4000	4000	5000
⅝	1500						3000	2750				
¾	1250	1250	1250				3000	2250	2250	4000	4000	5000
1	1000	1000	1000					2000	2000	4000	4000	5000
1¼	625							1625	1625	3000	3000	5000
1½								1250	1250	2500	2500	5000
2								1125	1125	2500	2500	5000
Coll-O-Crimp®	Pages 39-50	Pages 63-67	Pages 63-67	Pages 39-50	Pages 39-50	Pages 39-50	Pages 39-50	Pages 39-50, 71-78	Pages 71-78	Pages 71-78	Pages 71-78	Pages 79-80
Reusable		Pages 104-105	Pages 104-105					Pages 106-109				
Hose Number	H104	H435	H436	H114	H145	H146	H300	H425	H095	H430	H439	H470

[1]Fuel and Lube Accepted
[2]Fluid Power Accepted

WEATHERHEAD®

DANA

Application Data

Hose Selection Chart

CHART 1. Hose Flow Rate vs Pressure Drop

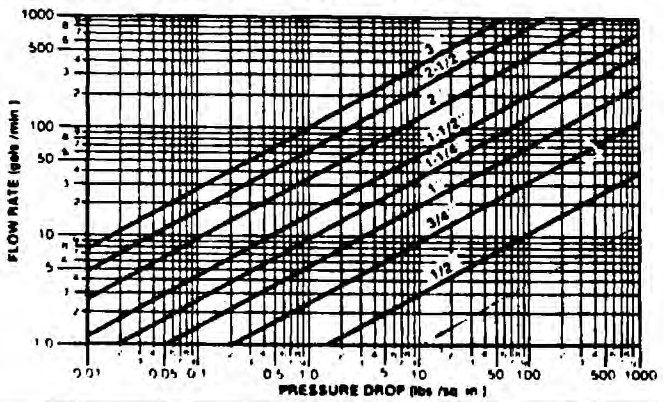
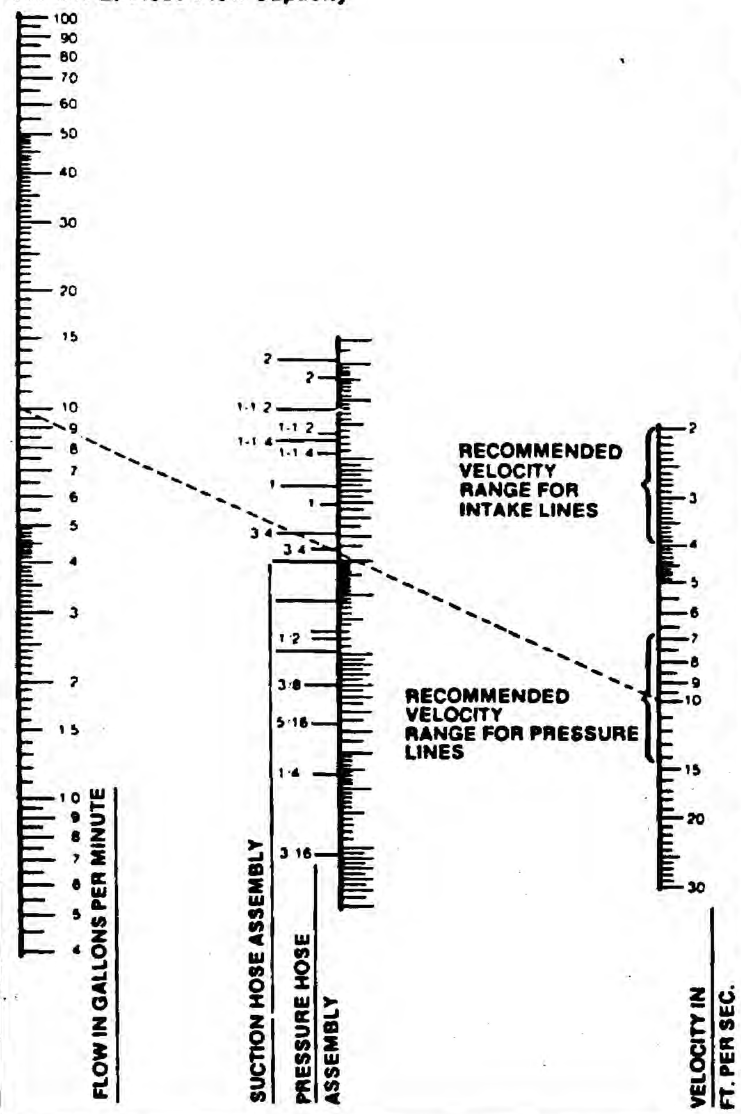


CHART 2. Hose Flow Capacity



There are several factors which enter into the selection of a hose sized such that it will provide the desired rate of flow at the required pressure; these are:

- **Hose size**
- **Hose length**
- **Hose ends & fluid conveyed**
- **Bends**
- **Static head pressure**

Hose Size

Undersized pressure lines produce excessive pressure drop with attendant energy loss and heating, and undersized suction lines cause cavitation at the pump inlet. Oversized hose assemblies, on the other hand, are excessively costly and generally too heavy.

In selecting hose for hydraulic systems, the following empirical values can be used to achieve minimum pressure drop consistent with reasonable hose size (see Chart 2):

Velocity of pressure lines . . . . . . .7 to 15 ft/sec
Velocity of short pressure lines . . . to 20 ft/sec
Velocity of suction lines . . . . .2 to 5 ft/sec

To use Chart 2, lay a straight-edge across the chart as shown by the dotted line. Always use the next larger size hose shown if the line passes between sizes listed.

Hose Length

Chart 1. gives the pressure drop in different-sized hoses based on hoses of 100-foot length, and is based on water as the fluid conveyed. For hoses of a different length, these values must be corrected. For example, a 100-foot length of ½" hose causes a pressure drop of 100 lbs./in² at a flow rate of 10 gal./min. If the hose in question is 50 feet long, the pressure drop derived from Chart 1. must be corrected by multiplying the value by the ratio of the actual length to 100 feet, or 50/100, or 0.5. Therefore, the actual pressure drop caused by a 50-foot length of ½" hose, at a flow rate of 10 gal./min., is 50 lbs./in² (0.5 x 100 = 50 lb./in²).

Hose Ends & Fluid Conveyed

In most cases, the end fitting openings are slightly smaller than the hose itself. However, this varies widely with hose end designs from "full-flow" ends which have the same ID as the hose, down to as much as ⅛-in. smaller ID than the hose bore. To allow for this, assume a 10-to 15% greater flow rate than actually measured in the system when determining pressure drop.

Chart 1. is based on water as the fluid conveyed, and for other fluids it is necessary to correct for the difference in specific gravity and viscosity. Chart 3. lists common fluids, their specific gravities, viscosities, and corresponding correction factors.

To determine the pressure drop for a specific fluid, first determine the pressure drop from

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Chart 1. for the hose length then divide this by the correction factor found in Chart 3.

For example, the 50-foot length of ½" hose just described had a pressure drop of 50 lbs./in² at a flow of 10 gal./min. of water. To determine the pressure drop if #2 fuel oil is the fluid conveyed, divide by 0.752 (from Chart 3) . . . 50 ÷ 0.752 = 66.5 lbs./in² pressure drop. If, on the other hand, the fluid conveyed is Type #3 gasoline, the pressure drop would be 50 ÷ 1.19 = 42 lbs./in².

Chart 3. Fluid Flow Correction Factors

Liquid	Specific Gravity	Viscosity Centistokes (CS)	Viscosity Centipoises (CP)	Correction Factor R
Acetic Acid 100%	1.05		1.3	0.975
Acetic Acid 70%	1.07		2.7	0.843
Ammonia liquid 100%	0.66	0.30		1.290
Ammonia liquid 26%	0.907		1.3	0.943
*Asphalt @ 120 F	1.40		300	0.350
*Beer	1.01	1.15		0.990
Benzene (Benzol)	0.88	.744		1.08
Brine Calcium Chloride 25%	1.23	3.80		0.78
Brine Sodium Chloride 25%	1.19	2.07		0.88
Butyl Alcohol	0.81	3.64		0.783
*Castor Oil	0.96	900		0.27
*Crude Petroleum (Typical)				
1 Pennsylvania Crude @ 100 F	0.80		3	0.78
2 California Crude @ 150 F	0.915		9	0.64
3 #33 API Crude @ 100 F	0.86	7.2		0.685
4 Texas Crude @ 150 F	0.875		3	0.792
5 Mexican Crude @ 150 F	0.96		550	0.287
Decane - n	0.73	1.24		0.975
Ethyl Alcohol 100%	0.789		1.25	0.93
Ethyl Alcohol 95%	0.808		1.45	0.904
Ethyl Alcohol 40%	0.939		3.00	0.807
Ethyl Glycol	1.12		24.00	0.55
Formic Acid	1.22			0.94
*Fuel Oils				
No 1 @ 100 F Sp Gr .82 to .95 Visc 30 to 40 SSU	0.88	2.45		0.85
No 2 @ 100 F Sp Gr .82 to .95 Visc 35 to 50 SSU	0.88	4.50		0.752
No 3 @ 100 F Sp Gr .82 to .95 Visc 55 SSU max	0.88	8.6		0.66
No 5 @ 100 F Sp Gr .82 to .95 Visc 60 to 450 SSU	0.88	55.0		0.47
No 6 @ 122 F Sp Gr .82 to .95 Visc 430 to 2900 SSU	0.88	38.0		0.493
*Gasoline (representative)				
Type #1	.74	.88	-	1.04
Type #2	.72	.64	-	1.11
Type #3	.68	.46	-	1.19
Glycerine (Glycerol) - 100% @ 150 F	1.26	-	75.0	0.45
Glycerine & Water - 50%	1.13	-	6.5	0.717
Heptane - n	.684	0.60	-	1.16
Hexane - n	.66	0.49	-	1.21
Hydrochloric Acid - 31.5%	1.16	-	1.92	0.92
Isobutyl Alcohol	0.817	-	3.90	0.745
Isopropyl Alcohol	0.785	-	2.20	0.828
Kerosene	0.80	2.23	-	0.892
Lubricating Oil (Machine Oil)	0.90	-	198	0.35
Lubricating Oil (Automotive)	0.893	-	110	0.39
Methyl Alcohol (Methanol) 100%	0.79	.74	0.60	1.072
Methyl Alcohol - 90%	0.824		0.77	1.03
Methyl Alcohol - 40%	0.937		2.00	0.863
*Milk	1.03	1.15	-	0.99
Motor Oil	0.893		110	0.39
Naphthalene	1.15	0.9		1.04
Nitric Acid 95%	1.50		1.13	1.07
Nitric Acid 60%	1.37		2.35	0.913
Nonane - n	0.718	.97	-	1.02
Octane - n	0.70	.77		1.068
Olive Oil	0.91	93.0	-	0.41
Pentane - n	0.63	0.37	-	1.24
Propyl Alcohol	0.804	2.8	-	0.828
Rapeseed Oil	0.91	180	-	0.36
Sodium Hydroxide 50%	1.53	-	95.0	0.443
Soya Bean Oil	0.974	86	-	0.418
Sperm Oil	0.88	21	-	0.55
Sugar Solution 20%	1.08	1.9	-	0.895
Sugar Solution 40%	1.18	5.3	-	0.728
Sugar Solution 60%	1.29	44.0	-	0.475
Sulfuric Acid - 100%	1.83	14.6	-	0.59
Sulfuric Acid - 95%	1.83	14.5	-	0.593
Sulfuric Acid - 60%	1.50	4.4	-	0.755
Toluene	0.866	-	0.6	1.092
Turpentine	0.86	1.83	-	0.90
Water (Fresh)	1.0	1.10	-	1.00
Water (Salt)	1.03	1.10	-	1.00
Xylene (Xylol)	0.87	0.93	-	1.03

*These figures are approximate or averages of those values available

Bends

If a hose of a given length is bent, the pressure drop will increase by some definite amount . . . the sharper the bend and the smaller the radius of bend the greater the pressure drop. The effect of a bend may be neglected if it is slight or if there are but few bends in a long length of hose. This is because the additional pressure drop caused by these bends is not significant when compared to the total pressure drop.

However, a dock hose may have four sharp 90° bends in a 25-foot length, and if pressure drop is important, these bends must be considered because they constitute a significant portion of the overall pressure drop.

The curves in Chart 4. show the effects of resistance due to 90 degree bends. This data can also be used as a guide for smooth bends less or greater than 90 degrees. For example, a 45 degree bend has about 4/10 the resistance of a 90 degree bend.

Problem: Determine the equivalent length, in terms of hose inside diameters, of a 90 degree and a 180 degree bend whose relative radii are 12 inches.

Solution: Referring to the "total resistance curve," the equivalent length for a 90 degree bend is 34.5 hose diameters. The equivalent length of a 180 degree bend is 34.5 diameters for one 90 degree bend, 18.7 diameters for resistance due to length, and 15.8 ÷ 2 diameters for bend resistance. Adding these 34.5, 18.7 and *15.8 ÷ 2 equals 61.1 diameters for a 180 degree bend. Note that this loss is less than the sum of losses through two 90 degree bends separated by tangents.

Static Head Pressure

Static head is the difference in height between the inlet and outlet ends of a hose. Before using Chart 1., it is necessary to correct for static head pressure because the values in Chart 1. are pressure losses due to friction only.

To correct for static head pressure, the difference in height is determined and multiplied by 0.433 to convert the head to an equivalent pressure in PSI (one foot of water exerts 0.433 PSI pressure).

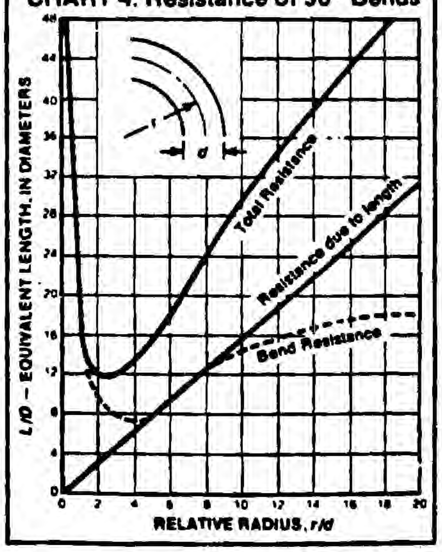
If the inlet is higher than the outlet, the pressure equivalent is added to the pump pressure. If the outlet is higher than the inlet, the pressure equivalent is subtracted from the pump pressure. In both cases, it is assumed that the pump pressure is the pressure available at the inlet end and that the pump is outside of the hose system.

Installation Design

Hose should not be twisted or put in torsion either during the installation or while in service. Sharp or excessive bends may cause the hose to kink or rupture.

Be sure to allow enough slack to provide for changes in length which will occur when pressure is applied. This change in length can vary from +2% to −4%.

CHART 4. Resistance of 90° Bends



Design the installation so the hose assembly is accessible for inspection and easy removal.

Bend radius is important. A good working rule is that the minimum bend radius should be five or more times the OD dimension of the hose.

*In a continuous bend of 180 degrees the second 90 degree bend produces approximately one-half the resistance of the first bend.

Application Data

Chemical Compatibility Chart

These tables alphabetically list commonly used materials of various chemical composition. After each agent listing you will find the basic hose core and fitting materials rated according to their chemical resistance to each individual agent. The chart is intended to be used as a guide only.

⚠ **WARNING - Selection of Hose:** Selection of the proper hose for the application is essential to the proper operation and safe use of the hose and related equipment. Inadequate attention to selection of the hose for your application can result in serious bodily injury or property damage. In order to avoid serious bodily injury or property damage resulting from selection of the wrong hose, you should carefully review the information in this catalog.

⚠ **WARNING - Proper Selection of Hose Ends:** Selection of the proper end fittings for the hose end application is essential to the proper operation and safe use of the hose and related equipment. Inadequate attention to the selection of the end fittings for your application can result in serious bodily injury or property damage. In order to avoid serious bodily injury or property damage resulting from selection of the wrong end fitting, you should carefully review the information in this catalog.

Where unusual conditions exist, or where questions arise, consult WEATHERHEAD® Division of Dana Corporation for assistance on your hose application problems.

FLUID	HOSE MATERIAL									HOSE END FITTINGS		
	Nitrile	Neoprene	Teflon	Nylon	Polyurethane	EPDM	CPE	Hypalon	Hytrel	Brass	Steel	316 Stainless
Acetaldehyde Solvent	X	X	G	F	X	G	-	X	X	-	-	G
Acetic Acid (concentrated)	X	F	G	X	X	G	G	-	-	X	X	F
Acetic Acid (diluted)	X	G	G	F	X	G	G	G	G	X	X	G
Acetic Anhydride	F	G	G	X	X	G	F	G	X	X	F	F
Acetone	X	F	G	G	X	G	G	F	F	G	G	G
Air	G	G	G	G	G	G	G	-	-	G	G	G
Alcohols (Meth. & Ethanol)	G	G	G	G	X	G	G	G	G	G	F	G
Aluminum Chloride	G	G	G	X	G	G	X	F	-	X	X	F
Aluminum Floride	G	F	G	G	G	G	X	-	-	X	X	X
Aluminum Hydroxide	G	G	G	G	G	G	-	-	-	G	-	G
Aluminum Sulfate	G	G	G	-	G	G	X	G	-	X	X	G
Alums	G	G	G	-	G	G	X	-	-	X	X	F
Ammonium Chloride	G	G	G	X	-	G	G	G	G	X	G	F
Ammonium Hydroxide	F	F	F	F	X	G	F	G	X	X	G	G
Ammonium Phosphate	G	G	G	G	-	G	G	-	-	X	X	G
Ammonium Sulfate	G	G	G	G	-	G	G	G	G	X	F	F
Amyl Acetate	X	X	G	G	X	G	X	X	F	F	F	G
Amyl Alcohol	F	C	G	G	X	G	G	G	G	G	F	F
Aniline	X	F	G	X	X	G	X	X	X	F	G	F
Aniline Dyes	F	F	G	X	X	G	X	-	-	F	X	F
Animal Oils & Fats	G	X	G	G	X	F	F	F	G	-	-	G
Anti-Freeze (Glycol Base)	G	G	G	G	-	G	-	G	-	-	-	-
Asphalt	F	F	G	X	X	X	X	X	-	G	G	G
Barium Chloride	G	G	G	G	G	G	G	-	-	X	F	F
Barium Hydroxide	G	G	G	G	X	G	G	G	-	F	F	G
Barium Sulfide	G	G	G	F	G	X	G	G	X	F	X	G
Beet Sugar, Liquors	G	G	G	-	-	G	G	-	-	-	G	G
Benzene, Benzol	X	X	G	G	X	X	F	X	F	G	G	G
Benzaldehyde	X	X	F	G	X	G	X	X	-	-	-	G
Black Sulfate Liquor	G	G	G	F	X	G	F	F	-	X	G	G
Borax	F	F	G	G	G	G	G	-	G	G	G	G
Boric Acid	G	G	G	F	G	G	X	G	G	X	X	F
Brake Fluid (petroleum base)	G	G	-	G	-	X	-	X	-	-	-	-
Brine	G	G	G	-	-	G	G	-	-	X	-	F
Butane	G	G	G	G	X	X	-	G	G	G	G	G
Butyl Acetate	X	X	G	G	X	F	F	X	F	G	G	G
Butyl Alcohol, Butanol	G	G	G	F	X	G	G	-	-	G	G	G
Calcium Bisulfite	G	G	G	F	G	-	X	-	-	X	X	X
Calcium Chloride	G	C	G	X	G	F	G	-	G	X	F	F
Calcium Hydroxide	F	G	G	F	X	G	G	-	-	F	G	G
Calcium Hypochlorite	F	F	G	F	X	G	G	F	F	X	F	F
Caliche Liquors Na Nitrate	G	G	G	-	-	G	G	-	-	G	G	G
Cane Sugar Liquors	G	G	G	-	X	G	G	-	-	F	G	G
Carbolic Acid (Phenol)	X	X	G	X	X	X	X	X	X	F	X	F
Carbon Dioxide (Dry)	F	F	G	G	G	F	G	G	G	G	-	G
Carbon Dioxide (Wet)	F	F	G	G	G	F	-	-	-	F	-	G
Carbon Disulphide (Bisulfide)	X	X	G	F	G	X	-	-	-	G	G	G
Carbon Monoxide (Hot)	F	F	G	-	F	F	G	G	G	X	F	G
Carbon Tetrachloride	X	X	G	G	X	X	F	X	F	F	F	G
Carbonic Acid	G	G	G	G	F	G	X	-	-	X	X	F
Castor Oil	G	G	G	G	F	F	G	G	F	G	G	G
Cellosolve Acetate	X	X	G	F	X	G	X	-	-	X	X	G
Cellulube-90,150,220,300, 550,1000	X	X	G	G	X	G	-	-	-	G	G	G
Chlorinated Solvents	X	X	G	F	-	X	X	-	-	G	G	F
Chloracetic Acid	X	X	G	-	X	F	X	F	-	X	X	F
Chloroform	X	X	G	-	X	X	X	X	X	G	-	G
Chlorosulphonic Acid	X	X	G	-	X	X	X	X	X	X	F	X
Chromic Acid	X	X	G	X	X	X	X	F	X	X	X	F
Citric Acid	G	G	G	G	X	G	X	-	G	F	X	F
Coke Oven Gas	F	F	F	G	X	X	-	-	-	F	G	G
Copper Chloride	G	G	G	G	G	G	X	-	G	X	X	G
Copper Sulfate	G	G	G	G	G	G	X	-	G	X	X	G
Cottonseed Oil	G	G	G	G	G	F	G	G	G	G	G	G
Creosote	F	X	G	X	F	X	F	F	-	F	-	G
Cresol	X	X	G	-	X	X	X	X	-	-	-	G
Diesel Fuel	G	F	G	G	F	X	G	F	F	G	G	G
Downtherm A & E	X	X	G	-	-	X	X	-	-	X	F	G
Ethers (Ethyl Ether)	X	X	G	G	F	-	X	X	-	G	G	G
Ethyl Acetate	X	X	G	G	X	G	F	X	F	F	G	G
Ethyl Cellulose	F	F	G	G	F	F	G	-	-	F	G	F
Ethyl Chloride	F	X	G	G	F	G	X	X	X	F	F	G
Ethylene Dichloride	X	X	G	G	X	X	X	X	X	G	X	X
Ethylene Glycol	G	G	G	F	F	G	G	G	G	F	G	G
Ethylene Oxide	X	X	G	F	X	F	X	X	G	-	-	-
Fatty Acids	G	X	G	G	-	X	-	-	-	-	-	G
Ferric Chloride	G	G	G	F	F	G	X	G	-	X	X	X
Ferric Sulfate	G	G	G	G	-	-	X	-	G	X	X	F
Formaldehyde	Γ	F	G	X	X	G	X	G	F	F	F	G
Formic Acid	F	F	G	X	X	G	X	G	X	F	X	G
Freon 12*	G	F	F	G	X	X	F	G	X	G	F	G
Fuel Oil	G	F	G	G	F	X	G	-	-	F	G	G
Furfural	X	F	G	G	-	F	F	F	-	F	G	G
Gasoline (refined)	G	F	G	G	F	X	G	F	G	G	G	G
Gasoline (unleaded)	G	X	G	G	X	X	G	F	X	G	G	G
Gasoline (10% Ethanol)	G	X	G	G	X	X	-	-	X	X	G	G
Gasoline (10% Methanol)	F	X	X	-	X	X	-	-	X	X	G	G

*: Use approved Freon Hose.

CODES: G-Good resistance F-Fair resistance X-Incompatible -No data available

Chemical Compatibility Chart

Application Data

FLUID	HOSE MATERIAL									HOSE END FITTINGS		
	Nitrile	Neoprene	Teflon	Nylon	Polyurethane	EPDM	CPE	Hypalon	Hytrel	Brass	Steel	316 Stainless
Glycerine, Glycerol	G	G	G	G	X	G	G	-	G	G	G	G
Greases	G	F	G	G	G	X	G	F	G	G	G	G
Green Sulfate Liquor	G	G	G	-	G	G	X	-	-	X	X	G
Heptane	G	F	G	G	F	X	G	F	G	G	G	G
N-Hexane	G	F	F	G	F	X	G	F	G	G	G	G
Houghto Safe 271 to 640	G	G	G	G	-	X	-	-	-	G	G	G
Houghto Safe 5046, 5046W	G	G	G	G	-	-	-	-	-	G	G	G
Houghto Safe 1010,1055,1115,1120,1130	X	X	G	G	X	G	G	-	-	G	G	G
Hydraulic Oil (Petroleum Base)	G	G	G	G	G	X	G	F	G	G	G	G
Hydrobromic Acid	X	X	G	X	X	F	X	G	X	X	X	X
Hydrochloric Acid	X	X	G	X	X	G	X	G	X	X	X	X
Hydrocyanic Acid	F	F	G	-	-	G	X	G	-	X	F	G
Hydrofluoric Acid	X	X	G	X	X	F	X	G	-	X	X	X
Hydrofluosilicic Acid	F	X	G	-	-	G	X	G	-	X	X	X
Hydrogen Peroxide	X	X	G	-	-	F	X	F	-	F	X	G
Hydrogen Sulfide	X	F	G	-	-	G	X	G	G	F	F	F
Hydralube H-2	G	F	G	G	-	-	-	-	-	G	G	G
Isopropyl Alcohol	F	G	G	G	X	G	G	G	G	G	F	G
Iso-Octane	G	F	G	G	X	X	G	G	G	-	F	-
Kerosene	G	F	G	G	G	X	G	F	F	G	G	G
Lacquer	X	X	G	G	X	X	F	-	-	G	X	G
Lacquer Solvents	X	X	G	G	X	X	F	X	F	G	X	G
Lactic Acid	G	G	G	G	X	G	X	G	X	F	X	F
Lindol	X	X	G	G	-	G	-	-	-	F	G	G
Linseed Oil	G	G	G	G	F	F	G	G	-	F	G	G
Lubricating Oils	G	G	G	G	F	X	G	F	G	G	G	G
Lime Sulfur	X	G	G	-	-	G	-	-	-	X	-	G
Magnesium Chloride	G	G	G	G	G	G	G	-	-	X	F	F
Magnesium Hydroxide	F	F	G	X	X	G	G	-	-	G	G	G
Magnesium Sulphate	G	G	G	-	-	G	G	-	-	F	G	G
Mercuric Chloride	G	G	G	X	-	G	X	-	-	X	F	X
Mercury	G	G	G	G	G	G	G	G	G	X	G	G
Methane	F	F	G	-	F	X	G	-	-	-	-	-
Methyl Chloride	X	X	G	G	X	X	F	-	-	G	G	G
Methyl Ethyl Ketone	X	X	G	G	X	G	G	X	G	F	F	F
Methyl Isobutyl Ketone	X	X	G	G	X	G	X	X	-	-	-	G
Methyl Isopropyl Ketone	X	X	G	G	X	G	X	X	-	F	F	F
Mineral Oil	G	G	G	G	G	X	G	G	G	G	G	G
Naphtha	G	F	G	G	F	X	G	X	G	F	G	G
Naphthalene	X	X	G	G	F	X	G	X	F	F	G	F
Nickel Acetate	X	F	G	-	-	G	-	-	-	-	-	-
Nickel Chloride	G	G	G	-	X	G	X	-	-	X	X	F
Nitric Acid (concentrated)	X	X	G	X	X	X	X	X	X	X	X	F
Nitric Acid 10%	X	F	G	X	X	G	X	G	X	X	X	G
Nitric Acid 70%	X	X	G	X	X	X	X	X	X	X	X	G
Nitrobenzene	X	X	G	G	X	X	F	X	X	X	G	F
Olaic Acid	F	F	G	G	F	X	X	F	G	F	F	G
Oxalic Acid	F	G	G	F	-	G	X	-	-	F	F	F
Oxygen	F	F	F	F	G	G	X	-	-	G	G	G
Palmitic Acid	G	F	G	-	X	F	F	X	X	X	F	F
Perchloroethylene	X	X	G	-	X	X	F	X	X	X	F	F
Petroleum Oils	G	F	G	G	G	X	G	-	G	G	G	G
Picric Acid (molton)	X	X	G	-	-	X	X	-	-	X	X	F
Picric Acid (solution)	X	X	G	-	F	G	X	F	-	X	X	F
Potassium Chloride	G	G	G	G	G	G	G	-	-	X	F	G

FLUID	HOSE MATERIAL									HOSE END FITTINGS		
	Nitrile	Neoprene	Teflon	Nylon	Polyurethane	EPDM	CPE	Hypalon	Hytrel	Brass	Steel	316 Stainless
Phosphate Esters	X	X	X	G	X	G	G	-	-	-	-	-
Phosphate Esters (Alkyl)	X	X	G	G	X	G	G	X	G	G	G	G
Phosphate Esters (Anyl)	X	X	G	F	-	-	-	X	-	G	G	G
Phosphate Ester (Blend)	X	-	G	F	-	-	-	X	-	G	G	G
Potassium Cyanide	G	G	G	G	G	G	G	-	-	-	G	G
Potassium Dichromate	F	G	G	-	G	G	X	G	-	X	G	G
Potassium Hydroxide	G	G	G	G	X	G	X	G	F	F	X	G
Potassium Sulphate	G	G	G	F	G	[illegible]	[illegible]	-	-	F	F	G
Propane	Use H366 Hose											
Pydraul F-9, 150	X	X	G	G	X	G	G	-	-	G	G	G
Pyridine	X	X	X	-	-	F	X	X	-	-	-	F
Sea Water	G	G	G	F	X	G	-	G	G	F	X	G
Skydrol (All)	X	X	G	G	X	G	G	-	-	G	G	G
Soap Solution	G	F	G	G	G	G	G	G	G	G	G	G
Soda Ash, Sodium Carbonate	G	G	G	G	-	G	G	-	-	F	F	G
Sodium Bisulphate	G	G	G	G	-	G	G	-	-	X	F	G
Sodium Chloride	G	G	G	G	G	G	G	G	G	X	F	G
Sodium Cyanide	G	G	G	G	-	G	G	-	-	X	-	G
Sodium Hydroxide	F	F	G	G	X	G	F	-	-	F	X	G
Sodium Hypochlorite	F	F	G	-	X	F	F	G	G	X	X	F
Sodium Nitrate	G	G	G	G	-	G	G	-	-	F	F	F
Sodium Perborate	G	G	G	-	X	G	X	G	-	F	F	G
Sodium Peroxide	G	G	G	-	X	G	X	G	-	X	F	G
Sodium Phosphates	X	X	G	G	G	G	X	-	-	F	F	F
Sodium Silicate	G	G	G	G	-	G	G	-	-	F	F	G
Sodium Sulphate	G	G	G	G	G	G	G	-	-	F	F	G
Sodium Sulphide	G	G	G	G	G	G	G	-	-	F	X	G
Sodium Thiosulphate	G	G	G	G	G	G	G	-	-	F	X	G
Soybean Oil	G	G	G	G	G	X	G	G	G	G	G	G
Stannic Chloride	G	G	G	G	-	G	X	F	-	X	X	X
Steam 450°	X	X	G	X	X	X	X	X	X	F	F	G
Stearic Acid	F	F	G	G	G	F	G	F	G	F	F	G
Sulphur	F	G	G	G	-	G	G	-	-	X	X	G
Sulphur Chloride	F	G	G	-	-	X	G	X	X	X	X	X
Sulphur Dioxide	F	F	G	F	-	G	X	-	-	X	F	G
Sulphuric Acid (10% Cold)	G	G	G	F	X	G	X	G	F	X	X	G
Sulphuric Acid (10% Hot)	X	F	G	F	X	F	X	G	X	X	X	G
Sulphuric Acid (75% Cold)	X	F	G	X	X	F	X	-	X	X	X	X
Sulphuric Acid (75% Hot)	X	X	G	X	X	X	X	-	X	X	X	X
Sulphuric Acid (95% Cold)	X	X	G	X	X	X	X	F	-	X	X	X
Sulphuric Acid (95% Hot)	X	X	G	X	X	X	X	X	X	X	X	F
Sulphuric Acid (Fuming)	X	X	G	X	X	X	X	X	X	X	F	F
Tannic Acid	F	G	G	G	G	G	X	G	G	F	X	F
Tar	G	F	G	X	-	X	X	-	-	F	F	G
Tartaric Acid	F	F	G	-	G	F	X	G	-	F	X	F
Tetrachloroethane	X	X	G	G	X	X	-	-	-	-	-	G
Toluene	X	X	G	G	X	X	F	X	F	G	G	G
Transmission Oil	G	G	G	G	-	X	-	-	-	-	-	-
Trichloroethylene	X	X	G	F	X	X	F	X	X	G	G	G
Turpentine	F	X	G	G	X	X	G	X	-	F	F	G
Varnish	X	X	G	G	X	X	F	-	-	G	G	G
Water	G	G	G	F	G	G	G	G	G	F	F	G
Water (Glycol)	G	G	G	F	-	G	G	G	-	F	G	G
Water (Petroleum)	G	F	G	F	X	X	G	F	G	G	F	G
Xylene	X	X	G	G	X	X	F	X	F	G	G	G
Zinc Chloride	F	F	G	G	-	G	X	-	G	X	X	F
Zinc Sulfate	G	G	G	G	-	G	X	-	-	X	F	G

CODES: G-Good resistance F-Fair resistance X-Incompatible -No data available

NOTES: All data given herein is believed to be accurate and reliable but presented without guaranty, warranty, or responsibility of any kind, express or implied, on our part. Chemical resistance will vary with the wide diversity of possible mixtures and service conditions. It is not therefore possible to give any guarantee whatsoever in individual cases.

APPLICATION

Hose | Medium Pressure

H069 General Purpose Truck Hose
SAE 100R5
SAE J1402 TYPE II†
DOT AII†

Typical Application: Medium pressure hydraulic, air, oil, fuel or grease lines on fork lifts, trucks & off highway vehicles: -16 thru -48 sizes are not DOT approved.
Inner Tube: Nitrile
Reinforcement: 1 Fiber Braid - 1 Steel Braid
Cover: Fiber Braid
Temp. Range: -40°F to +300°F (-40°C to +149°C) sizes -4 thru -16
-40°F to +250°F (-40°C to +121°C) sizes -20 thru -48
†sizes -4 thru -12 only

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H06904	3/16	33/64	3000	12000	3	30	16	50',250'	
H06905	1/4	37/64	3000	12000	3-3/8	30	20	50',250'	
H06906	5/16	43/64	2250	9000	4	30	23	50',250'	Coil-O-Crimp
H06908	13/32	49/64	2000	8000	4-5/8	30	27	50',250'	069 'E' Series
H06910	1/2	59/64	1750	7000	5-1/2	30	36	50',250'	Pages 68-70
H06912	5/8	1-5/64	1500	6000	6-1/2	30	41	50',250'	Reusable
H06916	7/8	1-15/64	800	3200	7-3/8	20	46	50',250'	069 'D',
H06920	1-1/8	1-1/2	625	2500	9	20	51	50'	069 'T' &
H06924	1-3/8	1-3/4	500	2000	10-1/2	15	59	50'	247 'N' Series
H06932	1-13/16	2-7/32	350	1400	13-1/4	11	90	50'	Pages 94-9[illegible]
H06940	2-3/8	3-7/8	350	1400	24		143	50'	
H06948	3	3-9/16	200	800	33		209	50'	

H169 Hydraulic Hose

USCG Fluid Power and Fuel/Lube Systems Accepted
MSHA Accepted
Typical Application: Medium pressure air, fuel, grease, oil, truck and power steering lines.
Inner Tube: Neoprene
Reinforcement: 1 Fiber Braid - 1 Steel Braid
Cover: Neoprene Perforated
Temp. Range: -40°F to +212°F (-40°C to +100°C)

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H16904	3/16	33/64	3000	12000	3	30	16	50',250'	
H16905	1/4	37/64	3000	12000	3-3/8	30	20	50',250'	Coil-O-Crimp
H16906	5/16	43/64	2250	9000	4	30	23	50',250'	069 'E' Series
H16908	13/32	49/64	2000	8000	4-5/8	30	27	50',250'	Pages 68-70
H16910	1/2	59/64	1750	7000	5-1/2	30	36	50',250'	Reusable
H16912	5/8	1-5/64	1500	6000	6-1/2	30	41	50',250'	069 'D' &
H16916	7/8	1-15/64	800	3200	7-3/8	20	46	50',250'	247 'N' Pages 94-99
H16920	1-1/8	1-1/2	625	2500	9	20	51	50'	
H16924	1-3/8	1-3/4	500	2000	10-1/2	15	59	50'	
H16932	1-13/16	2-7/32	350	1400	13-1/4	11	90	50'	

H104 Hydraulic Hose
SAE 100R1 TYPE AT

USCG Fluid Power and Fuel/Lube Systems Accepted
MSHA Accepted
Typical Application: Most commonly used for medium pressure hydraulic lines. Especially popular for farm implement hydraulic lines.
Inner Tube: Neoprene
Reinforcement: 1 Steel Braid
Cover: Neoprene
Temp. Range: -40°F to +212°F (-40°C to +100°C)

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H10404	1/4	17/32	2750	11000	4	30	15	50',250' 500'	
H10406	3/8	11/16	2250	9000	5	30	22	50',250' 500'	
H10408	1/2	13/16	2000	8000	7	30	29	50',250' 500'	Coil-O-Crimp 'U' Series Pages 39-50
H10410	5/8	15/16	1500	6000	8	30	34	50',250'	
H10412	3/4	1-3/32	1250	5000	9-1/2	20	40	50',250'	
H10416	1	1-13/32	1000	4000	12	20	50	50',250'	
H10420	1-1/4	1-23/32	625	2500	16-1/2	20	78	50'	

Medium Pressure | Hose

H114 HYDRAULIC HOSE
Exceeds SAE 100R1

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H11404	1/4	1/2	3000	12000	4		11	50',250' 500'	Coll-O-Crimp "U" Series Pages 39-50
H11406	3/8	41/64	3000	12000	5		18	50',250' 500'	
H11408	1/2	13/16	3000	12000	7		25	50',250' 500'	

MSHA Accepted
Typical Application: 3000 PSI continuous working pressure hose. Ideal for farm implement high temperature hydraulic lines. Will work with synthetic or petroleum/water based fluids.
Inner Tube: Hytrel
Reinforcement: 1 Steel Braid
Cover: Neoprene
Temp. Range: -40°F to +250°F (-40°C to +121°C)

H145 HYDRAULIC HOSE
Exceeds SAE 100R1

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H14504	1/4	1/2	3000	12000	2		13	50'250' 500'	
H14506	3/8	41/64	3000	12000	2-1/2		18	50',250' 500'	Coll-O-Crimp "U" Series Pages 39-50

MSHA Accepted
Typical Application: Ideal for use in high pressure lines on off the road construction equipment, farm equipment, and other high pressure applications where a small bend radius is needed.
Inner Tube: Nitrile
Reinforcement: 1 Steel Braid
Cover: Neoprene
Temp. Range: -40°F to +250°F (-40°C to +121°C)

HOSE

Hose | Medium Pressure

H146 3000 PSI Hydraulic Hose
Exceeds SAE 100R1

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H14604	1/4	33/64	3000	12000	4		11	50',250'	Coll-O-Crimp 'U' Series Pages 39-50
H14606	3/8	11/16	3000	12000	5		18	50',250'	
H14608	1/2	51/64	3000	12000	7		25	50',250'	

Typical Application: 3000 PSI continuous working pressure hose. Ideal for farm implement high temperature hydraulic lines. Will work with synthetic or petroleum/water based fluids.
Inner Tube: Hytrel
Reinforcement: 1 Steel Braid
Cover: Abrasion Resistant Fiber Braid
Temp. Range: -65°F to +250°F (-54°C to +121°C)

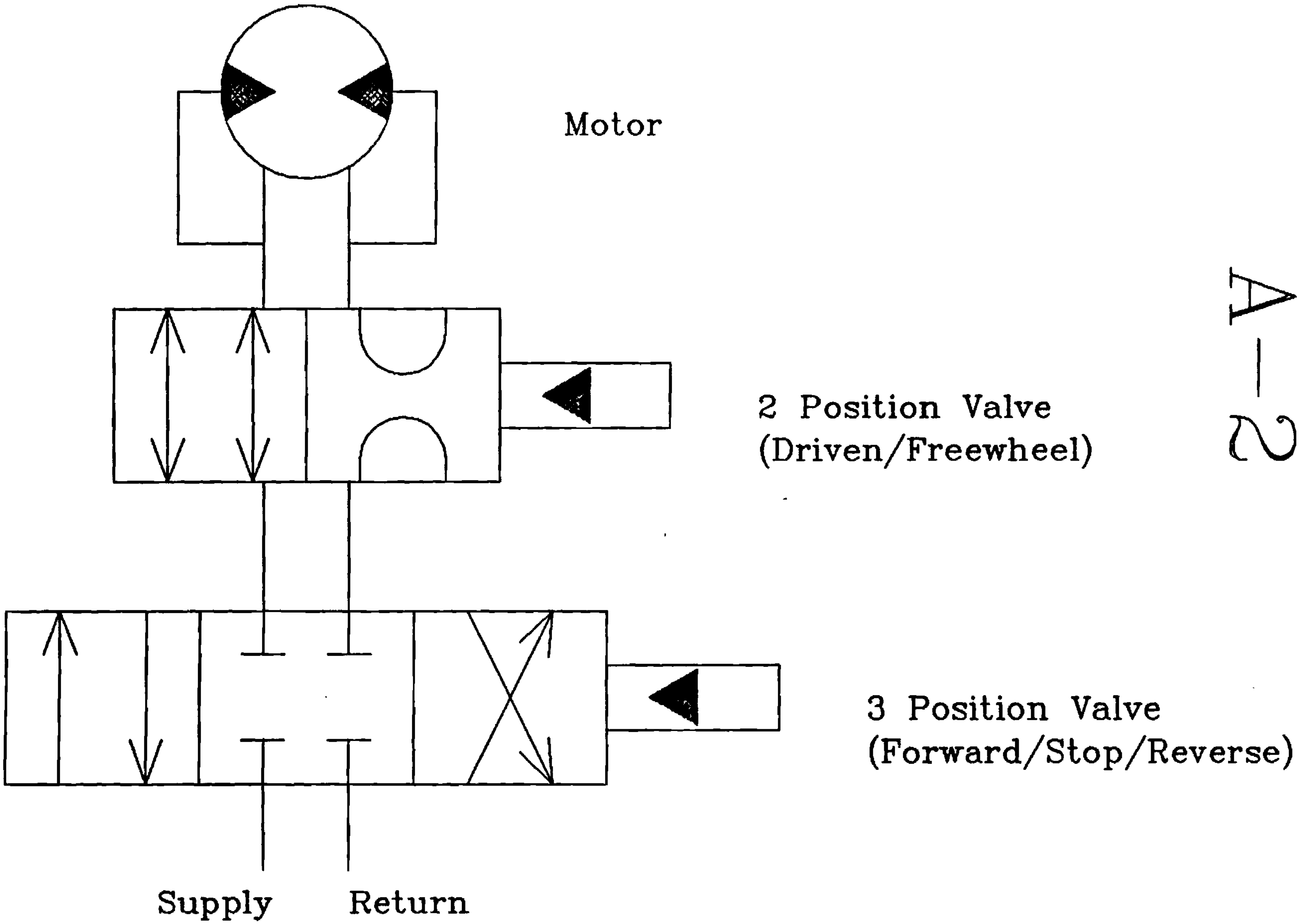
H300 3000 PSI Hydraulic Hose

Catalog Number	Hose I.D.	Hose O.D.	Work. Pres. (PSI)	Min. Burst Pres. (PSI)	Min. Bend Radii	Inch. Merc. Vac.	Wt. Per 100 Ft.	Avail. Lengths	Hose Ends
H30004	1/4	9/16	3000	12000	4		15	50',250'	
H30006	3/8	23/32	3000	12000	5		25	50',250'	Coll-O-Crimp 'U' Series Pages 39-50
H30008	1/2	29/32	3000	12000	7		45	50',250'	
H30010	5/8	1-1/32	3000	12000	8		55	50',250'	
H30012	3/4	1-3/16	3000	12000	9-1/2		65	50',250'	

MSHA Accepted
Typical Application: 3000 PSI constant working pressure hose. Ideal for farm implement construction and industrial equipment where high presssure hydraulic needs exist.
Inner Tube: Neoprene
Reinforcement: 4-6 size 1 steel braid; 8 size 2 steel braids; 10 and 12 size 2 spiral and 1 steel braid.
Cover: Neoprene
Temp. Range: -40°F to +212°F (-40°C to +100°C)

APPENDIX G

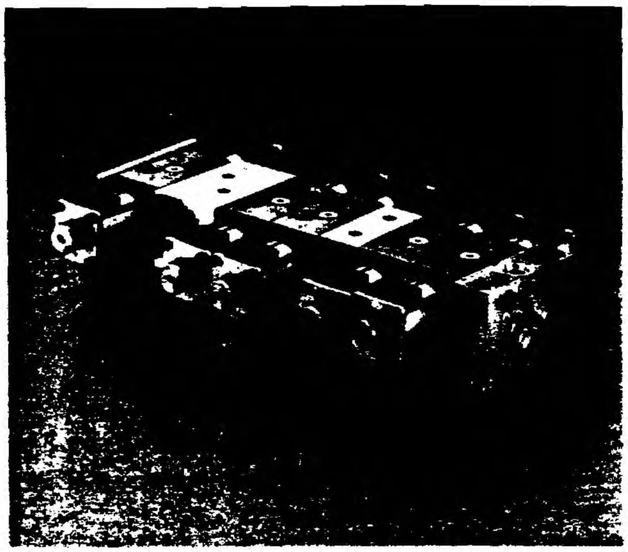
Valve Detail



G1 Power Supply – Group 5

ME4182c
Revision 1

Pressure Compensated, Electrohydraulic, Proportional Directional Control Valves



Application

Series six MCV valve systems are ideal for applications that require precise control of load, speed and direction. They provide precise and variable speed control without lurching during start–up. Series six valves provide smooth acceleration and deceleration, thus providing precise and predictable metering throughout the flow range.

