Heavy Movable Structures, Inc.

SIXTH BIENNIAL SYMPOSIUM

October 30 - November 1, 1996

Doubletree Resort Surfside
Clearwater Beach, Florida

Hydraulic Systems for Movable Bridges - Seminar Lesson Plan

by

Jim Phillips, E.C. Driver & Associates, Inc.
Warren Sharp, Vanderbilt University
Charles Simons, Mannesmann Rexroth
Lou Wendel, Flender Corporation
David Fair, Rexroth Electronics
Parveen Gupta, Rexroth Corporation

Hydraulic Systems For Movable Bridges
Seminar Lesson Plan

8:00 am Introduction by Jim Phillips

About the Presenters

Warren Sharp	Mr. Sharp is currently the Corporate Training Director for Activation and concurrently a Ph.D. student and teacher at Vanderbilt's Management of Technology Program. Mr. Sharp earned a degree in Mechanical Engineering from Georgia Tech and has spent 27 years as a fluid power Sales Engineer.
Charles Simons	Mr. Simons is currently a Project Manger for the Mannesmann Rexroth Corporation in their Hydraudyne Cylinders Division. Mr. Simons earned a Bachelors of Science in Mechanical Engineering from HMS Utrecht, The Netherlands.
Jim Phillips, P.E.	Mr. Phillips is a Vice President with E.C. Driver and Associates, Inc., Tampa, Florida. He earned a Bachelors of Science in Civil Engineering from the University of Florida and is also a Certified Fluid Power Engineer.
Lou Wendel	Mr. Wendel has been employed in the hydraulics industry for 23 years since attended the Industrial Engineering School of Louisiana State University in New Orleans, Louisiana. His career has mostly been involved in low speed high torque hydraulic motor drive applications. Mr. Wendel is currently employed by the Flender Corporation.
David Fair, P.E.	Mr. Fair is a Manager with the Rexroth Electronics. He earned his B.S.E.E. from Lehigh University. Mr. Fair has over 25 years of experience in electrical engineering and been with the Rexroth Corporation since 1987.
Parveen Gupta	Mr. Gupta is Marketing Manager - Civil Engineering Department for the Rexroth Corporation. Since earning his B.S.M.E. and M.S.M.E. degrees from Lehigh University, he has been involved in fluid power systems design, sales, and operations. Mr. Gupta has presented technical papers at several seminars and symposiums including the HMS Symposiums in 1992 and 1994.

I. 8:05 am - 9:00 am
Fluid Power Fundamentals
Presented by: Warren Sharp

A. Basic Concepts

1. Force, Area & Pressure
2. Work, Power, Efficiency
3. HP = (GPM x PSI) / (1714 x eff)

B. Four Answers
1. Zero
2. Whatever it takes to do the job
3. Relief valve setting
4. I don't know

C. A Simple Circuit
1. Reservoir
2. Fixed Volume Pump
3. Relief Valve
4. Cylinder with a Load
5. Two Ball Valves

D. Pumps
1. Fixed Volume
2. Variable Volume
3. Pressure Compensated
4. Load Sense
5. Torque Limited
6. Closed Loop

E. Directional Controls
1. 2-Way
2. 3-Way
3. 4-Way
4. "It doesn't matter how much you draw them, just so you draw them right."

F. Pressure Controls
1. Relief
2. Counterbalance
3. Unloading
4. Sequence
5. Pressure Reducing

G. Flow Controls & Checks
1. Needle Valve

2. Flow Control
3. Pressure Compensated Flow Controls
4. Check Valves
5. P.O. Checks

H. Regeneration

FLUID POWER FUNDAMENTALS
&
BASIC COMPONENTS

Heavy Movable Structures, Inc.
Movable Bridge Symposium
Clearwater, Florida
November 1, 1996

Warren Sharp

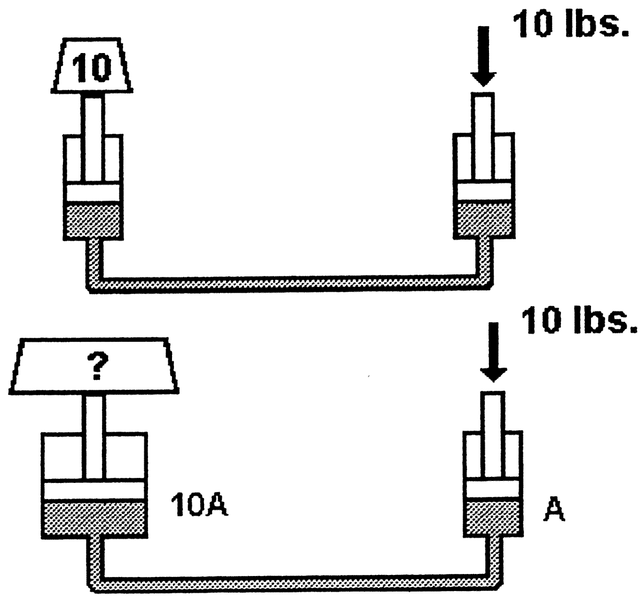
FLUID POWER?

Hydraulics & Pneumatics (oil & air)
Air is compressible.
Oil is "incompressible." (about 1/2% per 1000 psi)

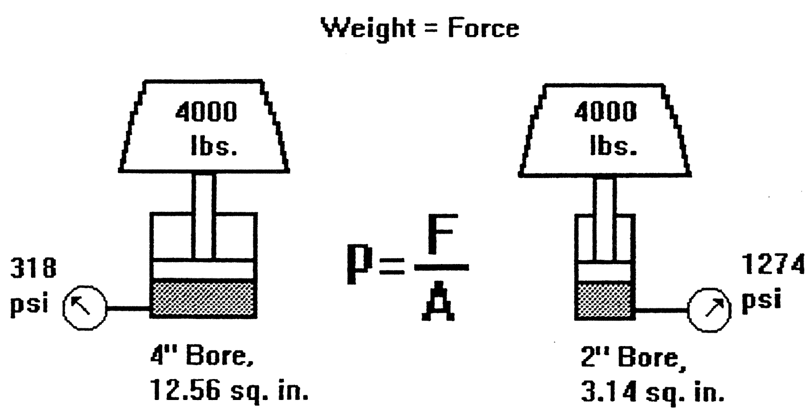
Three types of fluids for hydraulic systems:

1. Mineral
2. Synthetic
   a) Phosphate esters
   b) High Water Based Fluids (HWBF)
   c) Water/Oil emulsions
3. Vegetable

"Keep it "cool," keep it "clean."

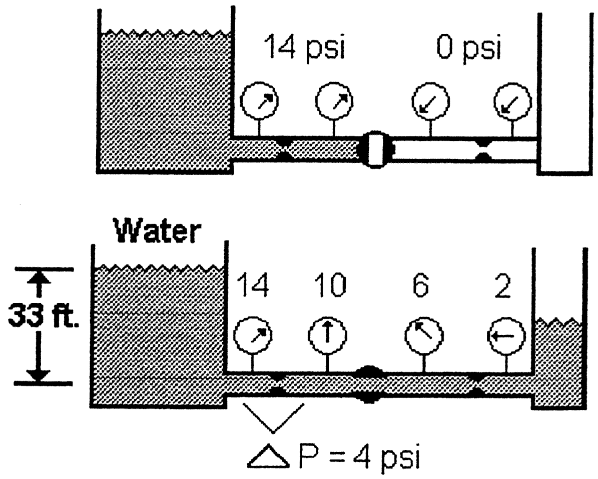


Force, Area, & Pressure



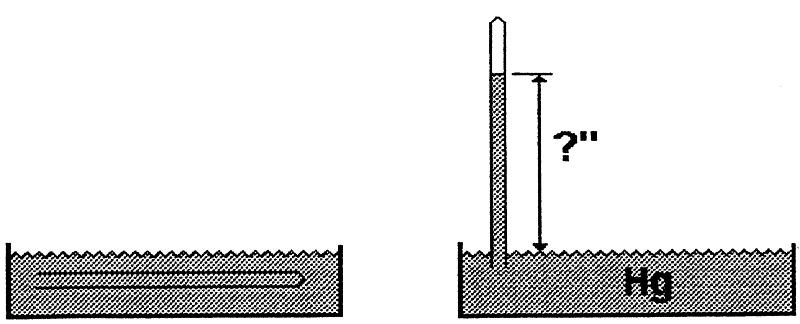
Force, Area, & Pressure

Why does fluid flow?



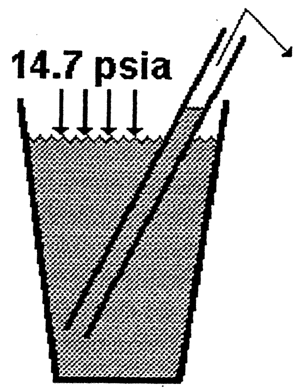
The Atmosphere

Does air weigh anything? If so, how much?



Torricelli's Experiment
(1608-'47)

A column of Hg, 1"x1"x30" weighs 14.7 pounds.
A column of water, 1"x1"x33' weighs 14.7 pounds.
A column of air, 1"x1" x ∞ weighs _____ pounds.

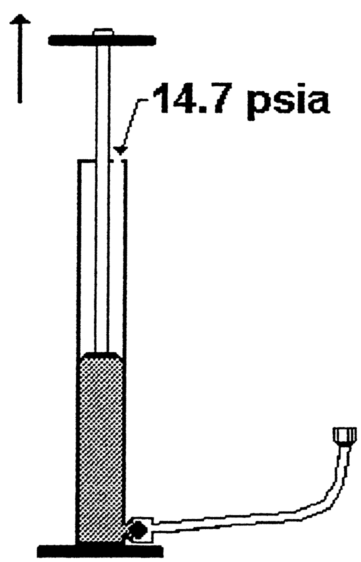


You can create a vacuum by removing the air.

Fluid is pushed up the straw (suction line) because of the pressure difference (Delta P).

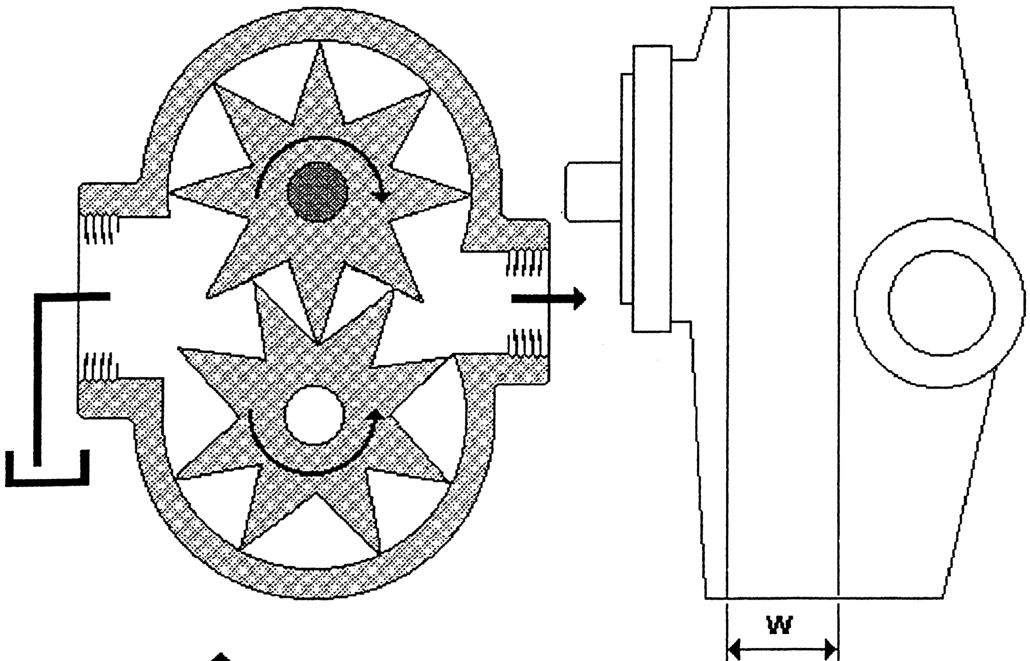
14.7 psi is all you have, therefore:

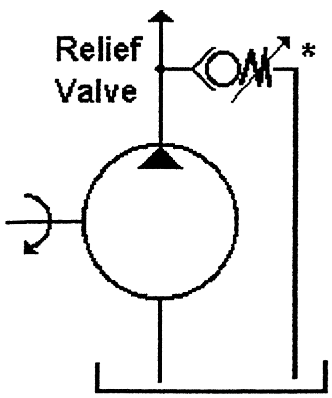
SUCTION LINES should be short, simple, and large.

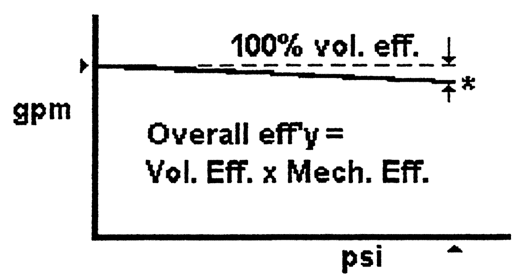


Pumping Principle:

You can create a vacuum by creating a volume







"Displacement" = in^3/rev

$$\text{GPM} = \frac{\text{cir x rpm}}{231\ in^3/gal}$$

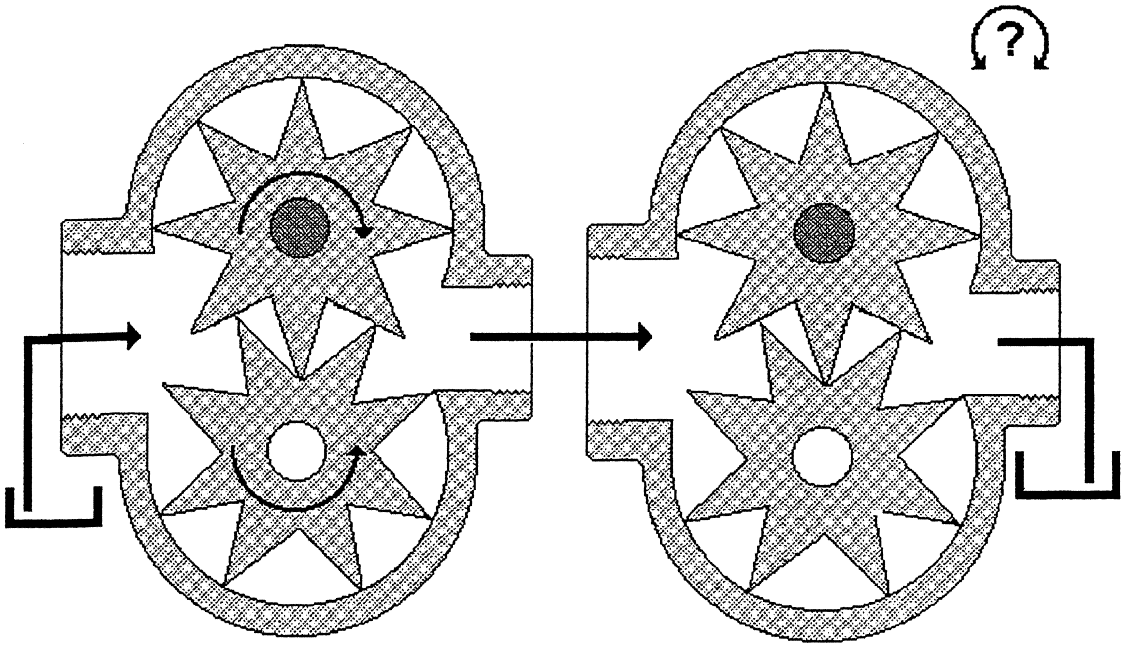
$$\text{HP} = \frac{\text{GPM x PSI}}{\text{1714 x eff.}}$$

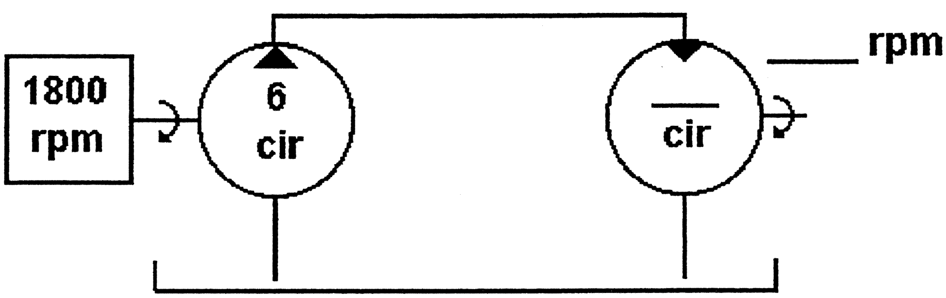
If eff. = 85%, then

$$\text{HP} = \frac{\text{GPM x PSI}}{1500}$$

GEAR PUMP

"Positive" or "Fixed" Displacement Pump

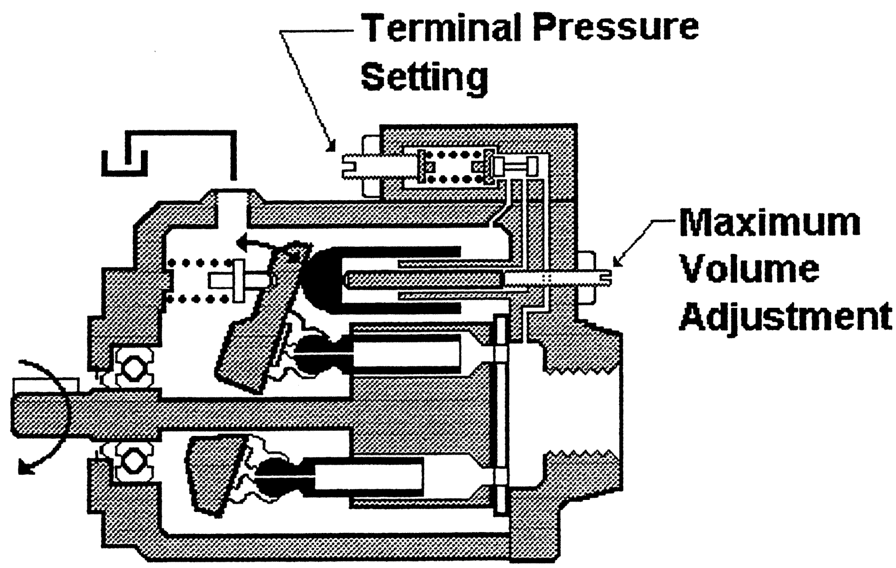


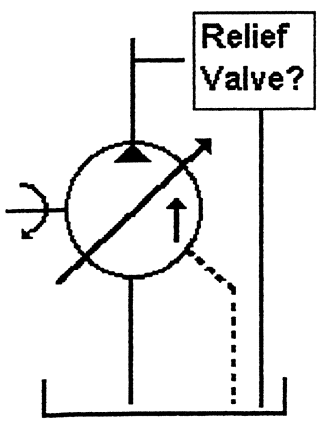


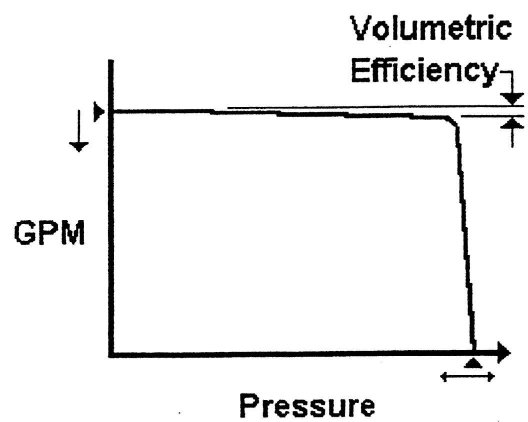
$$\text{RPM} = \frac{\text{gpm} \times 231\ \text{in}^3/\text{gal}}{\text{in}^3/\text{rev}}$$

$$\underset{\text{(in-lbs)}}{\text{Torque}} = \frac{\text{psi} \times \text{cir}}{2\pi}$$

HYDRAULIC MOTOR







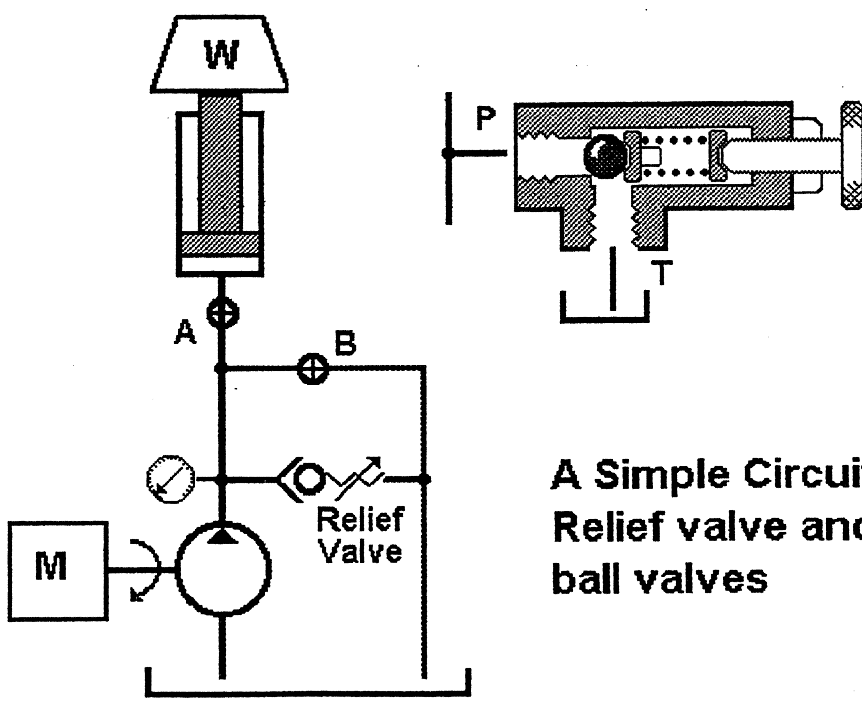
VARIABLE VOLUME PRESSURE COMPENSATED AXIAL PISTON PUMP

FOUR COMMON ANSWERS
(Sharp's Postulates

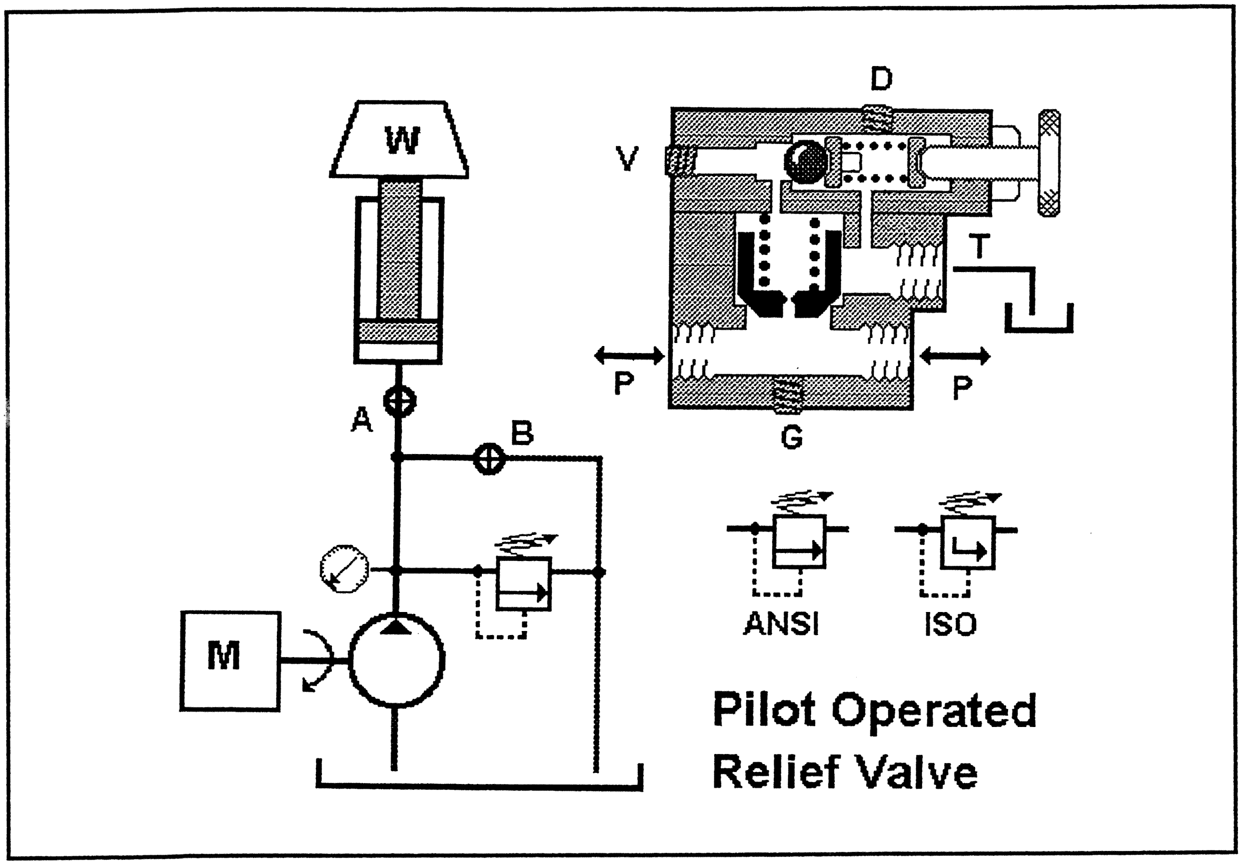
1. 0
2. Whatever It Takes To Do The Job (WIT2DJ)
3. Relief Valve Setting
4. I don't know.