Operation

The Series 6, MCV–ISO valves are controlled by proportional, push–type oil immersed solenoids, which provide precise metering characteristics, regardless of load variations.

Features

- Low Cost
- Pressure Compensation per Spool
- Precise low flow metering
- Compact and lightweight – saves space
- Adaptable to larger HPI proportional valves
- No external pilot or drain lines
- No null adjustment
- Adjustable flow control for each cylinder port
- Spade or Hirschmann connectors

Installation Data

Temp. Range	–40°F to 180°F (–40°C to 82°C)
Fluid Viscosity	35 to 1750 SSU recommended
Filtration	25 micron return line
Fluids	Petroleum base (consult factory for other fluids)
Mounting	Valve will function in any position; however, it is recommended valve be mounted horizontally for optimum performance.

Operating Specifications

Flow Ratings	.2 GPM (.76 LPM) min.— 7 GPM (26.5 LPM) max.
Max. Pressure	3000 PSI (204 Bar) maximum
Flow Settings For other input currents, please specify when ordering	Available in increments of .10 GPM *(.2 – 7 GPM) * "A" spool is limited to 5 GPM output flow Output flow is based on a solenoid current of: 2100 ma.–12 VDC (A & C spool) 1310 ma.–24 VDC (A & C spool)
Seals	Nitrile (Standard) Viton (Optional)

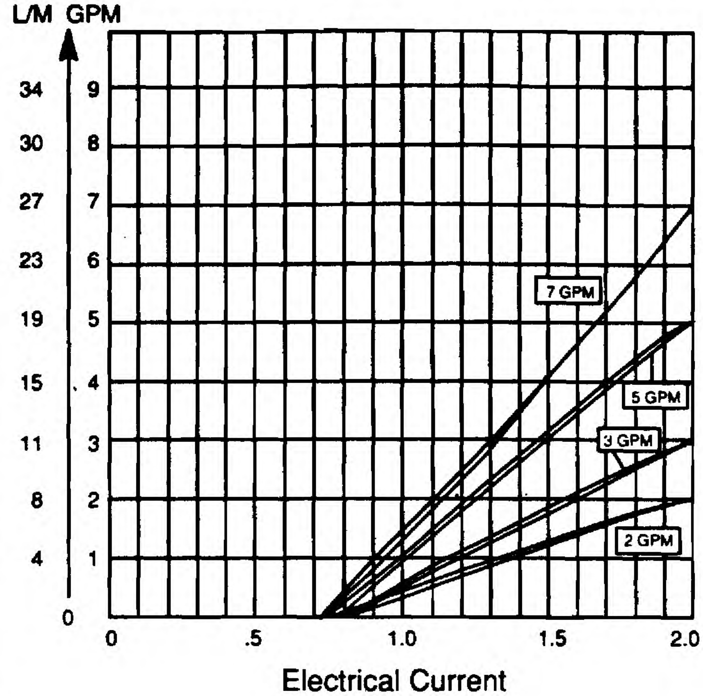
Electrical Specifications

Voltages	12 VDC or 24 VDC
PWM Control Required	To obtain precise, proportional control, pulse width modulation (PWM) is required: 40 Hz (P–Q Controllers) 65 Hz (OEM Controllers)
Current Range	Cracking to full flow: 12 VDC 500–2100 ma. (C spool) 800–2100 ma. (A spool) 24 VDC 310–1310 ma. (C spool) 500–1310 ma. (A spool) Output flow is based on a solenoid current of: 2100 ma.–12 VDC (A & C spool) 1310 ma.– 24 VDC(A & C spool)
Coil Resistance	At 68°F (20°C): 12 VDC 3.9 ohms 24 VDC 13.7 ohms At 140°F (60°C): 12 VDC 4.5 ohms 24 VDC 13.8 ohms
Duty	Continuous duty rated
Grounding	No grounding is required. No ground terminal is available
Connections	Standard: 1/2" male spade terminals
Optional	DIN 43650 appliance type plug with gasket for 6 mm to 8 mm cable diameter with one mounting screw. To order optional appliance plug separately, use P/N 15000–14

Performance Data

Hysterisis Curve (12 VDC)

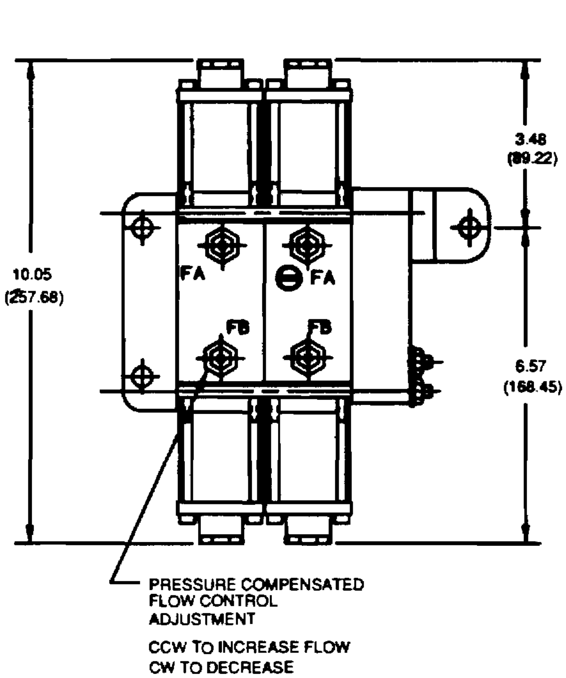
Various Flow Settings

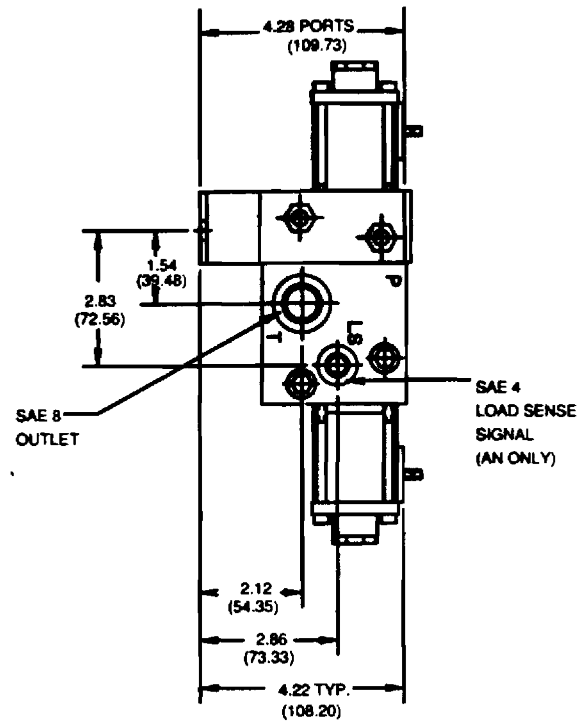


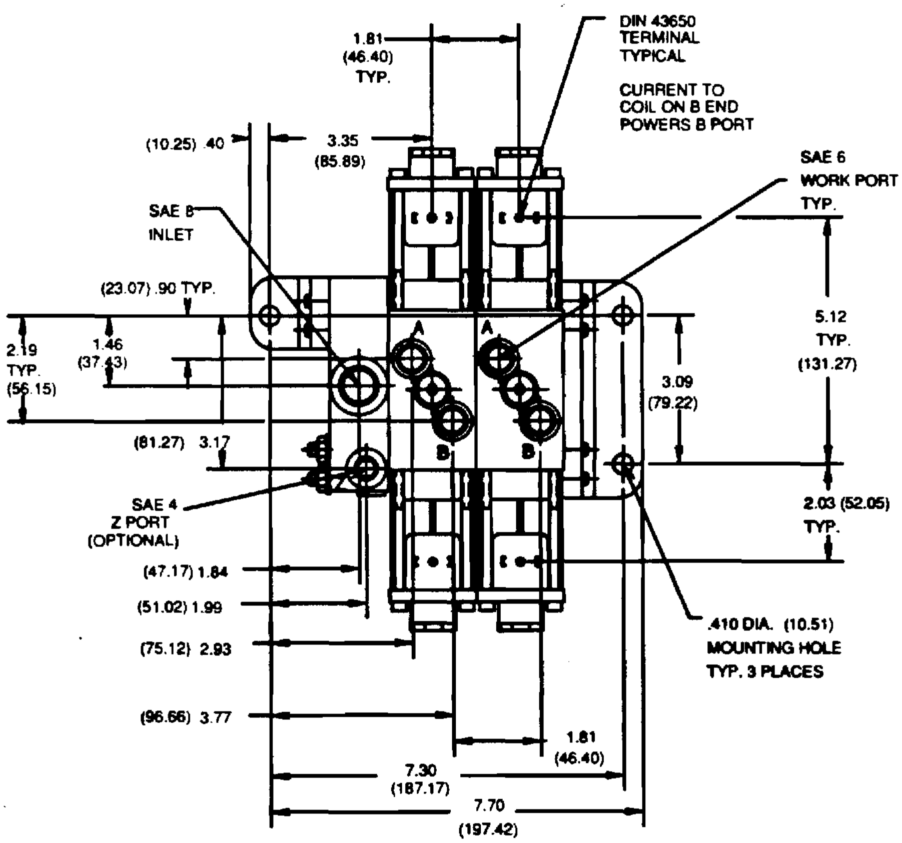
Dimensions

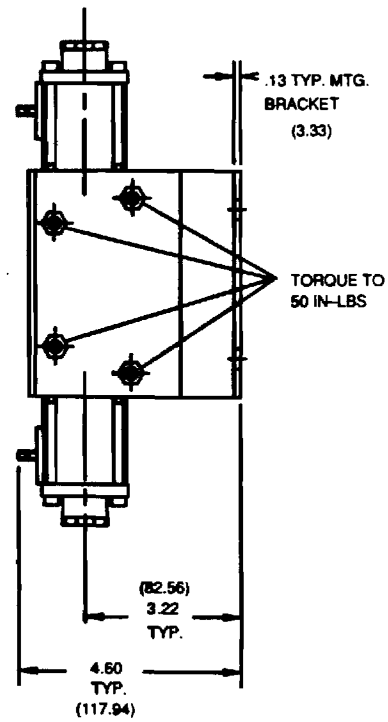
Millimeter equivalents for inch dimensions are shown in (**)

These drawings show a typical two function electrohydraulic proportional valve assembly which is used with load sense pumps and auxiliary shuttle option. Mounting brackets can be reversed 180°









Engineering Performance Data

D1V Series Pressure Drop vs. Flow

The chart to the right provides the flow vs. pressure drop curve reference for D1V Series valves by spool type.

Example:
Find the pressure drop at 6 GPM or a D1V with a number 1 spool.

Using the top chart, locate the numeral 1 in the spool number column. To the right of the numeral 1, locate the numeral 4 in the P–A column, and the numeral 1 in the B–T column.

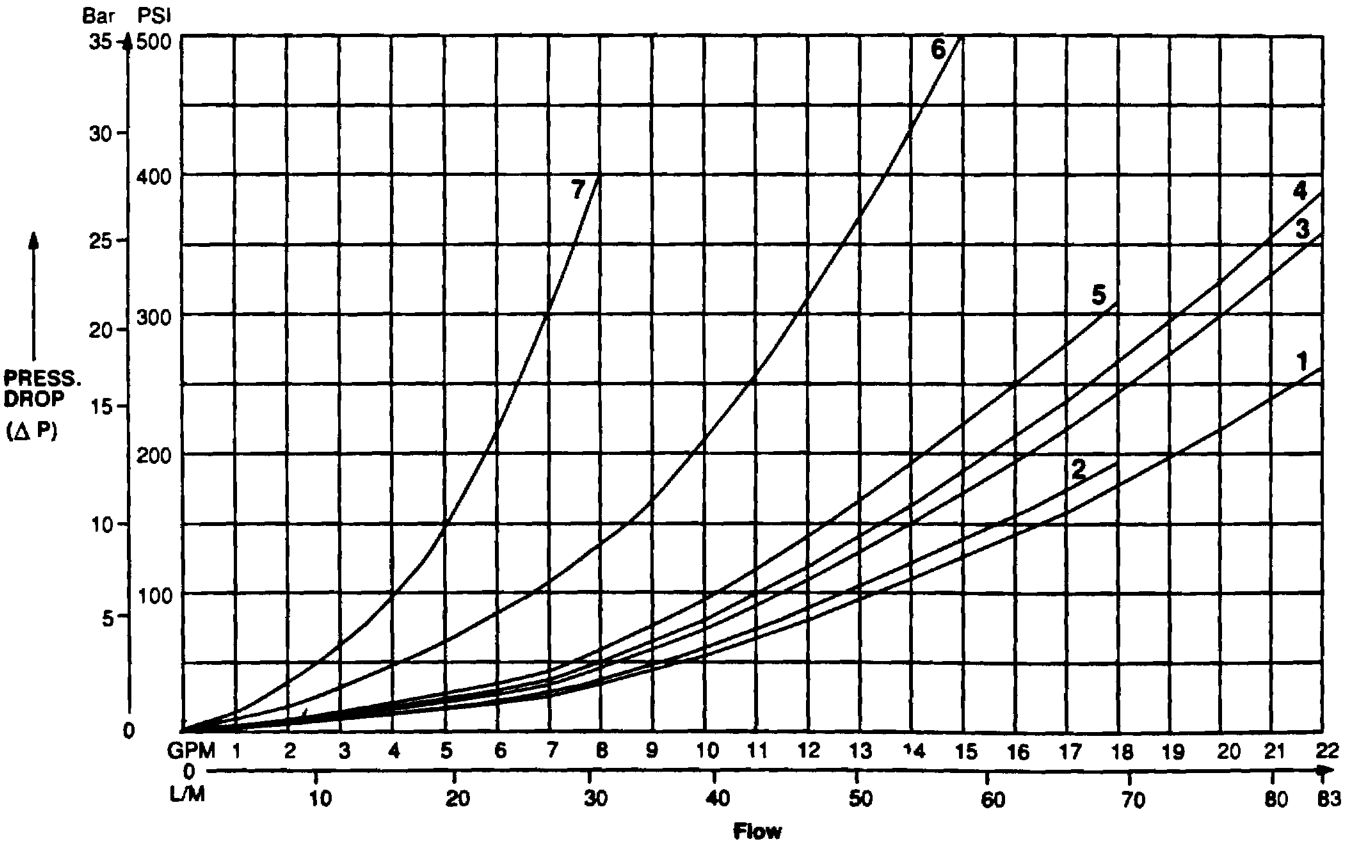
Using the bottom graph, locate curve 4; the pressure drop P–A is 28 PSI at 6 GPM. Then using curve 1, the pressure drop B–T at 6 GPM is 22 PSI. Total pressure drop through the valve is then 28 + 22 = 50 PSI.

Note: Pressure drops should be checked for all flow paths, especially when using non–symmetrical spools . (spools 3, 5, 7, 14, 15, 16, 21 & 22) and unbalanced actuators.

D1V Pressure Drop Reference Chart

Spool No.	Curve Number						
	P–A	P–B	P–T	A–T	B–T	B–A	A–B
1	4	4	–	1	1	–	–
2	3	3	4	1	1	–	4
3	4	4	–	1	1	–	–
4	4	4	–	1	1	–	–
5	3	4	–	1	1	–	–
6	3	3	–	1	1	–	–
7	4	3	6	1	1	–	6
8	2	2	6	4	4	–	6
9	2	2	6	4	4	–	6
10	4	4	–	–	–	–	–
11	4	4	–	1	1	–	–
14	3	4	–	1	1	–	–
15	4	4	–	1	1	–	–
16	4	3	–	1	1	–	–
20	5	5	–	2	2	–	–
21	4	7	–	1	–	4	–
22	7	4	–	–	1	–	4
30	5	5	–	2	2	–	–

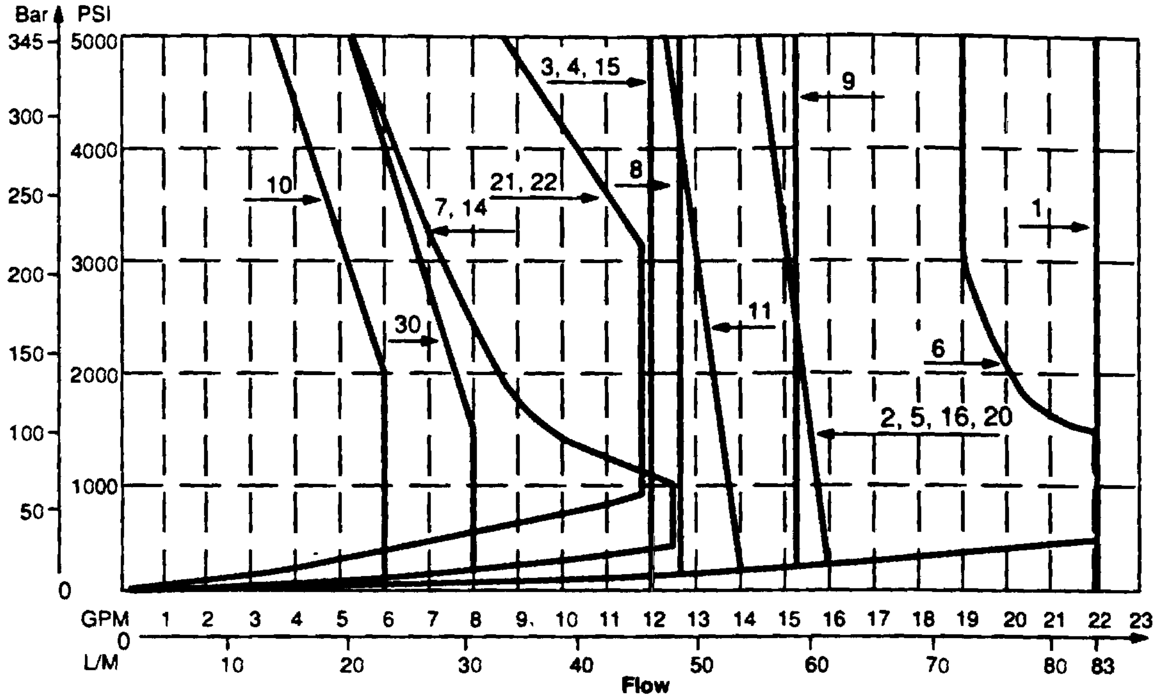
Pressure Drop Chart



Curves were generated using 100 SSU hydraulic oil. For any other viscosity, pressure drop will change as per chart.	VISCOSITY CORRECTION FACTOR							
	Viscosity (SSU)	75	150	200	250	300	350	400
	% of ΔP (Approx.)	93	111	119	126	132	137	141

Switching Limit Chart

by Spool Number



Example:

Determine the maximum allowable flow of a D1V Series Valve (#30 spool) at 1200 PSI (83 Bar) supply pressure. Locate the curve marked "30". At 1200 PSI (83 Bar) supply pressure, the maximum flow is 8 GPM (30 L/M). At 2000 PSI (138 Bar) the flow is 7.5 GPM (28 L/M).

Notes:
1. For F & M style valves, reduce flow to 70% of that shown
2. For AC low–watt coils, reduce flow to 40% of that shown, 3000 PSI max.
3. For DC low–watt coils, reduced flow to 85% of that shown.

For Maximum Valve Reliability, Adhere to the Following Installation Information

Recommended Mounting Surface

Surface must be flat within .0004 inch T.I.R. and smooth within 32 micro–inch. Torque bolts to 50 in.–lbs. (5.6 N.m)

Mounting Pattern

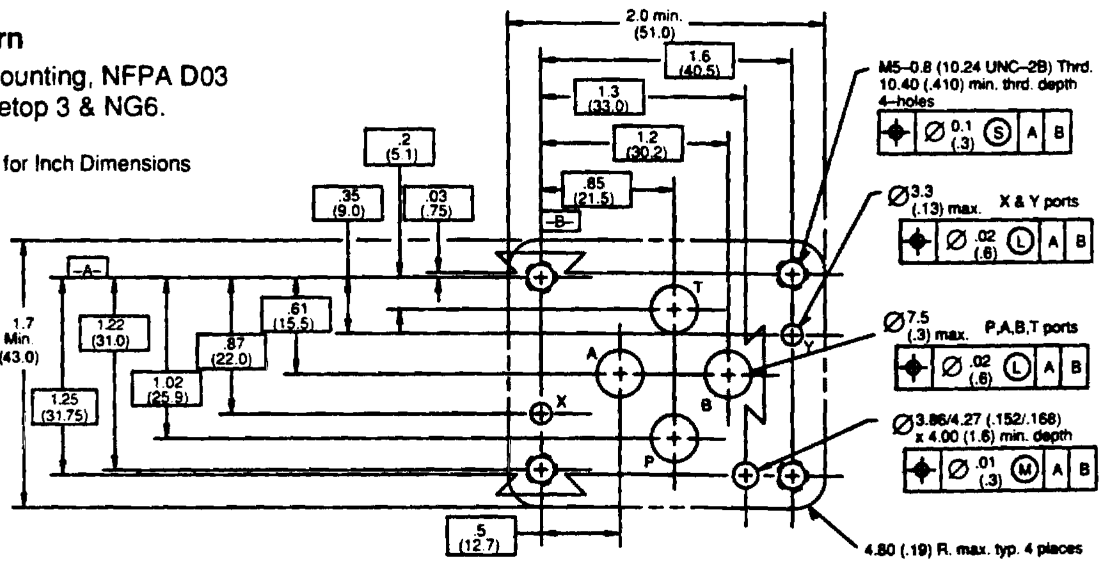
D1V – Subplate mounting, NFPA D03 (Formerly D01), Cetop 3 & NG6.

"Millimeter Equivalents for Inch Dimensions are shown in ("")"

Mounting Position

Detent – Unrestricted (Lever)
Detent – Horizontal (Solenoid, Pilot Operated)
Spring Offset – Unrestricted
Spring Centered – Unrestricted

E



Mounting Surface D1V Directional Control Valve Manifold M'TD. (NFPA D03, Formerly D01); Cetop 3 & NG6
Note: X and Y Ports are required for subplate piloted D1VP valves only

Engineering Performance Data

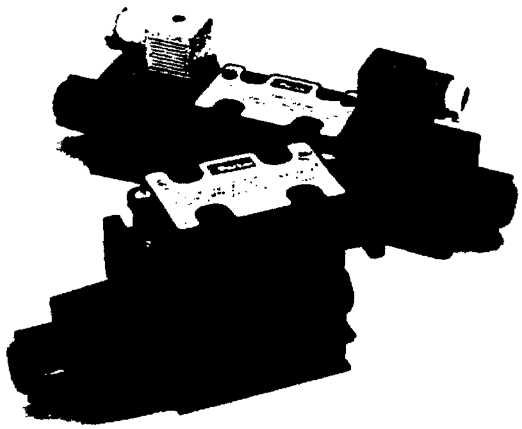
Mounting Pattern	NFPA D03, (Formerly D01); CETOP 3; NG 6
Maximum Pressure	Operating 5000 PSI (345 Bar) C.S.A. rating 4000 PSI Tank Line: 1500 PSI (103 Bar) – Standard 3000 PSI (207 Bar) – Optional 200 PSI (13.8 Bar) with monitor switch (10 option)
Nominal Flow	8.5 GPM (32 Liters/Min.)
Maximum Flow	See chart on page 3

Solenoid Electrical Characteristics *

Solenoid Code	Nominal Volts/Hz	In Rush Amps	Holding Amps	Watts
Y	120/60 110/50	2.00 2.10	0.49 0.58	25 27
YF**	120/60 110/50	0.90 0.94	0.18 0.20	10 9.5
T	240/60 220/50	1.00 1.05	0.26 0.31	25 27
R	24/60	10.50	2.70	27
RF**	24/60	5.50	0.85	10
L	6 VDC	—	5.00	30
LF†	6 VDC	—	4.00	24
K	12 VDC	—	2.50	30
KF†	12 VDC	—	2.00	24
J	24 VDC	—	1.25	30
JF†	24 VDC	—	1.00	24
D	120 VDC	—	0.25	30
DF†	120 VDC	—	0.20	24
Z	250 VDC	—	0.12	30
ZF†	250 VDC	—	0.10	24

*Based on nominal voltage @ 72°F.
**3000 PSI, 40% rated flow (low watt A.C.)
† 3000 PSI, 85% rated flow

C.S.A. file LR60407



Response Time

Nominal response time (milliseconds) at 5000 PSI, 8.5 GPM

Solenoid Type	Pull–In	Drop–Out
AC	13	20
DC	32	40

Solenoid Ratings

Insulation Class F
Allowable Deviation from rated voltage –10% +15%
Wet Armature Type

Explosion Proof Solenoid Ratings

U.L. (P07) Class I, Div. 1 & 2, Groups C & D. Class II, Div. 1 & 2, Groups E, F & G. As defined by the N.E.C.

M.S.H.A. (P06) Complies with 30 CFR, Part 18.

Solenoid Electrical Characteristics * P06 and P07

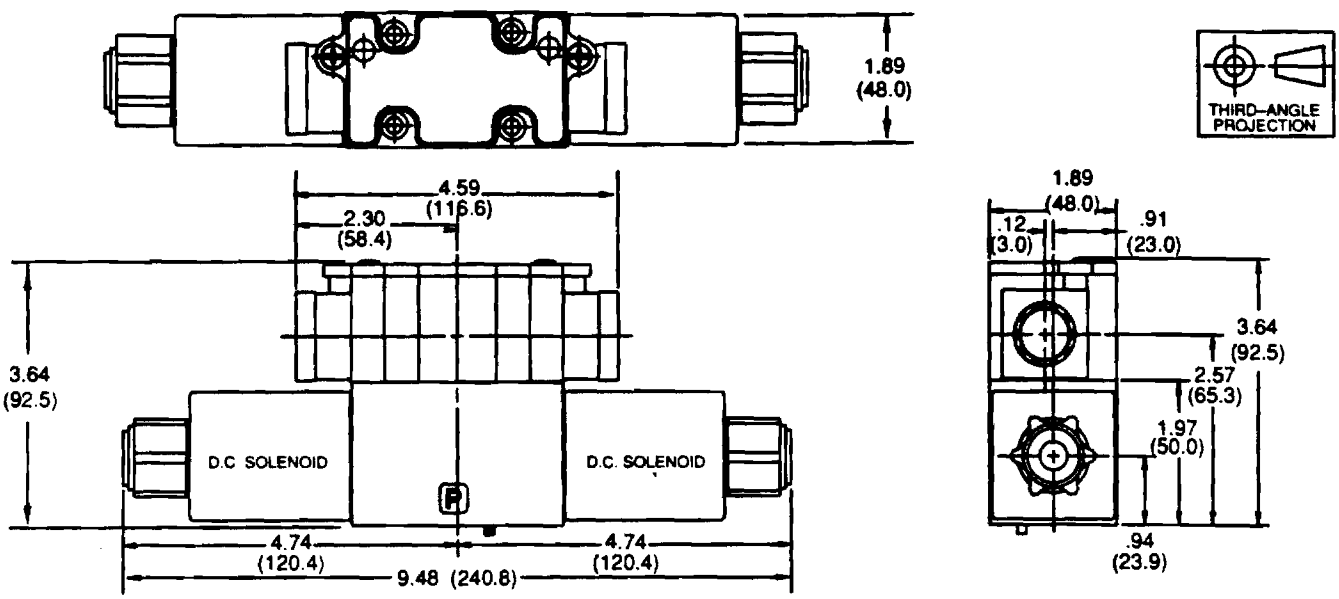
Solenoid Code	Nominal Volts/Hz	In Rush Amps	Holding Amps	Watts
Q	100/60	2.60	0.70	27
Y	120/60	2.20	0.58	27
T	240/60	1.10	0.29	27
R	24/60	11.10	2.90	27
L	6 VDC	—	5.50	33
K	12 VDC	—	2.75	33
J	24 VDC	—	1.38	33
D	120 VDC	—	0.28	33
Z	250 VDC	—	0.13	33

E

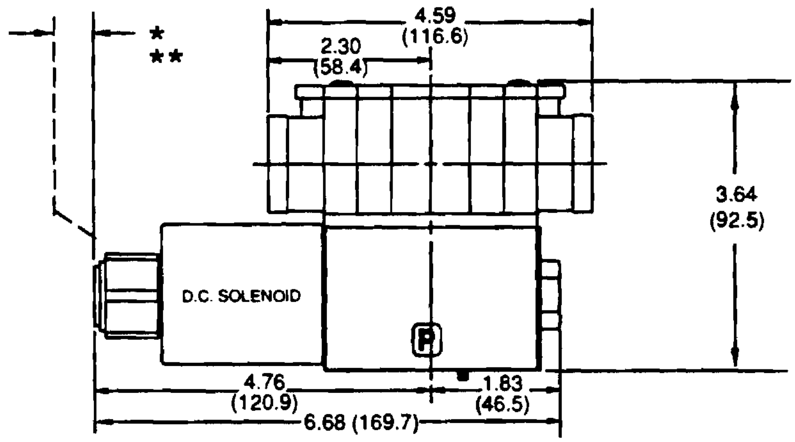
Dimensions

"Millimeter equivalents for inch dimensions are shown in (**)"

Conduit Box, D.C. Solenoid, Double



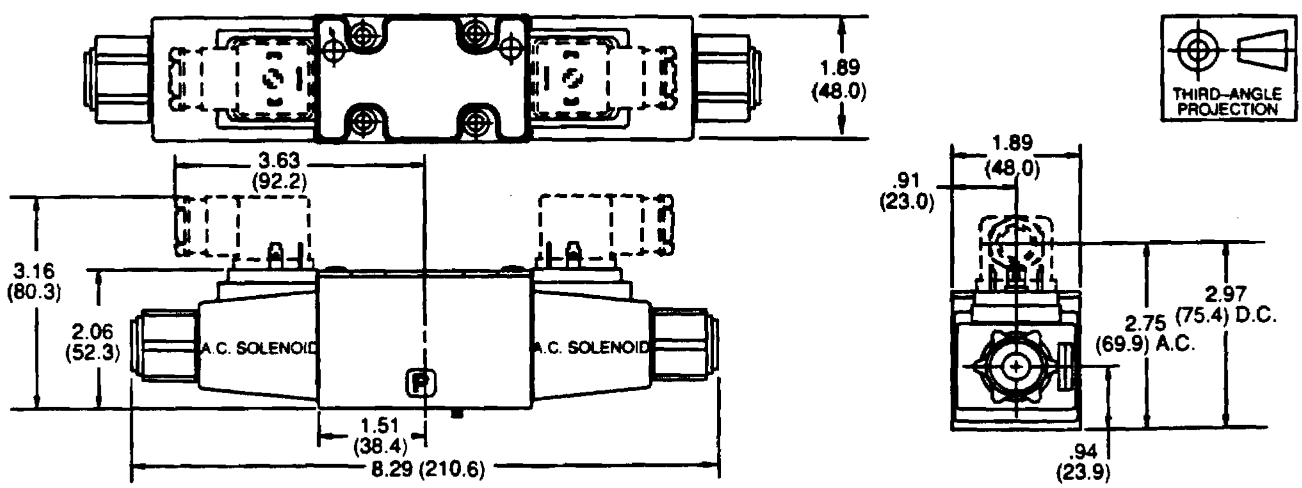
Conduit Box, D.C. Solenoid, Single



* Minimum clearance for coil removal
— D.C. 2.25 (57.2)

** Minimum clearance for solenoid removal
— D.C. 1.00 (25.4)

Hirschmann, A.C. Solenoid, Double



D	1V	W							
DIRECTIONAL CONTROL VALVE	BASIC VALVE	ACTUATOR	SPOOL	STYLE	ATTACHMENTS OR VARIATIONS	SEAL COMPOUND	SOLENOID	SOLENOID MODIF.	DESIGN SERIES

Basic valve: NFPA D03, (formerly D01)

Actuator: Wet armature solenoid

Design series: Note: not required when ordering

Spool

Code	Symbol	Code	Symbol
1	A B P T	10	A B P T
2	A B P T	11	A B P T
3	A B P T	14	A B P T
4	A B P T	15	A B P T
5	A B P T	16	A B P T
6	A B P T	20*	A B P T
7	A B P T	21	A B P T
*8, 9**	A B P T	22	A B P T
		30**	A B P T

* 8 & 20 spool have closed crossover
** 9 & 30 spool have open crossover

Seal Compound

Code	Type
Omit	Nitrile
V	Viton

Solenoid Modif.

Code	Description
F	Low amp coil

Note: Low amp A.C. coils are not CSA approved.

Attachments or Variations

Code	Description
Omit	Standard Valve No variations
4	CSA approval
5	Signal lights
6	Manaplug plug–in
10 ⊙	Monitor switch
56	Lights & manaplug
630	5 pin plug–in manaplug w/single solenoid valve
P06 ⊙	Exp. proof M.S.H.A.
P07	Exp. proof U.L.
P10 ⊙	Spade lug conn.
P14	Extended override & boot
P16 ⊙	High pressure solenoid tube

⊙ Not C.S.A. Approved

Solenoid

Code	Description
Y	120V / 60 Hz 110V / 50 Hz
YW	Standard Hirschmann #
YY	Hirschman with plug
T	240V / 60 Hz 220V / 50 Hz
TW	Standard Hirschmann #
TT	Hirschmann with plug
R	24V / 60 Hz
L	6 VDC
K	12 VDC
KW	Standard Hirschmann #
KK	Hirschmann with plug
J	24 VDC
JW	Standard Hirschmann #
JJ	Hirschmann with plug
D	120 VDC
DW	Standard Hirschmann #
DD	Hirschmann with plug
Z	250 VDC

\# Mating connectors must be ordered separately.

Style

Code	Description
B†	Sgl. solenoid, 2 position, spring offset. P to A and B to T in offset position.
C	Dbl. solenoid, 3 position, spring centered
D†	Dbl. solenoid, 2 position, detent
E	Sgl. solenoid, 2 position, spring offset to center. P to B and A to T when energized.
F	Sgl. solenoid, 2 position. Spring offset, energized to center. Position spool spacer on a side. P to A and B to T in spring offset position.
H†	Sgl. solenoid, 2 position, spring offset. P to B and A to T in offset postion.
K	Sgl. solenoid, 2 position. Spring offset to center. A side. P to A and B to T when energized..
M	Sgl. solenoid, 2 position, spring offset, energized to center position. Spool spacer on B side. P to B and A to T in spring offset position.

Subplate Note: See **"Installation Information, Directional Control Valves"** section of this catalog for subplate drawing and model numbers.

This condition varies with spool code

† Only spools 20 & 30

Unit Weight:
Single Solenoid
3.0 lbs. (1.36 kg)
Double Solenoid
3.5 lbs. (1.6 kg)

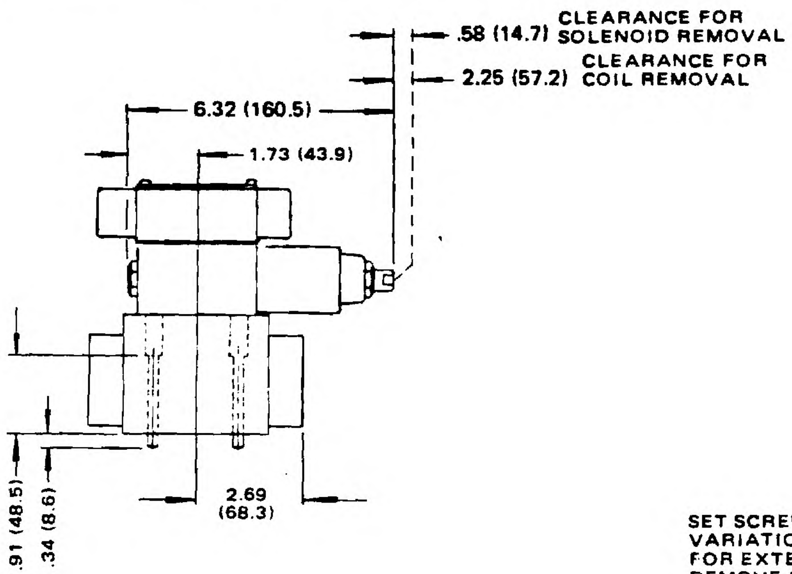
echnical Information

IMENSIONS

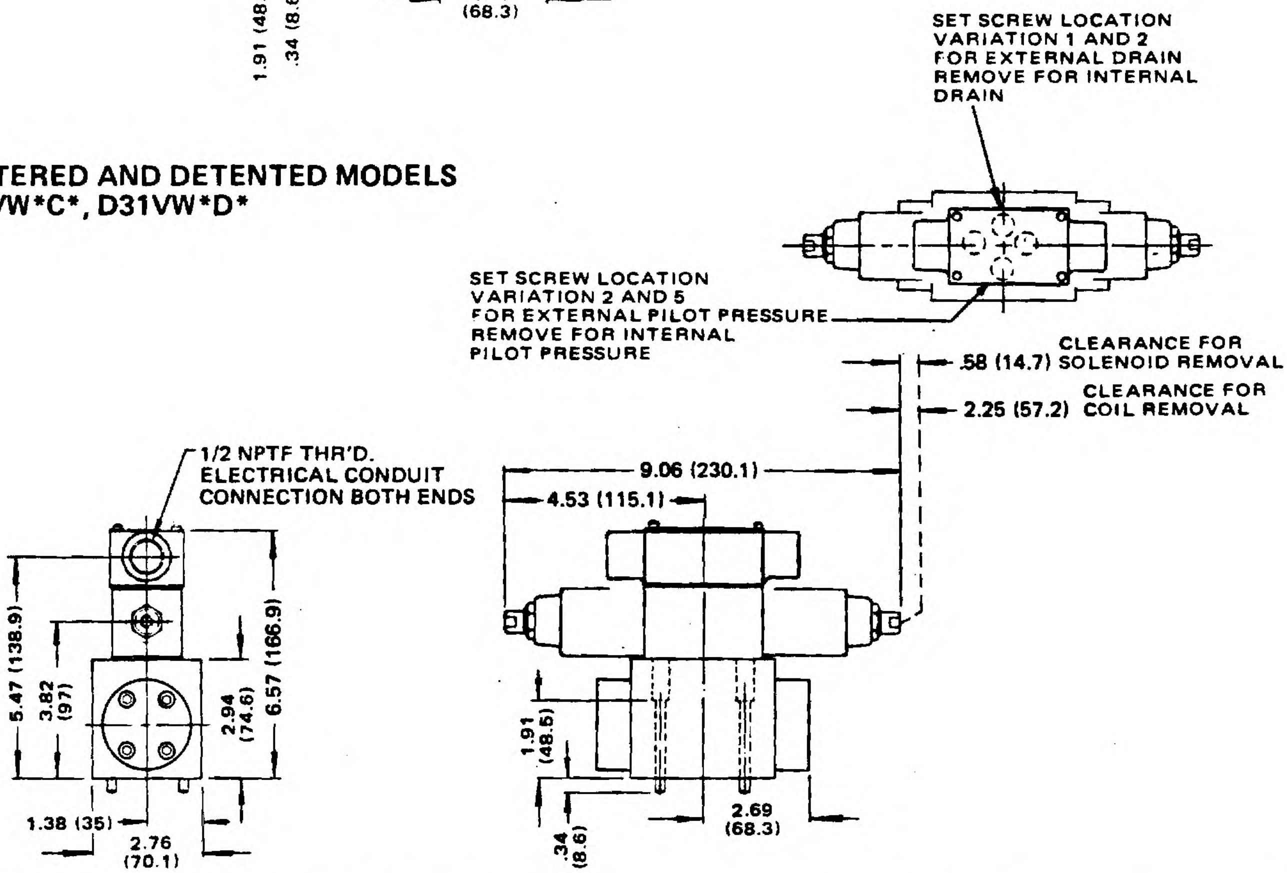
ILLIMETER EQUIVALENTS FOR INCH DIMENSIONS ARE SHOWN IN (**)"

C SOLENOID

INGLE SOLENOID, SPRING OFFSET MODELS
31VW*B*, D31VW*E*, D31VW*F*, D31VW*H*



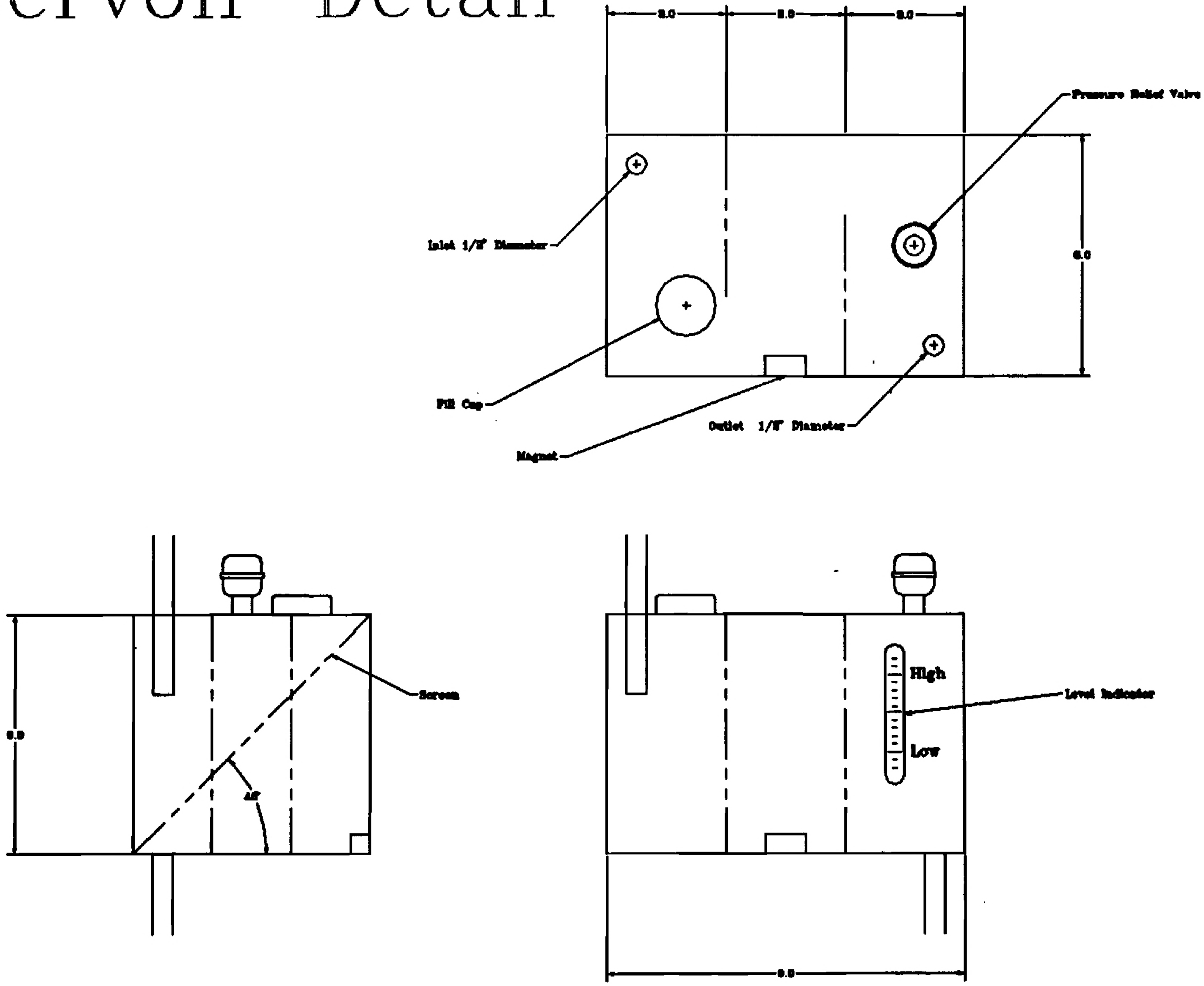
CENTERED AND DETENTED MODELS
D31VW*C*, D31VW*D*



E

APPENDIX H

Reservoir Detail



H1

Power Supply – Group 5

1/4" = 1"
ME4182a1
Revision 1

Appendix H2

Reservoir Description

The body of the reservoir is to be made from steel which is about 40 thousandths of an inch thick. This is so that the sides can be welded to the top and the bottom. The screen is 100 mesh steel screen. It should be placed at a 30° to 45° angle to the bottom. This will provide optimum performance of removing the air from the fluid. The baffles are to be made from the same steel as the body. Two baffels were used to increase the lenght the fluid has to flow without increasing the velocity of the fluid.