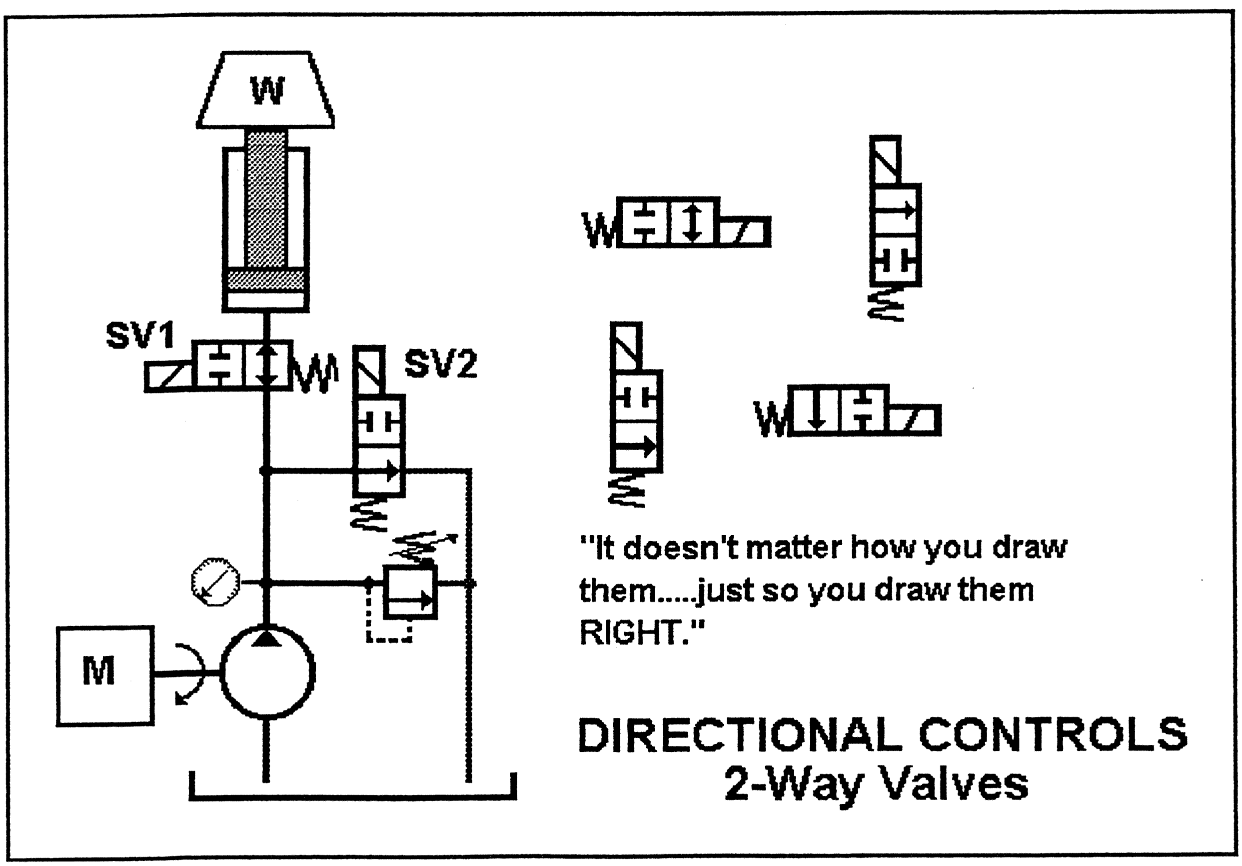
SHARP'S START-UP PROCEDURES

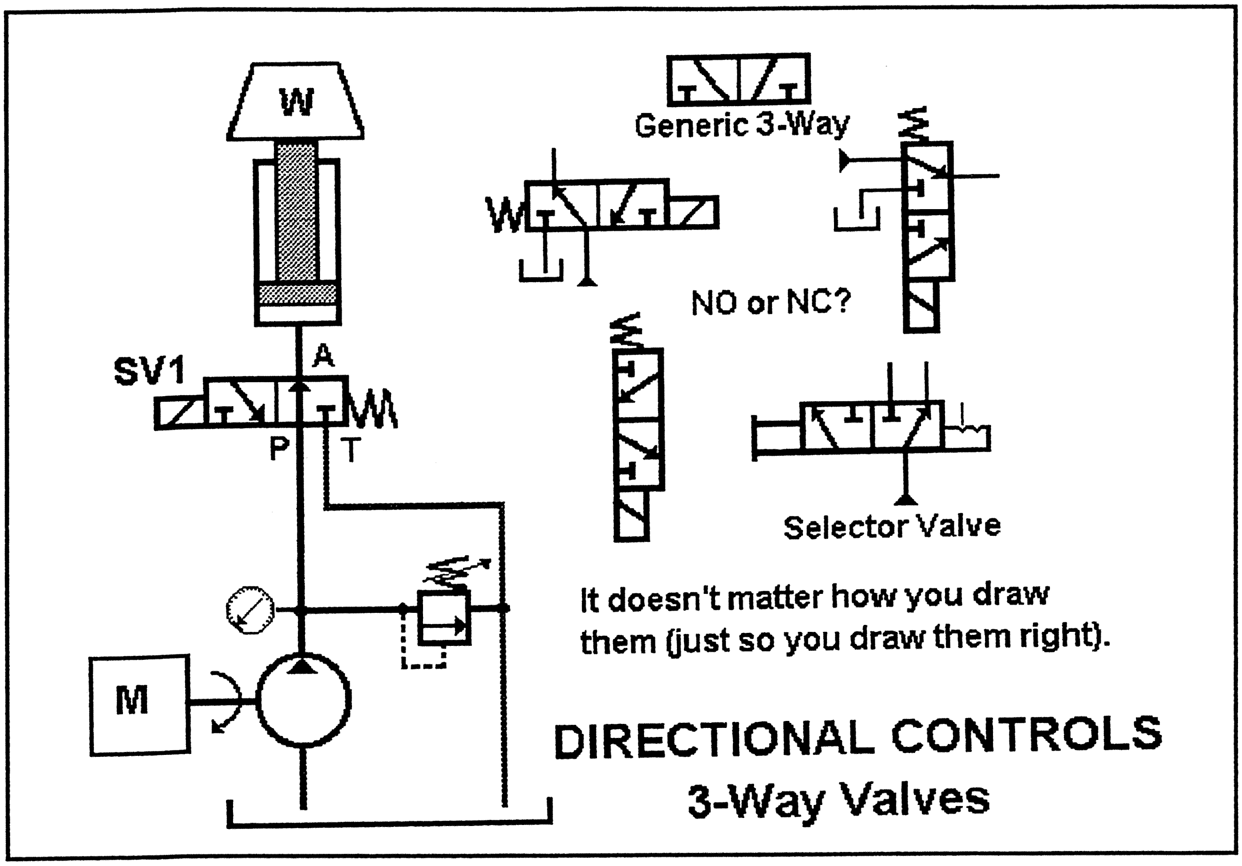
1. Never start up a power unit alone. Always start it up with a buddy, partner, or associate.
2. Be sure there is a free and clear passageway from the pump to the tank.

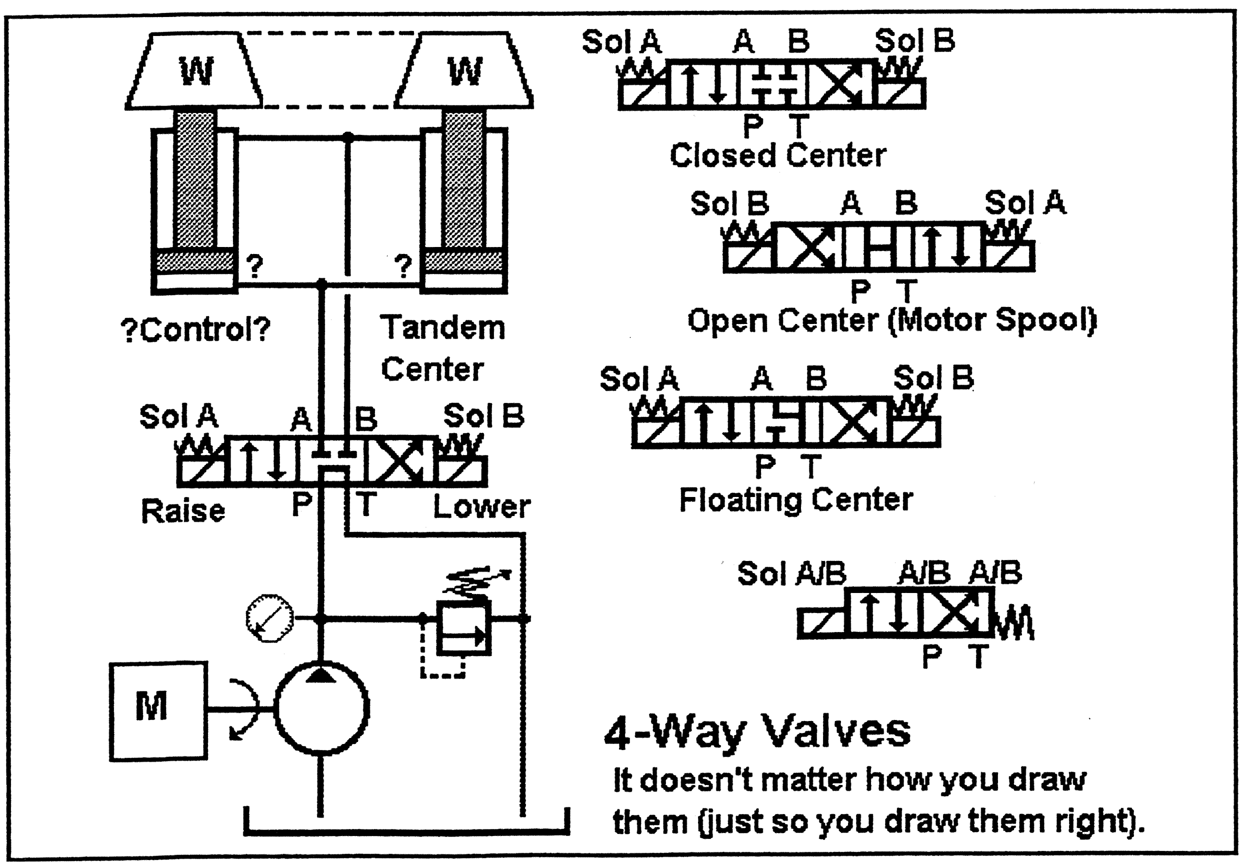


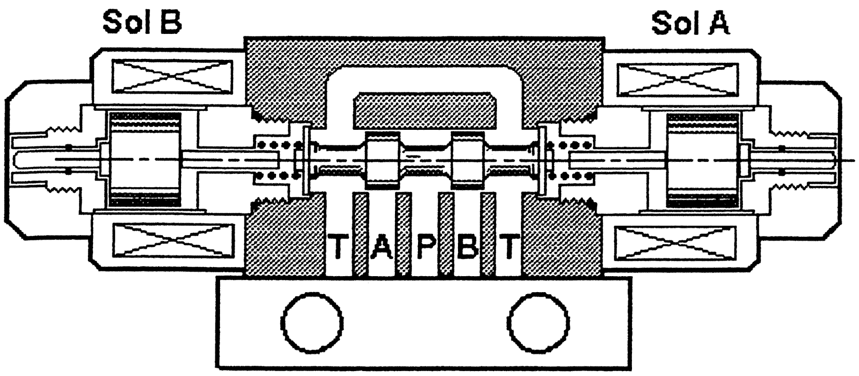
A Simple Circuit: Relief valve and ball valves

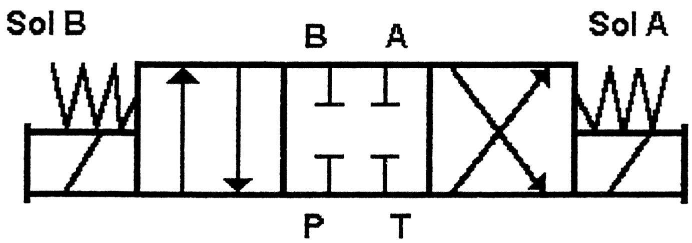

W
A
B
M
D
V
T
P
P
G
ANSI
ISO
Pilot Operated
Relief Valve


W
SV1
SV2
M
"It doesn't matter how you draw
them.....just so you draw them
RIGHT."
DIRECTIONAL CONTROLS
2-Way Valves

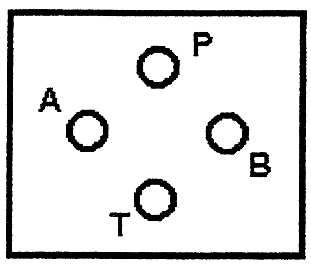

W
SV1
A
P
T
M
Generic 3-Way
NO or NC?
Selector Valve
It doesn't matter how you draw
them (just so you draw them right).
DIRECTIONAL CONTROLS
3-Way Valves


W
W
?
?
?Control?
Tandem
Center
Sol A
A
B
Sol B
Raise
P
T
Lower
M
Sol A A B Sol B
P T
Closed Center
Sol B A B Sol A
P T
Open Center (Motor Spool)
Sol A A B Sol B
P T
Floating Center
Sol A/B A/B A/B
P T
4-Way Valves
It doesn't matter how you draw
them (just so you draw them right).

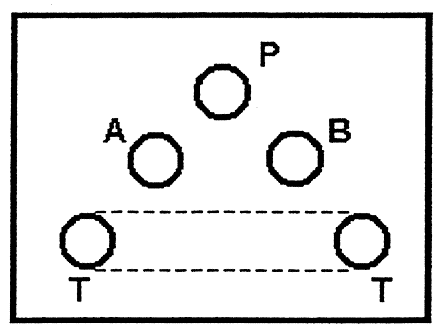




Mounting Patterns (Footprints)



1/4"
D03 (Formerly D01)

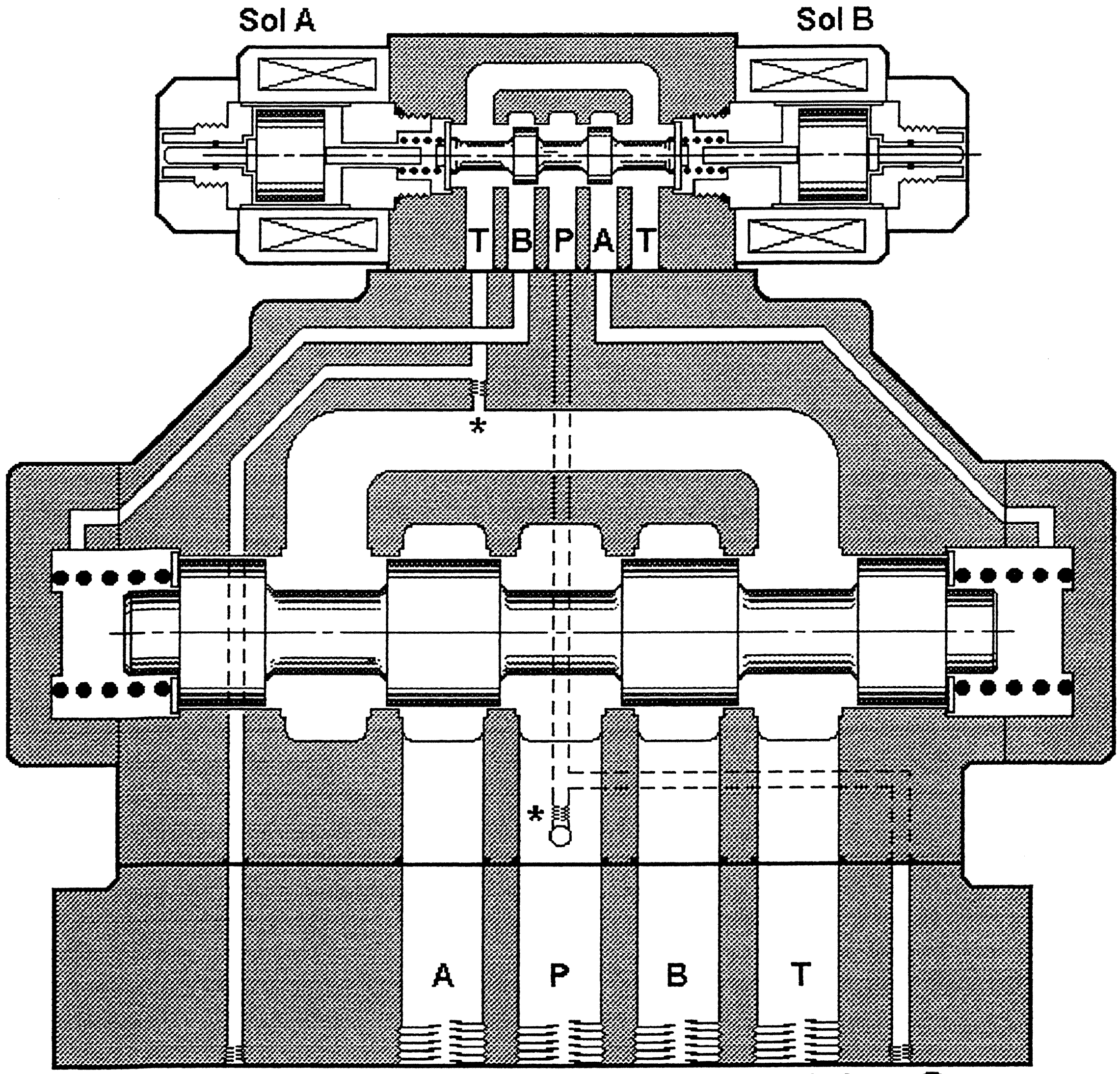


1/2"
D05 (Formerly D02)

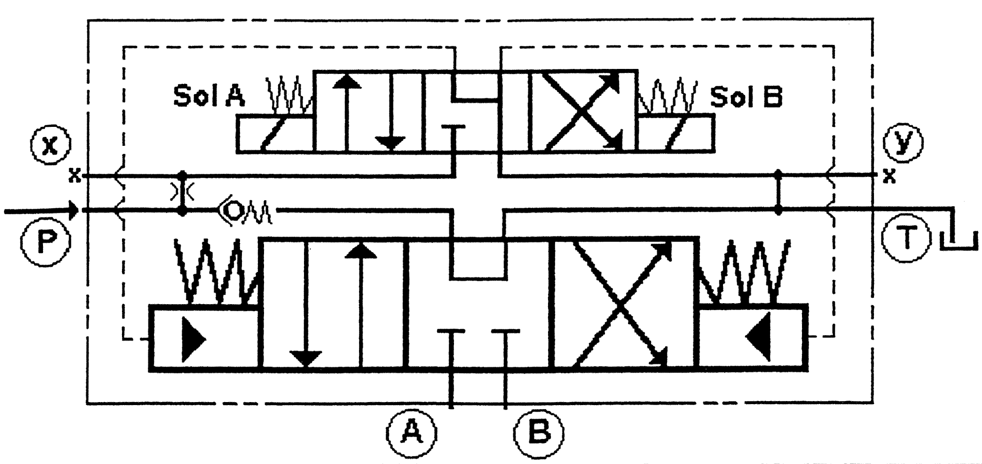
Definitions:

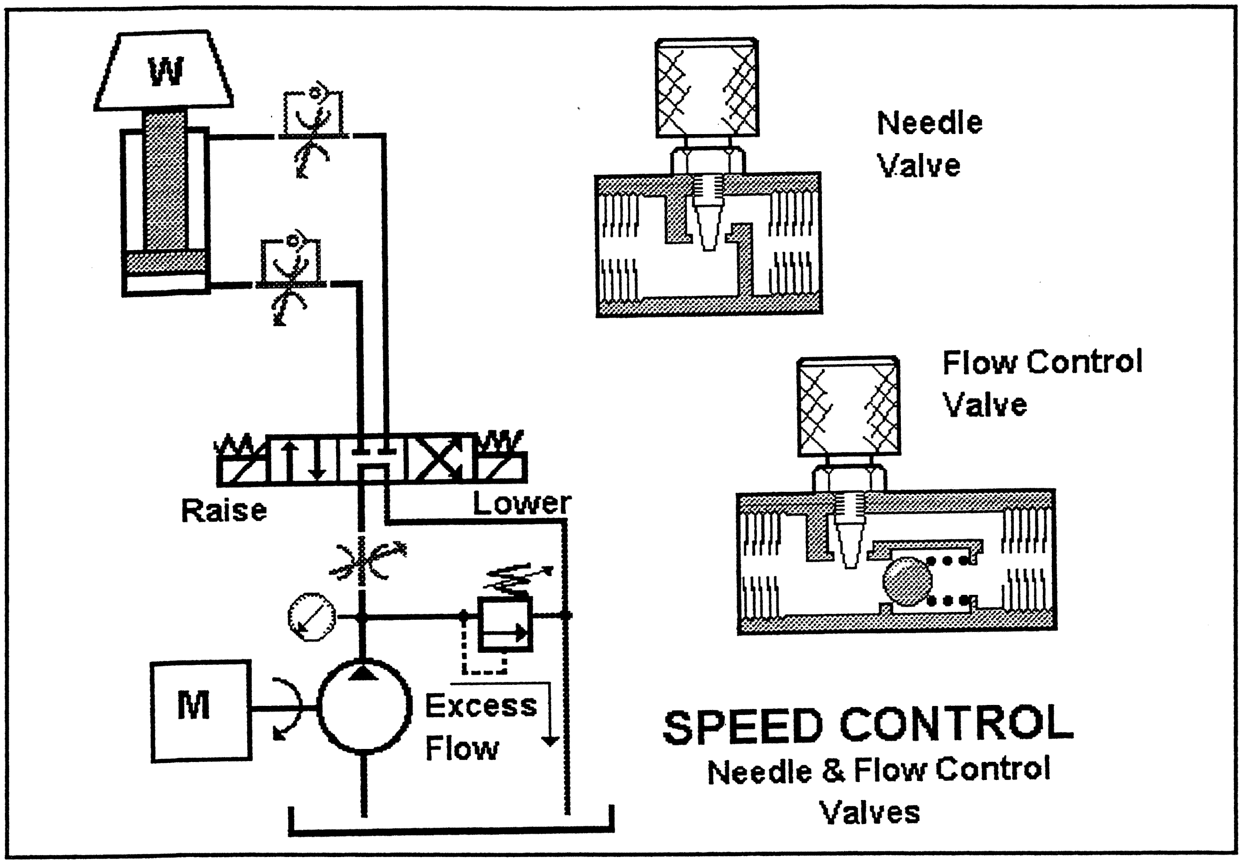
"Direct Acting" - Shifted directly by the solenoid, as opposed to being shifted by pilot pressure.

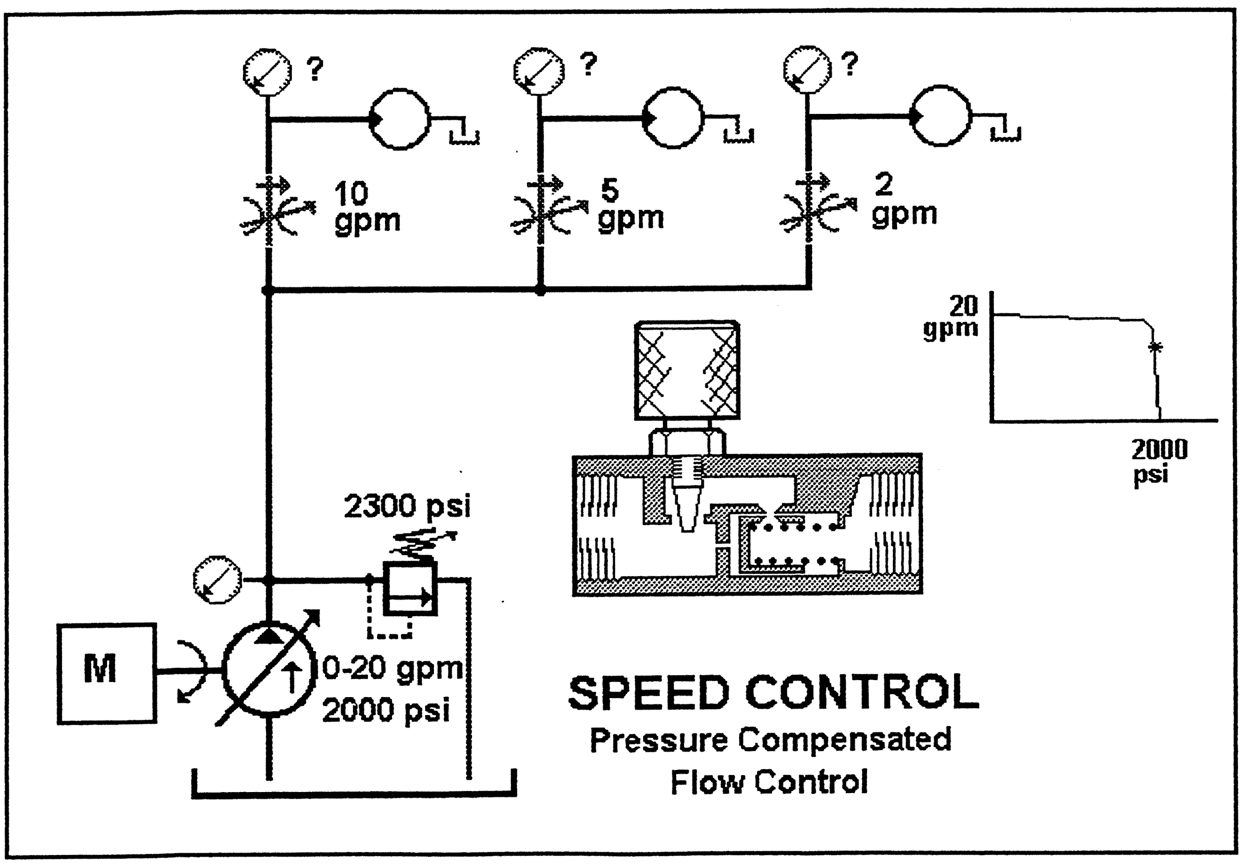
"Multi-purpose" - Any port can be pressurized and the valve will still function.

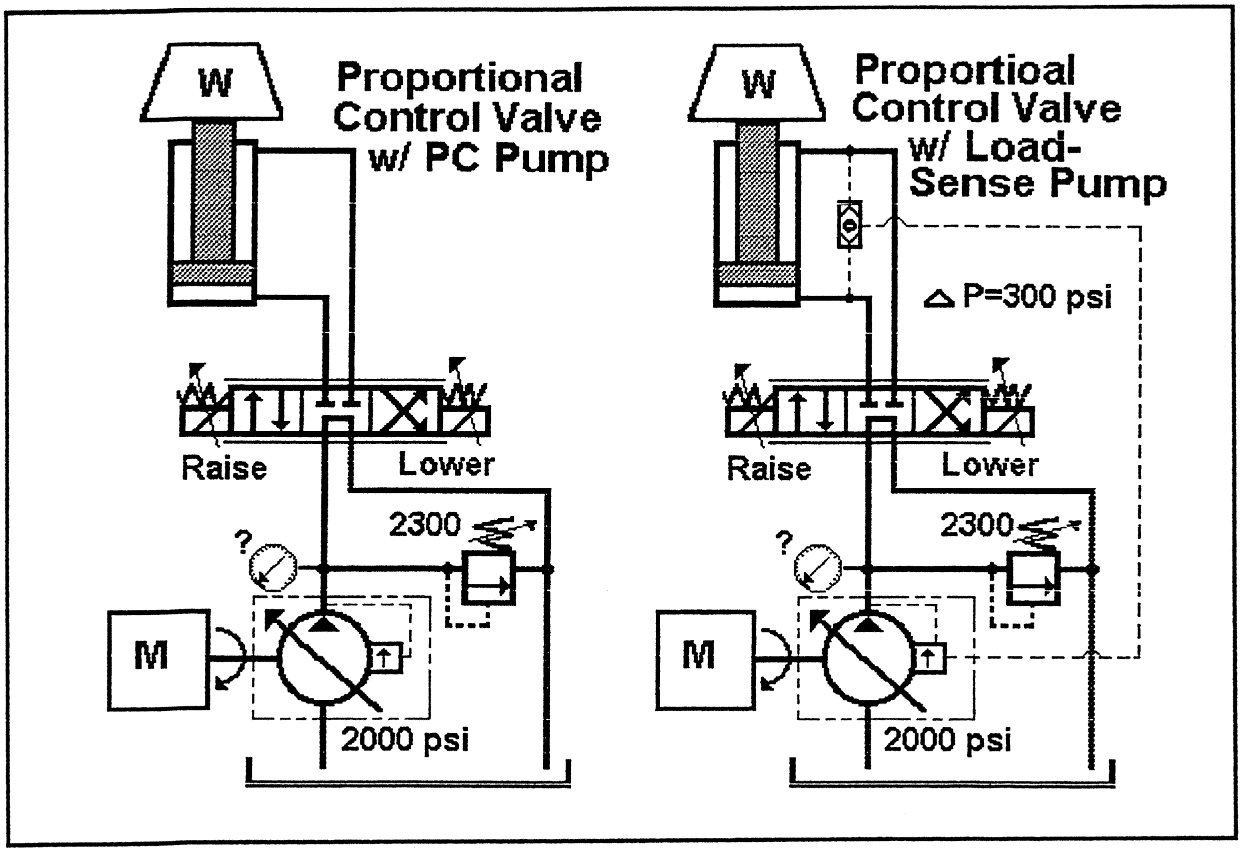


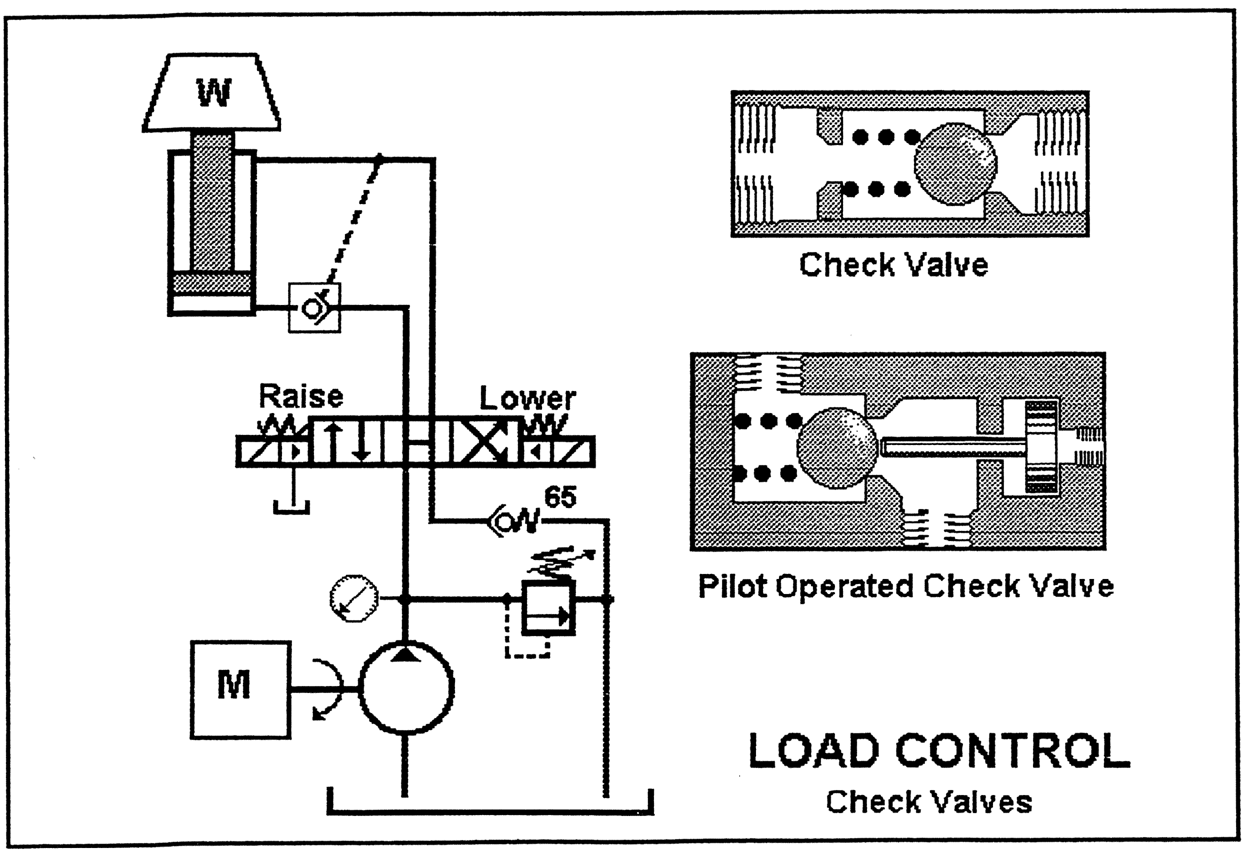
*Plugged for external pilot, external drain.
Open for internal pilot, internal drain.

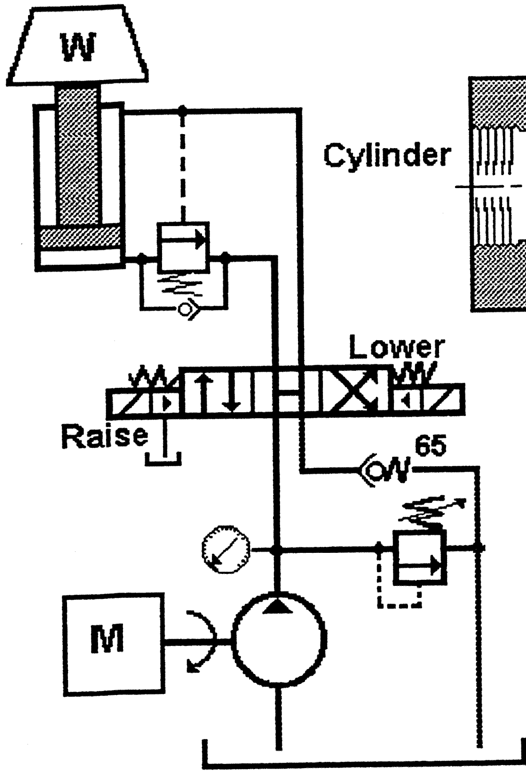


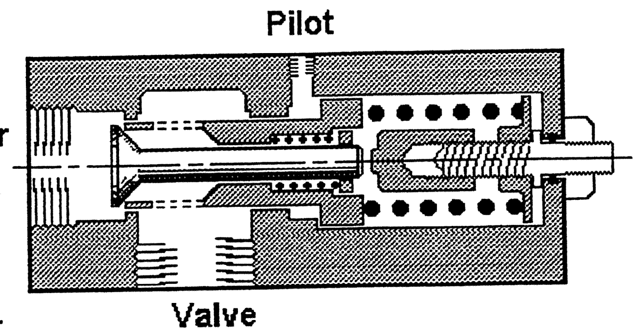

W
Needle
Valve
Flow Control
Valve
Raise
Lower
M
Excess
Flow
SPEED CONTROL
Needle & Flow Control
Valves


?
?
?
10
gpm
5
gpm
2
gpm
20
gpm
2000
psi
2300 psi
M
0-20 gpm
2000 psi
SPEED CONTROL
Pressure Compensated
Flow Control


W
Proportional
Control Valve
w/ PC Pump
Raise
Lower
2300
?
M
2000 psi
W
Proportioal
Control Valve
w/ Load-
Sense Pump
△ P=300 psi
Raise
Lower
2300
?
M
2000 psi

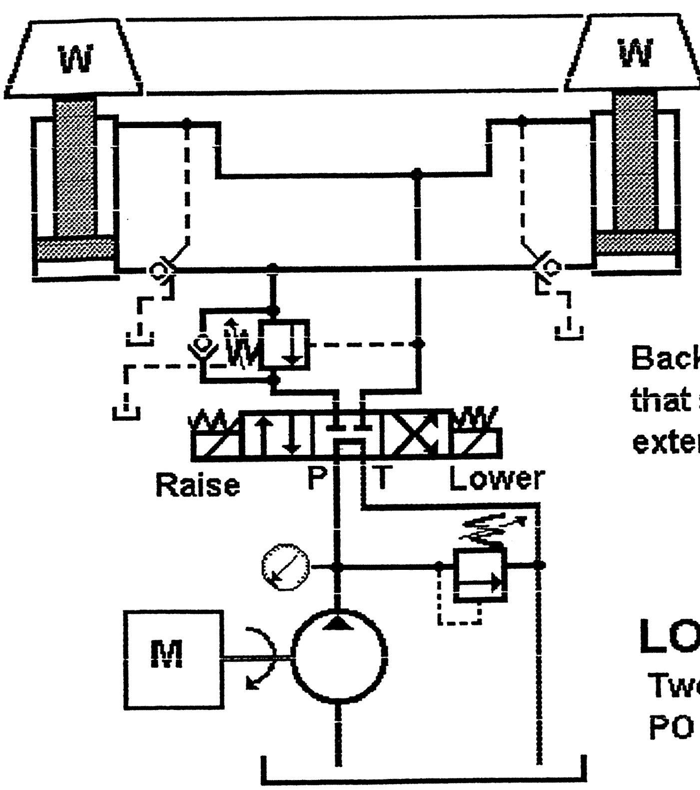

W
Raise
Lower
65
M
Check Valve
Pilot Operated Check Valve
LOAD CONTROL
Check Valves





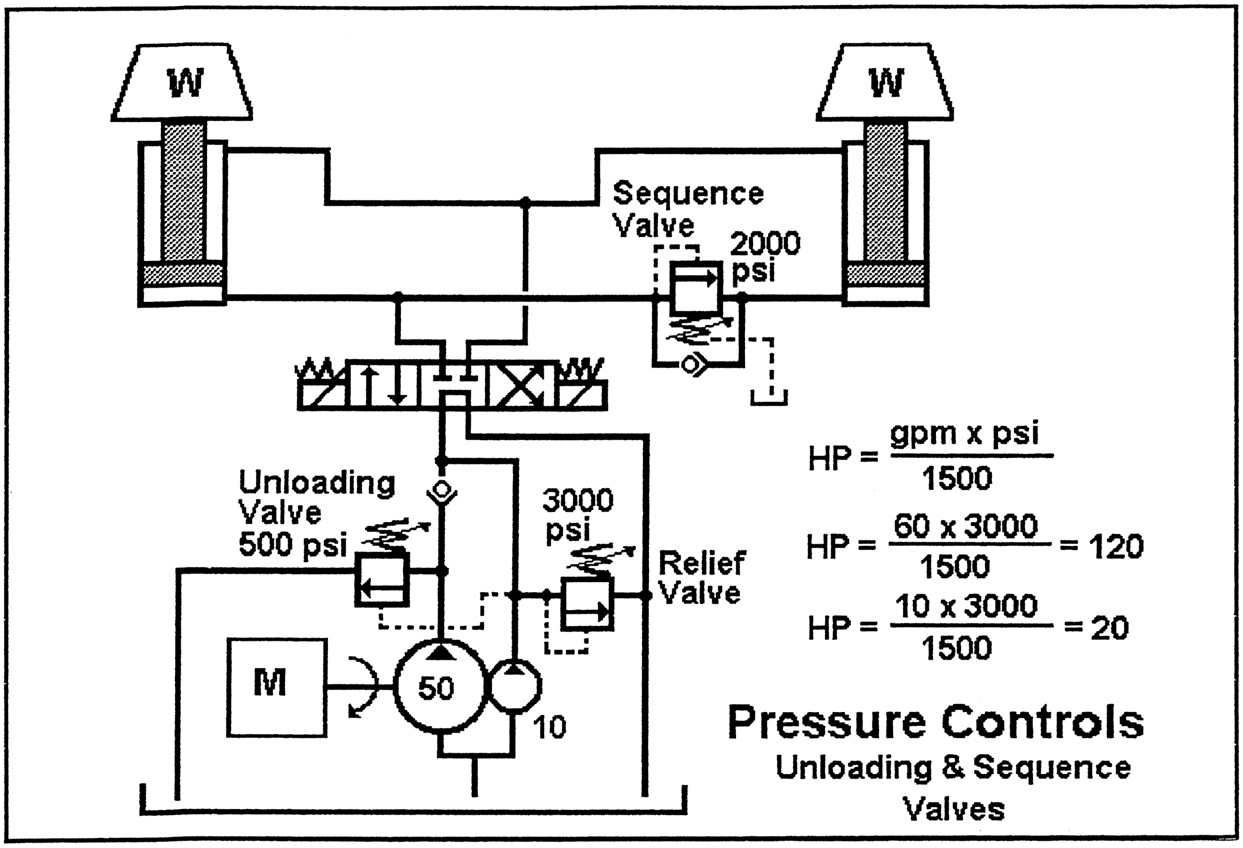
Typical pilot ratios: 3:1 & 4.5:1.
Pilot pressure required to lower load is a function of the load, cylinder bore, rod size, ratio, and set point. C-balance valves should be set at 1.3 x load induced pressure.

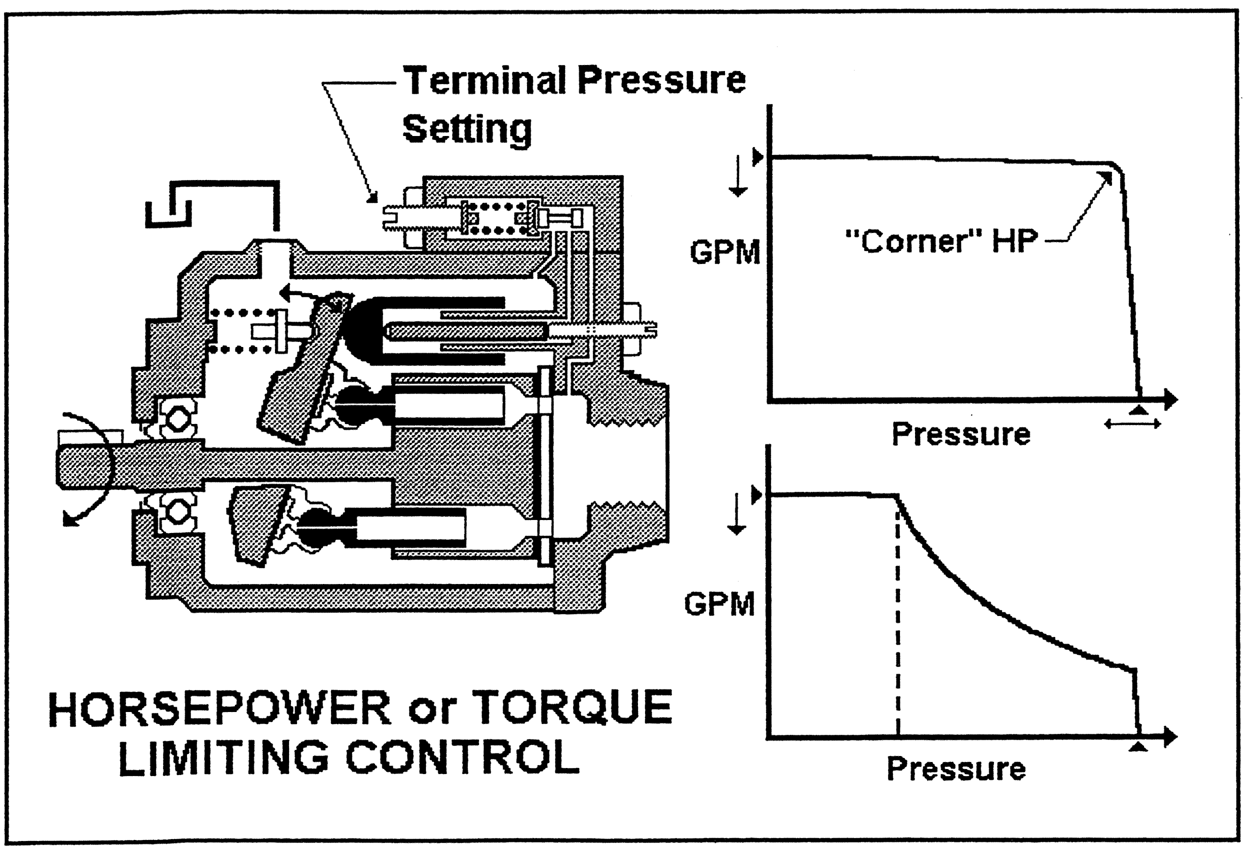
LOAD CONTROL
Counterbalance Valves

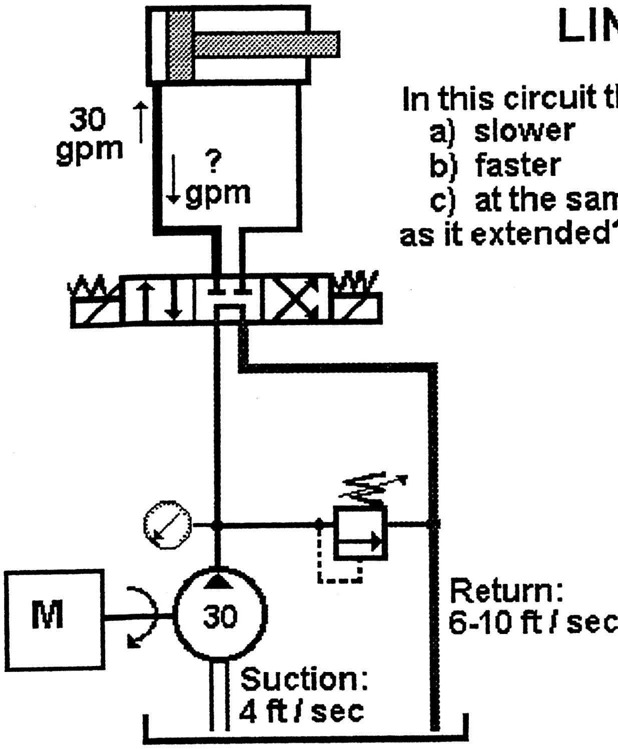


Back pressure may require that some components be externally drained.

LOAD CONTROL
Two Cylinders Using
PO Checks and C-Balance


W
W
Sequence
Valve
2000
psi
Unloading
Valve
500 psi
3000
psi
Relief
Valve
M
50
10
HP = gpm x psi / 1500
HP = 60 x 3000 / 1500 = 120
HP = 10 x 3000 / 1500 = 20
Pressure Controls
Unloading & Sequence
Valves


Terminal Pressure
Setting
GPM
"Corner" HP
Pressure
GPM
Pressure
HORSEPOWER or TORQUE
LIMITING CONTROL



LINE SIZES

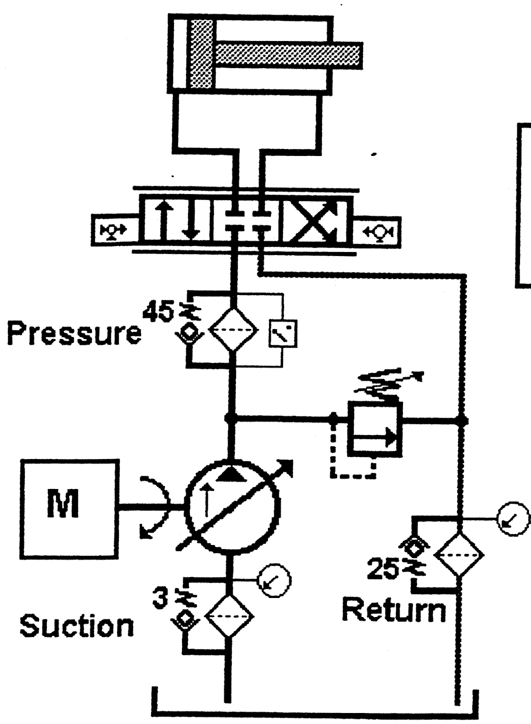
In this circuit the cylinder will retract
- a) slower
- b) faster
- c) at the same speed

as it extended?

Pressure Lines:

500 psi:	4 ft / sec
1000 psi:	6-8 ft /sec
2000 psi:	8-10 ft / sec
3000 psi:	10-15 ft / sec
4000 psi:	15+ ft /sec

$$\text{Inside Dia. in inches} = \sqrt{\frac{(.41)\,(\text{gpm})}{(\text{ft / sec})}}$$



FILTERS

Locations & Ratings

- 1 micron (µ) = .0000001 meter = .000039"
- Limit of visibility = 40 µ
- Particles >10µ should be removed

$$\text{Beta Ratio } (\beta) = \frac{\text{\# of particles IN}}{\text{\# of particle OUT}}$$

β_{10} = 2 → 50% efficiency

β_{10} = 5 → 80% efficiency

β_{10} = 20 → 95% efficiency

β_{10} = 75 → 98.7% efficiency

The Beta Ratio is also the # of particles that must be put into a filter in order to get <u>one</u> out.