The level indicator is available from Grainger and is made by Dayton. The model to be used includes a temperature guage and is model number No. 1A760 Reservoir Fluid Level Guage. The cost is less than $14. The filler cap is a standard container cap available from several vendors. The pressure relief valve is a standard relief valve and should be set for about 2 psi. The magnet is a permant type and should be mounted on the bottom of the reservoir where the velocity of the fluid is the least. The inlet and outlet ports will require fittings to connect them with the hydraulic lines.

ACTUATORS

MOTORS

FINAL DESIG

STE

B

ALPESHKUMAR

DAVID SH

JOHNSO

STEVE

EXECUTIVE SUMMARY

The purpose of the actuators and motors group is to provide a system for the chassis that will enable the wheels and the joints to move by selecting motors and the drive systems. This report investigates the possible drive systems by their torque and power requirements. The chassis needs six motors for the wheels and six motors for the joints. Char-Lynn model 101-1057 motors were chosen for the wheels and White Hydraulics model RS-04 motors were chosen for the articulation joints. The drive system for the wheels includes a spur gear system which connects onto a drive shaft. The drive system for the articulation joints includes a silent chain attached to the tube, run by a motor gear.

TABLE OF CONTENTS

LIST OF FIGURES

INTRODUCTION

Problem Statement

Our project was to develop a means of driving the six wheels of a remote control lunar rover, the ENABLER, and also to provide for the relative rotation of the six articulation joints that were used for steering and negotiating obstacles. The use of hydraulic power was required.

The wheel drive system should be lightweight, inexpensive, and as small as possible. Also, this drive system should be easy to manufacture.

The articulation joint drive should be simple, lightweight, easy to build, and inexpensive.

DESIGN OVERVIEW

Wheel Drive

The wheel drive involves five main parts: a hydraulic motor, a drive shaft, a small gear train, a motor bracket and a wheel plate that connects the drive system to the existing wheel hub. Only two components of this system, the motor bracket and the wheel plate, will have to be custom machined. The motor transmits power to the hub through a set of two steel spur gears. These gears give an overall drive ratio for the system of 1:1.465. At its maximum performance, the hydraulic motor will have an output torque of approximately 876 in-lbs. The lower gear is held onto the motor shaft by means of a specially cut end plug which can be screwed directly into the motor shaft. The upper gear is held in place by means of a key way and set screw. The drive shaft is supported on one end by the wheel plate, and on the other end by a small Torrington needle bearing. See Figure 1.

Articulation Joint Drive

Driving the articulation joints requires significantly more power than the wheels. The drive system will be subjected to varying and reversing loads, torques, and impacts. This necessitates the use of powerful motors that drive a lightweight, but strong, system. The system must also adapt to small misalignments of the drive train.

The drive system consists of the following items:

-Hydraulic Motor
-Motor Mount Bracket
-Drive Sprocket
-Drive Chain
-Chain Clamps

CONSTRAINTS

Wheel Drive

The items below, in Table 1, are the constraints on the wheel drive.

Table 1

Temperature	-5° to 100° F
Motor Controls	Forward, reverse, brake, freewheel
Motor Horsepower	.65 hp
Motor Torque	876 in·lbs
Final Drive Torque	1091 in·lbs
Motor Size	7"x4"x4", small as possible
Motor RPM	49.5 RPM
Manufacturability	All components should be manufactured by Georgia Tech equipment if possible
Mobility	Move on 4 wheels up a 17° incline
Supplied Pressure	2000 psi

Articulation Joint Drive

The items below, in Table 2, are the constraints on the articulation joint drive.

Table 2

Temperature	-5° to 100° F
Motor Controls	Forward, reverse, free spin
Motor Horsepower	.75 hp
Motor Torque	1350 in·lbs
Supplied Pressure	2000 psi
Motor Size	7"x4"x4", small as possible
Motor RPM	35 to 60 RPM
Manufacturability	All components should be manufactured by Georgia Tech equipment if possible
Mobility	180° cw and ccw from a 0° reference

SELECTIONS FOR THE WHEEL DRIVE

Motor

The selection of the motor is the most important part of our project. Any other analysis or selection was based on the type of motor to use. We have chosen the Char-Lynn H-series #101-1057 Hydraulic Motor to drive the wheels. The motor has a 1" SAE 6B splined drive shaft and 7/8-14 o-ring ports, with a 4-bolt flange for connection to the motor bracket. With the 0.6 GPM flow rate given, each motor will be able to obtain a RPM of 49.5 RPM. The rated torque of each motor is 876 in·lbs, making the motors 98.3% efficient. The motor will be able to run at a pressure of 2400 psi, which is well within the 2000 psi constraint. Our motor will be using a gear system to obtain the correct amount of RPM to the wheels. The gear ratio that we will use will be 1:1.465. This motor is the optimum motor because of its' abilities and it's size. Not many hydraulic motors fit the pressure, flowrate, and displacement requirements. This motor is relatively small and is easy to mount. This motor is easily accessible and relatively inexpensive, $130 per motor. We will need six of these motors, one for each wheel.

Motor Bracket

A motor bracket was designed to keep the motor stable during operation, as well as, provide a place to hold the wheel drive shaft. As seen in Figure 2, it is a slab of ANSI 6066 Aluminum 4" wide and 0.5" thick. The bracket will be mounted onto the upper sleeve with six ANSI UNC Type 8, 0.164 diameter SAE Grade 1 steel machine screws. Each screw will be 1" in length. On the lower half, the motor is connected by the use of the 4-bolt flange and four 3/8-16 UNC machine screws, each 1" in length. The drive shaft of the wheels is also stabilized on the motor bracket by going through the center with the bearing used for semi-frictionless rotation. The motor bracket will be manufactured by the equipment supplied at Georgia Tech. As seen in Figure 6 and 7, finite analysis was done on the bracket using the I-DEAS program to determine the dimensions. The dimensions were selected to obtain the maximum support for the motor and shaft while using size and weight as constraints.

Gear Drive

Since there isn't a motor available that can be connected directly onto the wheel drive shaft, a gear system is used to obtain the correct wheel RPM. A simple spur gear system with a gear ratio of 1:1.465 will be used. The two gears chosen are from the Lynn Gear Company. They both have a diametrical pitch of 8, a face width of 1 1/4", and a pressure angle of 14.5°. The motor shaft gear, the driver, will be gear #8FS15, which has 15 teeth and costs $27.10. The wheel drive shaft gear will be gear #*FS22, which has 22 teeth and costs $36.90. We will use six of each gear for the final project. The driver gear will connect onto the motor by use of a AISI 1040 steel plug as seen in Figure 3. The plug will be holding the gear by the use of a 1/4-20 UNC-2B machine screw that press fits the plug against the gear, which is on the splined shaft. The wheel drive shaft gear will connect onto the drive shaft by the use of a AISI 1040 steel key as seen in Figure 3. The key will have dimensions of 2"x1/4"x3/16". The gear will be press fitted onto the key. A gear system was chosen over other methods of drive due to simplicity, dependability, cost, and accessibility.

Drive Shaft

The drive shaft of the wheel drive connects the power from the motor to the wheels. We will be using a stock 15/16" diameter AISI 1040 steel shaft which is available from Bearings and Drives Incorporated which is located in Atlanta. As seen in Figure 4, the shaft will be connected to the hub mount on one end, then have a gear connected onto it on the other end, with support from the motor mount. The shaft is 7.856" long and will have two key ways cut into it to hold the AISI 1040 steel keys, as seen in Figure 4. The section cut into the shaft that holds the gear on will have dimensions of 2"x1/4"x3/32". The section cut into the shaft that holds the shaft onto the hub mount has dimensions 2.25"x1/4"x3/32". The gear will be press fitted onto the shaft and then the shaft will be press fitted into the wheel plate.

Bearing

The bearing maintains the smooth rotation of the wheel drive shaft. We have chosen a Torrington needle roller bearing #B-1516 which is open ended. The needle bearing has a 15/16" bore and an outer diameter of 1 3/16". The width of the bearing is 1" and it has a dynamic load rating of 4750 lbs and a static load rating of 13600 lbs. The bearing is designed to withstand a great magnitude in radial forces, which are the only forces acting on this bearing. The bearing will be press fitted into the motor bracket. The bearing costs $5.54 and is available from Bearings and Drives Incorporated.

Wheel Plate

The wheel plate was designed so that the wheels could be connected to the drive shaft. The wheel plate is connected onto the hub as seen in Figure 5 by eight equally spaced 8-SAE grade 1, ANSI UNC-8 machined screws. The wheel plate will be made out of 6066 Aluminum and will be manufactured at Georgia Tech. The drive shaft will connect onto the hub mount by way of a 1040 steel key. A groove will be cut into the wheel plate with dimensions 2.25"x1/4"x3/32". Then, the drive shaft will be press fitted into the wheel plate.

SELECTIONS FOR ARTICULATION JOINT DRIVE

Motor

The motor selection was based on the articulation joint constraints. The motor selected is the White Hydraulics RS-04 model. This is a 4-hole face mounting motor with a displacement of 4.62 in^3 and a rated torque of 1480 in-lbs. at 2000 psi. The motor shaft is 1" in diameter and 1.75" in length, and employs a key way and set screw for mounting accessories. The motor is to be mounted on a bracket that attaches to the upper bearing flange of the articulation joint. The inlet and outlet ports are 7/8" in diameter and are located on the top of the motor for easy access to the hydraulic supply lines. See Appendix H for the performance details and limits.

This motor met our space limitations as well as our torque requirements. It weighs only 13 lbs, which will not put undo stress on the mounting bracket or the bearing sleeve. Also, this motor can handle the overhung loads produced, therefore, an overhung adapter will not be necessary.

Because the motor shaft is larger than the maximum bore of the drive gear, a standard adapter will be used to step down the shaft size. A non-standard motor shaft was far too costly, and machining the motor shaft was nearly impossible. The additional length of the adapter will be compensated by cutting off the motor shaft.

Motor Mount Bracket

The bracket was designed based on a load capacity of 2160 lbs., and geometric constraints that limit the size of the bracket and dictate the position of the motor. The bracket will be subjected to reversing loads and moments. The mounting face is 3/8" thick, with slotted holes for accurate motor positioning, and is fixed to a curved, .25" thick base that bolts to the upper bearing flange inside the joint, using three 3/8" bolts (see Figure 11). Steel inserts will be fixed within the upper flange to accept the mounting bolts. Approximately 3/8" of material will have to be removed from the edge of the bearing flange in the area where the bracket will mount. This is necessary to properly position the motor relative to the drive components on the lower bearing flange. The bracket will be machined from 6066 aluminum. A finite element analysis was performed on this particular bracket design and verified the feasibility of its application (see Figures 13 to 14).

Originally, a fixture for restraining the hoses and cables of the power distribution system to the center of the tube body was integral to the motor mount bracket, but it was discovered that it was unnecessary due to the method of hose connection and routing used by the power group.

Drive System

The final articulation joint drive design employs a Morse 3/16" pitch silent chain. The chain, which is 1" wide and has 160 links, will be mounted to the inside surface of the lower bearing flange as an internal "pseudo-" spur gear. The chain will be driven directly by the motor through a 1.25" diameter spur gear (maximum bore 11/16") and will have to endure loads in excess of 2100 lbs. This results in a final gear ratio of 8:1 and a motor torque of 1350 in.-lbs.

The silent chain was chosen because of its light weight, small size, and high strength as opposed to steel gears or belts. Also, a chain does not need to be located as accurately as an internal or planetary gearing system would, making for greater reliability under use. The silent chain was chosen over other types of chains because its high tooth profile requires less wrap on the sprocket, and because the desired gear ratio could be preserved, whereas it could not with a roller chain of comparable pitch. See assembly drawings in Figures 8, 9, and 10.

Chain Clamps (Cleats)

The silent chain will be mounted to the lower bearing flange using specially-designed cleats. Six to eight cleats will be bolted to the flange and will restrain the chain in all directions. The cleats will be machined from 1040 steel and will be fastened to the lower bearing flanges by two 5/16"-1.5" bolts. Steel inserts will also be used here to anchor the mounting bolts in the aluminum. These cleats will have ~5 teeth machined into them which engage the drive chain. These cleats will make for easy chain replacement and maintenance. See figure 12 for clarity.

CONCLUSIONS AND RECOMMENDATIONS : ARTICULATION JOINTS

Analysis of the selected components shows that they are adequate for most performance specifications. However, our group has some concerns regarding the overall dependability of the total design under certain modes of operation.

One of our main concerns pertains to the sporadic motor rotations resulting from the low available flow rates from the power distribution unit. The selected motor had to meet certain torque requirements, which limited the range of possible alternatives. A motor capable of smooth operation at these low flow rates would not have generated the necessary torques. These sporadic rotations of the motor could cause severe impact loading on the articulation drive components (i.e. gear teeth, bolts, and drive chain). This loading could greatly reduce the service life of all related components.

Another concern regards the control of the articulation joint motors. Due to the varying torque loads on each motor, a motor rotation speed control combined with a position control would be necessary for optimum performance. This type of control would facilitate synchronization of joint rotations and smoother operation.

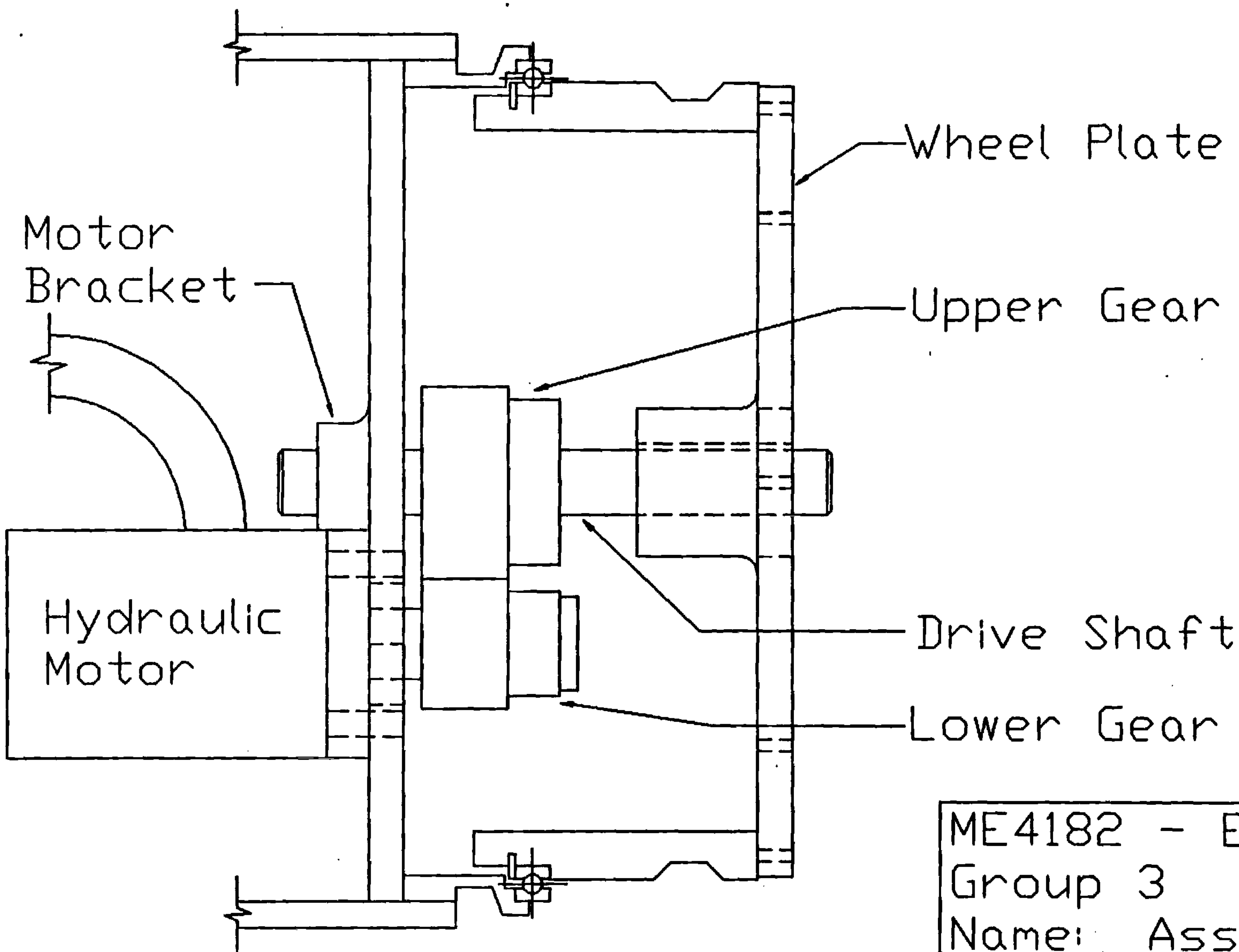
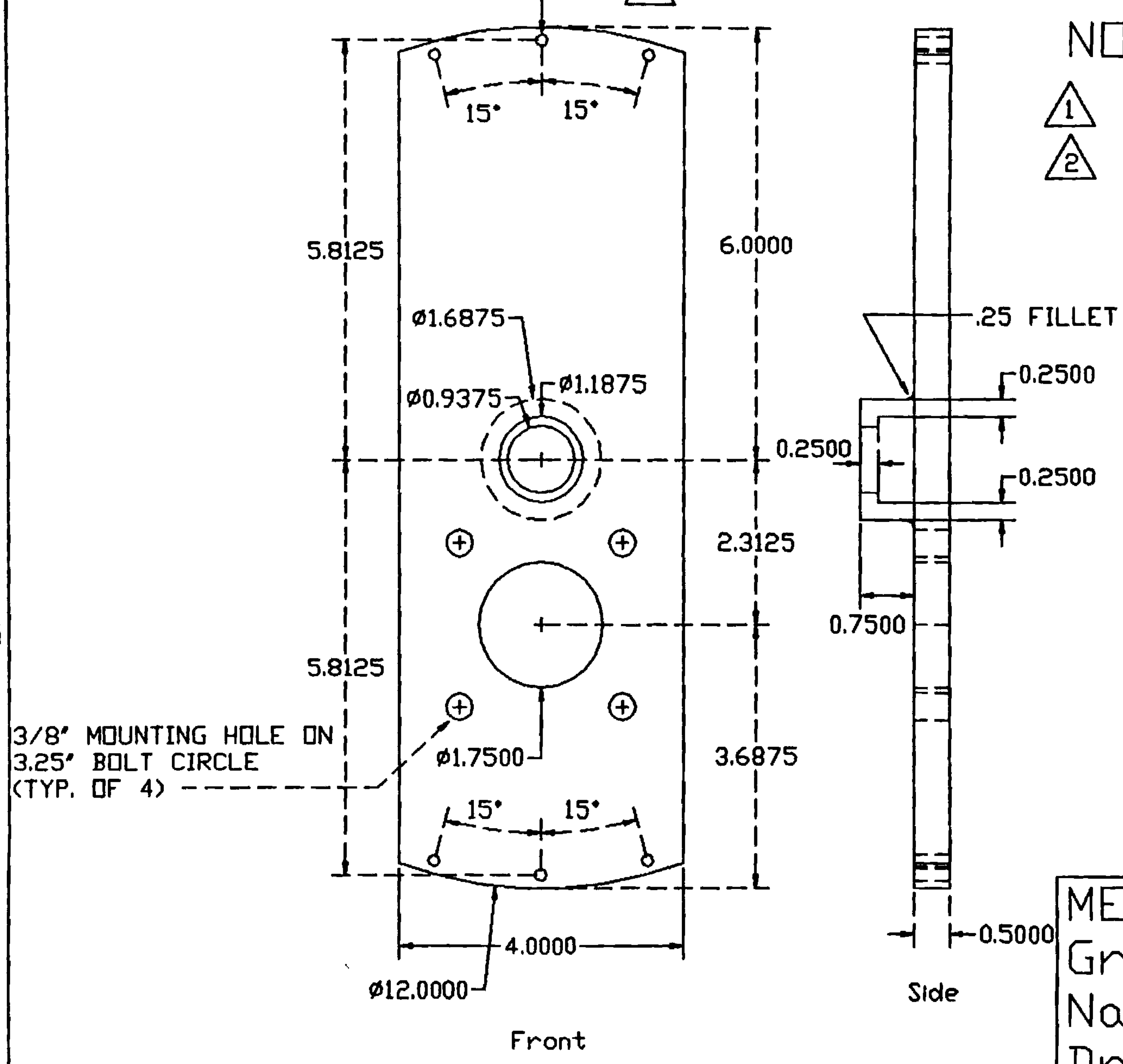


Figure 1

ME4182 - Enabler
Group 3
Name: Assembly
Drawn By: DSS
Date: 3/10/93
Scale: none

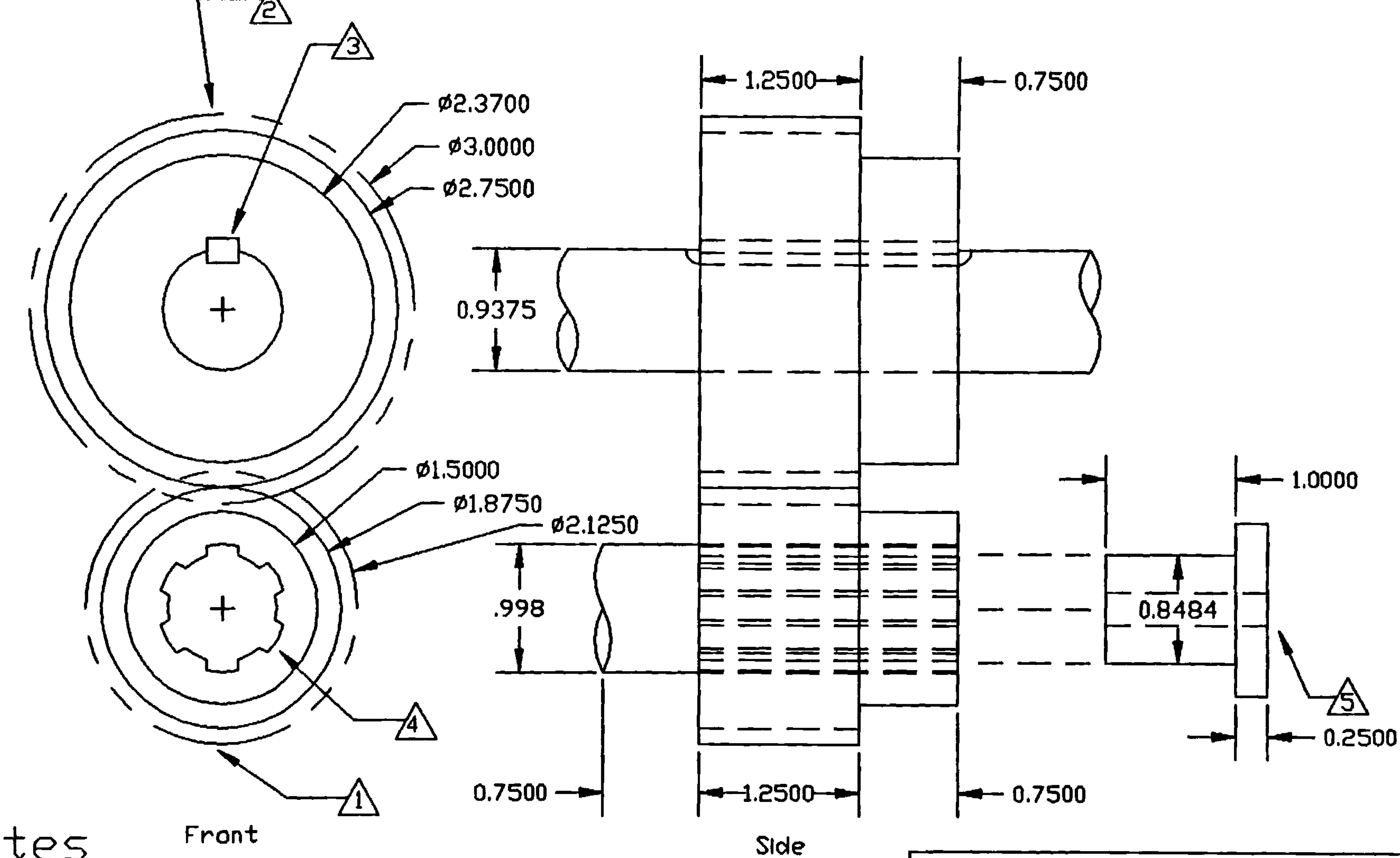


NOTES:

1. Material - ANSI 6066 Aluminum
2. ANSI UNC-8 Bolts on 5.8125" diameter bolt circle

Figure 2

ME4182 - Enabler
Group 3 - Wheels
Name: Bracket
Drawn By: DSS
Date: 3/10/93
Scale: none

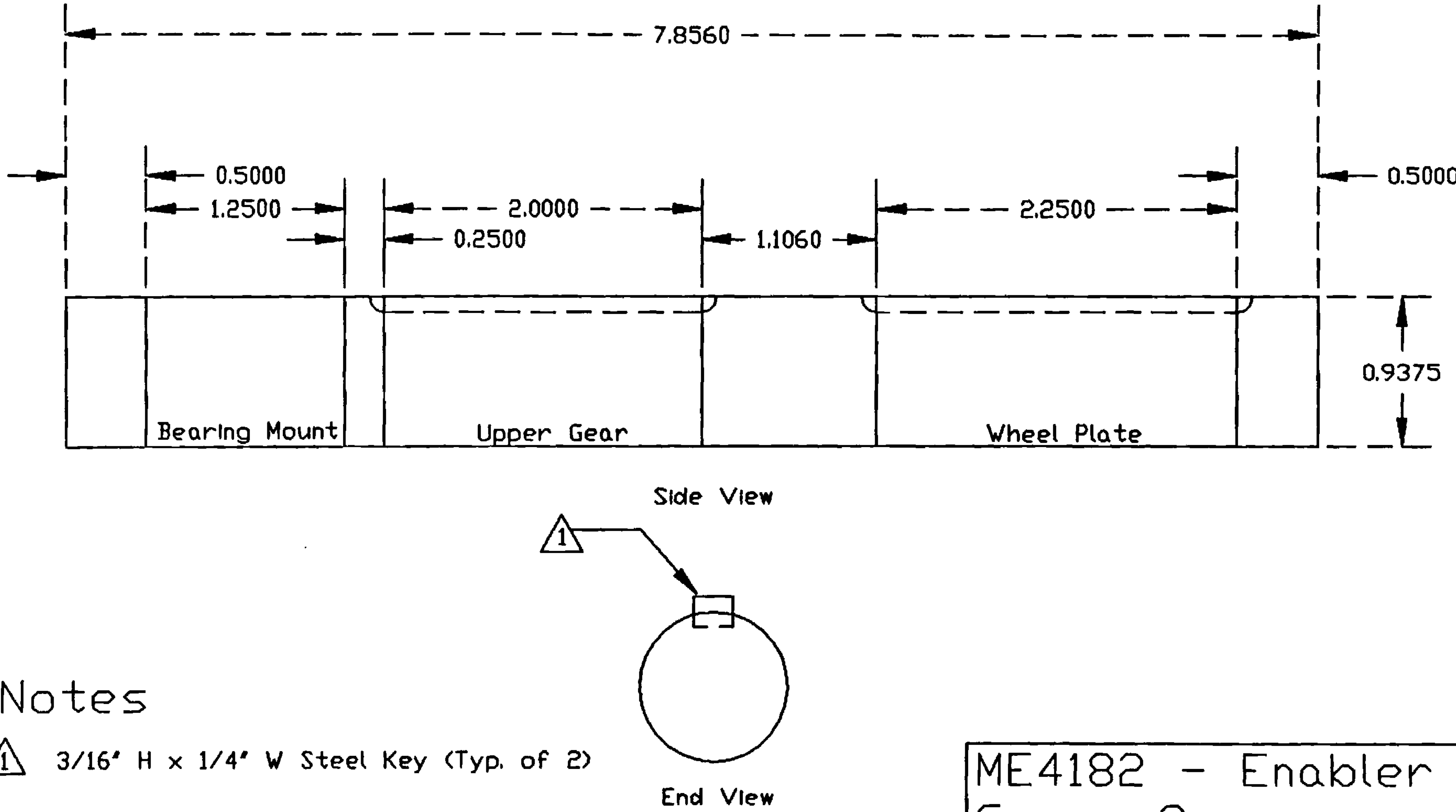


Notes

1. Lower Gear - 8 DP, 15 Teeth, AISI 1040 Steel, 14.5 P. Angle
2. Upper Gear - 8 DP, 22 Teeth, AISI 1040 Steel, 14.5 P. Angle
3. AISI 1040 Steel Key, 1/4" W X 3/16" H
4. SAE 6B Spline - Maximum Diameter of .998"
5. 1/4"-20 UNC-2B Threaded Hole

Figure 3

ME4182 - Enabler
Group 3
Name: Gearing
Drawn By: DSS
Date: 3/10/93
Scale: none

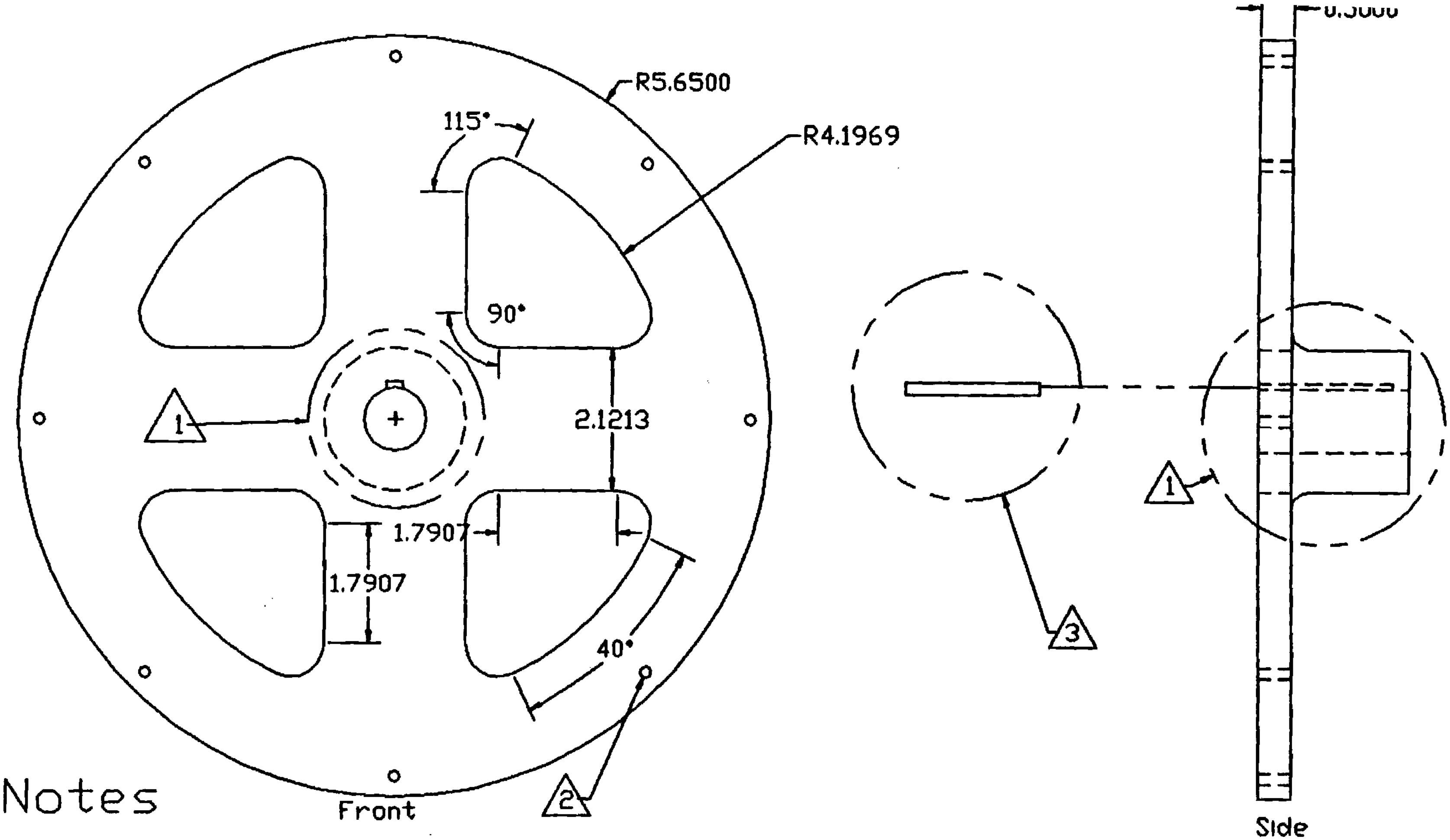


Notes

1. 3/16" H x 1/4" W Steel Key (Typ. of 2)
2. Shaft Material AISI 1040 Hardened Steel

ME4182 - Enabler
Group 3
Name: Shaft
Drawn By: DSS
Date: 3/10/93
Scale: none

Figure 4



Notes

1. See Detail Drawing 1
2. 8-SAE grade 1, ANSI UNC-8 Bolts, 10.65" DIA.
3. 2-14" L x 3/16" H x 1/4" W AISI 1040 Steel Key
4. Made From 6061 Aluminum

Figure 5

ME4182 - Enabler
Group 3
Name: Wheel Plate
Drawn By: JRS
Date: 3/10/93
Scale: none

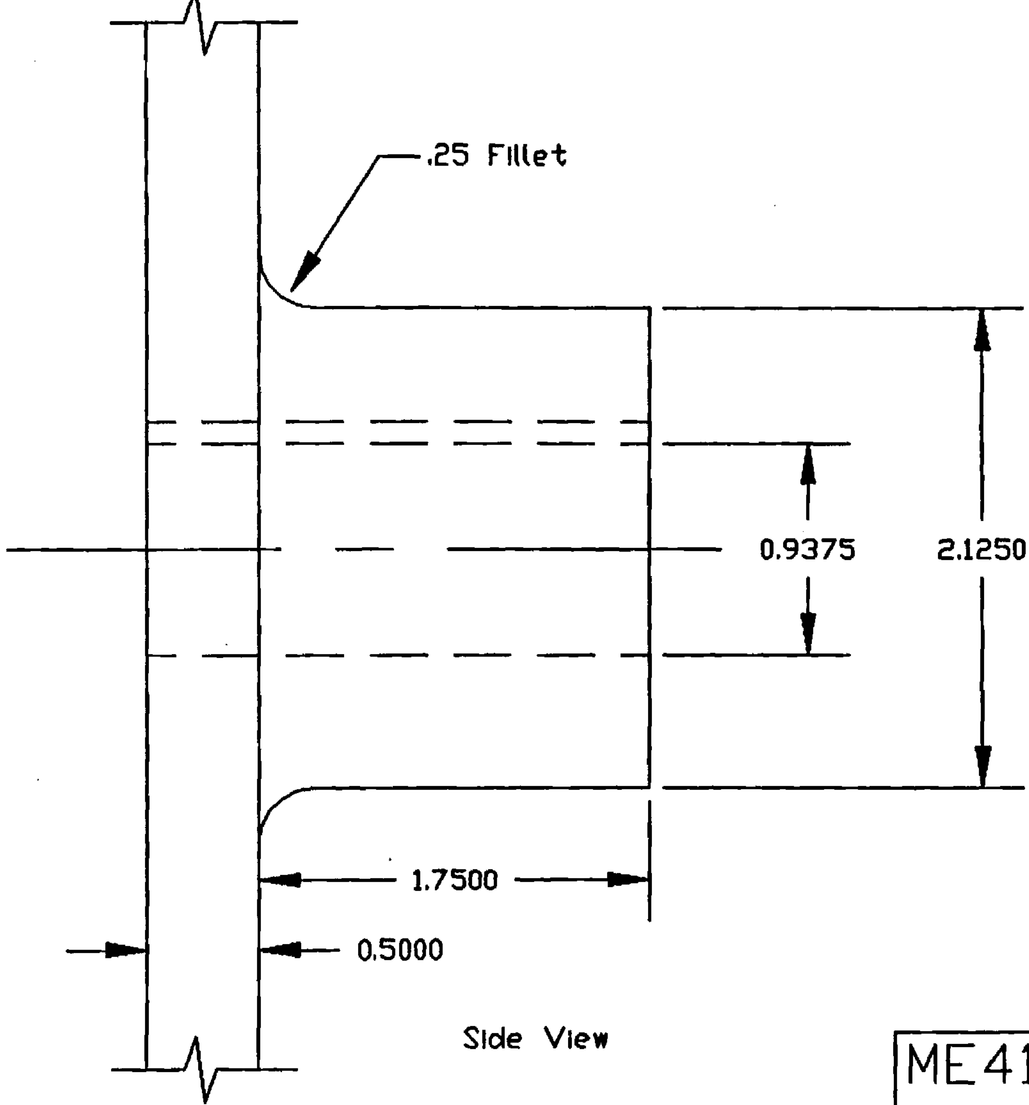
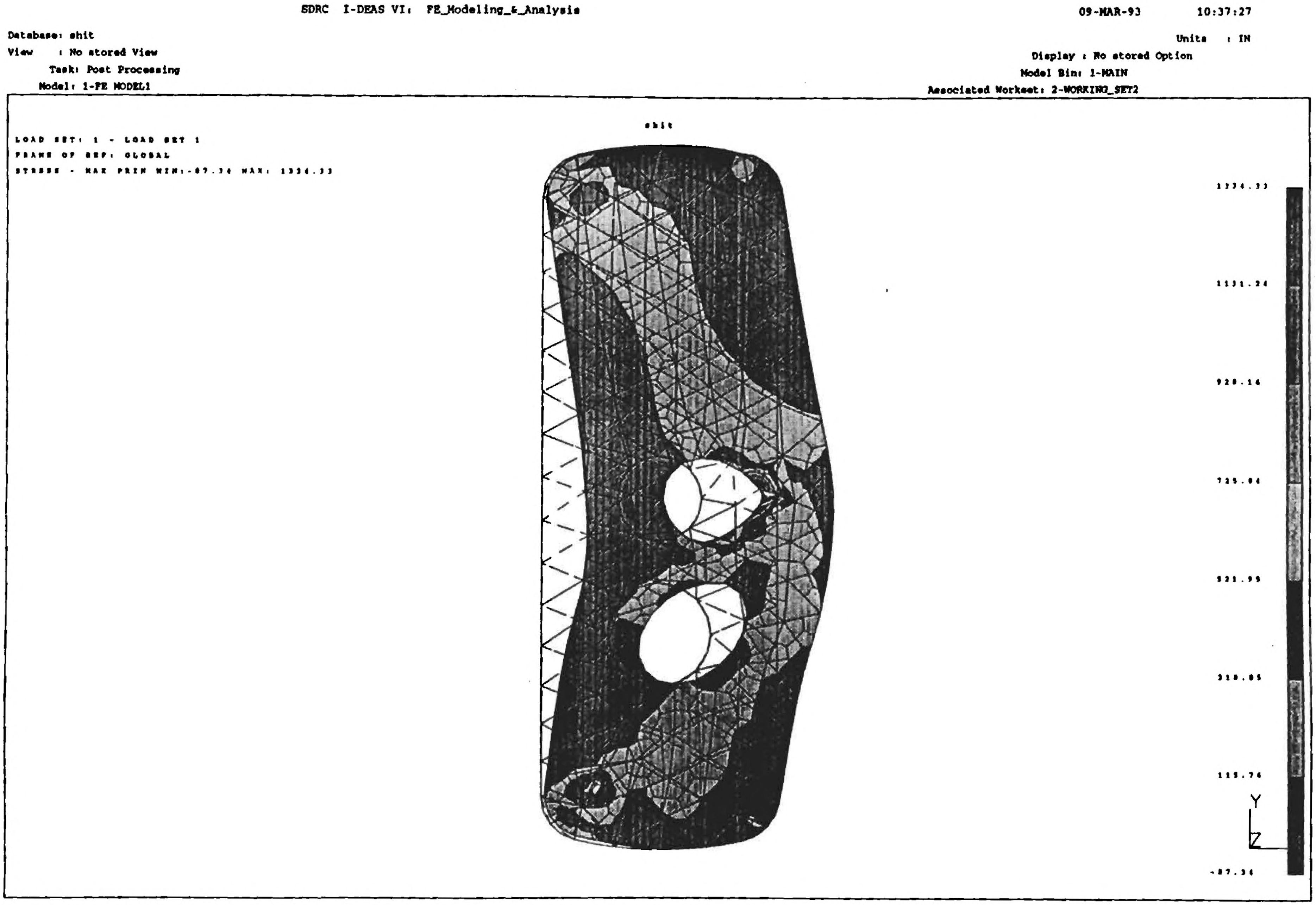


Figure 5 - Detail 1

ME4182 - Enabler
Group 3
Name: Detail 1
Drawn By: DSS
Date: 3/10/93
Scale: none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09-MAR-93 10:37:27
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View : No stored View
Task: Post Processing
Model: 1-FE MODEL1
Units : IN
Display : No stored Option
Model Bin: 1-MAIN
Associated Workset: 2-WORKING_SET2
shit
LOAD SET: 1 - LOAD SET 1
FRAME OF REF: GLOBAL
STRESS - MAX PRIN MIN:-87.34 MAX: 1334.33
1334.33
1131.24
928.14
725.04
521.95
318.85
115.76
-87.34
Y
Z

Figure 6

SDRC I-DEAS VI: FE_Modeling_&_Analysis 09-MAR-93 10:34:52

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View : No stored View Display : No stored Option

Task: Post Processing Model Bin: 1-MAIN

Model: 1-FE MODEL1 Associated Workset: 2-WORKING_SET2

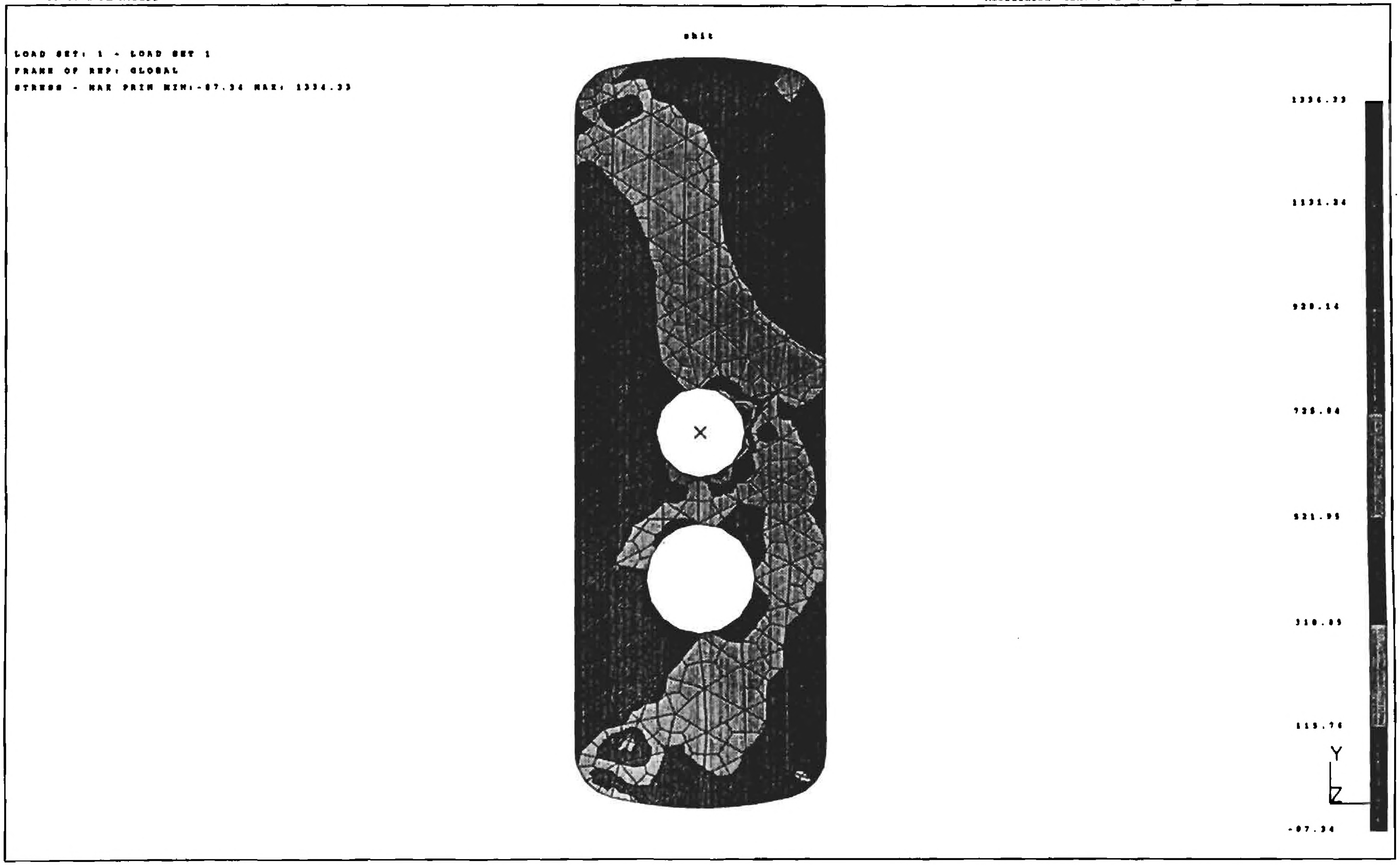
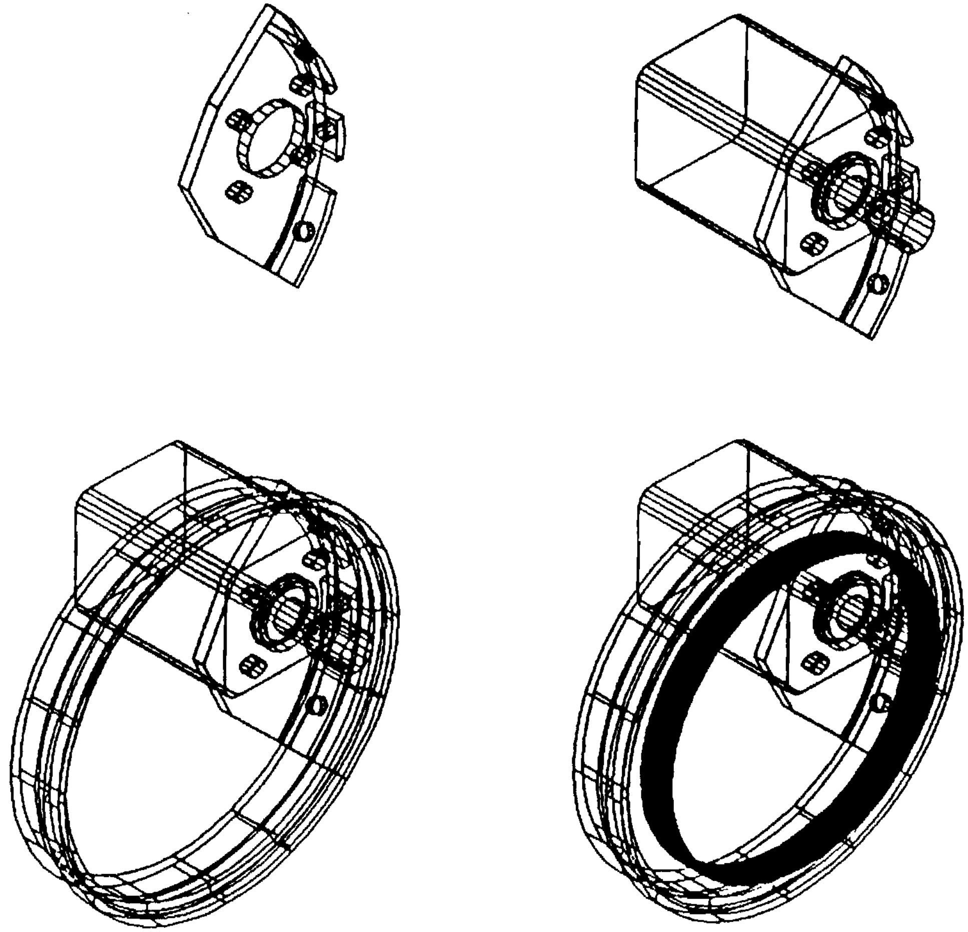


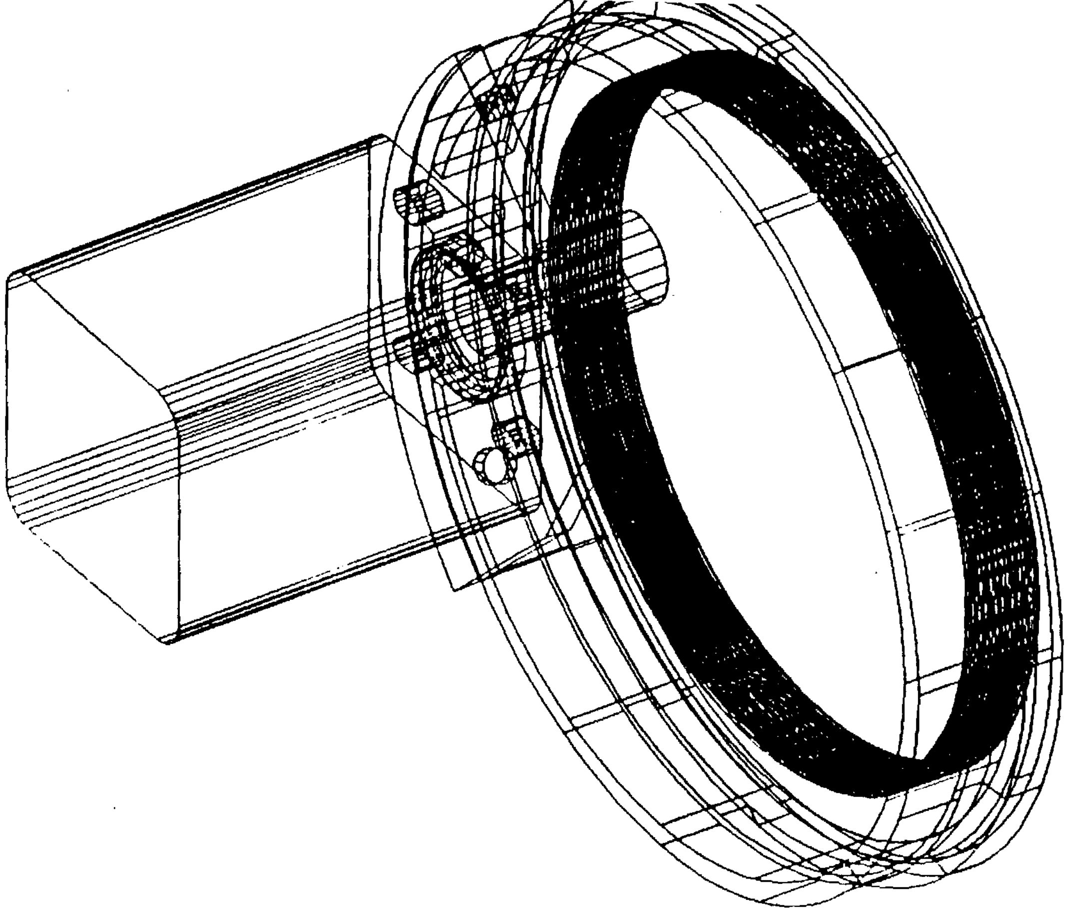
Figure 7



3-D ASSEMBLY

ME4182 - Enabler
Group 3
Name:3-D Assembly
Drawn By: ARM
Date: 3/10/93
Scale: none

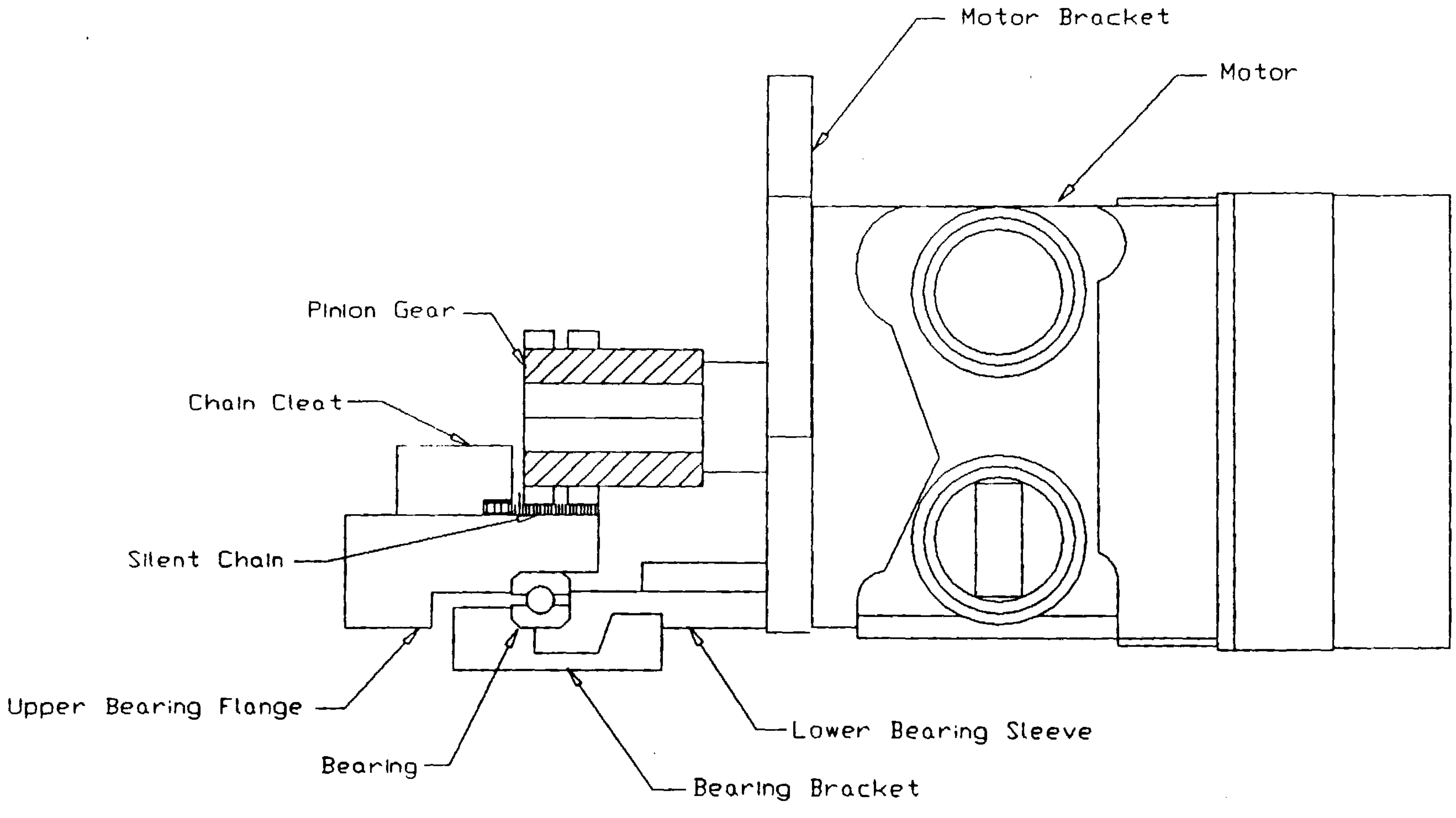
Figure 8



ME4182 - Enabler
Group 3
Namé:3-D Assembly
Drawn By: ARM
Date: 3/10/93
Scale: none

3-D ASSEMBLY

Figure 9



2-D ASSEMBLY

ME4182 - Enabler
Group 3:Art Joint
Name: Assembly
Drawn By: ARM
Date: 3/10/93
Scale: none

Figure 10

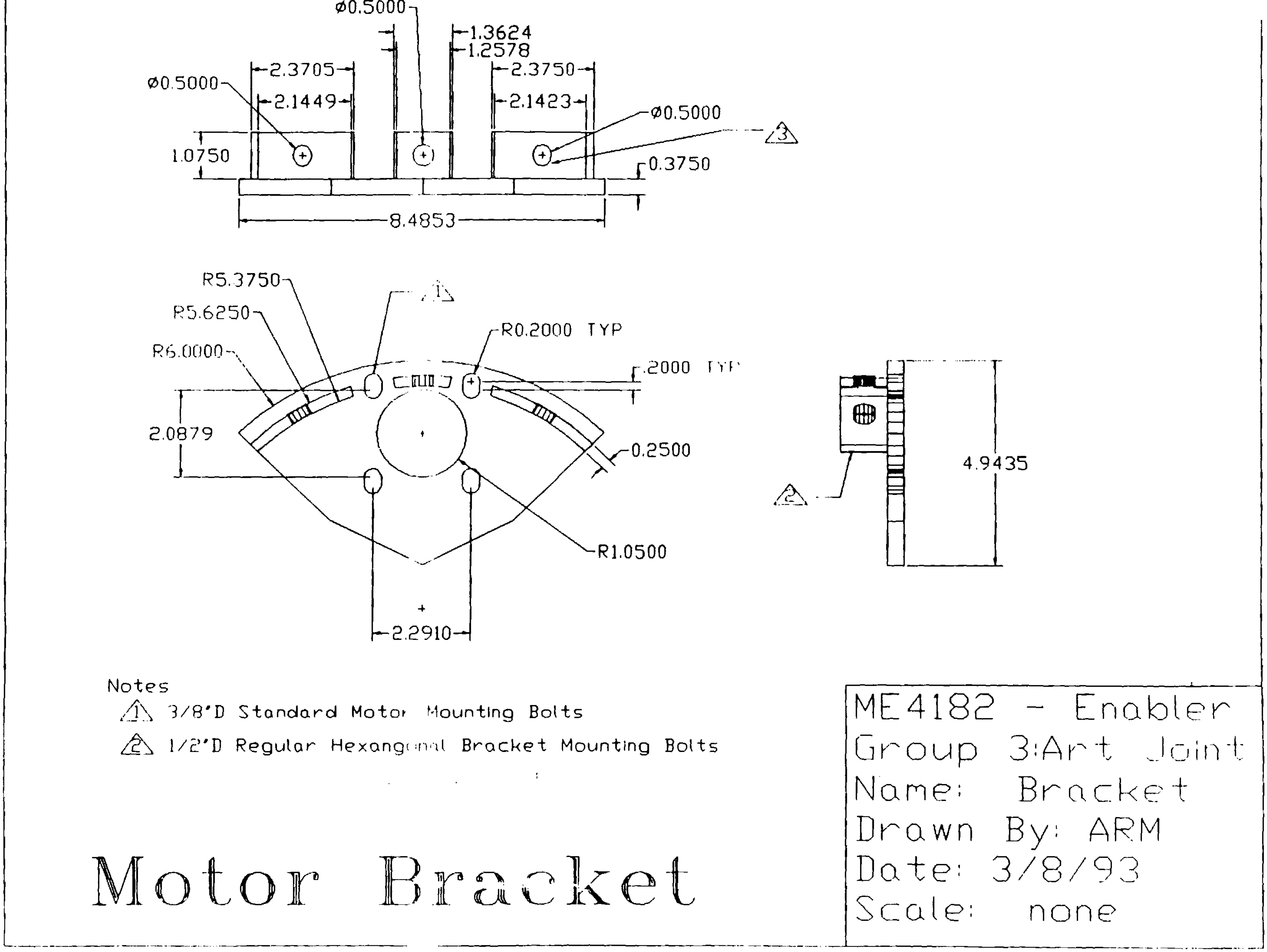

ø0.5000
1.3624
1.2578
2.3705
2.3750
ø0.5000
2.1449
2.1423
ø0.5000
3
1.0750
0.3750
8.4853
R5.3750
1
R5.6250
R0.2000 TYP
R6.0000
.2000 TYP
2.0879
0.2500
4.9435
2
R1.0500
2.2910
Notes
1 3/8"D Standard Motor Mounting Bolts
2 1/2"D Regular Hexangonal Bracket Mounting Bolts
Motor Bracket
ME4182 - Enabler
Group 3:Art Joint
Name: Bracket
Drawn By: ARM
Date: 3/8/93
Scale: none

Figure 11

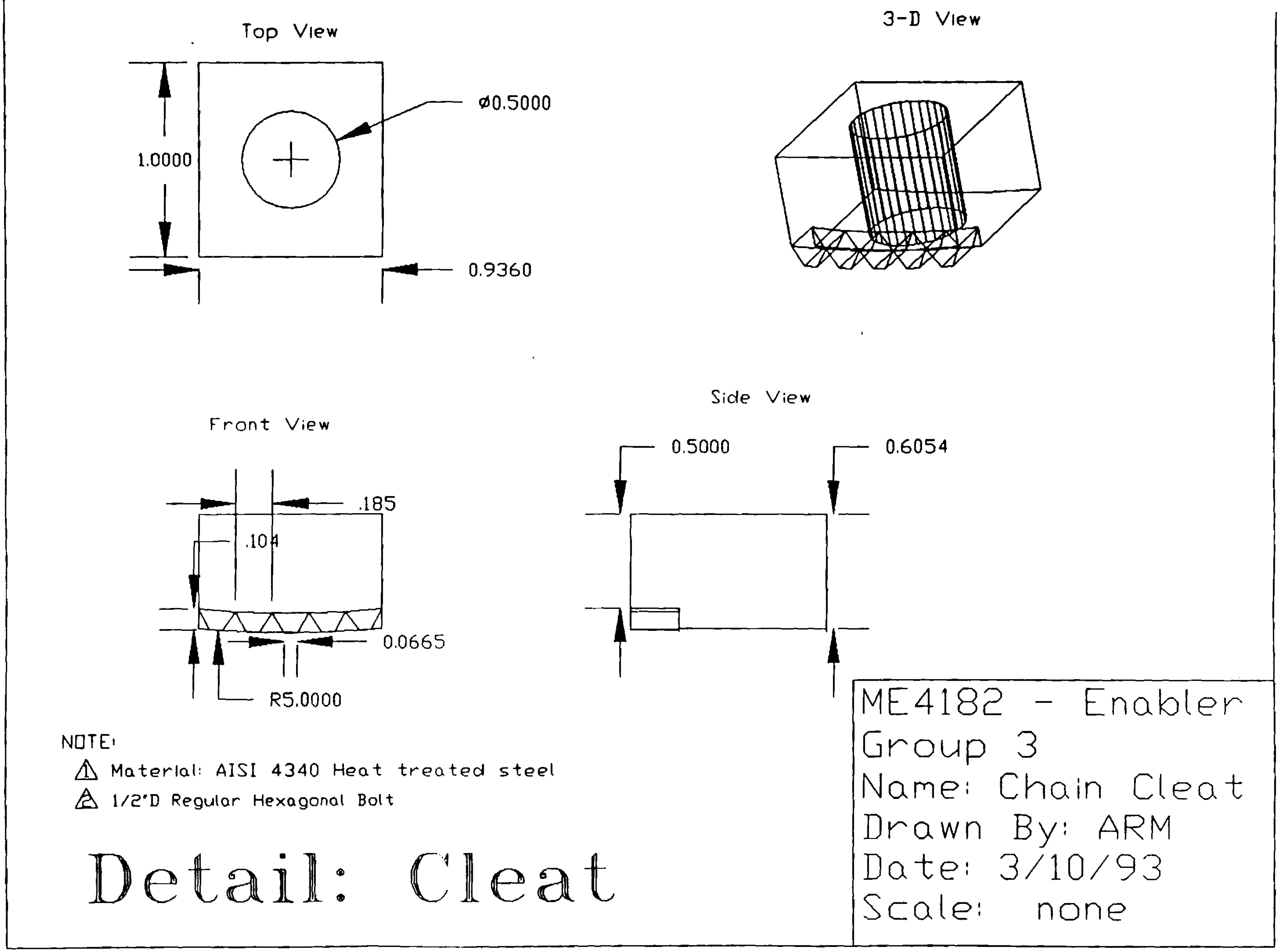

Top View
3-D View
ø0.5000
1.0000
0.9360
Front View
Side View
.185
.104
0.0665
R5.0000
0.5000
0.6054
NOTE:
Material: AISI 4340 Heat treated steel
1/2"D Regular Hexagonal Bolt
Detail: Cleat
ME4182 - Enabler
Group 3
Name: Chain Cleat
Drawn By: ARM
Date: 3/10/93
Scale: none

Figure 12

SDRC I-DEAS VI: FE_Modeling_&_Analysis 09-MAR-93 22:31:24

Database: bracket Units : IN

View : No stored View Display : No stored Option

Task: Post Processing Model Bin: 1-MAIN

Model: 1-FE MODEL1 Associated Workset: 1-WORKING_SET1

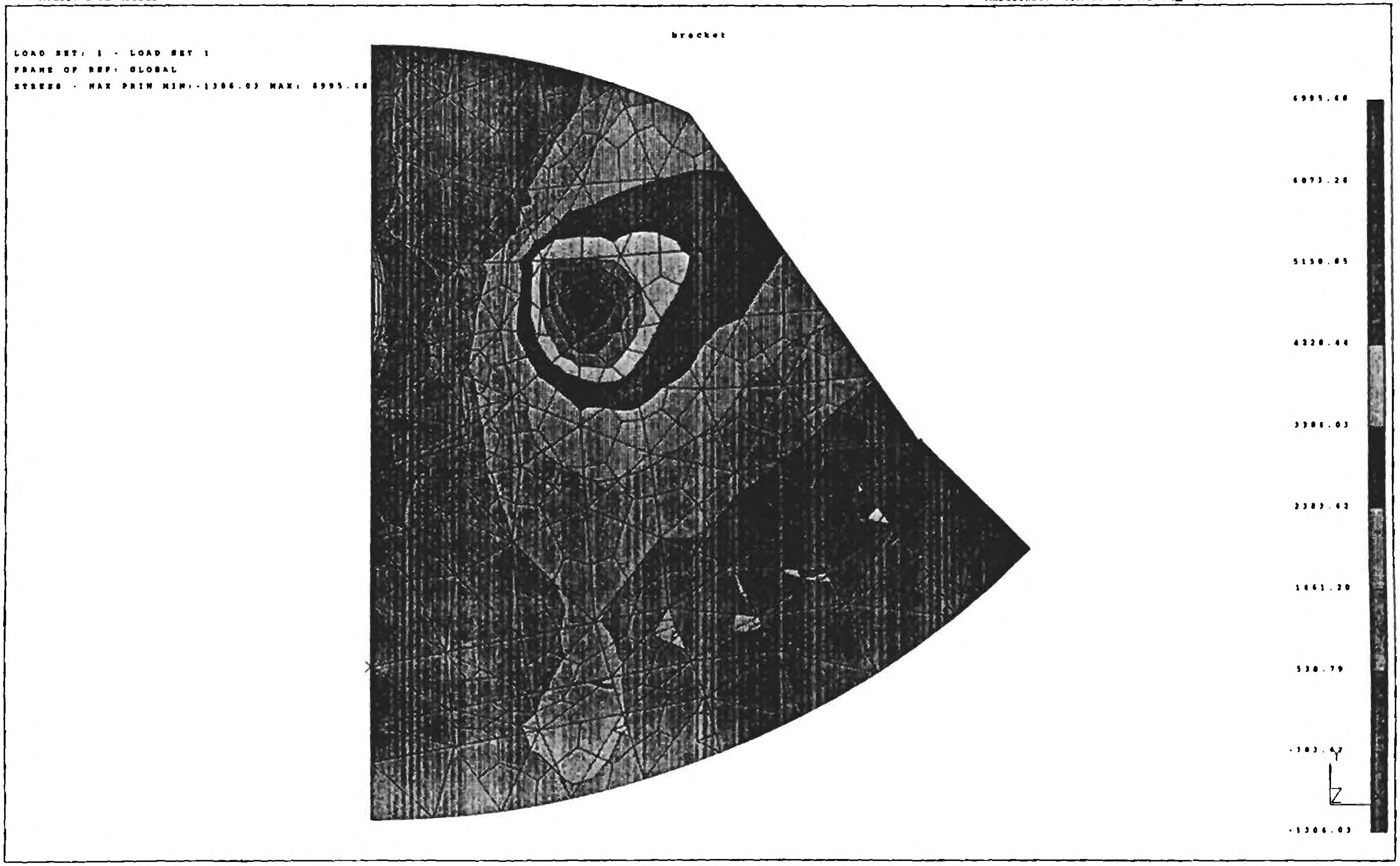
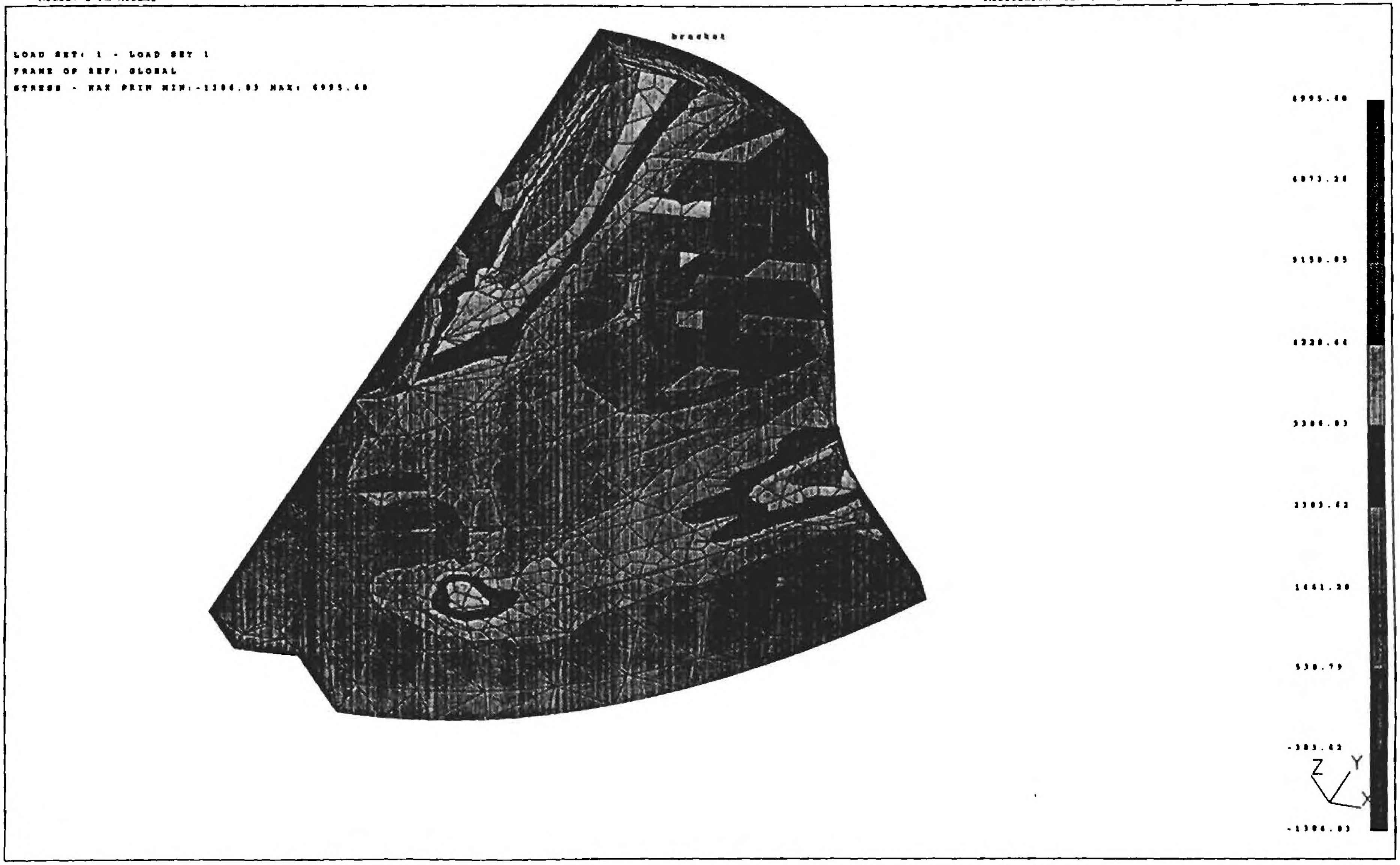


Figure 13

SDRC I-DEAS VI: FE_Modeling_&_Analysis 09-MAR-93 22:24:38

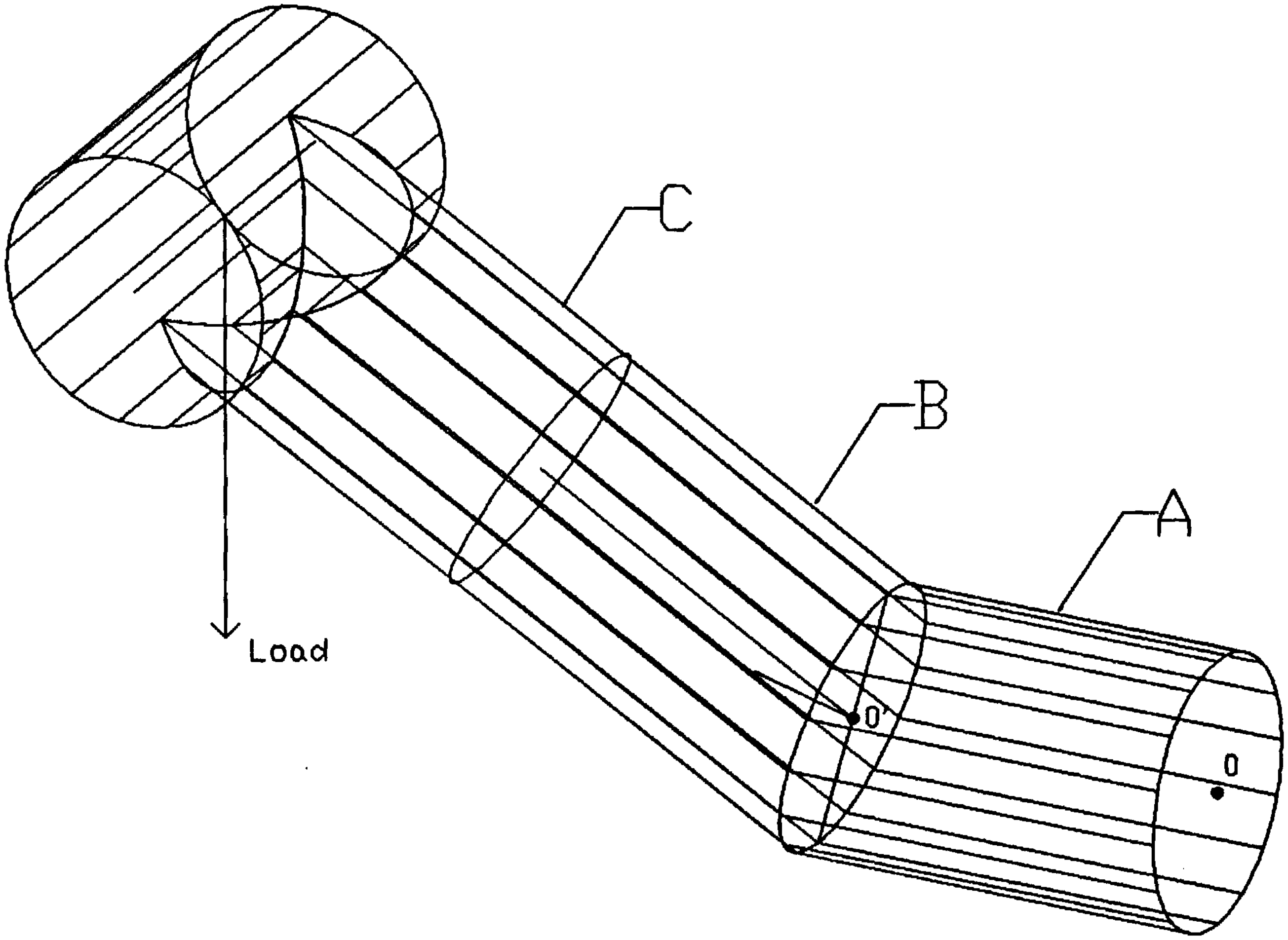
Database: bracket
View : No stored View
Task: Post Processing
Model: 1-FE MODEL1

Units : IN
Display : No stored Option
Model Bin: 1-MAIN
Associated Workset: 1-WORKING_SET1



Page 30

Figure 14



Drawing for Computer Program

APPENDIX A

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Hydraulic Pumps and Motors: Selection and Application for Hydraulic Power Control Systems; Lambeck, Raymond P., M. Dekker Publishing, 1983, New York, New York

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Linn Gear Company Products Catalog

The Torrington Company Service Catalog

APPENDIX B

Wheel Drive: Motor Justification and Analysis

Decision on What Type of Transducer to Use:

The Wheels group decided that the best way to power the ENABLER is a system of six, individual, reversible, hydraulic motors. The enslavement will be provided in the hydraulic system. That is, each side will be powered by the same hydraulic circuit. The braking of the wheel will come from closing off the hydraulic circuit. The motor will free-wheel when an open circuit is applied. The reasons for this decision are as follows:

1. This system can easily meet the performance, directional, enslavement, and free-wheeling criteria with the amount of power supplied by the hydraulic
2. Hydraulic motors are readily available through a variety of suppliers.
3. It may seem to be more costly and more massive to drive the ENABLER with a motor for each wheel than it is to drive the ENABLER with, for instance, a motor for each side. However, due to the geometry and the dynamics of the ENABLER it would be extremely difficult to mechanically transmit power from a motor to three different wheels.
4. Hydraulic motors are available to fit within a 12 in. diameter tube.

Motor Specifications:

The motors group used the above information to come up with a range of motor specifications that would satisfy the needs of the ENABLER. They are as follows:

- At least .58 horsepower per motor.
- Assume a gear ratio from 1 to 5.
- Motor R.P.M. from 33.78 to 168.9
- Displacement Range from .821 in^3 per revolution to 4.1 in^3 per revolution
- Operating pressure of 2000 psi
- Flow rate of .6 Gallons per minute.

Motor Performance Calculations:

Assumptions:

1000 lbs. vehicle weight.
Three Wheels in contact with the ground
Travel at 3 m.p.h. up a 17° grade.
Radius of wheel is 14.92 inches

Analysis:

The worst case performance situation is when the ENABLER must climb a 17° grade at 3 miles per hour on four wheels. The force at each wheel will be:

$F_W = (1000 \sin 17°) / 4 = 73.1$ lbs

Torque = F_W* Radius = 73.1* 14.92 = 1090.65 inch pounds.

Wheel RPM at 3 m.p.h. is 33.79

Horsepower= T * RPM/63025 = 1090.65 * 33.8 /63025 = .58 horsepower.

Using available flow rate and pressure to each motor:

Horsepower=pressure*flow/1714=2000*0.6/1714=.70 Meets Constraint

APPENDIX C

Wheel Drive: Motor Bracket Analysis

The motor mount was statically analyzed. The forces were calculated on the motor mount under the following conditions:

A. The motor was operating at full pressure. The force on the motor from the fluid pressure is calculated knowing that the fluid is at 2000 psi and the port size is .875 inches in diameter. However, the effective sealing diameter is .5 inches. The force on the motor from the fluid pressure is 383 lbs.

B. The force from at the gear mesh was calculated using the gear geometry and the rated torque of the motor. The rated torque of the motor is 876 inch-pounds.

These are the two force generating conditions that load the motor mount. The loads are applied to the motor mount at the bearing hole and at the motor bolt holes. A 3-d statics analysis was performed to get the following loads. Note that the results are stated without proof. Please consult the diagram on the next page for the location of the forces.