FORMULAS

$$P = \frac{F}{A} \qquad F = P \times A$$

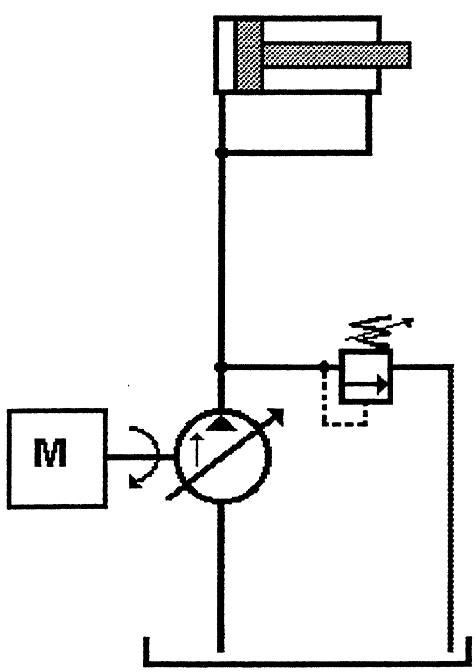
$$HP = \frac{GPM \times PSI}{1714 \times eff.} = \frac{GPM \times PSI}{1500} \quad \text{if efficiency} = .85$$

$$1\ HP = 2545\ BTU / Hr$$

$$HP = \frac{T \times RPM}{63{,}000} \quad \text{if Torque is in inch-lbs.}$$

$$T = \frac{PSI \times CIR}{2\pi}$$

$$GPM = \frac{CIR \times RPM}{231}$$



FINAL EXAM

What will happen when the pump is turned on?

II. 9:00 am - 9:30 am
Hydraulic Cylinder Design
Presented by: Charles Simons

A. Principles of Hydraulic Cylinders

1. Basics
2. Pressure and Force
3. Speed and Flow

B. Cylinder Construction

1. Cylinder Shell
   a. Strength Calculations
   b. Materials

2. Piston Rod
   a. Connections to Piston and Clevis
   b. Materials and Coatings

3. Piston
   a. Bearings, Composition and Materials
   b. Seals

4. Cylinder Head
   a. Bearings, Composition and Materials
   b. Seals
   c. Connection to the Shell

5. Oil Ports
   a. Sizing and Manifold Mounting

6. Mounting Styles
   a. Clevis Construction
   b. Spherical Bearings

7. Welding

C. Buckling and Deflection Calculations (Stability of the Cylinder)

D. Cushioning Construction and Calculations

Principles of Hydraulic Cylinders for Bridges

Charles A. Simons [1]

1. Introduction

What is a hydraulic cylinder?

The glossary for "Fluid Power", ANSI Standard B.92.21971, gives the following definition:

> "A hydraulic cylinder is a mechanism that converts hydraulic energy into a linear force and movement. In other words, it is a linear hydraulic motor. Usually this consists of a moving element such as piston with a piston rod or a plunger operating in a cylindrical bore."

As the hydraulic cylinders is a hydraulic motor, one may also speak of functions, types, models and specific qualities of the hydraulic cylinder. This motor also has a specific power, efficiency, medium, working temperature, adjustment and control.

Hydraulic cylinders are normally designed for working pressures of up to 350 - 400 bar and occasionally up to 600 bar and more for special purposes.

Apart from its function to a linear hydraulic motor, the cylinder is also a structural element. As structural element it must not only generate forces but also absorb forces.

The hydraulic cylinder, as a linear motor and structural element, is greatly influences by environmental circumstances. Elements such as temperature, humidity and corrosive influences largely determine to what extent the cylinder comes up to one's expectations and are therefore decisive with respect to quality. Therefore, in the design and construction of a cylinder, the following will must be closely considered:

- function as a hydraulic motor
- function as structural element
- Influences from immediate surroundings

2. Cylinder Types

Basically there are two types.
Judged by the nature of the generated force the cylinders may be classified as follows:

- Single acting cylinders
- Double acting cylinders

Single acting cylinders

The pressure medium moves the cylinder into one direction, outgoing or ingoing, and can therefore exert a pushing force or a pulling force. The single acting cylinder has only one oil connection. The return movement is effected as a result of gravity (own weight possibly with

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[1] Regional Sales Manager, Mannesmann Rexroth Hydraudyne Cylinders, Boxtel, the Netherlands

ballast) or by means of a return spring.

Double acting cylinders
The double acting cylinder has two oil connection ports and the pressure medium moves the cylinder in both directions. The cylinder exerts both a pushing and a pulling force.

3. Efficiency of the Hydraulic Cylinder

The mechanical efficiency of the cylinder depends on several factors.
Influential factors at constant viscosity:

a. Influence of surface roughness of piston rod, cylinder walls and bearings
b. Type of sealing and sealing material
c. Amount of occurring lateral forces
d. Function (type) and regulation of the cylinder such a single acting, double acting or regenerative regulation
e. Working pressure
f. Bore and phi-ratio
g. Cylinder model (single rod / through rod)

With respect to the above, the following may be stated:

1. High working pressures lead to higher efficiency

$$\varphi = \frac{piston\ area}{annulus\ area}$$

2. Larger bores at a certain working pressure have a higher efficiency
3. The lower the surface roughness and lateral forces, the higher the efficiency
4. Regenerative regulation gives a lower efficiency with a pushing function
5. Cylinder with a through rod has a lower efficiency
6. High phi-ratio gives a lower efficiency with a pulling function.
7. Full rubber sealing gives a lower efficiency than a rubber fabric sealing. Teflon compound sealing gives a higher efficiency than rubber fabric. Moreover, the efficiency is highly determined, not only by the material chosen, but also by the form of the sealing.

4. Temperature

The normal working temperature for the cylinder is between -20°C and +70°C. With service temperatures below or above these values, proper materials for cylinder main components and sealing have to be selected carefully. With low temperatures the several cylinder components will have to be made of materials with increased impact values. Furthermore, the cylinder has to be fitted with ice-scrapers and possibly sealing that are made from a special compound.

5. Construction of Hydraulic Cylinders

The quality of a hydraulic cylinder is determined by the several detailed constructions of the main components that a cylinder consists of and of their assembly. Various methods of assembly are possible and they depend on the intended applications, operating conditions, installation, regulations and possibilities of maintenance.

Cylinder for heavy industry with high pressures and/or long strokes, cylinders for bridges, sluices and offshore are characterized by a bolted head and bottom. For maritime application, a smooth exterior without projections is often chosen. For mobile, agricultural and related industry applications, the cylinder must be light and compact.

One of the first steps in using cylinders is to determine the main dimensions based on loads, available pressure and built-in situation. For longer strokes an important factor to determine the rod diameter is buckling.

<u>Buckling</u>

Buckling is a stability problem. As long as the construction returns to its original position after the load is taken away, the situation is stable, buckling is not expected. Buckling is a phenomena that not gradually starts, it occurs suddenly. With the calculations for buckling the following elements are important:

- buckling length
- mounting type
- angle
- slenderness

For the buckling calculation the buckling length has to be combined with the mounting style. The mounting styles have a considerable impact on the stability of the total cylinder, see fig. 1.

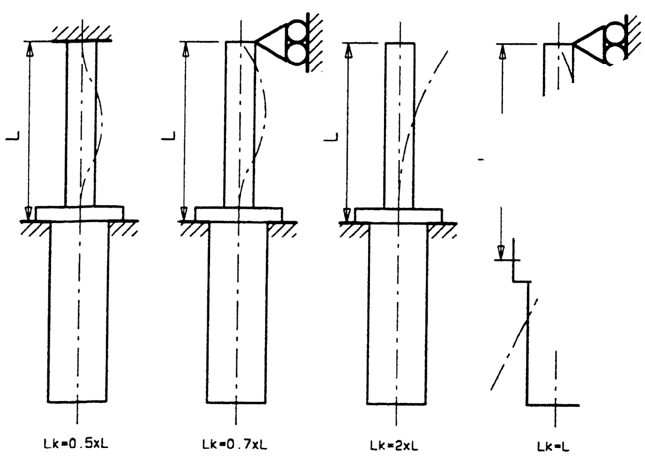


fig. 1, Buckling situations

The rod and the shell have different inertia moments. The total or combined inertia moment of the cylinder can be calculated and expressed in a reduction of the buckling length. This buckling length reduction is calculated according "Falck"

$$\textit{Buckling length reduction factor } (X) = \frac{1}{\dfrac{L_{total}}{L_{shell}} * \sqrt{\dfrac{I_{shell}}{I_{rod}}}} + \frac{1}{\dfrac{L_{total}}{L_{rod}}}$$

for vertical cylinders the maximum buckling load calculated according "Euler" is as follows:

$$FE = \frac{\pi^2 * Elasticity\ Modulus * \frac{\pi}{64} * D_{rod}^4}{buckling\ length * X}$$

Also, if the stroke length of a horizontally mounted cylinder becomes to long in relation to the diameter, the own weight of the cylinder will have a influence on the bending forces in the rod, the bearing load and the buckling stability. The deflection results into a bending moment in the rod, which reduces the buckling stability. Extensive computer calculations are necessary to calculate these second order effect influences.

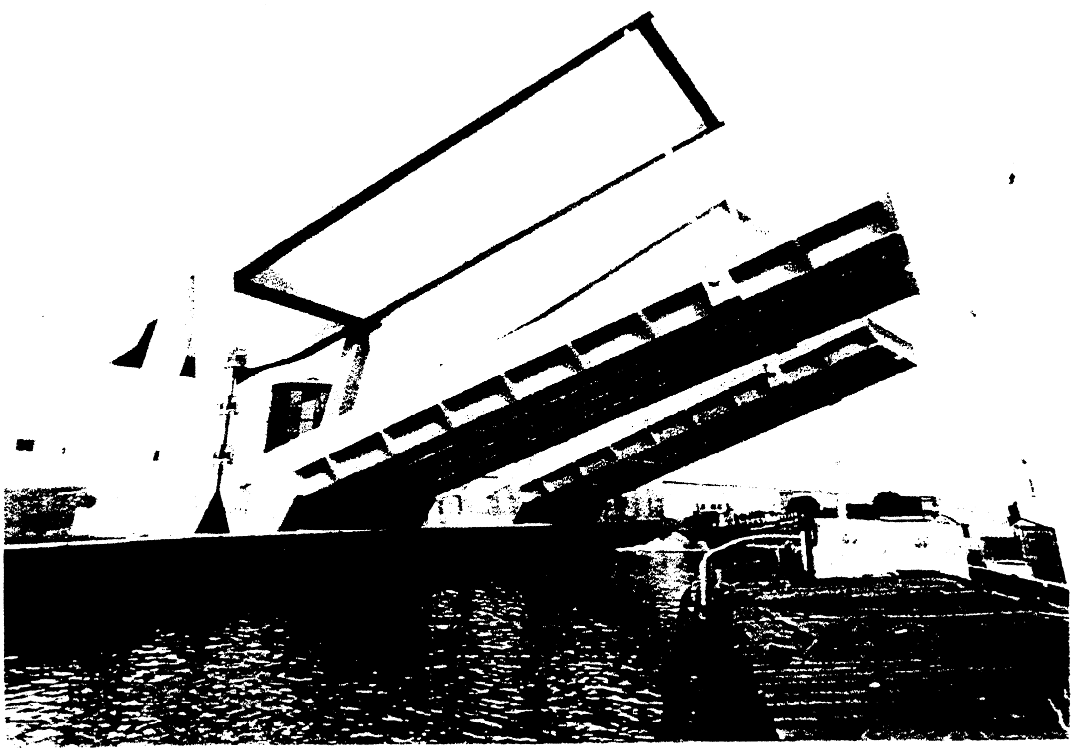
6. Application Based Standardization

Based on the extensive experience in the field of hydraulic cylinders for Civil applications the ABS (application based Standardization) cylinder concept is developed. ABS means translating the field experience into a standard design to create a fit for use long lasting and cost effective product, based on an active co-operation with manufacturers, consultants and end-users. The specific requirements in the bridge application field has resulted in design features as is shown in the following pages.

Boxtel, August 1, 1996

5. APPLICATION DEMANDS

Bridge cylinder



Bridge lay-out

Key principles
- Maximum availability
- Outdoor and wet environment
- Wind load and speed control
- Bridge vibrations
- According international standards

6. DESIGN CRITERIA
Bridge cylinder

Design bridge cylinder based upon customer and Hydraudyne experience

- Maximum availability
  - Failsafe design
  - Serviceability/worldwide service

- Outdoor and wet environment
  - Corrosion-resistant
  - Watertight
  - Piston rod coating corrosion and abrasion resistant
  - Design temperature -20°C, +70°C

- Wind load and speed control
  - Self adjusting cushioning both sides
  - Hydraulicaly pré stressed from both sides

- Bridge vibrations
  - Enlarged bearings

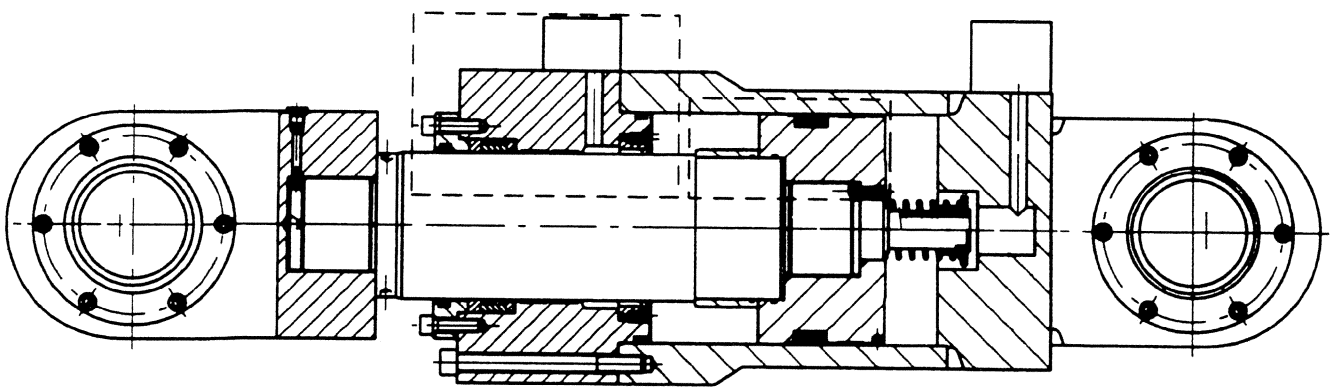
- According international Standards
  - Design according ASME, DIN or I.S.
  - Material certification