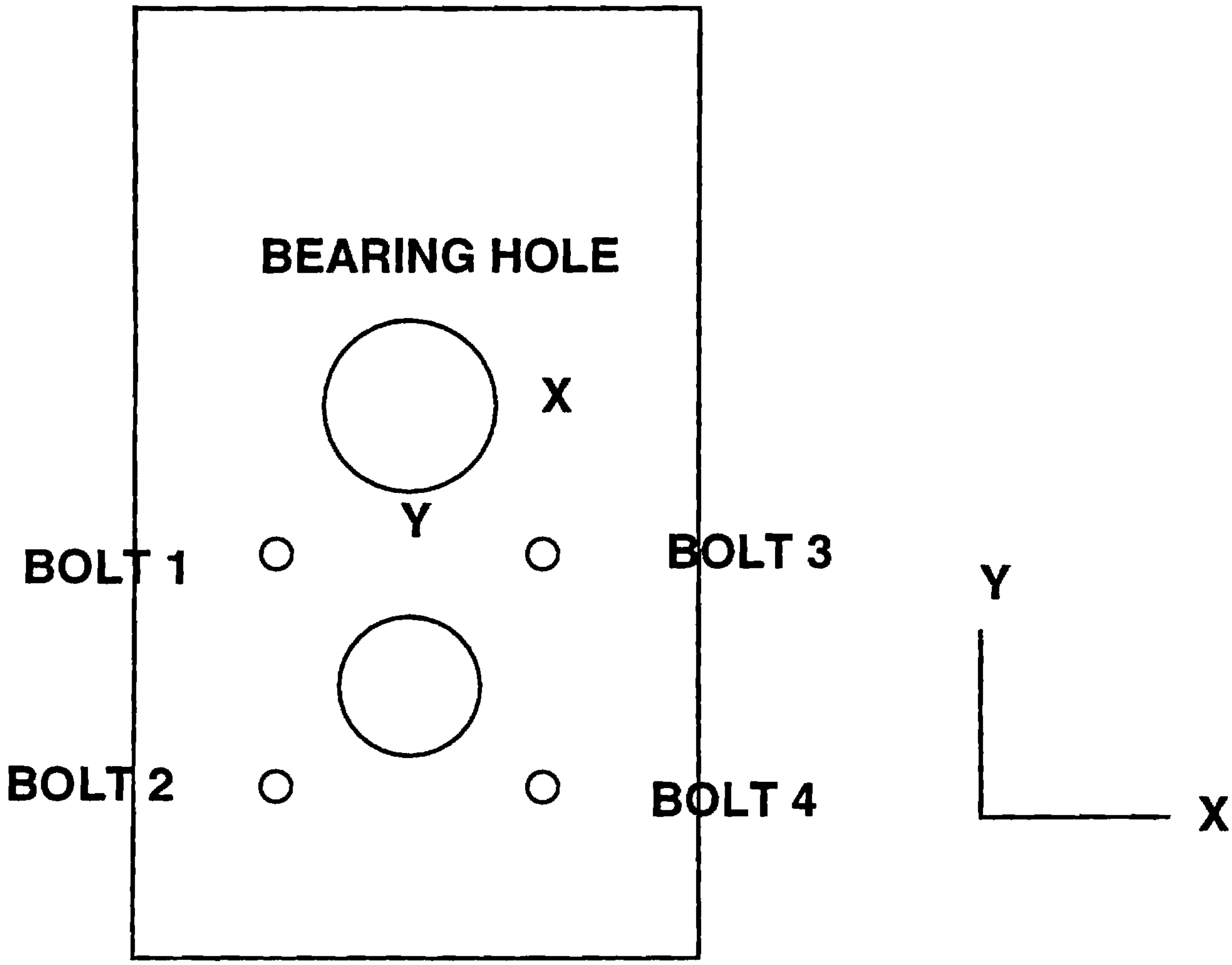
FORCE RESULTS (In Pounds)

Location	X direction	Y Direction
Bolt 1	292.6	-97.2
Bolt 2	174.6	-97.2
Bolt 3	292.6	-215
Bolt 4	174.6	-215
Bearing hole	934.4	241.7

A finite element analysis was performed on the motor bracket. Our original design thickness was one inch. A finite element analysis was performed on the motor bracket. The finite element analysis resulted in a maximum stress concentration of 1334.3 psi. The high stress was only in the area of the area to the right of the bearing hole. Note that these stresses are low compared to the allowable stress of Aluminum (about 20000 psi.) We decided from this analysis that our bracket was too thick. We cut down our bracket thickness to .5 inches. Please see figures five and six at the end of this appendix for the finite element analysis plots. The first plot shows the grossly exaggerated deformed geometry. The second plot shows the stress contours.

The motor mount bolts were chosen from the finite element analysis plot. The stresses in the area of the bolts are not more than 500 psi. We felt that we could use the same ASTM type 8 bolts for this application. We felt that 6 fastening points were necessary.

MOTOR BRACKET FORCE DIAGRAM



APPENDIX D

Wheel Drive: Gear Drive Justification and Analysis

Background - The wheels need to turn at 33.78 RPM's to maintain a speed of 3 M.P.H. Our motor at rated torque will turn 49.5 RPM's. The gear ratio needs to be 1.465 to 1.

Options

Chain Drive and Sprocket - Use a sprocket mounted along the centerline of the wheel and a chain connected to the motor shaft to power the wheel. This idea was rejected for one reason: (1) In order to get the necessary gear ratio the sprocket would have to be large enough so as to leave only about an inch between the drive and the driven sprockets.

Hydraulic Motor Only - Couple a hydraulic motor to a shaft along the centerline of the hub. Connect the shaft to the hub through spokes. This idea was rejected for two reasons: (1) In order to mount a motor at the center of a 12 inch hole the motor mounts would have to be large. (2) Neither our motor nor any other we could find could meet the performance specifications for this situation.

Hydraulic Motor on Centerline with Linear Speed Reducer - Mount a hydraulic motor along the centerline of the tube, have a shaft and spoke assembly to connect the hub to the motor, and place a linear reducer on the shaft to get the desired output characteristics. This idea was rejected for two reasons: (1) In order to mount a motor at the center of a 12 inch hole the motor mounts would have to be large. (2) Linear speed reducers are large, hard to find, and expensive.

Bevel Gear Design - Place a bevel gear on the end of the hub and mount the motor perpendicular to the centerline of the shaft. This system is good enough to be explored further in the next section.

Two Spur Gear Design - Have a spur gear on the shaft of the motor drive another spur gear on the shaft connected to the hub through sprockets. This system is good enough to be explored further in the next section.

Ring Gear Design - Have a ring gear (or bike chain)placed along the inside of the hub. Drive the ring gear through a spur gear mounted on the motor shaft. This system is good enough to be explored further in the next section.

Pro and Con Analysis of the Three Chosen Designs:

Bevel Gear Design.

Pro's	Cons
1. Fits geometry of the situation well.	1. How to attach the gear to the hub?
2. Little overhang load on motor.	2. Cost and Availability. Have found no bevel gears of this type without special ordering them.
3. No extra bearings, shafts or spokes.	3. Large drive gear to match large driven gear.
4. Low Mass	

The bevel gear design was not chosen in spite of its elegant simplicity. In order to mount the gear you would need to weld it to the aluminum hub. Steel can not be welded to aluminum, so the gear must be aluminum. There are none of these to be found without special ordering them.

Ring Gear Design.

Pro's	Cons
1. Fits geometry of the situation well.	1. How to attach the chain to the hub?
2. Overhang load on motor acceptable.	2. Large drive gear to match large driven gear.
3. No extra bearings, shafts or spokes.	
4. Low Mass	
5. Cheap - use chain and sprocket	
6. Readily available.	

If there was a way to attach a bicycle chain to the inside of the aluminum hub this would be the perfect answer. However we could find no good way to strongly attach a steel chain to aluminum.

2 Spur Gear Design.

Pro's	Cons
1. Fits geometry of the situation well.	1. High Mass
2. No overhang load on motor.	2. A lot of parts.
3. Gears can be made smaller.	
4. Easily manufactured here at Tech.	
5. Gears readily available from vendors.	

This was the layout that was chosen as the design to pursue. The parts are readily available and relatively cheap. Even though there will be a lot of parts in the design (bearings, shafts, gears and mounts), this design is the most available and therefore, the most feasible.

Motor Overhung Load Analysis Using Each Gear Design

The motor is rated to take 1216 (180.4 ft-lbs) pounds of force at 1.78 inches from the front face of the motor while spinning at a maximum speed of 45 rpm.

1. The Bevel Gear Design - The distance from the center of the drive gear to the front face of the motor will be at most 1 inch. This allows for a separating load of 2164 pounds. Over hung loading will not be a problem here.

2. The Two Gear design - Once again the distance from the center of the drive gear to the front face of the motor will be at most 1 inch. This allows for a separating load of 2164 pounds. Over hung loading will not be a problem here either.

3. The Ring Gear design - The maximum distance between the front face of the motor and the center of the drive gear will be 4 inches. This allows for a separating load of 541 pounds. The maximum motor torque is 876 inch-pounds. The drive gear will be 6.67 inches in diameter for this design. This means that the force from the motor, at the pitch point, will be 261 pounds -- hardly enough to damage the motor.

Analysis

First, determine σ so that it does not exceed 876 in·lbs torque. The gear material is AISI 1040; therefore obtaining a yield strength of 60 kpsi. We used a large factor of safety, 1.5, in order to absolutely minimize the risk of failure, so σ=40 kpsi. Since this will be a very low performance gear train, we used the Lewis strength formula, $\sigma=6W_t l/(F_t)^2$.
So, now we solve for W_t.

APPENDIX E

Wheel Drive: Bearing Analysis

The bearing in the motor mount does not receive much thrust force. Its main force is a radial force, therefore making roller bearings a better choice than ball bearings. When investigating the many types of roller bearings, the type that best suits pure radial forces are the cylindrical roller bearings. Many catalogs were looked at in the Georgia Tech library microfilm section and there were not many companies that made bearings with non-metric dimensions. The Torrington company had bearings that were in these dimensions, so we used these types to find the best one that fit the constraints. The first thing that needed to be calculated was the L_{10}=bearing life=$(C_o/P)^a$ where a=10/3 for roller bearings. The operation time=2000 hours(factor of safety is taken into account here). The average RPM used was 50 RPM.

$$L_{10}=2000*60*50=6000000=6(10^6) \text{ revolutions}$$

The next thing that needed to be found was the static load rating, C_o.

$$C_o=6^{(3/10)}*P \quad \text{where } P=963 \text{ lbs. from the gear force}$$
$$C_o=1648 \text{ lbs.}$$

Looking in the bearing catalog, using the 15/16" radius constraint, we found that the cylindrical bearing #B-1516 had a high C_o, making it an optimum choice. It was also easily accessible and very cheap.

APPENDIX F

Wheel Drive: Hub Mount Analysis

The torque on the wheel plate will, at maximum, only be 100 foot pounds. This loading is light compared to the loading of the motor bracket. Therefore, the thickness of .5 inches on the wheel bracket will hold up under the worst case loading.

Bolt Analysis:

The wheel plate is subjected to 1200 inch pounds of torque at a maximum. The bolt pattern diameter is 10.65 inches. This means that for 8 bolts, the force seen at each bolt is 28 lbs. We chose to use the ASTM type 8 bolts. The shear area of the bolts is .021 in^2 .This allows for a stress of 1341 psi in the bolts. The maximum allowable tensile stress in a SAE grade 1 bolt is 32,000 psi. Therefore, the maximum shear stress is approximately 16,000 psi. The bolts will definitely hold the load. We chose not to use a fewer amount of bolts because we felt that 8 fastening points were necessary. We chose not to use smaller bolts because these bolts had to be fastened through a .5 inch thickness plate. The length of smaller bolts would seem to be too large compared to the diameter.

APPENDIX G

Why Estimate Worst Torque Case?

Previous ME 4182 groups attempted to calculate the worst torque load on specific joint motor, but unfortunately many of the estimates were inaccurately done, or calculations were simply performed on the wrong position of the enabler to yield the critical case. This was our first task also because producing exact numbers of critical torque took four weeks of systematic observation, programming and analaysis of all the possible positions of the enabler. The reason it was so important to establish immediately what critical torques the positioning of the enabler placed on the motors because we needed to commence selection of articulation joint motor and power transmission systems. Intelligent estimates were made and these were substatiated using a systematic computer approach. The magnuitude of numbers, in terms of estimates, permitted us be somewhat confident that initial selections were appropriate and only small modifications would be necessary when more exacting calculations became prevalent. Overall, our time was used more effectively with initial estimates.

Articulation Joint Torque Analysis

Our original task was to determine a worst case torque loading on the articulation joint. Our goal was to find a relative worst case even if the actual numerical answers were incorrect. The worst case could be recognized merely by the relative magnitudes of the numbers. Of course, better assumptions would elicit more accurate answers that could later be applied to the selection of real components. The difficulty herein is that we did not know any specific dimensions, weights, or centers of gravity, therefore many assumptions had to be made:

-our assumed weight was 450 lbs. per axle for a total of 1350 lbs. (this turned out to be a good assumption)
-our assumed dimensions were 18 in. from joint to joint, and 18 in. from the front joint to the center of gravity in the leading T-section of the vehicle. The vehicle was also assumed to be 78 in. (2m) wide (which we later found to be incorrect).

Certain performance goals had also been set for the vehicle. Some goals were absolute and could not be altered while others were "wants" rather than "needs". We first tried to attain the "wanted" goals.

Absolute goals:
-operate using a hydraulic motor at a pressure of 3000 psi (later 2000 psi) at a maximum flowrate of 16-20 fps in .25" lines (.64 to 1.06 gpm).
-there must be a space with at least a 2 in. radius along the centerline of the vehicle.
-the design life should be 100 hours

"Wanted" goals:
-a maneuver time of 4 sec.
-continuous rotation of any combination of articulation joints
-enough power to raise the vehicle on three wheels (this last performance goal happened to be our worst case torque loading).

Using these "wanted" goals, we quickly discovered that they were far too stringent as required performance goals. The torque of more than 26,000 in.-lbs. at a rotation speed of 7.5 rpm created a need for 3.5 HP per joint. Because any maneuver required at least two joints to operate simultaneously and some required four joints, the total power requirement could be as high as 14 HP. This was unacceptable. Continuous joint rotation had also been rendered nearly impossible. It was also proposed that one of the joints be allowed to free-wheel, or be locked or unlocked. (This was proven to be incorrect, but it did show that in most maneuvers, one of the joints would require significantly less power than the other two that were near it). For these reasons, we fell back to the "needed" performance goals and re-figured our power requirements.

Secondary Goals

"Needed" Goals:
-180° joint rotation in either direction from some 0° reference.
-a maneuver time of 4 sec.
-enough power to raise any single section of the vehicle (i.e. always have four wheels on the ground)

Using the "needed" goals, we again discovered that our power requirement could not be met by the Power Distribution Group's equipment. The torques, however, could not be reduced further (the required torque for lifting one section was about 10,800 in.-lbs.). Our only recourse was to slow down the joint rotation speed. We also used as a guideline a limit of .75 HP per joint.

At this point we selected a motor based on rough estimates to use as a reference because we were unfamiliar with what capacities were required for a hydraulic motor that provided a certain torque at a certain speed. This gave us a good idea of what flowrates and pressures to expect from different motors under different conditions. Also, the operating pressure was changed to 2000 psi for all systems.

New Goals and Final Performance Specifications

With all of the new operating conditions, the following specifications were made by the articulation joint group:

-the vehicle must always have 4 wheels on the ground
-a flat turning maneuver may be accomplished in less than 5 sec.

through an increase in flow in the hydraulic system.
-lifting one section may take as long as 7.5 sec using normal flows. This is acceptable.
-The gear reduction will be between 8:1 and 10:1 depending on the required torques, not on the required speed. This is also limited by the maximum drive gear size allowed in the tubular body of the vehicle, although we are attempting to use the smallest gear possible (about 1" in diameter).

Process of Substantiating Initial Estimate of Critical Articulation Joint Torques:

As stated above, a systematic observation and analysis of the articulation joint position was integral in determining the critical torque placed on joint motors. The manual considerations were rejected when it was decided that computer implemenation was faster, and more accurate.

The possible positions of the enabler can be over whelming to a novice observer, and to systematiically observe and analyse these position through manual means is even more intimidating simply for the number of calculation which must be made. For a very short time of the quarter we attempted this task and came to the realization that, not only was the data collected extemely tedious, but they were inaccurate. We tried placing degree measurements around all the joints of the PVC model of the enabler, and we incrementally rotated the enabler to collect positional data of each joint for moment calculation to be made later. We believed that the measurements were inaccurate since the scale of the model was small and the lengths we measured were to be done from the center of the enabler (physically this was impossible). We therefore stopped this approach and considered the implemation of computers to simulate the positions of the enabler. This manual approach, however, helped us develop quite the systematic method of rotating the enabler incrementally as to observe all possible positions.

AutoCAD and Qbasic were implemented to observe and analyse the positions and torques on specified joint motors. CAD made it easier to visualize the positions of the enabler using the AME solid moding and the ROTATE command. View as the ones on Figure 15 (please note the labeling of the drawing, as it will help visualize what the later text will describe) were drawn specifically for this purpose and to familiarize us with 3-D drawing. We observed that there was symmetry in many of the positions of the enabler, and we were able to eliminate the range of necessary calculations based on sheer repeatition.

The moments produced by the ends being lifted will only be considered since only one motor may be used to lift, as opposed to the center joint, which will be lifted by two motors on either side of the itself. Essentially the calculations of the motion of one-half the enabler is considered with only the end being lifted a partial range of possible positons, due to symmetry and other possible simplifications. The enabler end is considered to be sitting off a cliff and the motors are statically to maintain

The incremental change of all calculations is 1 degree. It is also assumed that the main load, specifically the power distrubution system, is place in the center of either end of the enabler. The actual range of motion we decided to actually analyse reduced to 0-90^o rotation of joint A relative to joint O. Joint B is rotated -180 to 180 about joint A. Joint C is considered to remain stationary because the main load is in the center of the joint and any rotation of joint C will not change the load position relative to joint B.

In terms of the Qbasic program used for the position and moment calculations (see Program 1) and the origin of the system is consider to be the center of joint A closest to joint O.

Program 1

```
 the X-axis is center of the mid-section (A) plane.
 B is 30 deg. relative to A (neg. - neg.)
'otation will be of section A,
 s effect on section B & C
al Torque =-6075 in-lb
al J dis = 13.5 in.

NT "Assume the X-axis is center of the mid-section (A) plane."
NT "Assume B is 30 deg. relative to A (neg. - neg.)"
NT "First rotation will be of section A,"
NT "and it's effect on section B & C"

= rotation with affine transformation given
0 to 90 deg. rotation represents rotation of A
eta = -90 TO 90
theta * 3.141593 / 180

1a,ib2a,ib3a,jb1a,jb2a,jb3a etc are affine components
 the basis vectors of b (perp to a 15 deg plane) in the a basis.
1ar is rotation of basis vectors of b about x axis in a basis.

ese vectors are components of i unit vector of B basis in A basis.
 COS(15 * 3.141593 / 180)
 0
 SIN(15 * 3.141593 / 180)
ese vectors take components of i, and rotate.
= ib1a
= -SIN(rad) * ib3a
= COS(rad) * ib3a


ese vectors are components of j unit vector of B basis in A basis.
 0
 1
 0
ese vectors are j components rotated.
= 0
= jb2a * COS(rad)
= jb2a * SIN(rad)

oss product of ib1ar's.jb1ar's will yield kb1ar's
= ib2ar * jb3ar - ib3ar * jb2ar
= ib3ar * jb1ar - ib1ar * jb3ar
= ib1ar * jb2ar - ib2ar * jb1ar

ce program reaches this point A has rotated 1 deg clockwise
int D is 18+9 inches from point, initally 30 degrees from horizontal.
gment will rotate -180 to 180 relative to segment A.
gma = -180 TO 180

 sigma * 3.141593 / 180
COS(15 * 3.141593 / 180) * 27
0
SIN(15 * 3.141593 / 180) * 27
 ica
 -SIN(rad2) * kca
 COS(rad2) * kca

mplete affine transformation of point d occurs here
= icar * ib1ar + jcar * jb1ar + kcar * kb1ar + 18
```

```
= icar * ib3ar + jcar * jb3ar + kcar * kb3ar



w take moment about A center by D x -450 lbf k.
-450 * jcaff)


e j component of the moment will be applied by joint itself (cutwise).


(mi) > ABS(maxm) THEN maxm = mi
(jcaff) > ABS(maxj) THEN maxj = jcaff



sigma
theta
"Critical Torque ="; maxm
"Critical J dis ="; maxj
```

The center line of the enabler will serve as the X-axis for each independent basis. Initially, the position of the load will be calculated relative to basis/origin of joint B. Rotation matrices will be implemented to vectorally calculate new positions of joint centers and affine transformations will be used to relate vector to basis at joint A.

All moments the load produces will be about the Z-axis of the system origin o. For the moment calcualation, the torque the motor produces will only be a component of the moment the load produces at its position relative to system origin at joint A. The other moment component will be produced transmitted by the bearing at joint A closest to center.

The output of the program stated that a critical moment was apparent when the joint was positioned at its maximum bend of 30 degrees on a horizontal plane relative to the center joint. The critical moment was 6075 in-lbs.

APPENDIX H

Articulation Joint Drive: Motor Justification and Analysis

Decision of What Type of Transducer to Use:

It was determined that the best solution to the problem of driving the ENABLER is a system of six reversible hydraulic motors. Hydraulic motors are relatively cheap, small, and capable of generating large torques.

Motor selection

Motor selection was based on the following parameters:

- Power = .75 horsepower
- Pressure = 2000 psi
- Maximum flow rate = 1.06 GPM
- Maximum torque = 10,800 in-lbs
- Gear reduction = 8:1
- Must fit within limited space

The above constraints were used to calculate the necessary motor displacement as follows:

$T_{motor} = T_{max}$ / Gear reduction = 10800 / 8 = 1350 in-lbs

Motor displacement = $T_{motor} * 6.28 / P = 10800 * 6.28 / 2000 = 4.24\ in^3$.

The motor selected is a 4-hole face mount with 7/8" ports and has a displacement of 4.62 in^3, which satisfies the above requirement. This motor also has a rated torque of 1480 in-lbs at 2000 psi, which is also adequate for the expected loading. The dimensions of this motor are as follows: length = 5.47", height = 3.8", width = 3.72", weight = 13 lbs. As well as meeting the power requirements, the motor is also small enough to fit in the ENABLER tubing without extending into the restricted center space.

Next, the maximum necessary flow rate was calculated from the maximum torque criteria as follows:

$Q_{max} = HP * 1714 / P = .75 * 1714 / 2000 = .643$ GPM.

This value falls well below the maximum available flow rate of 1.06 GPM. Note, however, that this value is for the maximum torque condition. At lower torques, the flow rate would need to be lowered to maintain a desired rotation speed.

The next consideration in motor selection was the motor rotation speed at the maximum torque. This value was calculated as follows:

$RPM = HP * 63025 / T = .75 * 63025 / 1350 = 35$ rpm.

This results in a joint speed of:

Motor rpm / Gear reduction = 35 / 8 = 4.4 rpm.

This value is a little lower than desired, but it is limited by the available power, and not the motor capabilities.

APPENDIX I

Selection Between Different Possible Power Transmission Systems:

A number of different possible power transmission systems were considered, and a selection was made. Selection was based on the following necessary characteristics:

1) Must be light weight
2) Must be minimal in size, since a 4" virtual space must be left in the center of the enabler for power fluid and control lines.
3) Must be high in strength
4) Accepts misalignment of driver
5) Flexible for adaptible design
6) Position Accuracy for Control

	Chain Drive	Internal Gear Drive	Friction Drive
Light Weight	x	x	x
Size (minimal)	x		x
High Strength	x	x	
Misalignment	x		x
Flexible	x		x
Position Accuracy	x	x	

The internal gear drive was rejected because the necessary size of the internal gear was very in limited avaibility, and the actual radial thickness restricted our gear ratio to less than 8:1. The weight of the internal gear was very questionable in comparison to other available power transmission systems. Metal gears will not perform well under misalignment.

The friction drive is a very novel idea, where a rubber wheel on motor shaft contacts the skin of the actual enabler as a drive. The primary concern about this system was the friction necessary to eliminate slipping. The highest frictional coeffient of a rubber car tire was 1.3. We will be using a small tire (one similar to a skateboard tire) and the contact area compared to a car tire is very small, thus the frictional coeffiect must be much higher. A frictional coefficent of this magnitude can not be sustained, so slippage is enevitable.

Decision: Since the chain drive meets all necessary requirements as a power transmission system. We will attempt to select the necessary device components for stock supplies for its implementation. This will minimize cost.

APPENDIX I

Articulation Joint: Motor Mount Bracket Analysis

CAD analyses were performed using I-DEAS software. While the I-DEAS program worked well, limitations on computer disk space required the simplification of the FEA models. The simplified models still provided adequate data, as verified by Dr. Robert Fulton.

After the basic configuration of the bracket was initially designated based purely on geometry, it had to be tested. The area of most concern was the joint between the curved base and the vertical face, so the acceptable simplification involved filling all of the holes and gaps in the FEA model. This particular model was cut in half to further simplify the model (see Figure 13).

Using a gear force of 2160 lbs. at a distance of 1.25" from the mounting face of the motor, the force per mounting bolt was 540 lbs. in shear and 590 lbs. in tension. These calculations also helped to determine the number and size of the bolts used to mount the bracket to the flange.

Stress contour plots are seen in Figures 13 and 14. The maximum stress incurred is around 7000 psi. This is well below the allowable stress of about 20,000 psi for 666 aluminum.

Evaluation of Manipulator Boom Designs for Lunar Vehicles

A THESIS
Presented to
The Academic Faculty

by

Clemens M. Saur

In Partial Fulfillment
of the Requirements for the Degree
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Summary

This paper addresses manipulator boom geometries for lunar vehicles utilizing non–orthogonal revolute tubular joints; also called stovepipe articulation mechanisms. Various functional design mechanisms were analyzed and improved through evaluation of kinematic performance measures and impact studies on related areas such as human-machine and control factors.

The key to achieving above average characteristics in flexibility, controllability, and human–machine factors is the arrangement of the first three stovepipe articulation mechanisms in such a way to simulate a spherical joint with hemispherical reach, and all subsequent joints in revolute parallel orientation. This configuration has several advantages which are not limited to the suggested four–degree–of–freedom configurations. These are: inverse kinematic solutions of motion, continuous work–envelope, low center–of–gravity for the first three joints, and easily to conceptualize motions.

A detailed discussion of the kinematic characteristics and usage of the additional degrees–of –freedom, a derivation of the inverse kinematic equations, and a step by step description of an actual model implementation with focus on the initial instrumentation panel, are the heart of this thesis. Additional discussions on future topics describe the scope of related problems to such areas as telerobotics and control architecture. A concept of an initial development control module for studying basic boom movements is included. This development control module is based on specifications required by the lunar work vehicle called "Enabler", which is developed by the Georgia Institute of Technology, School of Mechanical Engineering.

CHAPTER I

Introduction

Constructing a lunar outpost for future space exploration missions is not just mere science fiction, but is the most critical development planned by NASA for the next 30 years. A lunar outpost has a multitude of functions, from oxygen production for refueling of Mars bound research vehicles to ensuring a constant energy supply for earthlings well in to the future:

- Missions to Mars require an enormous amount of rocket fuel. Yet, escaping Earth's gravitational field uses up most of the launch vehicle's reservoirs. Rather than engineering rockets greater than the size of Apollo, a more economical solution is send rocket loads into low altitude earth orbits and refill their payloads with oxygen shipment generated from lunar soil extraction.

- Energy consumption is a measure of standard of living on earth. When petroleum production will be exhausted in 40 years, only coal, hydro-electric, solar and nuclear energy are left. Fusion is the most promising technology for nuclear power generation of the future to meet the expected demand. Current fusion reactors can retrieve all of the energy invested in the reaction. In about 30 years fusion reactor-technology will be ready to deliver its promise. Fusion reactions depend on tritium or He^{3+} for the generation of energy. Tritium, an unstable element with a half life of 12.5 years, is a by product of weapons grade plutonium production, while He^{3+} is available in quantities that might serve earth's energy production for maybe 20 minutes. Fortunately solar winds have converted regular Helium into He^{3+} for millennia on the moon, leaving it in quantities much desired on earth. He^{3+} extracted from lunar regolith is the only answer to driving fusion reactions capable of meeting our energy demands here on earth. An economic feasibility study gave this concept a viable stamp of approval.

Consequently, construction of lunar manufacturing sites is a challenge of the highest priority; a manufacturing challenge unlike any other seen here on earth. The task at hand has to be executed with an efficiency that would be envied by the best companies in the world. Theories and methodologies for production technology will have to be reinvented. Lunar Industrial Engineering will be a critical part to the successes of this master plan from the very beginning.

While lunar missions will become omnipresent, their cost will remain prohibitive. Consequently, space exploration and manufacturing equipment must be designed for flexibility, in order to be capable of performing a wide variety of multifaceted dexterous tasks. Normal operating situations require relatively simple task capabilities, but when uncertainty or circumstances prevail, the machinery has to be self–reliant to avoid, prevent, or correct difficult situations.

One of the early designs for a conceptional multi-purpose lunar construction vehicle is Enabler. Enabler is an ingenious design for an all terrain, light–equipment work vehicle. A six-wheeled fully articulated rigid chassis and a four degree-of-freedom manipulator boom allow it abundant motion capabilities and make it ideally suited for exploration and assembly operations. Enabler, the first in a sequence of many critical projects to making this enormous task possible, while ingenious in design has problems of its own.

Enabler's boom is an engineering breakthrough in capacity–to–weight ratio for rigid structures. It consists of non–orthogonal stove-pipe articulation joints with tubular links in between. The tubular members more evenly distribute loads, thereby minimizing the necessary wall thickness and consequently reducing the boom's overall weight. However, for dexterous boom movements above and below vehicle grade, at least four degrees–of–freedom are required. This requirement causes severe human–machine problems, since the rotational stove–pipe movements are difficult to conceptualize and the resulting end–effector work–envelope is of generally unequal density.

The complexity of even simple design requirements found in a lunar vehicle like Enabler are similar to problems addressed in designing complex manufacturing systems. Features and controls are designed for maximum flexibility while addressing limiting factors and requirements. Given the time constraints of this research project, a complete analysis of the total Enabler concept is beyond the scope of this document. Instead, this research will focus on the design geometry of the boom with respect to its design and performance criteria. The boom is a high precision positioning device for the end–effector tool carriage. It has to be operated in real–time via telerobotic means, which limits the possible amount of complexity due to controllability constraints. Yet, like the vehicle, the boom has to be self-reliant and should be able of performing tasks in more than one way. This increased flexibility adds complexity and reduces overall capacity. The goal of the thesis presented here is to define all the important industrial engineering concepts that affect the performance of a lunar manipulator boom and consequently stipulate its geometry. Three critical areas were identified and will be discussed in detail. These are: work–envelope flexibility, absolute controllability, and human–machine issues.

Chapter Overview

This thesis is divided into seven chapters. Chapters one and two discussed the introduction and problem definition. Chapter three introduces somé key concepts addressed with this research and gives some relevant background information. The fourth chapter is the heart of the research and describes the requirements that lead to a desirable boom configuration. It continues with a detailed static kinematic analysis and concludes with some basic pointers for advanced dynamic and reactive enumerative control strategies. Chapter five provides a detailed discussion of factors in human–machine issues that need addressing due to telerobotics. This discussion provides useful future guidance on how an initial experimental control module could be expanded to work in a teleoperated environment. The sixth chapter gives a basic control flowchart for a simplistic experimental test set-up. This information is included so that physical models of suggested boom geometries can be evaluated easily. The final chapter, chapter seven, summarizes the uniqueness of the solutions obtained and the stresses how these findings will be the base to future research projects. The appendix includes information on varies analytical results, test procedures, and computer simulations.

CHAPTER II

Problem Definition

Reducing weight and increasing the capacity of cranes, lifts and robotic manipulator arms is nothing new. Even in a specialized field like lunar engineering, a multitude of proposals have been written on this subject. Several ingenious approaches have been suggested on achieving a high capacity–to–weight ratio. Taken to an extreme, the boom structure is designed to flex under its load and actuators and control algorithms counteract any undesirable overshoot, sway, or vibrations. The complexity of such a design is inconceivable, and is not controllable beyond a few degrees–of–freedom. An alternative approach to increasing capacity to weight ratios in manipulator booms is to keep the structure relatively rigid and reduce the weight of individual components, in particular the articulation joints. The stove–pipe articulation joint seems to promise this desirable alternative. Contrary to conventional joints, the stove–pipe mechanism uses only one instead of two bearing races, eliminates the heavy cross–pins, and distributes stresses more evenly through its tubular links. While the stove–pipe concept is nothing new, it has not found a great following because its overall capacity is limited by the strength of the single bearing interface. In lunar conditions, at one–sixth the amount of gravity found on earth, bearing strength is not the limiting factor anymore. The problem is how to utilize this new insight to describe an adequate geometry for a lunar manipulator boom structure.

Addressed Topics

Describing an adequate geometry for a lunar manipulator boom structure utilizing stove-pipe joints is a design problem. Problems of this type are often left to mechanical engineers. Yet the complexity of the situation stipulates that many of the issues that need to be addressed are industrial engineering related. The overall effectiveness of the design will be measured by its degree of useability by a human operator. Several physical performance issues like work envelope flexibility, or boom capacity, drive the design geometry, but more important are issues relating to absolute controllability in real time situations and human-machine interface problems due to teleoperation. The goal of this thesis is to show how all three areas affect boom geometry requirements and come up with possible design configurations that suits them all. The goal is not to solve problems associated with related human-machine issues like teleoperation, or control issues like active feedback control algorithms. Instead, a basic kinematic model, human-machine interface and control solution will be suggested that can aid in evaluating physical models for promising boom geometries.

Related & Future Topics

Given the limited scope of this research project, only some issues could be addressed in detail. Many related topics received only a brief mention or were left for future research. The following paragraphs illustrate some of the issues falling into these categories.

The manipulator boom is coupled to a lunar vehicle and consequently is not an independent system. The chassis and boom have to communicate to perform many tasks in tandem. Depending on the vehicle capabilities and characteristics, the boom requirements and limitations change. For simplicity, the concept vehicle Enabler[1] has been used as base vehicle whenever discussion about a complete system were required. Issues only briefly addressed relate to boom–chassis interference, boom–chassis mounting locations, and the number of booms required to perform a given task. While these topics seem to have a tremendous impact on overall boom geometry, they only limit the overall effectiveness of the system and are driven by task specifications. For the purpose of a basic proof–of–concept working model utilizing stove–pipe joints these topics are less important than others.

The kinematic system of equations that were derived to describe end–effector movements for varies related boom geometries was limited to static analysis only. Since the kinematic analysis and resulting inverse kinematic solutions are only partially given in closed form, they are not directly suitable for deriving a complete set of dynamic system equations. If required, these equations can be approximated or simulated via the existing results. This also applies to an eventual flexibility analysis. A useful flexibility analysis has to apply to the complete system including vehicle and wheel dynamics, and is consequently better suited for experimental modeling.

Human–machine interface issues were only discussed as far as they directly affect the geometry requirements of the boom. A general discussion of problems associated with telerobotic operation was included for future guidance. Issues relating to the implementation of multiple mode control, path planning, time delay compensation, and decision support system, among others, is left for future research.

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1 Enabler is a conceptual design of Professor J. Brazell at the Georgia Institute of Technology in Atlanta. Enabler's chassis uses stove–pipe mechanisms to achieve a fully articulated range of motions.

CHAPTER III

Literature Review

Lunar vehicles have not been in the news lately, at least not since the last mission to moon. Yet related research has been continuously ongoing at many U.S. and worldwide laboratories. Most research focuses either on telerobotic issues and time delayed responses, or autonomous control via hybrid neural networks. Complete system concepts for telerobotic vehicles have been few, and of those most have focused on exploration missions rather than on assembly and manufacturing tasks. Consequently, research on large scale, light weight, high capacity manipulator booms has been limited.

Conventional boom or crane designs are either to heavy or unworkable in lunar conditions. Temperature fluctuations of ± 150° Celsius, exposure to solar winds, and impacts by small meteorites cause most designs to break down in very short time spans. In designing boom configurations with high capacity and low weight, most research has focused on flexible structures with active feedback control to rapidly dampen undesirable vibrations. While this concept is noble in thought it still has to be proven workable beyond a few degrees-of-freedom.

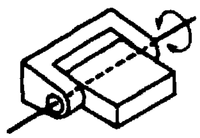
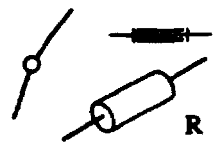
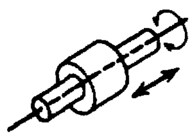
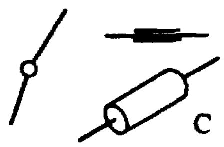
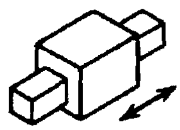
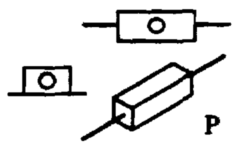
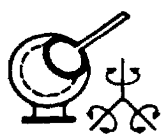
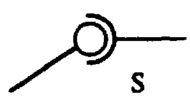
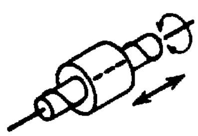
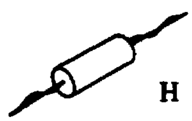
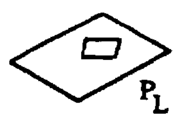
Kinematics & Robotics

Kinematics is the study of motion of machine members without consideration of forces and stresses produced [9]. The designers must select or devise mechanisms which will produce the required displacements and velocities and determine the resulting accelerations. Kinematics is the key to describing the motion of robotic manipulators. Robotic manipulators are mainly used for loading and unloading of machines, spray painting, assembly, spot welding, arc welding, inspection, die–casting, drilling and deburring of metal parts, etc…[8].

Articulation Joints

Eliminating flexible design configurations limits the topic of discussion to rigid structures. The functionality of the boom is driven by the articulation joints. These do not only stipulate the boom's overall geometry, but are also responsible for distributing all load forces throughout the structure. The more efficient the second task is performed, the lighter the overall load–carrying components can be designed. There are six basic types of articulation joints. These are revolute, prismatic, cylindrical, spherical (ball & socket), screw, and planate pair [5][6][10]. Typical robotic manipulators utilize only revolute and prismatic articulation joints [8]. These two mechanisms come in a variety of shapes but provide the overall same functionality.

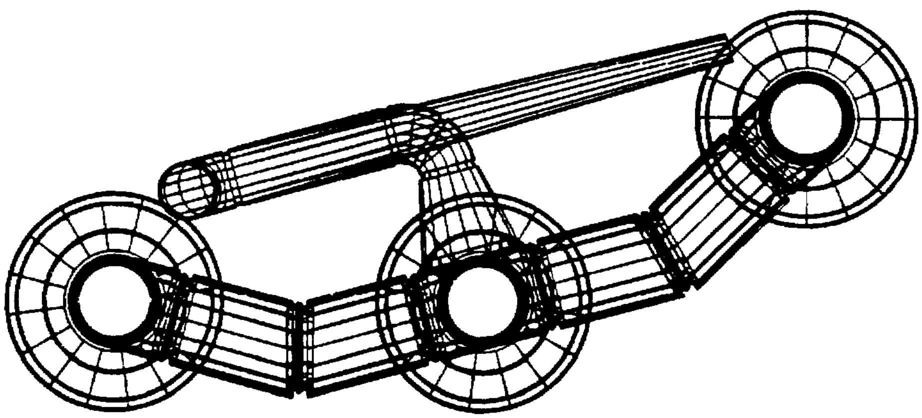
Table 3.1: Joint Mechanisms

Joint Type	Geometric Form	Schematic Representation	Degrees of Freedom
1. Revolute (R)			1
2. Cylinder (C)			2
3. Prism (P)			1
4. Sphere (S)			3
5. Helix (H)			1
6. Plane (P_L)			3

This research has opted to focus on rigid boom structures utilizing a special type of revolute joint, called the stovepipe non–orthogonal articulating joint. This high-tech nomenclature describes a simple prosaic mechanism which has been in use for many years [4]. The goal of the manipulator boom design is to be as light weight as possible. Given any omni–directional loads, the tubular structure is the geometry that requires the smallest cross–sectional area to carry that load without yielding. The stovepipe mechanism directly joins two tubular members. It also provides an easy passage and protective enclosure for internal instrumentation and wire harnesses. The limiting factor of the stove–pipe joint is the strength of its single bearing interface. Consequently, this design has found few applications in load driven operations.

One concept that has developed this mechanism quite intensively in unloaded situations is NASA's AX–5 Space Suit[7]. NASA's AX–5 space suit differs radically from previous designs in that it lacks fabrics or other soft components. The hard shell aluminum suit maintains a constant volume and hence constant internal pressure, which minimizes the effort of working in it. Rotary bearings arranged in non–orthogonal directions near human joints provide the near full range of mobility of all body motions.

The lunar concept vehicle "Enabler" [3] is an applications that will try to utilize the stovepipe mechanism in severely loaded situation. In lunar conditions at one–sixth the amount of gravity found on earth this idea might proof viable. "The Enabler is intended for lunar exploration and construction. Employing a stovepipe chassis and manipulator boom allows the Enabler to traverse the moon's rugged surface and complete remotely controlled tasks"; (1993, Information Brochure), [3].



Enabler: Figure 3.1

Another noteworthy project, for high load to low weight crane structures, although not directly applicable is SPIDER [1]. SPIDER (Steward Platform Instrumented Drive Environment Robot) is a free standing ultra–light weight crane using a six degree–of–freedom Steward platform to control x, y , z, roll, pitch, and yaw.

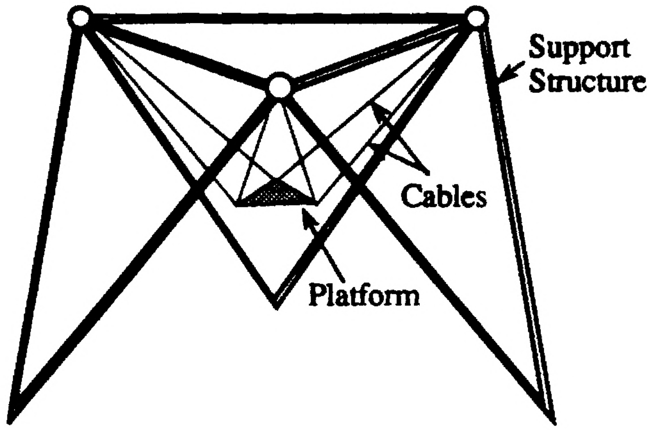


Figure 3.2: SPIDER

The advantage of the SPIDER is that it allows virtually vibration free precision positioning. The disadvantages are that it provides limited dexterity and can manipulate only vertical loads.

CHAPTER IV

Kinematic Analysis of Design Configurations

This chapter focuses on qualifying and analyzing manipulator boom geometries with respect to evaluation criteria which are based on perceived task requirements. This analysis will be restricted to stove-pipe articulation joints as opposed to conventional robotic joint mechanisms. Upon selection of a sufficient manipulator boom geometry a complete kinematic analysis that describes the complete set of boom articulation joints will be performed.

Designing equipment for self-reliance in a non-serviceable situation means that the system should be designed with enough flexibility to deal with any situations without having to anticipate all circumstances. It is also important to keep the system controllable and humanly operable. Applying this criteria to the boom of a lunar work vehicle means to aim for the highest degree of freedom within physical and economic limitations.

The theoretical highest degree of flexibility attainable for a boom is approximated by an infinitely flexible string. Yet dexterity is not enough if capacity is not available. Capacity and rigidity tend to be inversely related and therefore are conflicting goals. Likewise, capacity and weight, and therefore cost, tend to be directly related, which further complicates the relationship.