COST EFFECTIVE DESIGN

7. TECHNICAL ANSWERS

Bridge cylinder

General lay-out

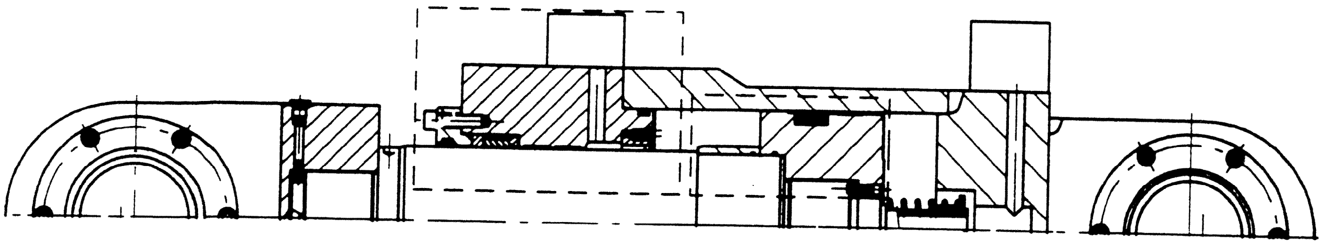


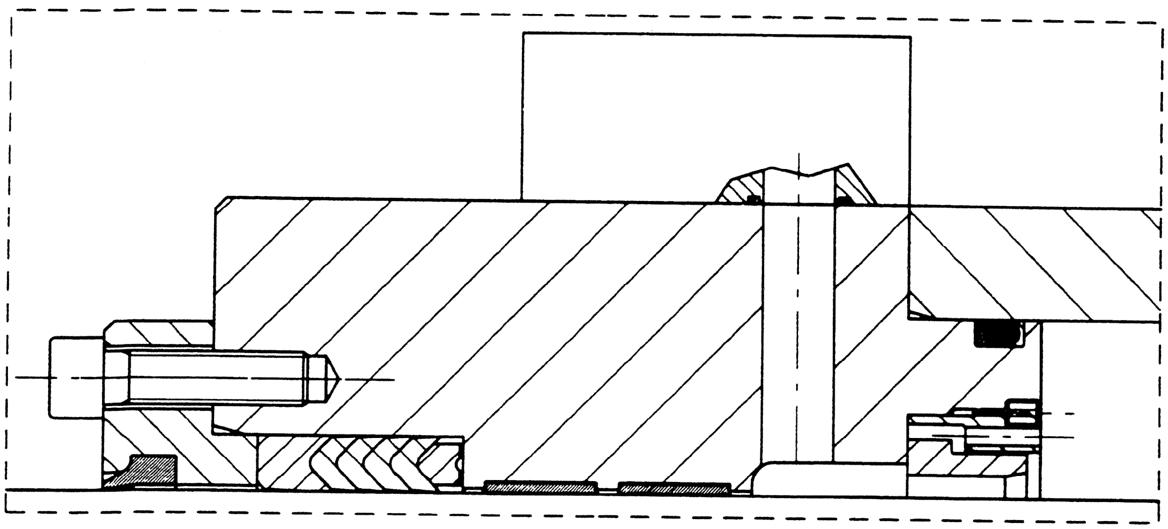
Features:

- Pistonrod material high strength carbon steel coated with corrosion and abrasion resistant ceramic coating
- Shell seamless tube with honed running surface
- Spherical bearing with dirt/water sealed construction
- Bolted connections of sealing flange and end cover for easy servicing
- Easy accessible failsafe multilip seal on pistonrod and heavy duty bi-directional seal on piston
- Manifold on both sides for prestressing against variable wind load

7a. TECHNICAL ANSWERS

Bridge cylinder



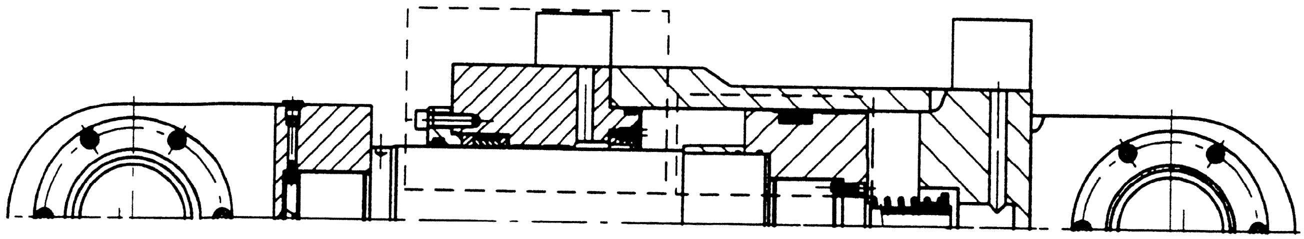


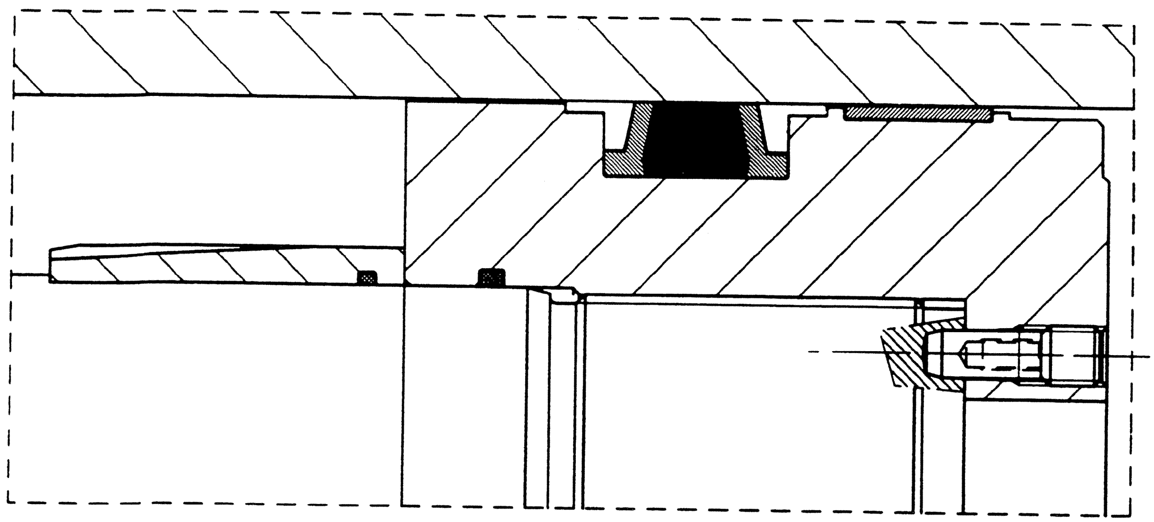
Detailed view rod bearing + seals assembly

- Pistonrod material high strength carbon steel coated with ceramic coating
- Sealing flange with heavy duty mud/ice scraper
- Heavy duty multi-lip seal for long lifetime high reliability
- Bearing strips for maximum reliability
- Bolted connections of sealing flange and end cover for easy servicing
- Manifold direct on cylinder for maximum reliability

7b. TECHNICAL ANSWERS

Bridge cylinder

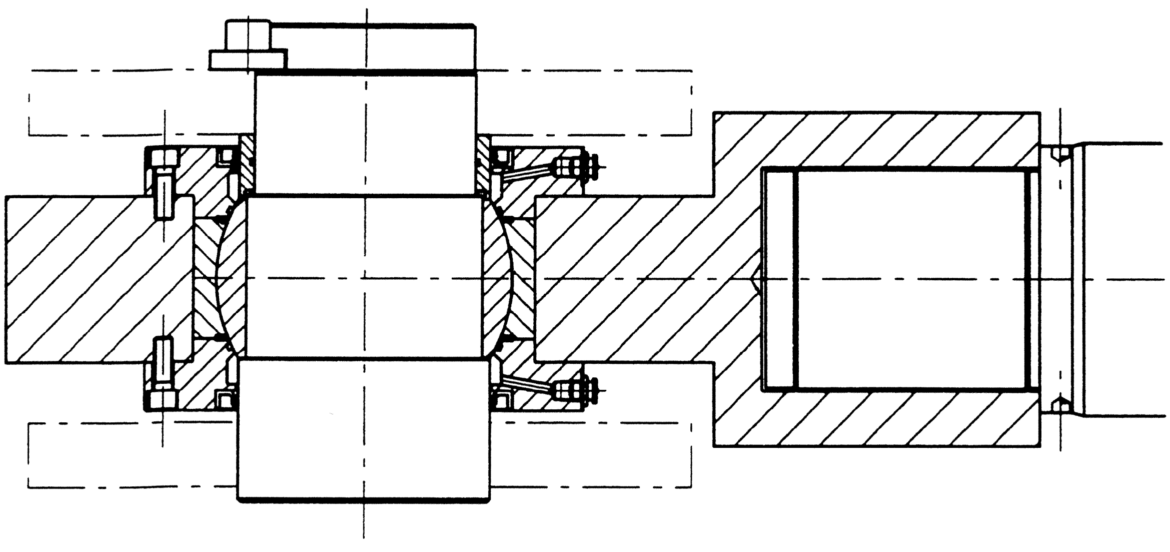




Detailed view piston assembly

- Failsafe bi-directional seal
- Bearing strips are based on high specific pressure
- Mechanical piston locking
- Shell material high strength carbon steel with honed running surface

7c. TECHNICAL ANSWERS
Bridge cylinder

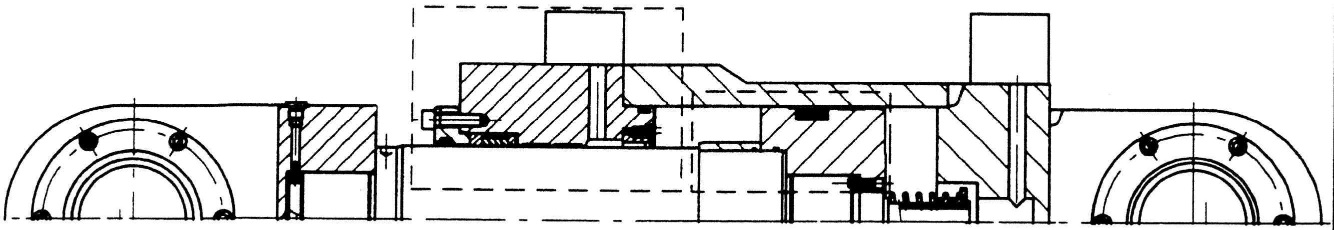


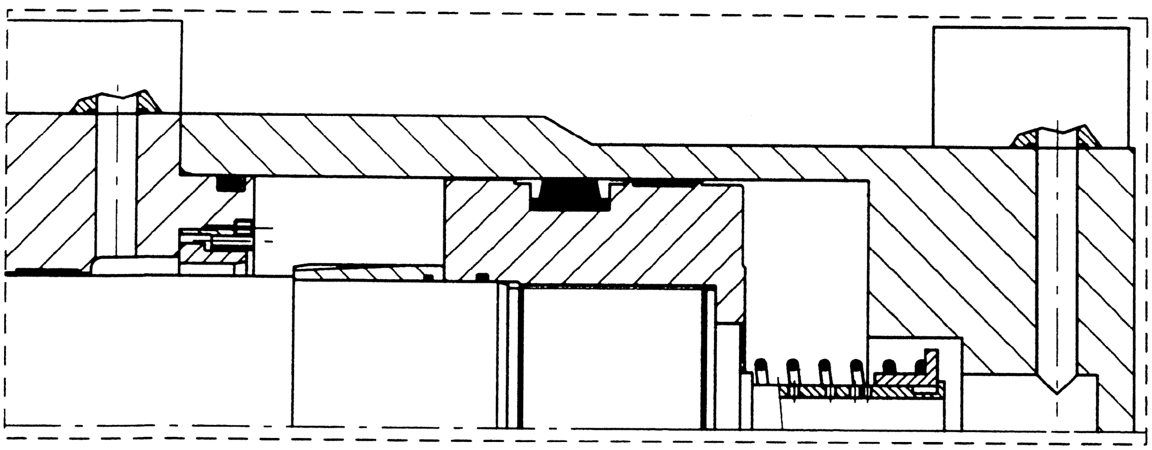
Detailed view spherical bearing assembly

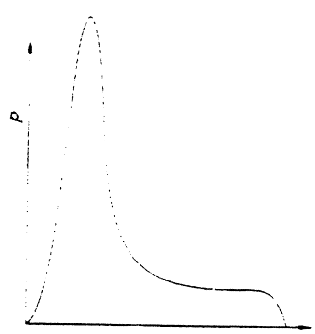
- Water tight clevis/rod connection corrosion proof
- Rotational movement/misalignment
- Maintenance free spherical bearing with safety-factor > 3 or Fmax ≤ 33 % Co
- Dirt/water sealed construction, corrosion proof

7d. TECHNICAL ANSWERS

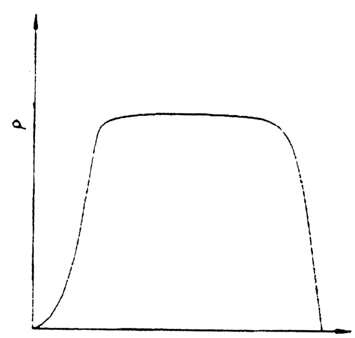
Bridge cylinder







Traditional

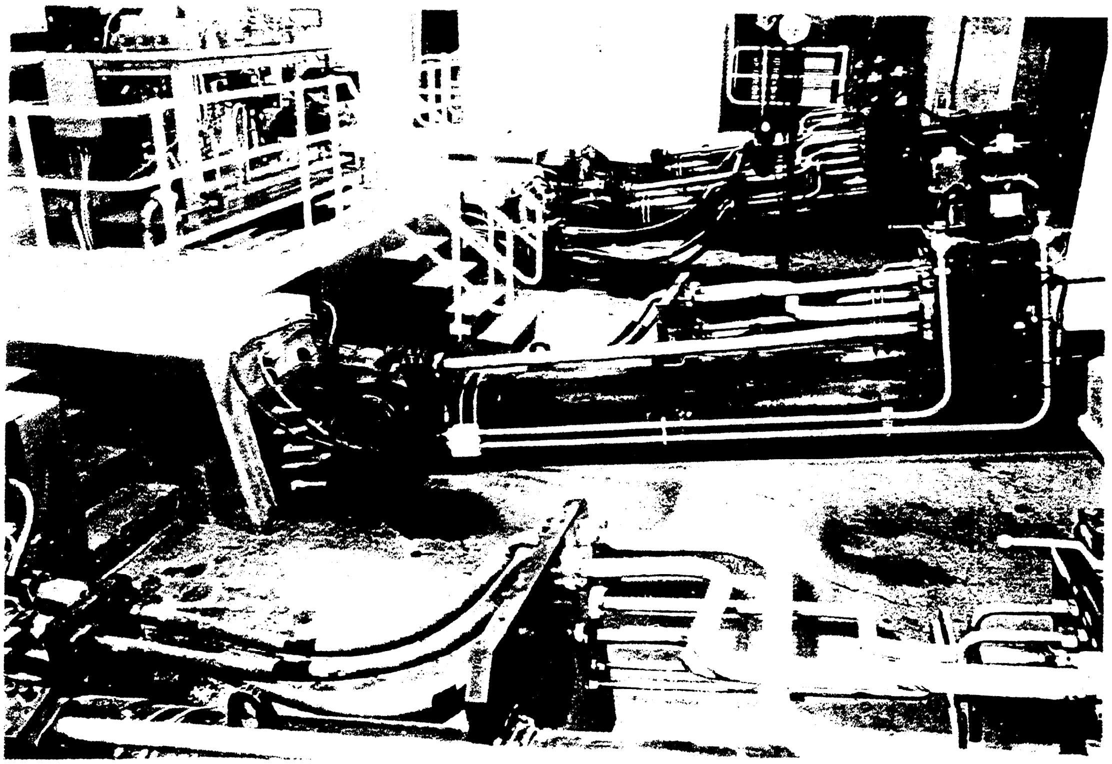


ABS-concept

Detailed view cushioning

- Self adjusting cushioning with integrated check valve
- Speed independent design
- Self centering cushion ring