The stove-pipe joint design offers a higher than average weight to strength ratio, making it an ideal contender for the complex requirement of capacity and rigidity. This phenomena arises due to more efficient distribution of loads throughout the cross-section area of all structural members. The cylindrical geometry of the stove-pipe joint has fewer stress raisers than found in a square or more complex geometry. This minimizes the required wall-thickness of cylindrical pieces, thereby reducing their weight and cost.

While the strength–to–weight ratio of the stove-pipe design are high, the overall capacity is limited by the strength of the bearing interfaces. The forces exerted on the bearing assembly are both compressive and tensile in nature. For earthbound applications the tensile strength of the bearing races tend to be the limiting factor, yet with one-sixth gravity in lunar conditions the overall performance improves significantly. The tremendous advantages of the stove-pipe design make it the focus of this research. Other joint-actuator configurations are briefly mentioned in the literature review.

Overview

This chapter is divided into six separate sections. The first section is a brief review of kinematic principles. The second section describes typical task requirements that might be performed by the manipulator boom of a lunar vehicle. The third section introduces the performance criteria that are of interest. Section four establishes a guideline for determining an adequate number of degrees–of–freedom in boom movements. Section five discusses various boom and boom–chassis configurations and describes the evaluation criteria and results in detail. Section six, the last section, derives the inverse kinematic equations and discusses other basic kinematic characteristics. This section is followed by a brief summary which illustrates the interaction of the basic required system components.

Kinematics

In robotics, kinematics analysis is used to describe the motion of the end effector with respect to its position, orientation, and speed, as well as the rate of change of these variables. Given simple mechanisms, easy geometric solutions that describe the kinematic structure can be found. Yet in more complex, multiple-degree-of-freedom mechanisms, advanced matrix manipulations are better suited to describe these kinematic structures. In direct kinematic analysis, knowledge of each actuators position and speed is used to predict the end-effector's overall motion. In inverse kinematic analysis, knowledge of the end-effector's trajectory and speed is used to anticipate the rotation of each articulation joint. The former is relatively easy, since the rotation of each actuator is independent from one another, giving rise to three equations describing continuous space. The latter posses considerable difficulties as discussed below.

Inverse Kinematic Equations give the joint rotations in terms of three dimensional coordinates such as rectangular, cylindrical, or spherical systems. Yet if the number of articulation joints is greater than three, the number of available equations will be less than the number of unknown variables. If several variables are linked so that the number of independent variables is reduced to no more than three, a reverse kinematic equation might be possible, although not guaranteed. Even in this case, the additional degree of freedom is lost unless a heuristic for its use is developed. Since solvable cases are the exception, development of heuristics to *approximate* the inverse kinematic solutions of complex systems is essential. Approximation solutions can take considerable computing power and time to simulate. Consequently, the advantage of a direct inverse kinematic solution should not be overlooked, especially with regard to its benefits for real-time control.

The Denvenit–Hartenberg convention is a technique for the systematic rotation and translation of displacement matrices. All joints are treated as a one-degree-of-freedom revolute or prismatic mechanism, linked in series. For each rotation, or translation, an appropriate Denvenit–Hartenberg convention is applied until the last link in the system has been reached. Evaluation of all manipulation matrices, while observing matrix notation and manipulation rules, expresses the position of the end-effector with respect to its base coordinates. For forward kinematic analysis no other method is needed. For inverse kinematic analysis, however, Denvenit–Hartenberg conventions are helpful but not sufficient.

Matrix rotations about X, Y, or Z axis by angle Θ are:

$$\mathrm{Rot}(X,\Theta) = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(\Theta) & -\sin(\Theta) & 0 \\ 0 & \sin(\Theta) & \cos(\Theta) & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{4-1}$$

$$\mathrm{Rot}(Y,\Theta) = \begin{bmatrix} \cos(\Theta) & 0 & \sin(\Theta) & 0 \\ 0 & 1 & 0 & 0 \\ -\sin(\Theta) & 0 & \cos(\Theta) & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{4-2}$$

$$\mathrm{Rot}(Z,\Theta) = \begin{bmatrix} \cos(\Theta) & -\sin(\Theta) & 0 & 0 \\ \sin(\Theta) & \cos(\Theta) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{4-3}$$

Translation by a distance L is given by:

$$L(\delta x,\delta y,\delta z) = \begin{bmatrix} 1 & 0 & 0 & \delta x \\ 0 & 1 & 0 & \delta y \\ 0 & 0 & 1 & \delta z \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{4-4}$$

where translation is by δx in the X–axis, δy in the Y–axis, and δz in the Z–axis.

Task Requirements

The first lunar rover was designed to help astronauts collect earth samples beyond walking distance of their landing site. These exploratory missions were dangerous and unnecessarily exposed astronauts to high risk situations. Future operations will leave first-time exploration missions to tele-operated vehicles. The boom will perform geological tasks and transmit the results to the control center for data mapping. Upon identifying proper geological sites the tele-operated vehicles will assemble necessary construction and living habitats. The astronauts function is to monitor progress and intervene or assist in critical situations. In order to perform tele-operated exploration and construction missions the boom has to assist in the following five areas: "object manipulation, data collection, vehicle stability, and limited self interference". Each of these five areas is discussed in detail below.

Object Manipulation: Object manipulation can be divided into two categories, geographical sampling and assembly operations. Geographical sampling operations requires a broad work-envelope above and below grade of the vehicle. Reaching into a crater or up a hill-side are two examples. Testing for sedimentary conditions requires drilling operations which depend on vertical straight line motions. Assembly operations are far more difficult. Assembly means putting two or more parts together. Orientation and alignment are critical issues as is obstacle interference. Unless a task can be performed in one step, such as snapping a component in place, more than one manipulator will be required for holding and mounting operations.

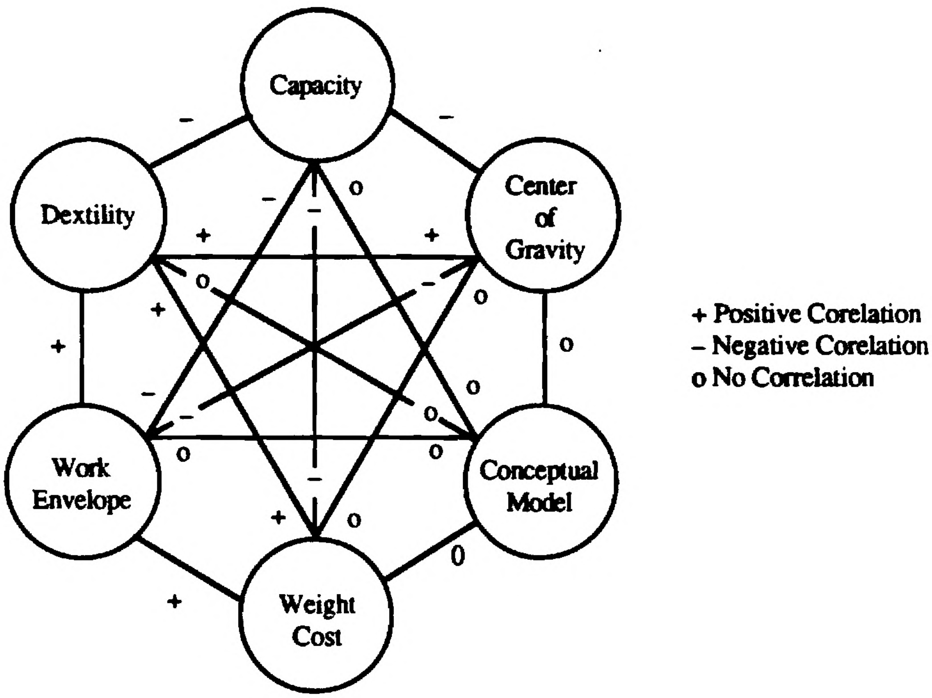
Data Collection: The inherent difficulty of tele-operations is the lack of adequate information. The mobility of the end-effector is an ideal location for data collection about the environment. The end-effector has a convenient view of the immediate surrounding, both during vehicle traversing and boom operations. An end-effector camera offers close-up views of points of interest as well as views far beyond the obstacles.

Vehicle Stability: The boom can act as a counterweight to assist chassis movements, as needed during crevice crossing, or act as an outrigger in more delicate situations. However, the boom is the source of vehicle instability and should consequently have a low center of gravity to avoid tipping over. In cases were vehicle stability is out of control the boom can serve as a last resort to minimize vehicle damage or assist in vehicle recovery. The boom can be used as an outrigger to reduce the load on a vehicle axis during service operations or add to chassis stability in a dual-boom operation configurations.

Self Interference and Storage: Boom-chassis self interference should be minimal as to maximize the useful work envelope. Likewise the boom should be contained within the vehicle boundaries during storage and transportation to avoid unnecessary assembly.

Performance Criteria

Many variables influence the effectiveness of potential boom configuration. Flexibility, controllability, and human machine factors are three general categories including a magnitude of other design criteria. The following relationship diagram illustrates the interaction of some of the most important evaluation variables:



Performance Criteria: Figure 4.1

The above diagram is an overview of a list of performance measures deemed most important by the author. These will be discussed in detail in the following sub-sections. The correlation of the various criteria differs as indicated by the symbols between them.

Degrees–of–Freedom of Manipulator Boom

Given the task requirements and a prediction for the important decision criteria, the next goal is to come up with possible boom configuration. This goal can best be approached by thinking about the following two questions: What kind of geometry does it take to perform all task requirements, and what kind of geometry best satisfies the evaluation criteria? The primary variable affecting boom geometry is the number of degrees-of-freedom of the kinematic structure. How many degrees-of-freedom are necessary and how many degrees-of-freedom are too many? That will be the discussion of this section.

This report defines flexibility as the ability to perform a task in more than one way. Yet, unless a task can be performed in at least one way, added flexibility is useless. Consequently when evaluating the flexibility of a boom configuration, criteria like work envelope volume, density, and continuity are just as important as redundancy or multiple solutions. However, multiple solutions are not measures of ability to circumvent obstacles. Therefore this report will define flexibility as a function of dexterity which is the ability to avoid obstacles near the end-effector. This biased definition is based on the assumption that most interferences occur near the work location rather than near the base of the boom. The second type of interference is easy to correct with vehicle repositioning. Last but not least, boom flexibility will be greatly affected by the chassis mounting location. Boom-chassis interference is the single largest confinement to the boom work-envelope

Typically robotic manipulator joints are of either prismatic and revolute type. The stove-pipe joints is capable of revolute movements only. Consequently a one degree-of-freedom stove-pipe structure is capable of drawing the outline of a circle. A two degree-of-freedom stove-pipe configuration defines the surface of a sphere, the area of a circle, or the surface of a V-shaped belt. Three degrees-of-freedom is the minimum number of joints required to describe a volumetric work-envelope. Only certain arrangement of three degree-of-freedom stove-pipe configurations will describe a solid void–less spherical space. For more than one singular solution within such a solid spherical space further degrees-of freedom have to be added.

Each additional degree of freedom enhances boom flexibility through multiple solutions.

Multi-Degree-of-Freedom Revolute Robots: Table 4-1

D.O.F.	Capabilities (Revolute Type)
1	Circular Workline
2	Surface Workarea
3	Voluminous Work Envelope With Possible Duplicate Solutions
4	Voluminous Work Envelope With Multi-Directional Approach
5	Voluminous Work Envelope With Reach Around Capabilities
6	Voluminous Work Envelope With Snake Like Obstacle Avoidance

A high degree of flexibility is desirable, however it is subject to reduced boom capacity. Boom capacity is limited by the maximum moment arm supported about the boom-chassis mounting location. The heavier the boom and the further its center of gravity is removed from its base-mount, the lower the overall boom capacity.

Further increases in flexibility without added degrees-of-freedom can be realized from the boom–chassis mounting position. Turning and twisting of the chassis articulation joints could be used to affect the positioning of the boom–base, thereby expanding boom flexibility. At the same time the boom-vehicle mounting location can restrict the overall boom work-envelope due to chassis-boom interference. The boom-vehicle mounting location is of critical importance. Rearrangement of the vehicle with respect to its task location does not add any degrees-of-freedom but repositions the work-envelope to better suit the task at hand. Lastly the added flexibility arising from end-effector actuators is of importance.

The primary need for additional degrees of freedom in boom movements arises in obstacle avoidance capabilities. A large number of obstacles requires a higher degree of boom flexibility for collision avoidance. Lacking knowledge about the possible work environment makes a proper assessment for boom flexibility requirements difficult. Yet, being tele-operated and given the harshness of lunar environment, operating resolution is also limited. Furthermore, assembly tasks can be designed with respect to the limited work flexibility.

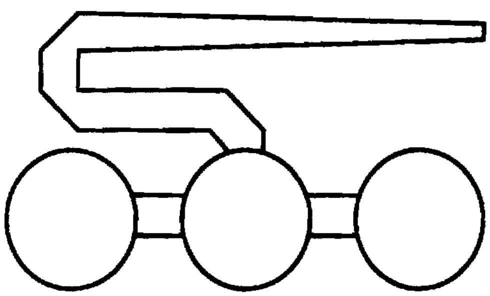
Admitting that vehicle repositioning is cumbersome and not always possible, a fourth degree of flexibility in boom movements might be warranted to give multi–directional access to a work location. This flexibility helps avoid minor interferences and deal with unforeseen circumstances. A fourth degree of freedom adds, complexity, weight, and controllability concerns. Now that there is more than one solution to a given path how does the operator utilize or direct the boom differently. The end effector is still reaching the same locations; the individual boom components can move differently to avoid collisions with the environment. If the operator is unable to conceptualize the movement of boom components during simple tasks then a simulation for path planning is required. In this case the real-time control characteristics of the system are lost. In any case, the boom-envelope has to be continuous with end-effector straight-line tracking capabilities. The trade-offs due to the additional weight of the fourth articulation joint can be minimized by placing heavy component near the base of the boom. Yet since each component is modular with its own actuators for design reliability and serviceability reasons, the first couple of boom components have to be as short as possible.

Manipulator Boom Geometries

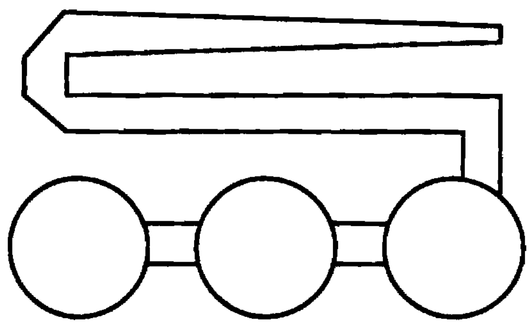
The boom-vehicle mounting location can greatly affect the amount of chassis-boom self-interference. While one goal is to maximize the useful work envelope another requirement, as stated before, is for the boom to be contained within the boundaries of the vehicle, for easy storage and transportation without disassembly. This report will distinguish between the two basic types of boom-vehicle mounting locations at the end of the vehicle and its middle. Both locations have advantages & disadvantages as illustrated next.

BOOM CHASSIS CONFIGURATIONS

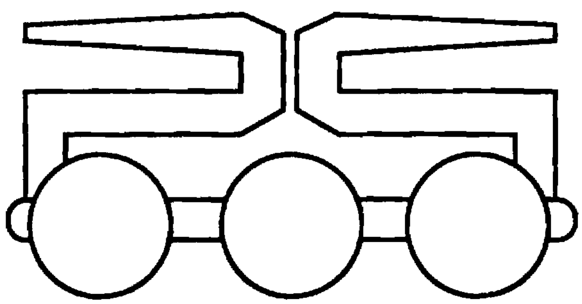
The platform for the manipulator arm of a lunar work vehicle is designed to operate in extremely rough terrain. Crevice crossing and obstacle climbing are two key operations which are absolutely essential. The most optimal design configuration for such platform is a six wheeled rover with three in line axis and whose center of gravity can be shifted about the center axis. The boom plays an important part during this operation since it the necessary counterweight. Consequently the maximum possible moment arm generated by the boom about the center axis is of critical importance. Whether the task assignment calls for a single or dual boom configuration the overall redistribution of weight about the center axis must be possible. The following are conceptual drawings of possible boom–vehicle combinations. Notice that the boom dimensions do not exceed the dimensions of the vehicle boundaries.



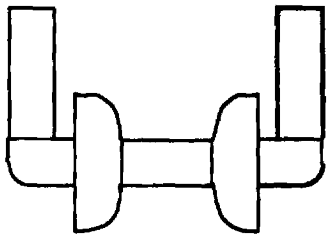
Single Center Mounted: Figure 4–2



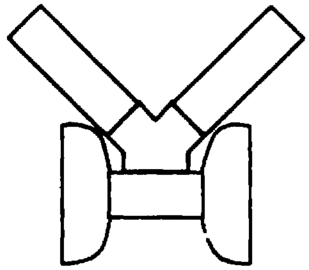
Single End Mounted: Figure 4–3



Dual End Mounted: Figure 4–4



Dual Axis Mounted: Figure 4–5



Dual Center Mounted: Figure 4–6

Evaluation of the boom-chassis combinations is possible on a limited basis only. Depending on whether or not the requirement call for a dual–boom configuration for complex assembly operations or single–boom configuration the ideal mounting locations vary. The following table gives an indication of the relative advantages and disadvantages:

Boom–Chassis Configurations: Table 4-2

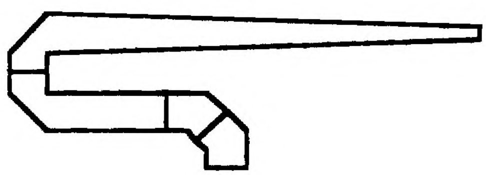
Boom–Chassis Configuration	Advantages	Disadvantages
Single Center Mounted	Simple Design Good Counterbalance	One Handed Tasks Only
Single End Mounted	Long Manipulator Reach	Limited Counterbalance Large Moment Arm
Dual End Mounted	Work Sharing Good Stability Good Counterbalance	Collision Avoidance Limited Overlap Limited Reach
Dual Axis Mounted	Work Sharing Good Stability	Small Overlap Severe Collision Avoidance
Dual Center Mounted	Work Sharing Extensive Overlap	Collision Avoidance Linked Base Limited Stability

The center mounted single–boom configurations has definite advantages over the end mounted single–boom configuration which is incapable of adequately rearranging weight about the center axis of the vehicle. For dual–boom configurations, further study is necessary for a definite conclusion.

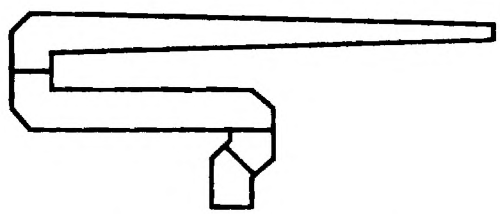
BOOM CONFIGURATIONS

Making the boom fit within the horizontal vehicle boundaries makes the vehicle convenient to transport without requiring de–assembly or re–assemble. The validity or necessity of this statement is not questioned and all suggested boom geometries abide by its limitations. Two categories of boom geometries were evaluated. These are center mounted and end mounted configurations as shown below.

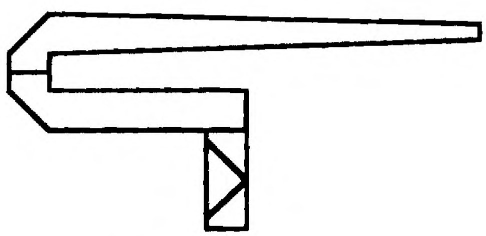
Center mounted boom configurations are ideally suited, but not limited to single–boom operations. Some possible boom geometries are illustrated below:



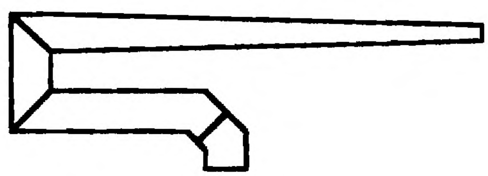
Boom Configuration #1: Figure 4–7



Boom Configuration #2: Figure 4–8



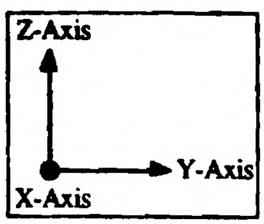
Boom Configuration #3: Figure 4–9



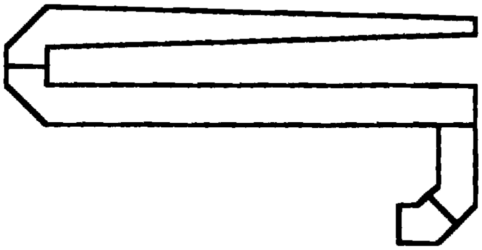
Boom Configuration #4: Figure 4–10

Coordinates:

Notation:
L=Vehicle Length

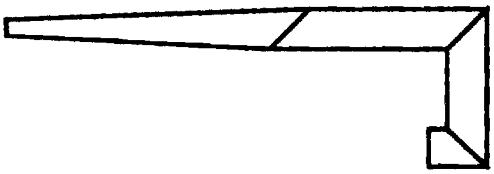


End mounted boom geometries are primarily limited to dual–boom configurations for reasons mentioned before. Some promising geometries are illustrated below:

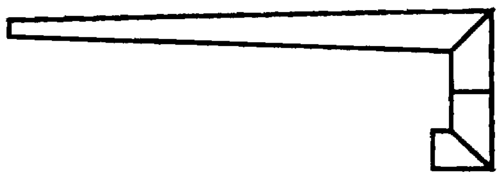


Boom Configuration #5: Figure 4–11

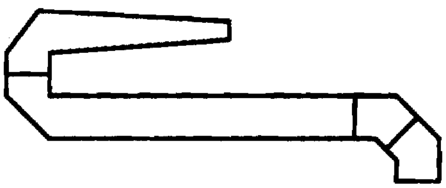
Boom Configuration #6: Figure 4–12



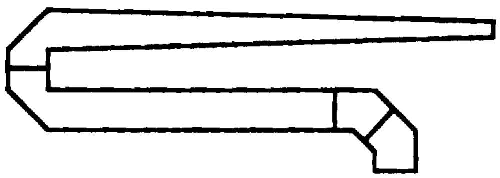
Boom Configuration #7: Figure 4–13



Boom Configuration #8: Figure 4–14



Boom Configuration #9: Figure 4–15



Boom Configuration #10: Figure 4–16

Evaluating the various boom geometry illustrations is qualitative and quantitative in nature. Two considerations, center of gravity and intuitiveness of motion, put considerable pressure on cramping the first couple of joints in close proximity of each other. While the optimal dimensions for any one configuration is unknown, the boom is required to remain within the vehicle boundaries, for transportation and storage reasons. This criteria makes a relative description of boom segment lengths possible. The following table gives measures of relative boom segment dimensions for evaluation purposes. "L" is used to describe the limiting dimension of the overall vehicle length.

Initial Boom Dimensions: Table 4–3

Configuration	Segment 1		Segment 2		Segment 3		Segment 4		Total Length
	Z=	Y=	Z=	Y=	Z=	Y=	Z=	Y=	
1[2]	1.5	0.3	0.8	–0.4	5.0	0.8	0.8	11.0	18.4 = 1.67 L
2[2]	1.5	0.3	0.8	–0.4	0.8	0.8	0.8	11.0	18.9 = 1.72 L
3	2.5	0.3	0.8	–0.4	0.8	0.8	0.8	11.0	19.9 = 1.81 L
4	1.5	0.0	4.0	–4.0	1.0	1.0	8.0	8.0	19.9 = 1.81 L
5	1.5	0.0	2.0	-2.0	0.8	–11	0.8	11.0	26.4 = 2.40 L
6	1.5	0.0	2.0	-2.0	3.4	-3.5	4.3	4.3	15.3 = 1.39 L
7	1.5	0.0	2.0	-2.0	3.5	-3.5	4.3	–4.3	15.3 = 1.39 L
8	1.5	0.0	1.0	-1.0	1.5	0.0	8.0	–8.0	15.7 = 1.43 L
9[2]	1.5	0.3	0.8	–0.4	0.8	0.8	0.8	5.0	17.4 = 1.58 L
10[2]	1.5	0.3	0.8	–0.4	0.8	0.8	0.8	11.0	23.4 = 2.13 L

BOOM GEOMETRY EVALUATION

As briefly outlined before, there are several factors that influence the performance of a potential boom geometry. Unfortunately not all variables are quantitative, and consequently some are evaluated on user judgment. For the quantitative measures it is important that the test measures are evaluated in a normalized manner, otherwise comparative conclusions are difficult to substantiate. Eight different variables have been identified for evaluation and are briefly discussed below. A detailed discussion of some individual concepts follow this section.

The vehicle work envelope is evaluated by its volume, maximum, and minimum reach. The vehicle

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[2] The first three joints were arranged in close proximity in a configuration that mimicks the motion of a hemisspherical ball–point joint. See section on stove-pipe representation of a ball–point joints for a detailed discussion.

capacity is limited by the booms worst moment arm, or its normalized center of gravity. Dexterity, as defined previously, is the ability to approach the work location from various directions, and could be quantified by the mean square–volume contained within the multiple solutions as measured from the end-effector. This simply states that a short last boom segment gives a greater flexibility than a long one. Rather than quantifying this value numerically, a relative measure will be estimated. The type of control is either direct inverse kinematic, 1st order approximate, or simulated. This category will be filled out based on extensive mathematical research and is simply based on the authors results. This evaluation is only valid as long as no other direct or approximate solutions are found. Intuitiveness of motion is self explanatory. This measure will become important in later discussions of the human control interface. Self interference is a function of vehicle position and dimensions. It is difficult to enumerate and therefore based on conceptual estimates.

Table 4–4: Center Mounted Configurations[3]

Configuration	Work Envelope Volume	Maximum Radius	Minimum Radius	Center of Gravity	Type of Control	Dextility	Intuitiveness of Motion	Self Interference	Comments
1	13 L^3 94%	1.66 L	.36 L	.26	Direct	B–	A	B	Horiz. Hemis. 4th DOF
2	14 L^3 96%	1.67 L	.39 L	.27	Direct	B–	A–	B	Vert. Hemis. 4th DOF
3	17 L^3 96%	1.64 L	.40 L	.29	Appx.	B+	B	C	Redundant 5th Degree
4	17 L^3 94%	1.79 L	.41 L	.31	Siml.	?	F	B+	Top Heavy

Table 4–5: End Mounted Configurations

5	36 L^3 97%	217%	40%	.33	Siml.	B	C	A	Dual Boom Only
6	7 L^3 87%	138%	42%	.30	Siml.	?	F	A	Small Work Volume
7	5 L^3 61%	138%	79%	.30	Siml.	?	F	A	Small Work Volume
8	4 L^3 47%	143%	94%	.26	Siml.	?	F	A	Worst Work Volume
9	8 L^3 92%	158%	50%	.31	Direct	A	A	B	Best Dexterity
10	27 L^3 99%	212%	9%	.29	Direct	B	A	B	Long Reach

3 The "Work Envelope Volume" is expressed as a measure of L^3= (Vehile Length)^3 dimensions. The number beneath it is the percentage of its "Volume Efficiency".

The center of gravity is based on a boom length of unity and is a measured from the boom base. This is a rather vague measure, since both the absolute boom weight and the boom component placement had to be estimated.

Based on the above evaluation table, all of the configurations with direct control were given above average qualitative grades. While their quantitative values differ, within each category of center mounted vs. end mounted, only one configuration stands out, depending on whether the work envelope or the center of gravity is more highly valued. Further specifications are necessary to evaluate this truce.

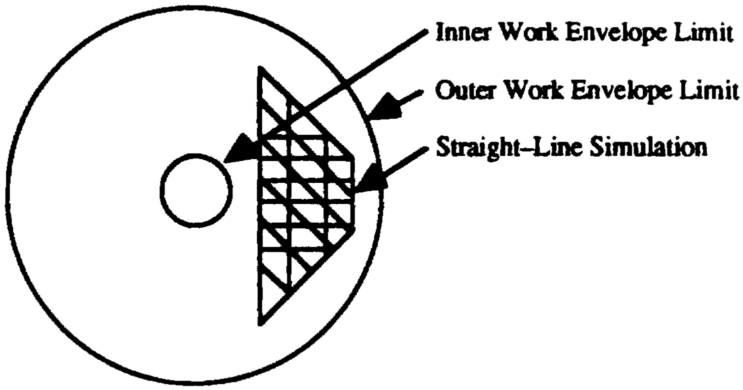
WORK-ENVELOPE VOLUME

The work envelope volume for any one of proposed geometries is simply its cross sectional area through its base joint integrated over 2π radians. Finding the cross sectional area however requires finding the inner and outer limits of the work envelope. Since there is no easy evaluation mechanism for a four degree of freedom kinematic structure, a plot of random joint rotations whose end effector values fall within a narrow cross sectional plane generates a reasonable estimate. The end-effector coordinates are calculated by the forward kinematic equations given by the Denvenit Hartenberg transformations. Depending on the height of the accepted cross-sectional area, between 100,000 and 500,000 iterations are necessary for a clean outline. During the simulation it is easy to also calculate the minimum and maximum range of the end effector from its origin. While these values are of only limited use, since they are direction dependent, they give first order approximations of volume utilization.

CONTINUOUS PATH TRACKING

Work envelope continuity is a prerequisite for a valuable boom configuration. Unless the end-effector can move from any one point inside the outer and inner limitation of the work envelope to another, its work use ability will be nil. Unfortunately without reverse kinematic equations, it is impossible to prove that a work-envelope is continuous and without voids. However it takes only a single instance to prove that a work-envelope is non-continuous.

Straight-line tracking is a special case of continuous path-tracking. A straight-line simulation that could cover most of the known work-envelope without a single failure give some evidence that a work envelope is continuous without having to obtain inverse kinematic solutions. If the simulation fails to execute a path within the conceived work-envelope, the boom configuration in testing is probably not worth further consideration. Tasks like drilling operations are dependent on straight line tracking. A possible test for continuity could be the following path tracking exercise.



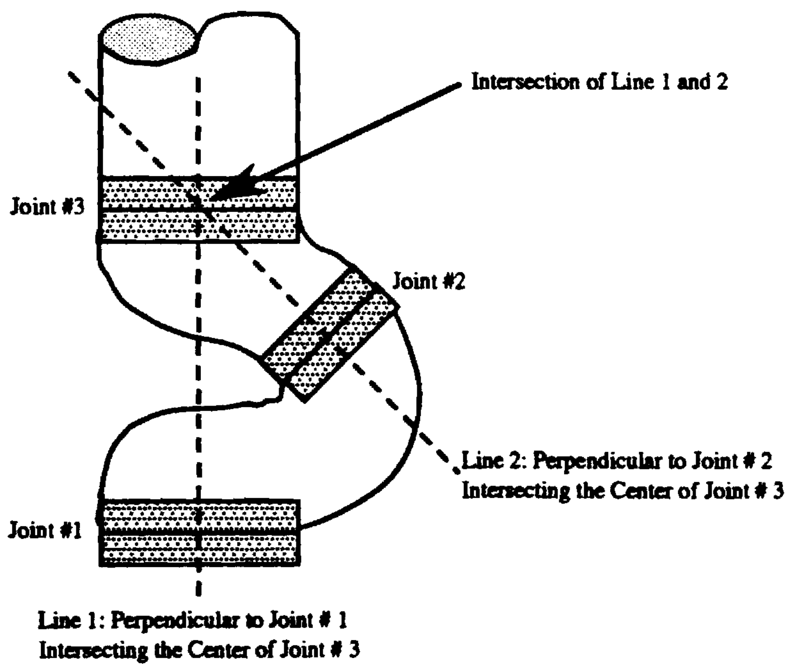
Possible Straight–Line Test Path: Figure 4–17

Note that is only necessary to analyze the plane perpendicular to that of the base joint, since the base rotation will add three-dimensionality to that of the cross-section. A computer simulation for continuous path tracking randomly varies the joint rotation of each articulation joint by a small finite step. If the resulting end-effector position is within an allowable tolerance from straight line deviation to the point of destination, then the new point is recorded as good. If not, the most recent previous good joint rotations are revisited and another random attempt is taken. If the program begins to cycle, the next step is to reduce the end-effector deviation from path line. If no further points can be found and cycling does not stop, the simulation has either reached the limits of the work-envelope, or encountered a non-linearity. Determining which points are within the work-envelope must be estimated from the random cross-sectional plots. This method is tedious and does not prove that a configuration has a continuous work-envelope.

Kinematic Solution

A three degree of freedom boom could have the same work envelope described by any one of the suggested boom configurations. However the suggested configurations have one additional degree–of–freedom to increase dexterity. This additional degree–of–freedom allows the end–effect to be placed at its destination via several different joint rotations. These multiple solutions cause difficulty in performing the inverse kinematic analysis.

In deriving various boom geometries one of the key thoughts was how to utilize the fourth freedom in motion. The ideal mechanism seemed to be a rotation about the line defined by the end-effector coordinates and its relative boom–base origin. This concept gave rise to a ball-point type rotating mechanism at the boom base. To approximate a ball-point type socket, three stove-pipe joints are arranged so that the combination of their center lines intersect at one common focus. This is illustrated in the following diagram:

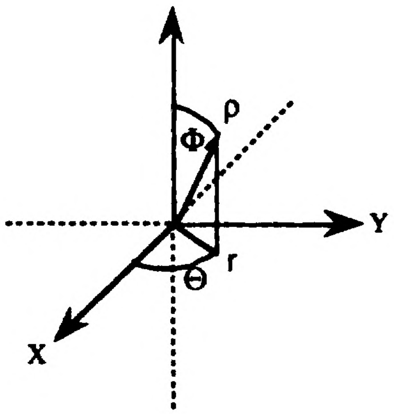


Stove–Pipe Configuration of a Hemispherical Ball–Point Socket: Figure 4–18

Since rotations about the first three joints has no effect on X, Y, and Z at the point of intersection, the intersection point will from hereon be defined as the boom origin. The most convenient way to represent the end-effector coordinates are in spherical coordinates. Insight into this coordinate system will be used to gain understanding necessary for a generating a complete inverse kinematic solution.

STOVE-PIPE REPRESENTATION OF SPHERICAL COORDINATES

Spherical coordinates are described in terms of radius, phi, and theta. The first three joints of the stove–pipe representation of a ball–point socket affect pitch, yaw, and roll only. The fourth joint is solely responsible for varying the radius. Pitch, a measure of phi, is only affected by stove–pipe joint number two.



Spherical Coordinate System: Figure 4–19

The effect of a rotation about the first three articulation joints is:

$$X = +\frac{\sqrt{2}}{2}\ \rho\ \mathrm{Cos}(\Theta_1)^{*}\ \mathrm{Sin}(\Theta_2) + \frac{1}{2}\ \rho\ \mathrm{Cos}(\Theta_2)\ \mathrm{Sin}(\Theta_1) + \frac{1}{2}\ \rho\ \mathrm{Sin}\Theta_1) \qquad (4\text{–}5)$$

$$Y = +\frac{\sqrt{2}}{2}\ \rho\ \mathrm{Sin}(\Theta_1)\ \mathrm{Sin}(\Theta_2) - \frac{1}{2}\ \rho\ \mathrm{Cos}(\Theta_1)\ \mathrm{Cos}(\Theta_2) - \frac{1}{2}\ \rho\ \mathrm{Cos}(\Theta_1) \qquad (4\text{–}6)$$

$$Z = -\frac{1}{2}\ \rho\ \mathrm{Cos}(\Theta_2) + \frac{1}{2}\ \rho \qquad (4\text{–}7)$$

The effect of a rotation about articulation joint number two in terms of Theta and Phi are:

$$X = +\frac{\sqrt{2}}{2}\rho\,\mathrm{Sin}(\Theta_2) \quad = \rho\,\mathrm{Sin}(\Phi)\,\mathrm{Cos}(\Theta) \tag{4–8}$$

$$Y = -\frac{1}{2}\rho\left(\mathrm{Cos}(\Theta_2) + 1\right) \quad = \rho\,\mathrm{Sin}(\Phi)\,\mathrm{Sin}(\Theta) \tag{4–9}$$

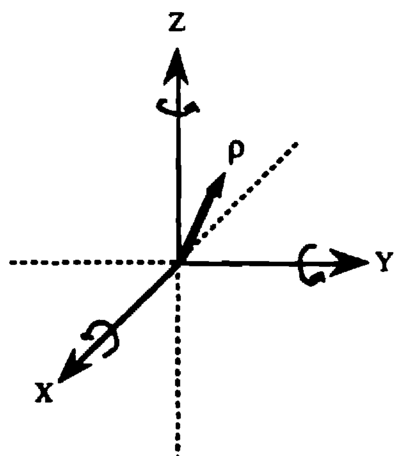
$$Z = +\frac{1}{2}\rho\left(1 - \mathrm{Cos}(\Theta_2)\right) \quad = \rho\,\mathrm{Cos}(\Phi) \tag{4–10}$$

$$\Phi = \mathrm{Cos}^{-1}\left(\frac{1 - \mathrm{Cos}(\Theta_2)}{2}\right) \tag{4–11}$$

$$\Theta = \mathrm{Tan}^{-1}\left(\frac{-\mathrm{Cos}(\Theta_2) - 1}{\sqrt{2}\,\mathrm{Sin}(\Theta_2)}\right) \tag{4–12}$$

STOVE-PIPE REPRESENTATION OF A BALL POINT SOCKET

A ball–point joints motion are defined in terms of roll, pitch, and yaw. Pitch is an upward or downward motion corresponding to a change in Φ. Yaw is a sideways motion corresponding to a change in Θ. Roll is a rotation about the first segment attached to the ball–point socket. The overall effect is best illustrated by rotation transformations about the X, Y, and Z–axis:



Transformation Rotations: Figure 4–20

The goal of the inverse kinematic equations will be to express the joint rotations in terms of the three matrix transformations about the X, Y, and Z–axis and the length vector ρ. Four equations and four unknowns makes the following inverse kinematic analysis possible:

$$\begin{bmatrix} \overline{X} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{Y} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{Z} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{I} & \overline{\rho} \\ 0 & 1 \end{bmatrix} = \begin{bmatrix} \overline{\Theta_2} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{I} & \overline{L_1} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{\Theta_3} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{I} & \overline{L_2} \\ 0 & 1 \end{bmatrix} * \begin{bmatrix} \overline{\Theta_1} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{I} & \overline{L_{34}} \\ 0 & 1 \end{bmatrix} \quad (4\text{–}13)$$

$\overline{X}$, $\overline{Y}$, and $\overline{Z}$ are transformations about the X, Y, and Z–axis as defined by the Denvenit–Hartenberg matrix transformation. $\overline{I}$ is an identity matrix, $\overline{\rho}$ a direction vector, $\overline{\Theta_1}$, $\overline{\Theta_2}$, and $\overline{\Theta_3}$ rotation transformations about the respective stove–pipe joints, and $\overline{L_1}$, $\overline{L_1}$, and $\overline{L_{34}}$ directional vectors. The directional vector $\overline{L_{34}}$ includes the last two segments of the boom thereby including the fourth articulation joint (Θ_4).

INVERSE KINEMATIC ANALYSIS

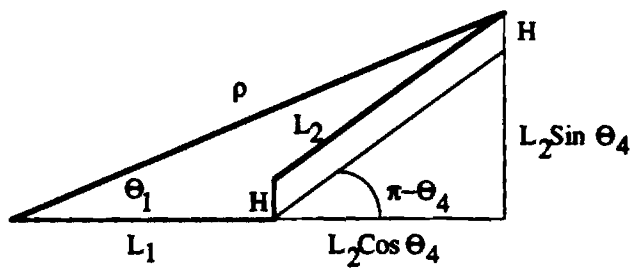
Deriving the reverse kinematic equations directly from the format kinematic analysis is not possible, since several variables are dependent on more than one other variable. The goal of the inverse kinematic analysis will be to identify the variables with one dependency first, and then extrapolate the remaining complex combinations. A general system of equations for a ball–point type socket of stove–pipe joints with two additional boom segments is as follows:

$$\begin{aligned} X = &- Y_4 \operatorname{Cos}(\Theta_1) \operatorname{Cos}(\Theta_2) \operatorname{Cos}(\Theta_3) \operatorname{Sin}(\Theta_4) - Z_4 \operatorname{Cos}(\Theta_1) \operatorname{Cos}(\Theta_2) \operatorname{Sin}(\Theta_3) + \frac{\sqrt{2}}{2} Y_4 \operatorname{Cos}(\Theta_1) \\ &\operatorname{Sin}(\Theta_2) \operatorname{Sin}(\Theta_3) \operatorname{Sin}(\Theta_4) - \frac{\sqrt{2}}{2} Z_4 \operatorname{Cos}(\Theta_1) \operatorname{Sin}(\Theta_2) \operatorname{Cos}(\Theta_3) - \frac{\sqrt{2}}{2} Y_4 \operatorname{Cos}(\Theta_1) \operatorname{Sin}(\Theta_2) \operatorname{Cos}(\Theta_4) + \frac{\sqrt{2}}{2} \\ &Z_3 \operatorname{Cos}(\Theta_1) \operatorname{Sin}(\Theta_2) + \frac{\sqrt{2}}{2} Y_4 \operatorname{Sin}(\Theta_1) \operatorname{Sin}(\Theta_2) \operatorname{Cos}(\Theta_3) \operatorname{Sin}(\Theta_4) + \frac{\sqrt{2}}{2} Z_4 \operatorname{Sin}(\Theta_1) \operatorname{Sin}(\Theta_2) \operatorname{Sin}(\Theta_3) + \frac{1}{2} \\ &Y_4 \operatorname{Sin}(\Theta_1) \operatorname{Cos}(\Theta_2) \operatorname{Sin}(\Theta_3) \operatorname{Sin}(\Theta_4) - \frac{1}{2} Z_4 \operatorname{Sin}(\Theta_1) \operatorname{Cos}(\Theta_2) \operatorname{Cos}(\Theta_3) - \frac{1}{2} Y_4 \operatorname{Sin}(\Theta_1) \operatorname{Cos}(\Theta_2) \operatorname{Cos}(\Theta_4) \\ &+ \frac{1}{2} Z_3 \operatorname{Sin}(\Theta_1) \operatorname{Cos}(\Theta_2) - \frac{1}{2} Y_4 \operatorname{Sin}(\Theta_1) \operatorname{Sin}(\Theta_3) \operatorname{Sin}(\Theta_4) + \frac{1}{2} Z_4 \operatorname{Sin}(\Theta_1) \operatorname{Cos}(\Theta_3) - \frac{1}{2} Y_4 \operatorname{Sin}(\Theta_1) \\ &\operatorname{Cos}(\Theta_4) + \frac{1}{2} Z_3 \operatorname{Sin}(\Theta_1) \end{aligned} \quad (4\text{–}14)$$

$$\begin{aligned} Y= &- Y_4 \operatorname{Sin}(\theta_1) \operatorname{Cos}(\theta_2) \operatorname{Cos}(\theta_3) \operatorname{Sin}(\theta_4) - Z_4 \operatorname{Sin}(\theta_1) \operatorname{Cos}(\theta_2) \operatorname{Sin}(\theta_3) + \frac{\sqrt{2}}{2} Y_4 \operatorname{Sin}(\theta_1) \\ &\operatorname{Sin}(\theta_2) \operatorname{Sin}(\theta_3) \operatorname{Sin}(\theta_4) - \frac{\sqrt{2}}{2} Z_4 \operatorname{Sin}(\theta_1) \operatorname{Sin}(\theta_2) \operatorname{Cos}(\theta_3) - \frac{\sqrt{2}}{2} Y_4 \operatorname{Sin}(\theta_1) \operatorname{Sin}(\theta_2) \operatorname{Cos}(\theta_4) + \frac{\sqrt{2}}{2} Z_3 \\ &\operatorname{Sin}(\theta_1) \operatorname{Sin}(\theta_2) - \frac{\sqrt{2}}{2} Y_4 \operatorname{Cos}(\theta_1) \operatorname{Sin}(\theta_2) \operatorname{Cos}(\theta_3) \operatorname{Sin}(\theta_4) - \frac{\sqrt{2}}{2} Z_4 \operatorname{Cos}(\theta_1) \operatorname{Sin}(\theta_2) \operatorname{Sin}(\theta_3) - \frac{1}{2} Y_4 \\ &\operatorname{Cos}(\theta_1) \operatorname{Cos}(\theta_2) \operatorname{Sin}(\theta_3) \operatorname{Sin}(\theta_4) + \frac{1}{2} Z_4 \operatorname{Cos}(\theta_1) \operatorname{Cos}(\theta_2) \operatorname{Cos}(\theta_3) + \frac{1}{2} Y_4 \operatorname{Cos}(\theta_1) \operatorname{Cos}(\theta_2) \operatorname{Cos}(\theta_4) \\ &- \frac{1}{2} Z_3 \operatorname{Cos}(\theta_1) \operatorname{Cos}(\theta_2) + \frac{1}{2} Y_4 \operatorname{Cos}(\theta_1) \operatorname{Sin}(\theta_3) \operatorname{Sin}(\theta_4) - \frac{1}{2} Z_4 \operatorname{Cos}(\theta_1) \operatorname{Cos}(\theta_3) + \frac{1}{2} Y_4 \operatorname{Cos}(\theta_1) \\ &\operatorname{Cos}(\theta_4) - \frac{1}{2} Z_3 \operatorname{Cos}(\theta_1) \end{aligned} \qquad (4\text{–}15)$$

$$\begin{aligned} Z= &- \frac{\sqrt{2}}{2} Y_4 \operatorname{Sin}(\theta_2) \operatorname{Cos}(\theta_3) \operatorname{Sin}(\theta_4) - \frac{\sqrt{2}}{2} Z_4 \operatorname{Sin}(\theta_2) \operatorname{Sin}(\theta_3) - \frac{1}{2} Y_4 \operatorname{Cos}(\theta_2) \operatorname{Sin}(\theta_3) \operatorname{Sin}(\theta_4) \\ &+ \frac{1}{2} Z_4 \operatorname{COS}(\theta_2) \operatorname{Cos}(\theta_3) + \frac{1}{2} Y_4 \operatorname{Cos}(\theta_2) \operatorname{Cos}(\theta_4) + \frac{1}{2} Z_3 \operatorname{Cos}(\theta_2) - \frac{1}{2} Y_4 \operatorname{Sin}(\theta_3) \operatorname{Sin}(\theta_4) + \frac{1}{2} Z_4 \\ &\operatorname{Cos}(\theta_3) - \frac{1}{2} Y_4 \operatorname{Cos}(\theta_4) + \frac{1}{2} Z_3 \end{aligned} \qquad (4\text{–}16)$$

Theta four is the rotation between the last two boom segments. Theta four is the only variable capable of adjusting the boom reach expressed by ρ. Calculating ρ from the user commands gives the following relationship:



Theta 4: Figure 4–21

The height between segment three and four of the boom reduces the value of theta four almost insignificantly, yet is not ignored in the following calculations:

$$\begin{aligned}\rho^2 &= \left(L_1 - L_2\mathrm{Cos}(\Theta_4)\right)^2 + H^2 + \left(L_2\mathrm{Sin}(\Theta_4)\right)^2 = \\ &= L_1{}^2 - 2L_1L_2\mathrm{Cos}(\Theta_4) + L_2{}^2\mathrm{Cos}(\Theta_4)^2 + H^2 + L_2{}^2\mathrm{Sin}(\Theta_4)^2 \\ &= L_1{}^2 - 2L_1L_2\mathrm{Cos}(\Theta_4) + H^2 + L_2{}^2\left(\mathrm{Sin}(\Theta_4)^2 + \mathrm{Cos}(\Theta_4)^2\right)\end{aligned} \tag{4–17}$$

$$\Theta_4 = \mathrm{Cos}^{-1}\left(\frac{L_1{}^2+L_2{}^2+H^2-\rho^2}{2L_1L_2}\right) \quad \text{where} \begin{cases} L_1 = Z_3 \\ L_2 = Y_4 \\ H = Y_3+Z_4 = Z_4 \end{cases} \tag{4–18}$$

Theta two is the main variable adjusting height, and the only variable affecting the angle Φ_{L1} of the first boom segment. Depending on the orientation of the dexterity variable 4_{th}, Φ_{L1} can range from 0 to $(2\Theta_i+\Phi)$, where Θ_i is the inside angle (Fig. 4–21) of boom segments one and two, and Φ is the angle determined by the end–effector height and reach. Θ_i and Φ are calculated as follows:

$$\Phi = \mathrm{Atan}^{-1}\left(\frac{H}{\rho}\right) \tag{4–19}$$

$$\Theta_i = \mathrm{Cos}^{-1}\left(\frac{L_1{}^2+\rho^2-H^2-L_2{}^2}{-2\sqrt{\rho^2-H^2}\,L_1}\right) \tag{4–20}$$

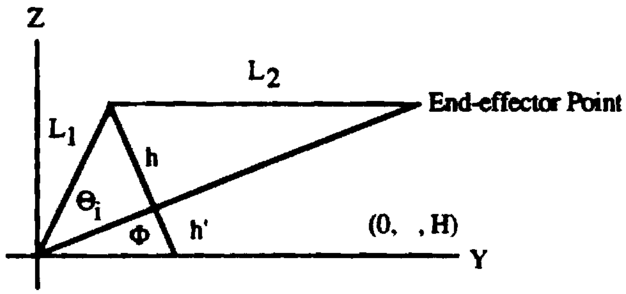
Aligning the first boom segment in the ZY-plane and the end–effector at height Z, gives the following coordinates for Y_{L1} and Z_{L1}:

$$X_{L1'} = 0 \tag{4–21}$$

$$Y_{L1'} = L_1\,\mathrm{Cos}(\Phi+\Theta_i) \tag{4–22}$$

$$Z_{L1'} = L_1\,\mathrm{Sin}(\Phi+\Theta_i) \tag{4–23}$$

The amount of dexterity available to the fourth degree–of–freedom which is limited by the hemispherical range of the first boom segment is given by the maximum amount of rotation of L_1' about the line intersecting the origin and the end-effector. The maximum rotation is achieved when the transformation $L_1'(Z, 4_{th})$ is zero. Graphically and mathematically this calculated as follows:

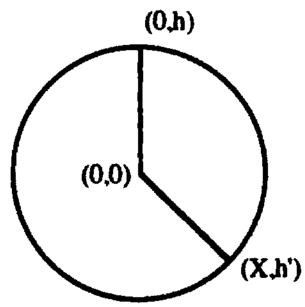


Maximum Dexterity: Figure 4–22

$$h = L_1 \, Sin(\Theta_i) \tag{4–24}$$

$$h' = L_1 \, Cos(\Theta_i) \, Tan(\Phi) \tag{4–25}$$

Graphically, the 4_{th} degree–of–freedom is the in angle between h and h':



4_{th} Degree–Of–Freedom: Figure 4–23

$$4_{th} = Acos\left(\frac{h(-h')}{h^2}\right) \tag{4–26}$$

If $|h'| > h$ the fourth degree–of–freedom is unrestricted and can vary between π and $-\pi$. The angle Φ_{L1}

depends on Θ_i, Φ, and 4_{th}. Mathematically the 4_{th} degree–of–freedom will be represented as a percentage of maximum permissible amount of rotation given by equation 4–26. From figure 4–22, $\Phi_{L1}(\Theta_i,\Phi,4_{th})$ is calculated by the following matrix rotations:

$$\Phi_{L1}(\Theta_i,\Phi,4_{th}) = \begin{bmatrix} \overline{X_{\Phi i}} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{\%4_{th}} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{-X_{\Phi}} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{I} & \overline{L_1'} \\ 0 & 1 \end{bmatrix} \tag{4-27}$$

Transformed coordinates of boom segment three:

$$X''_{L1} = -L_{Y1'}\,\mathrm{Sin}(\%4_{th})\mathrm{Sin}(\Phi) + L_{Z1'}\,\mathrm{Sin}(\%4_{th})\mathrm{Cos}(\Phi) \tag{4-28}$$

$$\begin{aligned} Y''_{L1} = {} & (L_{Y1'}\,\mathrm{Cos}(\Phi) + L_{Z1'}\,\mathrm{Sin}(\Phi))\mathrm{Cos}(\Phi) \\ & -(-L_{Y1'}\,\mathrm{Cos}(\%4_{th})\mathrm{Sin}(\Phi) + L_{Z1'}\,\mathrm{Cos}(\%4_{th})\mathrm{Cos}(\Phi))\mathrm{Sin}(\Phi) \end{aligned} \tag{4-29}$$

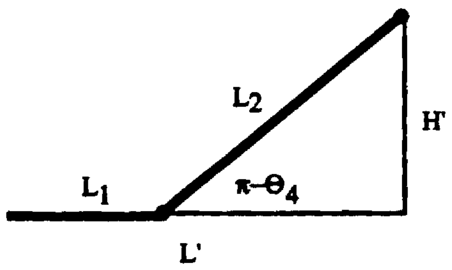
$$\begin{aligned} Z''_{L1} = {} & (L_{Y1'}\,\mathrm{Cos}(\Phi) + L_{Z1'}\,\mathrm{Sin}(\Phi))\mathrm{Cos}(\Phi) \\ & (-L_{Y1'}\,\mathrm{Cos}(\%4_{th})\mathrm{Sin}(\Phi) + L_{Z1'}\,\mathrm{Cos}(\%4_{th})\mathrm{Cos}(\Phi))\mathrm{Cos}(\Phi) \end{aligned} \tag{4-30}$$

$$\Phi = \mathrm{Atan}^{-1}\left(\frac{Z''}{\sqrt{X^2_{L1''} + Y^2_{L1''}}}\right) \tag{4-31}$$

From equation (4–11) it follows that:

$$\Theta_2 = \mathrm{Cos}^{-1}\left(1 - 2\mathrm{Cos}(\Phi_2)\right) \tag{4-32}$$

Theta three affects X, Y, and Z coordinates whereas theta one affects X, and Y coordinates only. Knowing the values of Θ_4 and Θ_2 makes it possible to solve for Θ_3 in terms of Z. Notice that rotation about Θ_4 and Θ_2 are always positive. This does not limit the dexterity of the boom in any way, but much rather prevents discontinuities. Given gradual small changes of boom direction, Θ_4 and Θ_2 would always stay positive or negative, depending on their initial settings. This phenomena is used to make a conclusion about theta three at a later stage. The Z value is a function of the rotation about joint number three. The vector of rotation is given as follows:

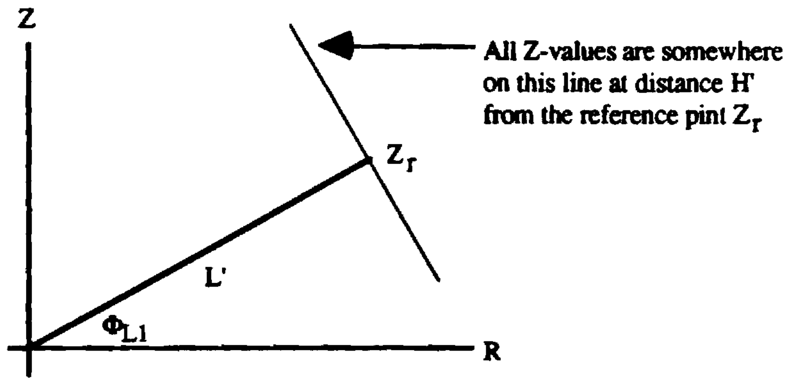


Boom Reach: Figure 4–24

$$H' = \sqrt{H^2 + L_2 \mathrm{Sin}(\Theta_4)^2} \quad (4\text{–}33)$$

$$L' = L_1 - L_2 \mathrm{Cos}(\Theta_4) \quad (4\text{–}34)$$

Theta three will be derived similar to the way the range of the fourth degree–of–freedom was calculated with the use of h and h'. Here we define H' as the distance of the end-effector to the perpendicular of the line through boom segment one. L' is the distance from the origin to the point of intersection of H' with boom segment one. Knowing Φ_{L1} and L' makes it possible to calculate a reference point for net rotation requirements as illustrated in the following graph:



Theta 3: Figure 4–25

$$Z_r = L'\mathrm{Sin}(\Phi_{L1}) \quad (4\text{–}35)$$

$$Z_c = F_z(0,\Theta_2,0,\Theta_4) = -\frac{\sqrt{2}}{2} Y_4 \mathrm{Sin}(\Theta_2)\,\mathrm{Sin}(\Theta_4) + \frac{1}{2} Z_4 \mathrm{COS}(\Theta_2) + \frac{1}{2} Y_4 \mathrm{Cos}(\Theta_2)\,\mathrm{Cos}(\Theta_4)$$

$$+\frac{1}{2}Z_3\,\mathrm{Cos}(\Theta_2)+\frac{1}{2}Z_4-\frac{1}{2}Y_4\,\mathrm{Cos}(\Theta_4)+\frac{1}{2}Z_3 \tag{4-36}$$

$$Z = Z \tag{4-37}$$

$$\Theta_3(Z_c,Z_T) = \mathrm{Asin}^{-1}\left(\frac{Z_T - Z_c}{H'\mathrm{Cos}(\Phi_{L1})}\right) \tag{4-38}$$

$$\Theta_3(Z,Z_T) = \mathrm{Asin}^{-1}\left(\frac{Z - Z_T}{H'\mathrm{Cos}(\Phi_{L1})}\right) \tag{4-39}$$

Since Θ_4 is always positive between 0–180 degrees and the same is true for Θ_2 the overall effect is that at Θ_3 equals zero the 4th degree is always arranged clockwise. Since Formula (4–38) measures the rotation to Z' from the vertical axis for clockwise arrangements the smaller difference of the two rotations is valid:

$$\Theta_3 = \Theta_3(Z,Z_T) - \Theta_3(Z_c,Z_T) \tag{4-40}$$

else

$$\Theta_3 = \Theta_3(Z,Z_T) + \Theta_3(Z_c,Z_T) \tag{4-41}$$

Theta one is the final variable left solving for and is simply obtained by checking the current position of X and Y given Θ_2, Θ_3, and Θ_4. The net rotation of Θ_1 is given by:

$$\Theta_1 = \mathrm{Tan}\left(\frac{Y - F_y(0,\Theta_2,\Theta_3,\Theta_4)}{X - F_x(0,\Theta_2,\Theta_3,\Theta_4)}\right) \qquad \text{if } F_x(0,\Theta_2,\Theta_3,\Theta_4) > 0 \tag{4-42}$$

When the current value of X is negative, the current boom end-tip is in the third or fourth quadrant.

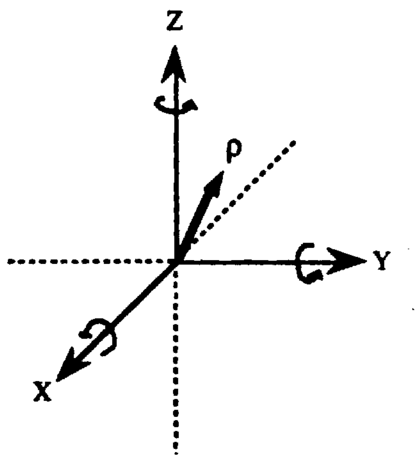
$$\Theta_1 = \pi - \mathrm{Tan}\left(\frac{Y - F_y(0,\Theta_2,\Theta_3,\Theta_4)}{X - F_x(0,\Theta_2,\Theta_3,\Theta_4)}\right) \quad \text{if } F_x(0,\Theta_2,\Theta_3,\Theta_4) < 0 \tag{4-43}$$

VELOCITY & ACCELERATION

The inverse kinematic solution specifies the amount of rotation each articulation joint has to move to reach the destination from the current position. If the trajectory is a complex path that has to be executed with high precision, then that trajectory can be split into a finite number of smaller segments for better control. Unless velocity and acceleration are critical, each individual net joint rotation can be executed in a specified equal amount of time. Using the most basic physics equations the resulting velocity and acceleration curves can be calculated. Given that actuators do not change instantaneous and are prone to overshoot, a more complex control theory type algorithm with a closed loop feedback system capabilities can be developed. This feature is not a critical requirement of the problem statement and is therefore left as a topic for a future project.

WORLD COORDINATES & BOOM COORDINATES

The base of the boom is not always perpendicular to the field of gravity. Depending on terrain and vehicle articulation joint configurations world coordinates and boom coordinates can vary significantly. Since feedback of the environment is received in terms of world coordinates, data input is also best done in that coordinate system. A simple transformation between the two systems is possible through the following matrix transformations;



World-Vehicle Coordinates: Figure 4–26

$$\begin{bmatrix} \bar{X} & \bar{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \bar{Y} & \bar{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \bar{Z} & \bar{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \bar{I} & \bar{\rho} \\ 0 & 1 \end{bmatrix}$$

$$= \text{WORLD COORDINATES} \qquad (4\text{–}44)$$

Since there is no gyroscope to measure rotation about the Z-Axis the equations simplify to:

$$\begin{bmatrix} \overline{X\Theta} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{Y\Theta} & \overline{0} \\ 0 & 1 \end{bmatrix} \begin{bmatrix} \overline{I} & \overline{\rho v} \\ 0 & 1 \end{bmatrix} = \text{WORLD COORDINATES} \qquad (4\text{–}45)$$

Given the following basic matrix manipulation:

$$[A][B][X]=[Y] \qquad (4\text{–}46)$$

$$[X] = ([A][B])^{-1}([A][B])[X] = ([A][B])^{-1}[Y] = [B]^{-}[A]^{-11}[Y] \qquad (4\text{–}47)$$

$$\begin{bmatrix} \overline{Y\Theta} & \overline{0} \\ 0 & 1 \end{bmatrix}^{-1} \begin{bmatrix} \overline{X\Theta} & \overline{0} \\ 0 & 1 \end{bmatrix}^{-1} \begin{bmatrix} \overline{I} & \overline{\rho w} \\ 0 & 1 \end{bmatrix} = \text{VEHICLE COORDINATES} \qquad (4\text{–}48)$$

where

$$\overline{X\Theta} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos(X\Theta) & -\sin(X\Theta) \\ 0 & \sin(X\Theta) & \cos(X\Theta) \end{bmatrix} \qquad \overline{X\Theta}^{-1} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos(X\Theta) & \sin(X\Theta) \\ 0 & -\sin(X\Theta) & \cos(X\Theta) \end{bmatrix}$$

$$\overline{Y\Theta} = \begin{bmatrix} \cos(Y\Theta) & 0 & \sin(Y\Theta) \\ 0 & 1 & 0 \\ -\sin(Y\Theta) & 0 & \cos(Y\Theta) \end{bmatrix} \qquad \overline{Y\Theta}^{-1} = \begin{bmatrix} \cos(Y\Theta) & 0 & -\sin(Y\Theta) \\ 0 & 1 & 0 \\ \sin(Y\Theta) & 0 & \cos(Y\Theta) \end{bmatrix}$$

The resulting transformation coordinates in X, Y, and Z for world and vehicle coordinates are:

World Coordinates:

$$X_W = X_V\text{Cos}(Y\Theta)+Y_V\text{Sin}(Y\Theta)\text{Sin}(X\Theta)+Z_V\text{Sin}(Y\Theta)\text{Cos}(X\Theta) \qquad (4\text{–}49)$$

$$Y_W = Y_V\text{Cos}(X\Theta)-Z_V\text{Sin}(X\Theta) \qquad (4\text{–}50)$$

$$Z_W = -X_V\text{Sin}(Y\Theta)+Y_V\text{Cos}(Y\Theta)\text{Sin}(X\Theta)+Z_V\text{Cos}(Y\Theta)\text{Cos}(X\Theta) \qquad (4\text{–}51)$$

Vehicle Coordinates:

$$X_V = X_W\text{Cos}(Y\Theta)-Z_W\text{Sin}(Y\Theta) \qquad (4\text{–}52)$$

$$Y_V = X_W\text{Sin}(X\Theta)\text{Sin}(Y\Theta)+Y_W\text{Cos}(X\Theta)+Z_W\text{Sin}(X\Theta)\text{Cos}(Y\Theta) \qquad (4\text{–}53)$$

$$Z_V = X_W\text{Cos}(X\Theta)\text{Sin}(Y\Theta)-Y_W\text{Sin}(X\Theta)+Z_W\text{Cos}(X\Theta)\text{Cos}(Y\Theta) \qquad (4\text{–}54)$$

BOOM FLEXIBILITY AND ERROR

Boom flexibility and geometric error due to construction imperfections give rise to skewness of end-effector coordinates. Both problems have vastly different origins, yet correcting their deviation is similar.

Boom flexibility is difficult to measure or estimate. The greater the load lifted by the end-effector and the greater the moment arm exerted about the boom-base, the larger the possible deflection. Fortunately there seems to be a direct correlation between vehicle tilt and boom flexibility. Vehicle tilt is a measure recorded by system gyroscopes. This information coupled with knowledge of the boom direction and experimental data could be used to correct the world coordinates by the estimated area, and consequently the vehicle coordinates of the boom.

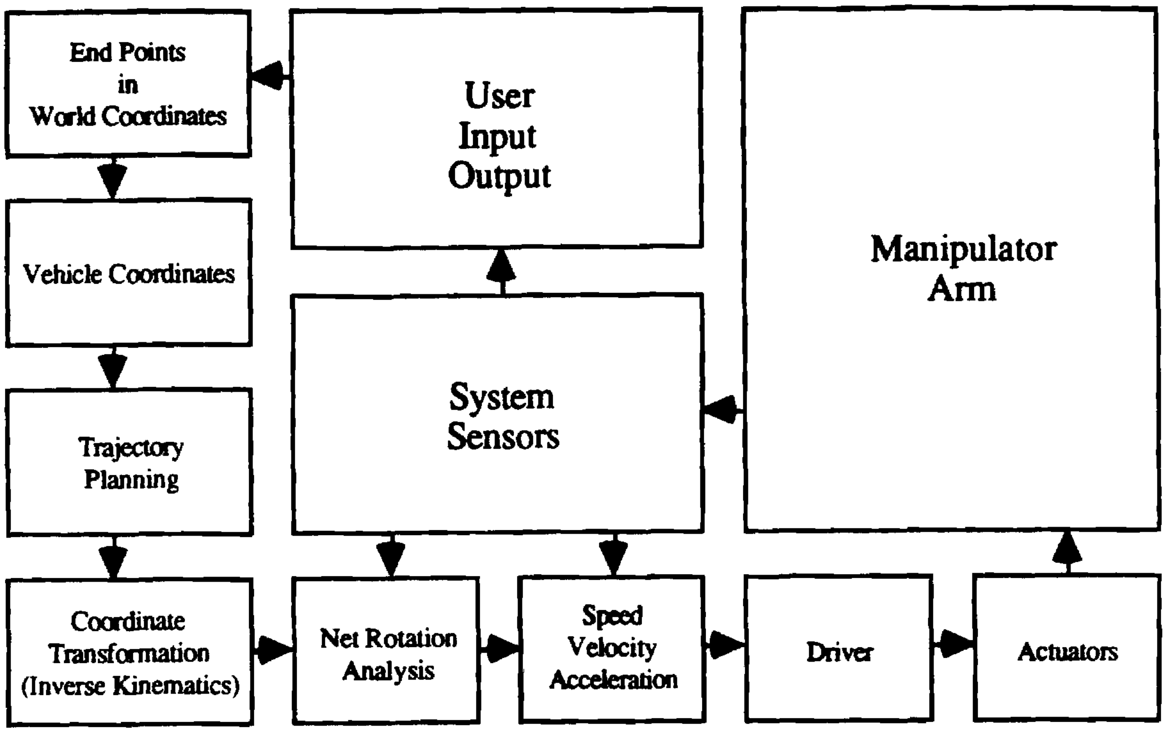
Imperfections in the boom geometry due to construction variations or, only approximate ball-joint configurations can be overcome by calculating a 1st order approximation of a perfect geometry. The difference in coordinates between the perfect geometry and actual geometry can be added to the desired boom–tip destination. Two or three further iterations should give an adequate result of the real articulation joint values.

INVERSE KINEMATIC SIMULATION FOR UNKNOWN SOLUTIONS

Statistical Analysis determines which joint rotation has a correlating effect on the overall straight line motion. If the coefficient of correlation is less than unity, the joints effect on straight line motion can be ignored. Smoothing of the data or fitting of regression models provides the desired motion instruction for each articulation joint. This process requires intensive data manipulation and does not suit itself well for manual override modes. While any system is manageable, the controllability is directly related to the computing power of the interface software. Given the extreme dexterity requirements of the design, a simple system will be more reliable and less vulnerable to breakdown, error, or service requirements. Furthermore there is no well known algorithm for properly utilizing the redundant degrees–of–freedom.

Kinematics Summary

Figure 4–12 is an overview of all the critical components necessary to control a kinematic structure with the methods described above. Further research into the human–machine interface and the control structure are necessary to implement this system.



Kinematic Analysis: Figure 4–27

CHAPTER V

Human Machine Interface

Designing equipment for use in space exploration and manufacturing means designing for multiple redundancy, dexterity and functionality. The resulting complex design, while mechanically ingenious is oftentimes humanly uncontrollable. The human machine interface is the necessary link that translates user inputs via a control system into necessary commands for direct drive actuators. In other words, the human machine interface organizes all system controls in such a way that they are convenient to use by one or more human operators.

The human-machine interface for a manipulator boom is one such system, which is complicated by the fact that the boom is functionally linked to the operation of vehicle movements, in addition to being tele-operated. Consequently a human–machine interface for the boom cannot be discussed, without also investigating vehicle operation and telerobotics.

The human interface solution proposed in this chapter should be viewed as a general discussion of *possible* solutions and requirements. Many of the human control issues related to telerobotics are still considered unsolved. The focus of this research concentrates on proposing a valuable geometry that is functional, controllable, and humanly–operateable.

Overview

This chapter is divided into six separate sections. The first section is intended to describe the problem domain in general, briefly summarizing the concepts involved in telerobotics. The second section describes task requirements for operation of a lunar vehicle, in addition to those already mentioned for operating the manipulator boom. The third section focuses on general human-machine issues involved in teleoperation. Factors that make this a difficult control problem will be emphasized. The fourth section describes the required functionality for a teleoperated lunar vehicles with respect to its task requirements. This section stresses different levels of operator control. Section five provides a feasible design of a human-machine interface for basic teleoperation. Section six will suggest an experimental development control module for initial analysis of a functional earth bound proof-of-concept model.

What is Telerobotics?

T.B. Sheridan [5], a pioneer in the field of human-machine interaction in telerobotics, defines the subject as follows:

> "Telerobotics is a form of *teleoperation* in which a human operator acts as a *supervisor* intermittently communicating to a computer information about goals, constraints, plans,... and orders relative to a task, getting back information about accomplishments, difficulties, and concerns." (p. 488) [italics added]

The important ideas in this paragraph are the two italicized words "*teleoperation*" and "*supervisor*". *Teleoperation* refers to a situation where the human controller is somehow removed from the actual robot. Tasks that are potentially dangerous, or inaccessible to humans such as radioactive areas, extreme heat spots, deep-sea oceanography, explosives, or outer space are ideally suited for teleoperated *supervision* from a distance. *Supervision* is the act of teleoperation assigned to the human operator. The robot itself is neither autonomous nor incapable. Rather than relying on direct input for movements, the robot needs only guidance in the form of human supervision to execute its task assignments.

If prior knowledge about a given environment were complete, the robot could be entirely preprogrammed to carry out its functions without the need for a human controller. Automated industrial robots in an assembly applications provide a good example of these conditions. However, in a telerobotics application, the demands of a particular task are often dynamic. Information about the environment is incomplete, and appropriate actions are often impossible to specify. These conditions require the presence of the human operator in the control loop.

This does not, however, mandate direct control of the robotic functions at all times. Lower-level assignments may be better suited to be performed without human intervention. For this reason, different levels of control must exist. In some situations, the operator may only want to provide high-level description of intentions and goals. In these cases the robot would interpret the commands and carry out the actions. In other cases, the operator may have to specify low level values for particular robotic actuators. The specific levels of control required depends on the physical work environments the limits of the robot capabilities and the complexity of the task requirement. A detailed discussion of various task requirements in addition to those already discussed in the previous chapter for the manipulator boom will follow in the next section.

Lunar Vehicles

The first lunar rover was designed to help astronauts collect earth samples beyond walking distance of their landing site [18],[19]. These exploratory missions were dangerous and unnecessarily exposed astronauts to high risk situations. Future operations will leave first-time exploration missions to teleoperated vehicles. The boom will perform geological tasks and transmit the results to the control center for data mapping. Upon identification of proper geological sites teleoperated vehicles will assemble necessary construction and living habitats. A teleoperated robot control can be grouped into three main categories: vehicle control, boom operations, and maintenance. While the primary focus of this research is limited to the boom design, the linked nature of these categories makes a brief discussion of the not yet mentioned area of vehicle control and maintenance necessary.

Given the harsh conditions of the lunar environment, getting from point A to point B requires the abilities of an all terrain work-vehicle. For normal operation and obstacle avoidance, rover functions are grouped into three main categories: contour following, crevice crossing, and obstacle climbing [3].

Contour following concerns the ability to read terrain surface features, simplifying vehicle steering and directional controls. This is achieved through stress-gauges along the wheel surfaces connected to an on-board computer which controls articulation joint movements to equalize load differentials between wheels on a common axis.

A small crater should not be the end of a mission. For example, Enabler's chassis was designed to be inherently rigid so that any wheel axis can be supported off-ground without having to bear any loads. The chassis has three axis of which only two are needed for regular stability. The third axis in combination with boom movements allows the redistribution of vehicle weight to permit features such as crevice crossings.

Similar to crevice crossing, the third axis and boom movements can be used to relief load of one axis for obstacle climbing. This is important for navigation through rough terrain with multiple small boulders.

Boom and chassis movements are not independent since the boom is the means of shifting weight during vehicle operations. Besides exploration and assembly operations, the boom supports such tasks as: object manipulation, data collection, vehicle stability, outrigger support, and interference avoidance. A detailed discussion of these tasks were discussed in chapter four.

If the rover fails, tips over, or is in a situations that has previously not been anticipated, procedures allowing for continued normal operation must be available. Internal status monitoring aids, trouble shooting routines, minor repairs, and operations in degraded mode are basic features that are to be addressed for all system variables.

General Human Control Issues

The mission which confronts lunar vehicles such as Enabler seem straight forward: perform simple construction tasks, traverse on open terrain, and collect data. To the human these tasks are easy to conceive and offer little intellectual challenge. As insignificant these assignments are for a human, we depend on a lot of conceptual knowledge not available to teleoperated robots. In teleoperated robotics the human controller depends on the information extracted from the surrounds by system sensors. Poor information gathering proficiency reduces conceptual knowledge about any state to levels difficult to control by remote human operators. The worse the data collection capabilities, the lower the autonomous modes of the robot. For autonomous operation the robot depends on "knowledge in the head" based on interpretation of the environment. Given poor spatial reasoning capabilities, limited processing power, and absence of basic physical understanding many tasks will be left to human interpretation and guidance.

This is not to say the machine autonomy is impossible, undesirable, or dispensable. The level of autonomy displayed by a robot depends on task understanding and the ability to translate this knowledge into machine code. For example, the actual contour of the lunar surface is unknown to the design engineers. Consequently precise vehicle movement cannot be anticipated. Yet machine usable decision rules and procedures can provide the necessary information to adapt to changes in the environment as they arise. It may not be possible to create a hazard recognition or traversal algorithm which approach performances levels achievable by a humans. Some tasks might even be totally unfit for automation. But contrary to factory automation problems which require precise control of both robot position and work piece location/orientation a similar level of control in lunar telerobotics is undesirable and economically inefficient. The level of supervision a machine requires to do a certain task in a particular environment is a question that needs to be answered. Concise sets of rules and procedures that are independent of "knowledge in the head" and common sense reasoning can reduce the level of necessary human supervision.

The level of autonomy displayed by a robot is a function of the number of feasible solutions and the amount of external data considered. Increases in supervision decrease the scope of the problem for the on-board computer until eventually only one or no solution exist. At this point autonomous operation ceases. In teleoperation the goal of the operator is to make decisions using high level functions and assist with movements that are too difficult to automate. Teleoperation, however, has its limits. Human operators decisions making depends on perception and feedback, both of which are severely degraded through teleoperation. Consequently, movements often seem unnatural and dexterity is lost. This is particularly true for manipulators. Degradation in perception is caused by any one of the following problems [12], [15]:

- Absence of depth perception
- Unfamiliar references
- Reduction in visual resolution
- Reduction in contrast
- Unnatural controls with limited ergonomic correlation
- Poor sense of direction due to awkward screen displays
- Absence of tactile sense
- Absence of gravitational forces like inclination, acceleration, and vibration
- Poor sense of self such as balance, attitude, and orientation.
- Audio feedback

Operating a lunar rover is further made difficult by the absence of familiar visual cues. The lunar landscape lacks familiar markers found on Earth necessary to gauge distance, magnitude, and ground characteristics. On top of that, a three seconds communication time delay between the moon and earth is going to make controls feel extremely sluggish. The real time behavior will exhibit characteristics more commonly associated with steering boats than those experienced in driving a vehicles or controlling direct drive machinery.

There is a number of approaches to deal with problems related to teleoperation. The simplest solution to performing a task is to change the necessary system variables by small incremental steps, one at a time. Each future increment is based on an evaluation of the previous step. If the previous step did not return the desired results, the plan of execution will be corrected no further action is taken. The assumption is that if the incremental steps are small enough, trouble can be avoided before disaster strikes. The obvious drawback to this approach is that the operation will be exceedingly slow. Another approach could separate tasks into a planning stage and an execution stage. Based on goals, site information, and machine limitations, the operator develops a plan for task execution. Once the plan is completed and validated, it is down loaded to the machine for implementation. Simulations based on knowledge of the robot's present state can be used to develop and validate such plans. Given sufficient information, the operator can anticipate problems and their solutions. The drawback is that discrepancy between expected state and actual state will accumulate in error until the system instructions are corrected with in process data collection. Tasks that are relatively insensitive to error accumulation are apt for such an open loop control strategy. A third approach which might be appropriate for terrain traversal is based on real time control. In this case, one accepts the sluggishness of the controls and simply acquires sufficient sensory information to provide the necessary look ahead capability. Given open terrain, or knowledge of previously traveled passages may be sufficient to anticipate and thereby avoid potential problems. A fourth approach might try to continually enhance the level of vehicle autonomy. While this attempt might prove difficult, it also depends, at least initially, on providing adequate sensory information to the operator. The site and task will dictate which method of control is the most appropriate. Lacking experience of lunar terrain the task at hand can upset any previous expectations of difficulty one way or another.

Specific Human Control Tasks

The previous sections identified potential problems with both autonomy and human control issues. Identifying the specific human involvement in the control loop for various tasks is the next necessary step to describing a comprehensive solution for the human control issues facing vehicles like Enabler. Although this will not directly specify specific or desirable displays and controls, it will provide indispensable guidance. The two major task areas for complete vehicle control are concerned with movement tasks. This includes boom and vehicle movements, and vehicle maintaining, like troubleshooting, repairing, or operating in a degraded state.

The highest level of movement tasks is "automated" control, ranging from no involvement by the human operator to high-level, symbolic instructions. The operator simply specifies a destination or task for the boom / vehicle and the machine is responsible for performing its own path planning. An internal model of the state of environment in addition to sets of rules and procedures help the robot execute the task in autonomy.

Examples of autonomous motion are the stabilization task of the boom during vehicle movements. Internal feedback from gyroscopic sensors provide information on ground inclination; wheel skid sensors provide information on traction; both of which are used to enhance vehicle stability and performance. Other examples are preprogrammed movement such as generic digging motion, exchanging tools and preventing boom–chassis collisions. Another level of autonomous movement is automatic correction of system variables due to external physical phenomena. Examples of this include automatic correcting for involuntary wheel spin and correction in end-effector position due to deflection or moment of the boom. Another level of automation is repetition of previously performed tasks such as soil samples, digging, or loading assignments. This class of tasks is called learned movements. Once stored in the automation "repertoire" it is recalled by a high level function through the human operator. Another level of automation is when the human controller gives the robot some kind of high-level symbolic instructions such as "return to base", or "climb this boulder" without specifying the exact path. The robot the executes the task based on its own path planning procedures.

"Guided movements" follow "automated movements" in the level of movement tasks. Here the operator gives the robot some high level instructions for a particular path without specifying enough information for precise execution. The human operator specifies intermediate points along the execution path and the robot follows this route in a bee–line type manner. This mode is ideal for guidance around obstacles, without having to worry about precision positioning or path efficient.

The next level of movement control is "direct control". This mode requires real-time operator attention in order to monitor terrain characteristics and avoid possible obstacles. With direct control the human operator is responsible for both path–planning and path–executing. Moving the vehicle in direct control is somewhat analogous to driving a car with the exception that the operator is at a remote location. Given the inherent complexity of movements and the multiple degrees–of–freedom in both the chassis, and the boom, the controls are simplified to behave in a more natural, and conceptual manner. All commands are given in real–time direct control mode.

The last level of movement control is "really direct control". Here the human operator directly and, perhaps individually, operates all drives for the chassis, boom, and wheels. Really direct control may be needed in difficult situations such being stuck, having rolled over, maintenance, degraded mode, or during other extremely complex tasks. Given the multiple degrees–of–freedom in both the chassis and the boom, really direct control of either is a tremendous challenge.

Maintaining the operational integrity of the robot is another major task that requires human operator involvement. Here the controller monitors the internal state of individual sensors, and the operational integrity of the mechanical parts. A decision support system with troubleshooting capabilities alerts the user of possible failures, tracks their causes, and identifies alternative feasible solutions. This assists the human operator in performing possible self-repairs and allows for easy recalibration of system constraints and variables. Problems that cannot be fixed should allow for continued operation in the degraded mode. The subsequent potential loss of range in motion should be reflected in the user interface with constraint on feasible motion commands. Those constraints could be initiated on a preprogrammed or human-originated bases.

Human-Machine Interface

Three basic types of information are necessary to assist a human operator in teleoperation. These are information providing knowledge about the state of the robot, knowledge about the state of the external environment, and decision support data. In the case of Enabler, the human-machine interface must provide controls for vehicle, boom, and end–effector movements in one of several interswitchable modes. The most basic feasible solution integrates these requirements into three displays, necessitating one monitor for each type of information.

Direct video displays about the state of an environment lack in-depth perception and need to be computer enhanced to give visual cues and reference points. In the case of terrain traversing a computer superimposed perspective grid map allows the operator to judge important objects and comprehend relative depth. While such information could be displayed by preference, it is an invaluable tool to operate in a three–dimensional situation using a two–dimensional display. This kind of information becomes extremely useful during guided movements and is easily constructed from information generated by range finders. Boom movements, likewise, are dependent on perspective knowledge of the environment. Without such information the most basic construction task would be extremely difficult and would have to proceed at an exceedingly slow speed. Video displays could be originated from mounting locations on the chassis, boom, or external observation posts.

When necessary these displays have the ability to show simulated data. One reason the operator may want to see simulated data is for path planning purposes. Simulation is a useful tool in previewing outcomes of various choices in actions. If the operator approves of a choice, its actions could then be executed. A second major reason for simulation capabilities is to accommodate for the inherent time lag in data transmission over large distances. Moon–earth communication experiences a three second time delay. Given enough computing power a simulation could generate perceived real–time images and compare them with actual data as it becomes available. If the difference in expected results versus actual values accumulates an error beyond acceptable predicted bounds, the task execution is stopped.

The second type of display shows information about the current state of the robot, or parts thereof such as the vehicle, individual joints, the boom, or the end–effector. View(s) displayed are based on perceived states as monitored by sensors linked to the onboard computers. This data is used to calculate the positions of overall vehicle components. Information derived from those calculations can be used to superimpose the state of the robot onto actual data about the external environment. Information like that provides immediate feedback to the operator about a given any task. Furthermore, these displays can be rotated in three dimensions to best represent the directionality of the users controls or sense of orientation.

A third type of display called the decision support system, monitors all system variables and lists any failures and out of range values. Data can be displayed in various levels of detail or by rank of importance. Error functions such as non–conformance to expected values, failures to execute, insufficient capacity, or low energy–level can be addressed with interpretation from the decision support system. This gives the operator a rapid understanding of how any functionality might be degraded. The decision support system then suggests possible solutions or maintenance routine to remedy the problems. If the failure is permanent the operator can choose to upgrade the system model to reflect the limited functionality in the control actuators. A further function of the decision support system is to anticipate the effects of operator input on such important variables as vehicle stability. For example, if movements or loads manipulated the end–effector have prospects of a tipping the robot, an active operator override would be necessary to keep the system from reversing its potentially dangerous movement command.

Each of the three monitors with their respective information operates in any one of the four basic types of modes discussed. The difference between the four modes is simply the means of data input by which operator commands are translated into machine code. Several technologies exist to communicate data from the human operator to the control system; all varying greatly in complexity. The most basic and easiest to implement methods for initial concept research are joy stick and screen driven control actuators. Joy sticks or a small replica of actual robotic actuators are useful for direct and really direct control, with the more advanced control features better apt to screen driven methods. The next section will describe a most basic experimental control module for testing of an actual manipulator boom.

Experimental Development Control Module

The comprehensive human control system for a complete lunar robotic rover is far beyond the scope of this discussion. Depending on the number of tool fixtures the type of boom configuration, the number of booms, and the number of lunar rovers assisting in a common task, the human control problem can take on proportions worthy of many more dissertation topics. Yet every human–machine control solution has a starting point. Given that the complex mechanical design has been completed, a basic control mechanism has been proposed, and a favorable boom geometry has been suggested, the next step is to run some basic practical experiments, ignoring telerobotic constraints such as impaired vision and time delay problems. Rather than using on–off control switches to manipulate individual actuator at a time as in really direct control, a basic boom control panel can utilize the inverse kinematic equations and provide a direct control mode. Direct control provides the user with easy to conceptualize directional control actuators. One possible computer driven solution, avoiding a complex joy stick interface is as follows:

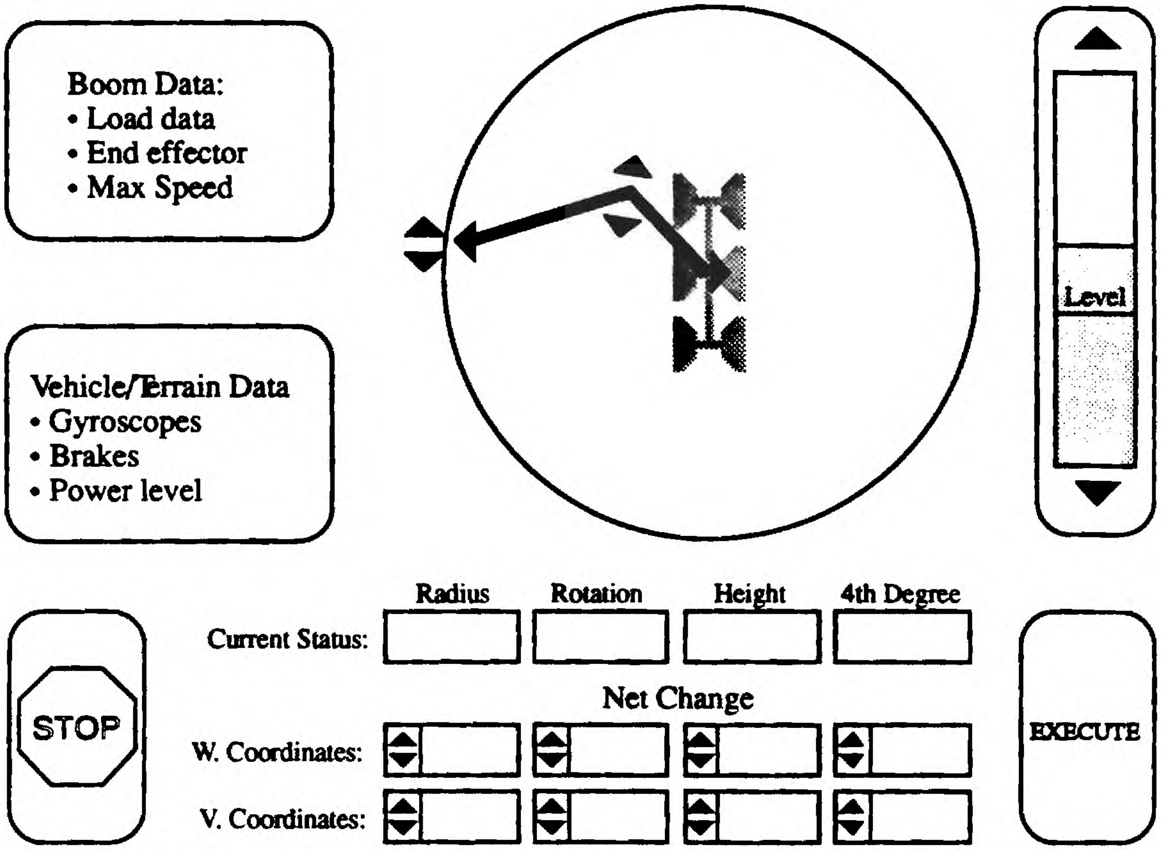


Figure 5–1: Experimental Development Control Module

The boom can be controlled by manipulating individual control directions such as rotation, radius, height, and directionality, or through relative directional inputs by means of vehicle or world coordinates. Relative directional inputs allow simultaneous movements in multiple directional coordinates. This is the predecessor method for continuous path planning capabilities.

The biggest advantage of a screen based user–interface is modularity and expandability. A joystick-type of control however might be easier to operate since visual feedback from model is directly translated into joy stick hand movements. A keyboard or a touch–screen user–input is not as direct and requires more input time in addition to being limited to singular sequential commands. The disadvantages of the joy-stick control is that its movements have to be translated into articulation joint rotations via a processor. Without elimination of the processor, the joy–stick controller is an added complexity to the system which should be added at a later stage. The screen based user–interface is easily modified to accommodate changes and additions in the model such as end–effector tools and boom–chassis communications.

The coordinate system selected for the most basic user–interface is cylindrical. This choice was made because one of the key kinematic requirements is straight line movements for drilling operations. Since most drilling operations are in the vertical direction, a height coordinate in world coordinates makes this task very convenient to execute. A simple mode expansion could included to accept other system inputs such as spherical or rectangular coordinates.

Entering commands on the suggested initial user–interface can be done by three methods. These are: movement of the boom segments via the directional arrows on the graphical display, entering of the net changes in world coordinates, or entering of the net changes in the vehicle coordinates. A entry will not be performed until the "Execute" bottom has been activated. Likewise, a running command can be interrupted using the "Stop" bottom. Initially, the executed commands, will prompt a calculation of the net rotations for each articulation joint and result in simultaneous shortest path movement of all joints over a given amount of time. Future modes will allow straight line tracking, which required small incremental steps and the corresponding transformation calculations.

The Experimental Development Control Module is only one in many ways to approach a simple user–interface. Added modes rapidly expand its usefulness and customize it to specific system requirements. Since the boom cannot be viewed as an individual system but as a part of the entire lunar vehicle, many functional changes which have not yet been identified, will have to be incorporated. The suggested control module is only intended to be a starting basis for future advanced modular human–machine solutions.

CHAPTER VI

Conclusions

Several philosophies exist on how to tackle design problems. One popular theory is that good products are developed by cross–disciplinary product teams. These consist of people with various expertise in areas such as finance, marketing, science, engineering, customers, and others. While the ultimate design task is the responsibility of the engineers, product specifications and limitations are decided upon by the group. This is to ensure that as the product goes from conceptualization to a finished product it still meets all customer requirements.

There are two types of specifications, one is a requirement, the other a trade–off. In deciding which specifications are more important, one has to evaluate the long term effect of that decision. An easy decision now might cause unnecessary complexity later on. Foresight can simplify and harmonize overall system performance.

Many of the suggested design configuration in this report have similar kinematic performance measures, such as dexterity, flexibility, and work envelope continuity. Their difference is in how their geometry affects related areas such as human–machine and control issues. Geometry requirements were only driven by task goals. Human–machine and control issues merely stated trade–off goals. An example: "If there is no inverse kinematic equations to describe the boom then an elaborate simulation mechanism will be required which makes real–time control difficult".

The key to achieving above average characteristics in flexibility, controllability, and human–machine factors is the arrangement of the first three stovepipe articulation mechanisms in such a way to simulate a spherical joint with hemispherical motions, and all subsequent joints in revolute parallel orientation. This configuration has several advantages. Not limited to, but explicitly discussed in a four–degree–of–freedom configuration, these advantages are:

- Inverse kinematic equations for any four degree of freedom configuration
- Easy to conceptualize boom movements
- Efficient use of fourth degree–of–freedom
- Three out of four actuators with extremely low center of gravity
- Uniformly dense, continuous work–envelope
- System implementation capabilities ranging from simple to ultra complex
- Applicable to single, dual, center, or end mounted boom configurations

The prosaic mechanism with spherical characteristics is so versatile that the performance controlling links are easily changed (stretched or shortened) to optimize the task specific assignments. This feature will enhance and simplify experimental testing and can be used to evaluate the usefulness of the fourth degree–of–freedom and the benefit of further dexterity.

Future Research Projects

The completed research on boom specific geometries is a starting point for many future research projects. Human–machine control issues and numerical control strategies have only been briefly discussed to determine cross-disciplinary requirements and describe the overall system scope. For the most part, the boom has been treated as an independent system, separate from the chassis, the end–effector, or a second dual boom. Simultaneous operation and interference between anyone of those is an interesting control problem that might be addressed with a knowledge and rule–based information system design. Implementation of higher control modes and human–machine interface levels require a similar solution approach. Path–planning, repetitious tasks, vehicle stability, and outrigger mode are among many ideas that have received only brief mention that still require detailed implementation description.

CONTROL ARCHITECTURE

The function of the control system is to translate operator inputs into usable machine code. Human operators direct telerobotic operations via one of several higher level symbolic functions depending on the mode of operation which varies from direct and autonomous commands. The control network consists of actuators, sensors, processors, and data libraries. Depending on the complexity of the task and the mode of operation the required amount of data manipulation is immense. The control system has to be able to response immediately to sudden changes in the work environment. Given the time–delay in telerobotic operations, local processing power is required to perform real–time corrective actions. Local levels of control are responsible for monitoring feedback data relating to position control, speed control, torque control, and tactile control. This task can only be achieved with synchronized coordination of actuator and sensor mechanisms. The dexterity of the lunar work vehicle, with its abundant number of actuators and sensors in the chassis, wheels, boom, and end–effector, necessitate multiple processors linked with handshake to coordinate such activities.

Hybrid Control Network: If information about the environment and state of events were known with absolute certainty, then several modes of control would be redundant. Yet there is a high degree of uncertainty, and furthermore the system is designed to be flexible and respond to human input. However, since the human operator is removed from the task and is separated by several seconds of time delay, the telerobot needs to have the autonomy to make decisions of its own. Local processing power is responsible to execute commands and checking the results by means of feedback mechanisms. The separation of tasks gives rise to a hybrid control problem [16][17].

Enumeration Levels: There are three levels of enumeration: local, global, and telebased enumerations. In local enumeration, local processing power is used to numerically control individual joints and actuators. The local enumeration level takes symbolic actuator commands and translates them into executable machine codes. Feedback from local sensors and global monitors is used to verify the correctness each execution step and adjust it if necessary.

For global enumeration, global processing power monitors the interaction of the overall internal system behavior with the surrounding environment. If undesirable differences in overall system behavior and expected results exist, global enumeration power is responsible for taking immediate corrective action and reversing undesirable trends. The global processor is also responsible for coordinating the activities of local controllers.

For telebased enumeration, telebased processing power takes existing and collected data about the environment to anticipate predictable undesirable movements and avert imminent catastrophes. Telebased enumeration power also translates high level symbolic instructions into lower level symbolic instructions before transmission.

Neural Knowledge Data Base: The best overall system performance is achieved when all three types of processing power are coordinated to achieve a common goal. In the case of rotary actuators, local level numeric control monitors position, speed, and torque. A simple PID (Proportional Integral Derivative) controller would do a good job to prevent excessive overshoot and achieve rapid steady state. Unfortunately, perfect execution of individual actuators does not guarantee overall desirable result. Due to overall system orientation, dynamics, and deformation individual PID gain values need constant adjustment for optimal performance. The existing local processing power can easily harnessed into a neural network with handshake capabilities. A global processor with access to a knowledge data base can monitor the overall system performance and react to undesirable system trends. Undesirable trends are characteristics that have previously been recorded and are stored in the knowledge data base. The function of the Neural Knowledge Data Base is to achieve better control characteristics based on previously learning experience. This capability enhances precision control as well as overall performance when uncertainty is high or the overall system responds nonlinear.

Artificial Intelligence: The level of robot autonomy stipulates the necessary amount of global processing power. Both reactive and inferred processing capabilities are necessary for strategic decision planning. The key to an intelligent dynamic control system are learning, adaptation, and anticipation levels. A neural knowledge data bases processes information for all three levels of control. Given a task the neural knowledge data base infers unknown facts from previous experiences and uses this information to develop a strategic plan. As the plan unfolds newly collected data is used to correct the plan and adapt to existing circumstances. This data in turn is used to update the knowledge based library and gives rise to the learning level. With near complete knowledge about a given state, the system can even anticipate future inferences. This capability is especially useful to limit self–interference and increase the average level of vehicle stability. The knowledge database consists of both numeric and rule–based information.

EXPERIMENTAL CONTROL ARCHITECTURE

An elaborate Hybrid Control Network capable of all modes of control is still many years into development. The experimental human machine development module offers user friendly direct control capabilities. Translation of direct commands into actuator specific instructions is made easy via the inverse kinematic equations. This makes near real–time control possible and is the ideal stepping stone to building more complex rule and knowledge–based control strategies.

Sensor Requirements: Sensors collect three types of information. One is about the state of the environment which includes video images, proximity data, and gyroscopic data. The second type of information is about the internal state of the vehicle like position, speed and torque information for each rotary actuator, etc... The third type of information is on interaction between the vehicle and its environment. This category includes strain gauges in the wheels to measure load distribution, and tactile feedback from the end–effector sensor, etc…

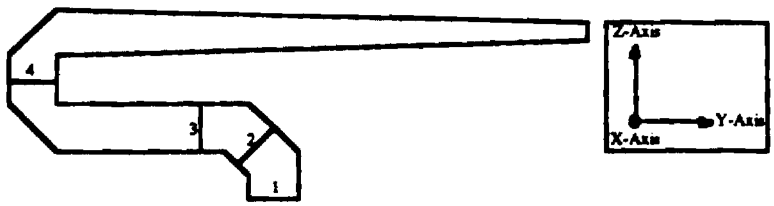
Unless the system utilizes a closed loop reactive control algorithm none of the sensor information is required. Optical encoders for position and speed control, and gyroscopic sensors for stability analysis might be among the first type of sensor that are necessary for a more thorough system analysis.

Smart Card Network: A smart card is capable of local or global processing power. When coupled to an individual actuator the smart card generates actuator specific signals to achieve low level symbolic functions. The experimental development module does not have any smart cards for global data coordination since the boom is treated as an isolated system with very limited data feedback. All of the direct mode symbolic instructions entered by the human operator are translated into actuator specific low level symbolic commands at the host computer level. Low level actuator specific information contains information about the actuator, its position, its speed, and its time of execution. From the host computer this information is passed on to local smart card for further processing. Motorola's HC11 family of chips provide sufficient processing power and multiple ports to makes them ideally suited for such local level processing power.

APPENDIXES A

Configuration Evaluation Results

BOOM CONFIGURATION #1,9, AND 10: ANALYSIS



1) $$\begin{bmatrix} \cos(q_1) & -\sin(q_1) & 0 & 0 \\ \sin(q_1) & \cos(q_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #1

2) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_1 \\ 0 & \sin(45) & \cos(45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #1
45° rotation along X-Axis

3) $$\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #2

4) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_2 \\ 0 & \sin(45) & \cos(45) & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #2
45° rotation along X-Axis

5) $$\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #3

6) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & ly_3 \\ 0 & -1 & 0 & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #3
(-90)° rotation along X-Axis

7) $$\begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #4

8) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #4

FORWARD KINEMATIC ANALYSIS RESULTS:

X= (C1*(C2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+N*S2*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+N*K*(N*C4*Y4+Z3)+Y2))+N*S1*(K*(S2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+C2*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+N*K*(N*C4*Y4+Z3)+Y2))+N*K*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+K*(N*C4*Y4+Z3)+Z2)+Y1))

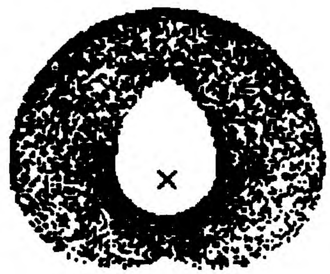
Y= (S1*(C2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+N*S2*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+N*K*(N*C4*Y4+Z3)+Y2))+C1*(K*(S2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+C2*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+N*K*(N*C4*Y4+Z3)+Y2))+N*K*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+K*(N*C4*Y4+Z3)+Z2)+Y1))

Z= (K*(S2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+C2*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+N*K*(N*C4*Y4+Z3)+Y2))+K*(K*(S3*N*S4*Y4+C3*(Z4+Y3))+K*(N*C4*Y4+Z3)+Z2)+Z1)

Notation:

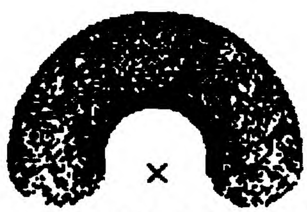
N = Constant = −1
Z = Length along z–axis
Y = Length along y–axis
K = Constant = $\frac{\sqrt{2}}{2}$
C = $\cos(\Theta_i)$
S = $\sin(\Theta_i)$

Configuration #1: Plot ZY



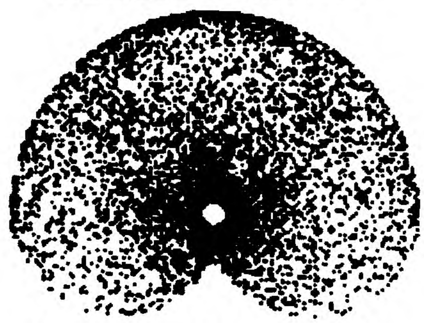
Maximum Reach: 166% of Vehicle Length
Minimum Reach: 36% of Vehicle Length

Configuration #9: Plot ZY



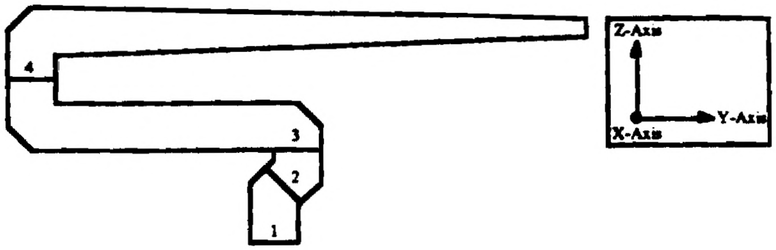
Maximum Reach: 158% of Vehicle Length
Minimum Reach: 50% of Vehicle Length

Configuration #10: Plot ZY



Maximum Reach: 212% of Vehicle Length
Minimum Reach: 9% of Vehicle Length

BOOM CONFIGURATION #2 ANALYSIS



1) $\begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #1

2) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(-45) & -\sin(-45) & ly_1 \\ 0 & \sin(-45) & \cos(-45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #
-45° rotation along X-Axis

3) $\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #2

4) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_2 \\ 0 & \sin(45) & \cos(45) & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #2
45° rotation along X-Axis

5) $\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #3

6) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_3 \\ 0 & 0 & 1 & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #3
0° rotation along X-Axis

7) $\begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #4

8) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #4
0° rotation along X-Axis

FORWARD KINEMATIC ANALYSIS RESULTS:

X= (C1*(C2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+N*S2*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))
+N*K*(Z4+Z3)+Y2))+N*S1*(K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(K*(S3*N*S4*Y4
+C3*(C4*Y4+Y3))+N*K*(Z4+Z3)+Y2))+K*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))+K*(Z4+Z3)+Z
2)+Y1))

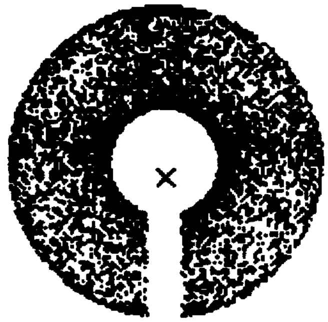
Y= (S1*(C2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+N*S2*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))
+N*K*(Z4+Z3)+Y2))+C1*(K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(K*(S3*N*S4*Y4+C
3*(C4*Y4+Y3))+N*K*(Z4+Z3)+Y2))+K*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))+K*(Z4+Z3)+Z2)+
Y1))

Z= (N*K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))
+N*K*(Z4+Z3)+Y2))+K*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))+K*(Z4+Z3)+Z2)+Z1)

Notation:

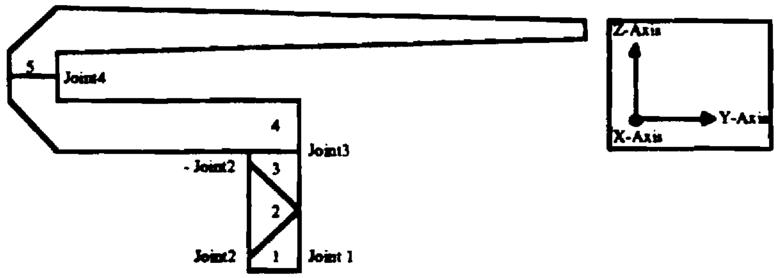
N = Constant = −1
Z = Length along z-axis
Y = Length along y-axis
K = Constant = $\frac{\sqrt{2}}{2}$
C = $\mathrm{Cos}(\Theta_i)$
S = $\mathrm{Sin}(\Theta_i)$

Configuration #2: Plot ZY



Maximum Reach: 167% of Vehicle Length
Minimum Reach: 39% of Vehicle Length

BOOM CONFIGURATION #3 ANALYSIS



1) $$\begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #1

2) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_1 \\ 0 & \sin(45) & \cos(45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #
45° rotation along X-Axis

3) $$\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #2

4) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & ly_2 \\ 0 & -1 & 0 & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #2
(-90)° rotation along X-Axis

5) $$\begin{bmatrix} \cos(-\theta_2) & -\sin(-\theta_2) & 0 & 0 \\ \sin(-\theta_2) & \cos(-\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #3

6) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_3 \\ 0 & \sin(45) & \cos(45) & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #3
45° rotation along X-Axis

7) $$\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #4

8) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ **Translation along section #4**
0° rotation along X-Axis

9) $\begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ **Rotation about articulation joint #5**

10) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_5 \\ 0 & 0 & 1 & lz_5 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ **Translation along section #5**

<u>FORWARD KINEMATIC ANALYSIS RESULTS:</u>

$$\begin{aligned} X= {} & C1*(C2*(C2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+S2*(K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+ \\ & N*K*(Z5+Z4)+Y3))+N*S2*((K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+K*(Z5+Z4)+Z3)+Y2))+ \\ & N*S1*(K*(S2*(C2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+S2*(K*(S4*N*S5*Y5+ \\ & C4*(C5*Y5+Y4))+N*K*(Z5+Z4)+Y3))+C2*((K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+ \\ & K*(Z5+Z4)+Z3)+Y2))+N*K*(N*(N*S2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+ \\ & C2*(K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+N*K*(Z5+Z4)+Y3))+Z2)+Y1) \end{aligned}$$

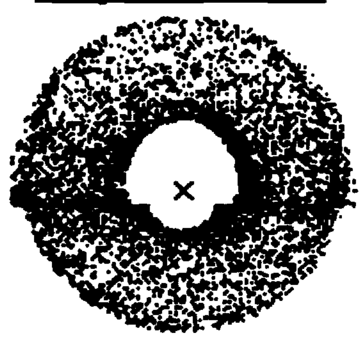
$$\begin{aligned} Y= {} & S1*(C2*(C2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+S2*(K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+ \\ & N*K*(Z5+Z4)+Y3))+N*S2*((K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+K*(Z5+Z4)+Z3)+Y2))+ \\ & C1*(K*(S2*(C2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+S2*(K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+ \\ & N*K*(Z5+Z4)+Y3))+C2*((K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+K*(Z5+Z4)+Z3)+Y2))+ \\ & N*K*(N*(N*S2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+C2*(K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+ \\ & N*K*(Z5+Z4)+Y3))+Z2)+Y1) \end{aligned}$$

$$\begin{aligned} Z= {} & K*(S2*(C2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+S2*(K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+ \\ & N*K*(Z5+Z4)+Y3))+C2*((K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+K*(Z5+Z4)+Z3)+Y2))+ \\ & K*(N*(N*S2*(C4*N*S5*Y5+N*S4*(C5*Y5+Y4))+C2*(K*(S4*N*S5*Y5+C4*(C5*Y5+Y4))+ \\ & N*K*(Z5+Z4)+Y3))+Z2)+Z1 \end{aligned}$$

<u>Notation:</u>

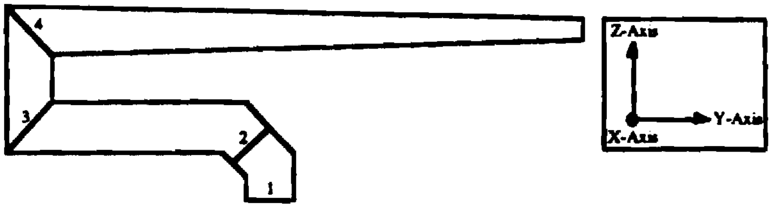
N = Constant = −1
Z = Length along z–axis
Y = Length along y–axis
K = Constant = $\frac{\sqrt{2}}{2}$
C = $\mathrm{Cos}(\Theta_i)$
S = $\mathrm{Sin}(\Theta_i)$

<u>Configuration #3: Plot ZY</u>



Maximum Reach: 40% of Vehicle Length
Minimum Reach: 164% of Vehicle Length

<u>BOOM CONFIGURATION #4 ANALYSIS</u>



1) $$\begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #1

2) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_1 \\ 0 & \sin(45) & \cos(45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #1
45° rotation along X-Axis

3) $$\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #2

4) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_2 \\ 0 & 0 & 1 & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #2
0° rotation along X-Axis

5) $$\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #3

6) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & ly_3 \\ 0 & -1 & 0 & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #3
(-90)° rotation along X-Axis

7) $$\begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #4

8) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #4

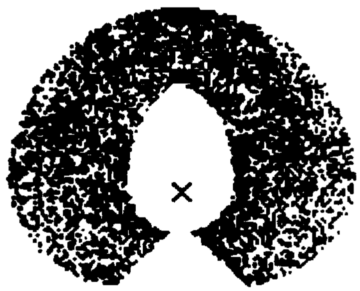
FORWARD KINEMATIC ANALYSIS RESULTS:

X= (C1*(C2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+N*S2*((S3*N*S4*Y4+C3*(Z4+Y3))+Y2))+
N*S1*(K*(S2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+C2*((S3*N*S4*Y4+C3*(Z4+Y3))+Y2))+N*K*((
N*C4*Y4+Z3)+Z2)+Y1))

Y= (S1*(C2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+N*S2*((S3*N*S4*Y4+C3*(Z4+Y3))+Y2))+
C1*(K*(S2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+C2*((S3*N*S4*Y4+C3*(Z4+Y3))+Y2))+N*K*((N*
C4*Y4+Z3)+Z2)+Y1))

Z= (K*(S2*(C3*N*S4*Y4+N*S3*(Z4+Y3))+C2*((S3*N*S4*Y4+C3*(Z4+Y3))+Y2))+
K*((N*C4*Y4+Z3)+Z2)+Z1)

Configuration #4: Plot ZY

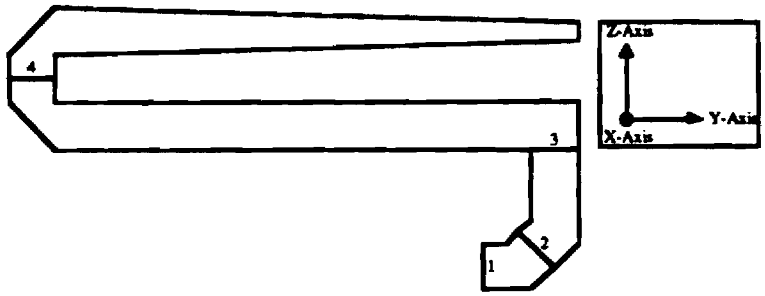


Maximum Reach: 179% of Vehicle Length
Minimum Reach: 41% of Vehicle Length

Notation:

N = Constant = –1
Z = Length along z–axis
Y = Length along y–axis
K = Constant = $\frac{\sqrt{2}}{2}$
C = $\mathrm{Cos}(\Theta_i)$
S = $\mathrm{Sin}(\Theta_i)$

BOOM CONFIGURATION #5 ANALYSIS



1) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

-90° rotation along X-Axis

2) $$\begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #1

3) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_1 \\ 0 & \sin(45) & \cos(45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #1
45° rotation along X-Axis

4) $$\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #2

5) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_2 \\ 0 & \sin(45) & \cos(45) & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #2
45° rotation along X-Axis

6) $$\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #3

7) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_3 \\ 0 & 0 & 1 & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section
0° rotation along X-Axis

$$8)\quad \begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Rotation about articulation joint #4

$$9)\quad \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #4

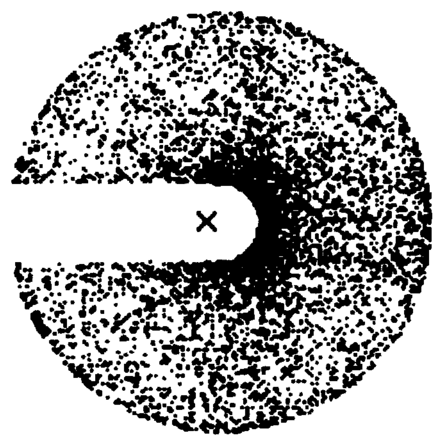
FORWARD KINEMATIC ANALYSIS RESULTS:

X= (C1*(C2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+N*S2*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))+ N*K*(Z4+Z3)+Y2))+N*S1*(K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(K*(S3*N*S4*Y4+ C3*(C4*Y4+Y3))+N*K*(Z4+Z3)+Y2))+N*K*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))+K*(Z4+Z3)+ Z2)+Y1))

Y= (K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))+ N*K*(Z4+Z3)+Y2))+K*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y3))+K*(Z4+Z3)+Z2)+Z1)

Z= N*(S1*(C2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+N*S2*(K*(S3*N*S4*Y4+ C3*(C4*Y4+Y3))+N*K*(Z4+Z3)+Y2))+C1*(K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(K *(S3*N*S4*Y4+C3*(C4*Y4+Y3))+N*K*(Z4+Z3)+Y2))+N*K*(K*(S3*N*S4*Y4+C3*(C4*Y4+Y 3))+K*(Z4+Z3)+Z2)+Y1))

Configuration #5: Plot ZY

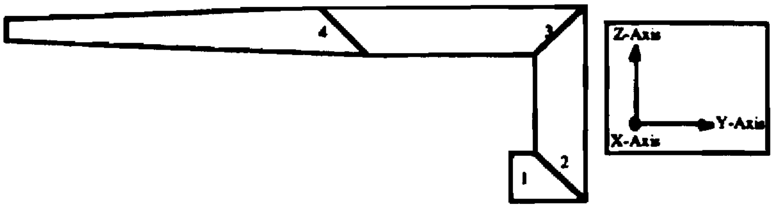


Maximum Reach: 217% of Vehicle Length
Minimum Reach: 40% of Vehicle Length

Notation:

N = Constant = −1
Z = Length along z-axis
Y = Length along y-axis
K = Constant = $\frac{\sqrt{2}}{2}$
C = $\mathrm{Cos}(\Theta_i)$
S = $\mathrm{Sin}(\Theta_i)$

<u>BOOM CONFIGURATION #6 ANALYSIS</u>



1) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ -90° rotation along X-Axis

2) $\begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #1

3) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_1 \\ 0 & \sin(45) & \cos(45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #1
45° rotation along X-Axis

4) $\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #2

5) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & -1 & ly_2 \\ 0 & 1 & 0 & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #2
90° rotation along X-Axis

6) $\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #3

7) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & -1 & ly_3 \\ 0 & 1 & 0 & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #3
90° rotation along X-Axis

8) $\begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #4

$$9)\quad \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #4

FORWARD KINEMATIC ANALYSIS RESULTS:

X= (C1*(C2*(C3*N*S4*Y4+N*S3*(N*Z4+Y3))+N*S2*(N*(C4*Y4+Z3)+Y2))+N*S1*(K*(S2*(C3*N*S4*Y4+N*S3*(N*Z4+Y3))+C2*(N*(C4*Y4+Z3)+Y2))+N*K*((S3*N*S4*Y4+C3*(N*Z4+Y3))+Z2)+Y1))

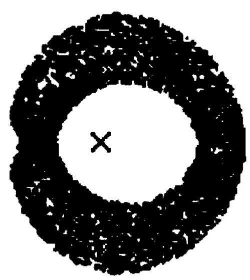
Y= (K*(S2*(C3*N*S4*Y4+N*S3*(N*Z4+Y3))+C2*(N*(C4*Y4+Z3)+Y2))+K*((S3*N*S4*Y4+C3*(N*Z4+Y3))+Z2)+Z1)

Z= N*(S1*(C2*(C3*N*S4*Y4+N*S3*(N*Z4+Y3))+N*S2*(N*(C4*Y4+Z3)+Y2))+C1*(K*(S2*(C3*N*S4*Y4+N*S3*(N*Z4+Y3))+C2*(N*(C4*Y4+Z3)+Y2))+N*K*((S3*N*S4*Y4+C3*(N*Z4+Y3))+Z2)+Y1))

Notation:

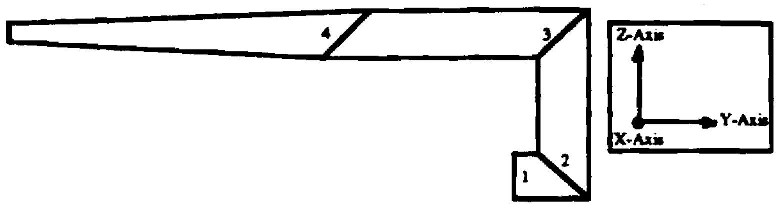
N = Constant = −1
Z = Length along z−axis
Y = Length along y−axis
$K = \text{Constant} = \frac{\sqrt{2}}{2}$
$C = \text{Cos}(\Theta_i)$
$S = \text{Sin}(\Theta_i)$

Configuration #6: Plot ZY



Maximum Reach: 138% of Vehicle Length
Minimum Reach: 42% of Vehicle Length

BOOM CONFIGURATION #7 ANALYSIS



1) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ -90° rotation along X-Axis

2) $\begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #1

3) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_1 \\ 0 & \sin(45) & \cos(45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #1
45° rotation along X-Axis

4) $\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #2

5) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & -1 & ly_2 \\ 0 & 1 & 0 & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #2
90° rotation along X-Axis

6) $\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #3

7) $\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_3 \\ 0 & 0 & 1 & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Translation along section #3
0° rotation along X-Axis

8) $\begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$ Rotation about articulation joint #4

$$9)\quad \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #4

FORWARD KINEMATIC ANALYSIS RESULTS:

X= (C1*(C2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+N*S2*(N*(Z4+Z3)+Y2))+
N*S1*(K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(N*(Z4+Z3)+Y2))+N*K*((S3*N*S4*Y4
+C3*(C4*Y4+Y3))+Z2)+Y1))

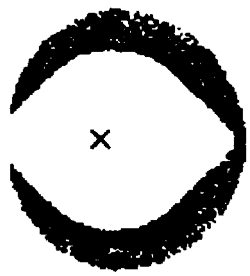
Y= (K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(N*(Z4+Z3)+Y2))+
K*((S3*N*S4*Y4+C3*(C4*Y4+Y3))+Z2)+Z1)

Z= N*(S1*(C2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+N*S2*(N*(Z4+Z3)+Y2))+
C1*(K*(S2*(C3*N*S4*Y4+N*S3*(C4*Y4+Y3))+C2*(N*(Z4+Z3)+Y2))+N*K*((S3*N*S4*Y4+C
3*(C4*Y4+Y3))+Z2)+Y1))

Notation:

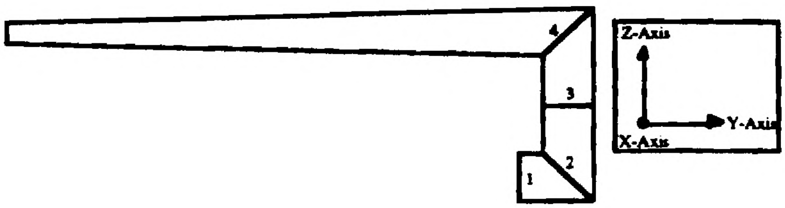
N = Constant = −1
Z = Length along z–axis
Y = Length along y–axis
K = Constant = $\frac{\sqrt{2}}{2}$
C = $\cos(\Theta_i)$
S = $\sin(\Theta_i)$

Configuration #7: Plot ZY



Maximum Reach: 138% of Vehicle Length
Minimum Reach: 79% of Vehicle Length

BOOM CONFIGURATION #8 ANALYSIS



1) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ -90° rotation along X-Axis

2) $$\begin{bmatrix} \cos(\theta_1) & -\sin(\theta_1) & 0 & 0 \\ \sin(\theta_1) & \cos(\theta_1) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #1

3) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_1 \\ 0 & \sin(45) & \cos(45) & lz_1 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #1
45° rotation along X-Axis

4) $$\begin{bmatrix} \cos(\theta_2) & -\sin(\theta_2) & 0 & 0 \\ \sin(\theta_2) & \cos(\theta_2) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #2

5) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_2 \\ 0 & \sin(45) & \cos(45) & lz_2 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #2
45° rotation along X-Axis

6) $$\begin{bmatrix} \cos(\theta_3) & -\sin(\theta_3) & 0 & 0 \\ \sin(\theta_3) & \cos(\theta_3) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #3

3) $$\begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & \cos(45) & -\sin(45) & ly_3 \\ 0 & \sin(45) & \cos(45) & lz_3 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Translation along section #3
45° rotation along X-Axis

8) $$\begin{bmatrix} \cos(\theta_4) & -\sin(\theta_4) & 0 & 0 \\ \sin(\theta_4) & \cos(\theta_4) & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$ Rotation about articulation joint #4

$$9)\quad \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & ly_4 \\ 0 & 0 & 1 & lz_4 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Translation along section #4

FORWARD KINEMATIC ANALYSIS RESULTS:

X= C1*(C2*(C3*N*S4*Y4+N*S3*(K*C4*Y4+N*K*Z4+Y3))+N*S2*(K*(S3*N*S4*Y4+
C3*(K*C4*Y4+N*K*Z4+Y3))+N*K*(K*C4*Y4+K*Z4+Z3)+Y2))+
N*S1*(K*(S2*(C3*N*S4*Y4+N*S3*(K*C4*Y4+N*K*Z4+Y3))+C2*(K*(S3*N*S4*Y4+
C3*(K*C4*Y4+N*K*Z4+Y3))+N*K*(K*C4*Y4+K*Z4+Z3)+Y2))+
N*K*(K*(S3*N*S4*Y4+C3*(K*C4*Y4+N*K*Z4+Y3))+
K*(K*C4*Y4+K*Z4+Z3)+Z2)+Y1)

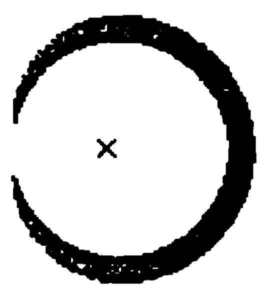
Y= (K*(S2*(C3*N*S4*Y4+N*S3*(K*C4*Y4+N*K*Z4+Y3))+C2*(K*(S3*N*S4*Y4+
C3*(K*C4*Y4+N*K*Z4+Y3))+N*K*(K*C4*Y4+K*Z4+Z3)+Y2))+K*(K*(S3*N*S4*Y4+
C3*(K*C4*Y4+N*K*Z4+Y3))+K*(K*C4*Y4+K*Z4+Z3)+Z2)+Z1)

Z= N*(S1*(C2*(C3*N*S4*Y4+N*S3*(K*C4*Y4+N*K*Z4+Y3))+N*S2*(K*(S3*N*S4*Y4+
C3*(K*C4*Y4+N*K*Z4+Y3))+N*K*(K*C4*Y4+K*Z4+Z3)+Y2))+
C1*(K*(S2*(C3*N*S4*Y4+N*S3*(K*C4*Y4+N*K*Z4+Y3))+C2*(K*(S3*N*S4*Y4+
C3*(K*C4*Y4+N*K*Z4+Y3))+N*K*(K*C4*Y4+K*Z4+Z3)+Y2))+
N*K*(K*(S3*N*S4*Y4+C3*(K*C4*Y4+N*K*Z4+Y3))+
K*(K*C4*Y4+K*Z4+Z3)+Z2)+Y1))

Notation:

N = Constant = -1
Z = Length along z-axis
Y = Length along y-axis
K = Constant = $\frac{\sqrt{2}}{2}$
$C = \mathrm{Cos}(\Theta_i)$
$S = \mathrm{Sin}(\Theta_i)$

Configuration #8: Plot ZY



Maximum Reach: 143% of Vehicle Length
Minimum Reach: 94% of Vehicle Length

APPENDIXES B

Symbolic Equation Generator

QUICK BASIC: BOOM FORMULA GENERATOR

```
'Matrix Multiplication Program
DIM A$(4,4):DIM B$(4,4):DIM C$(4,4)

'Initialize B$ Matrix
FOR C1=1 TO 4
FOR C2=1 TO 4
IF C1=C2 THEN
B$(C1,C2)="1"
ELSE
B$(C1,C2)="0"
END IF
NEXT C2
NEXT C1

1 'Read Multiplication Matrix
FOR C1=1 TO 4
FOR C2=1 TO 4
READ A$(C1,C2)
IF A$(C1,C2)="END" THEN
END
END IF
NEXT C2
NEXT C1

'Multiplication Procedure
FOR C1=1 TO 4
FOR C2=1 TO 4
SIGN=0
FOR C3 = 1 TO 4
IF A$(C1,C3)="0" OR A$(C1,C3)="" THEN
GOTO 2
ELSEIF B$(C3,C2)="0" OR B$(C3,C2)="" THEN
GOTO 2
ELSEIF A$(C1,C3)="1" THEN
IF SIGN >0 THEN
C$(C1,C2)=C$(C1,C2)+"+"+B$(C3,C2)
ELSE
C$(C1,C2)=B$(C3,C2)
END IF
ELSEIF B$(C3,C2)="1" THEN
IF SIGN >0 THEN
C$(C1,C2)=C$(C1,C2)+"+"+A$(C1,C3)
ELSE
C$(C1,C2)=A$(C1,C3)
END IF
ELSEIF SIGN>0 THEN
C$(C1,C2)=C$(C1,C2)+"+"+A$(C1,C3)+"*"+B$(C3,C2)
ELSE
C$(C1,C2)=A$(C1,C3)+"*"+B$(C3,C2)
END IF
SIGN=SIGN+1
```

```
' Denvenit Hartenberg Boom Formula Generator Continued

2 NEXT C3
IF SIGN>1 AND C$(C1,C2)<>B$(C1,C2) AND C4 >0 THEN
C$(C1,C2)="("+C$(C1,C2)+")"
END IF
NEXT C2
NEXT C1

'Output Screen
C4=C4+1
IF C4<8 GOTO 3
PRINT "Matrix Multiplication #";
PRINT C4
FOR C1=1 TO 3
FOR C2=4 TO 4
PRINT "Matrix Term (";
PRINT C1;C2;
PRINT " = ";
PRINT C$(C1,C2)
WRITE #1,C$(C1,C2)
NEXT C2
NEXT C1

3 'Next Matrix Multiplication
FOR C1=1 TO 4
FOR C2=1 TO 4
B$(C1,C2)=C$(C1,C2)
C$(C1,C2)=""
NEXT C2
NEXT C1
PRINT
GOTO 1

'Database Information
'SET 1
DATA "1","0","0","0"
DATA "0","1","0","Y4"
DATA "0","0","1","Z4"
DATA "0","0","0","1"

'SET 2
.
.
.

DATA "END"
```

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