8. REFERENCES BRIDGE CYLINDERS



- More than 50 projects with more than 100 cylinders
- Bores from 200 mm up to 600 mm
- Strokes up to 8 mtrs.
- For all leading contractors worldwide

Galata	-	Turkey
Erasmus	-	Holland
Berendrecht	-	Belgium
520 bridge	-	U.S.A.
Jupia	-	Brasil
Stone ferry bridge	-	U.K.
Jann Berghaus	-	Germany

III. 9:45 am - 10:15 am
Hydraulic Cylinder Bridge Drives
Presented by: Jim Phillips

A. Movable Bridge Systems

1. Cylinder Geometry
    a. Push Open
    b. Pull Open
    c. Push-Pull Tandems

2. Basic Cylinder Circuit

3. Circuit Details Controlling Loads
    a. Counterbalance Valves
    b. P.O. Check Valves
    c. Relief Valves

4. Cylinder Loads
    a. Operating
    b. Holding
    c. Power Loss

5. Power Requirements
    a. Maximum Pressures (AASHTO 2.5.18)
    b. System Efficiency

B. Design of Supports

1. AASHTO Criteria for Connections
    a. 150% Relief Valve (AASHTO 2.1.14)
    b. Operating Loads
    c. Holding Loads

2. Strength Considerations
    a. Structural Elements
    b. Machinery Elements

C. Cushions

1. HYDRAULIC CYLINDER GEOMETRY
a. CYLINDERS WHICH PUSH A BASCULE OPEN

ADVANTAGE: LARGER BLIND END AREA WORKS AGAINST HIGHEST LOADS (MAXIMUM WIND, PLUS UNBALANCE)

CYLINDER PLACEMENT CONSIDERATIONS

- MOMENT ARM SELECTION: OPTIMIZE MOMENT ARM AND STROKE FOR EFFICIENT DESIGN

  $$Stroke = f\ (R) \qquad\qquad F = \frac{T}{Reff}$$

  SMALL MOMENT ARM = LARGE CYLINDER FORCE (*May produce uplift on trunnion bearings*)

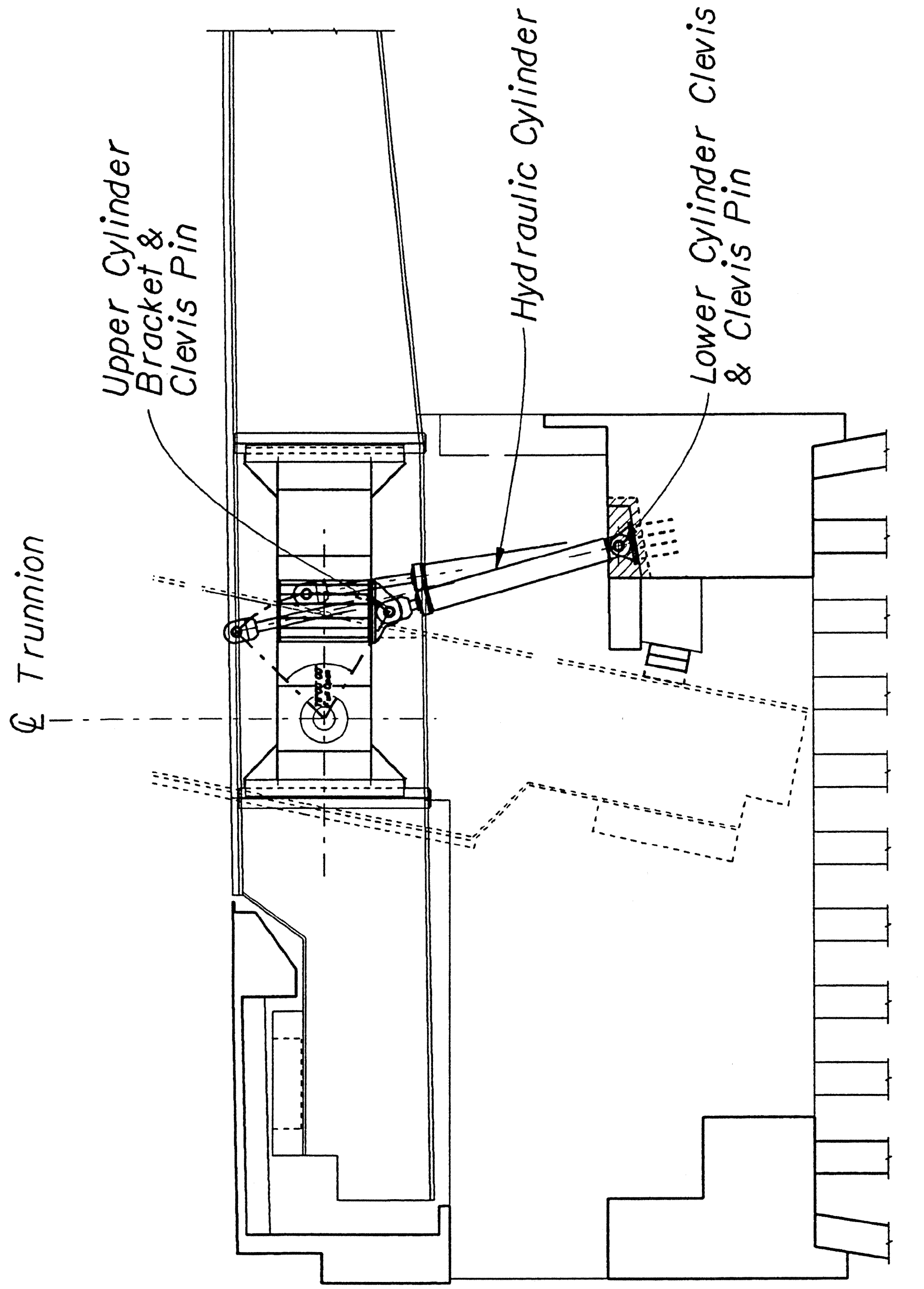
  LARGE MOMENT ARM = HIGH FLOW RATE, LOW EFFICIENCY

- CYLINDER SIZE TYPICALLY CONTROLLED BY HOLDING FORCE ON BLIND END @ MAXIMUM PRESSURE (3000 psi)

- CHECK ROD SIZE FOR BUCKLING & INCREASE IF REQ'D

  $$Buckling = f\ (Rod\ Diameter,\ Stroke,\ Mounting)$$

- INCREASE BORE AS REQUIRED FOR ROD END PRESSURE vs HOLDING LOAD OR OPERATING LOADS IF BUMPER BLOCKS ARE USED FOR HOLDING (*make certain cylinder does not bottom out before bumper blocks are engaged*)



PUSH DRIVE FOR BASCULE SPAN

HYDRAULIC SYSTEMS FOR MOVABLE BRIDGES

Heavy Movable Structures, Inc. - 6th Biennial Symposium
Clearwater Beach, Florida, November, 1996

b. HYDRAULIC CYLINDER GEOMETRY
CYLINDERS WHICH PULL A BASCULE OPEN

ADVANTAGE: MAXIMUM BUCKLING LOADS OCCUR WITH ROD RETRACTED - LIMITING OR ELIMINATING BUCKLING AS A CONTROLLING ELEMENT

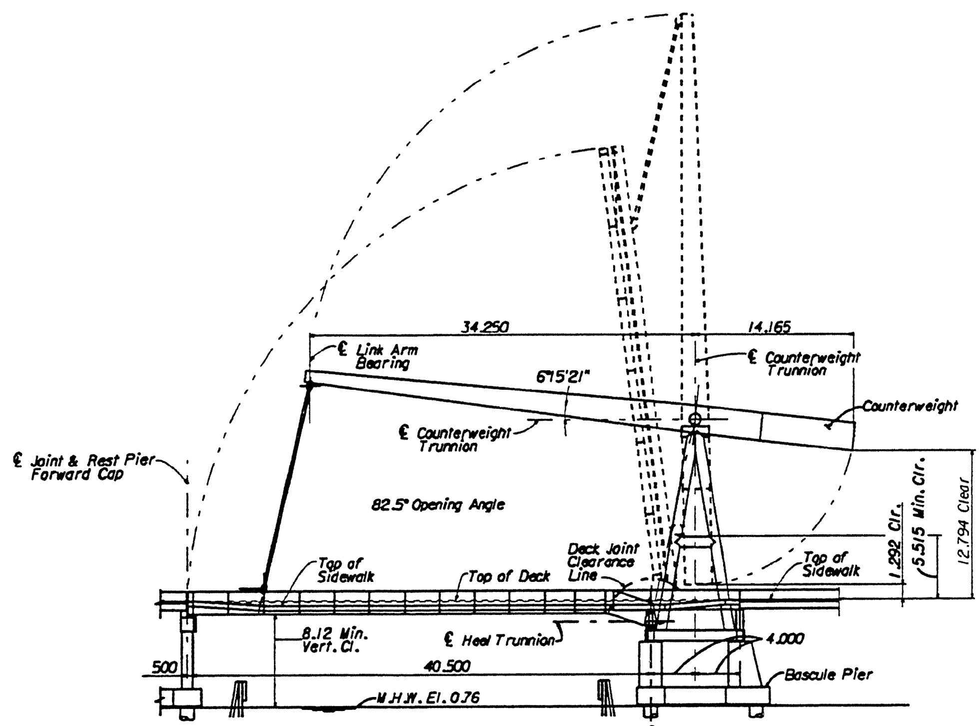
CYLINDER PLACEMENT CONSIDERATIONS

- CYLINDER SIZE TYPICALLY CONTROLLED BY HOLDING FORCE ON ROD END @ MAXIMUM PRESSURE (3000 psi)

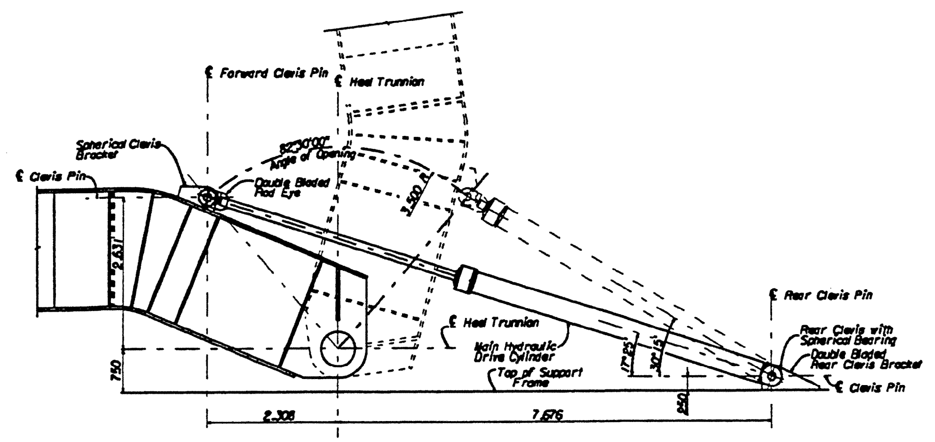
- CHECK ROD SIZE FOR BUCKLING UNDER COMPRESSIVE LOADS & INCREASE IF REQUIRED

  Consider self weight of cylinder

  Consider compression at relief valve setting controlling maximum blind end pressure with cylinder extended.



ELEVATION



PULL DRIVE FOR BASCULE SPAN

HYDRAULIC SYSTEMS FOR MOVABLE BRIDGES

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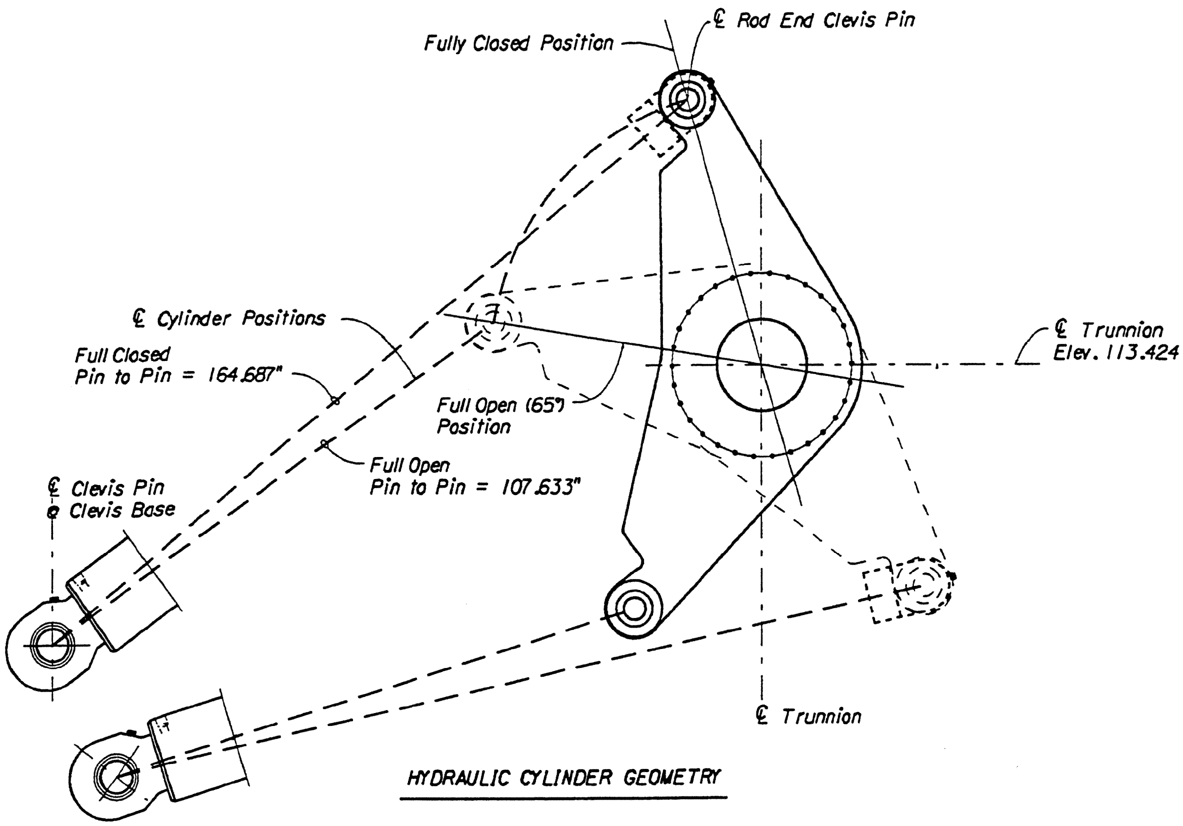
C. HYDRAULIC CYLINDER GEOMETRY CYLINDERS WHICH OPERATE A BRIDGE IN PUSH-PULL TANDEMS

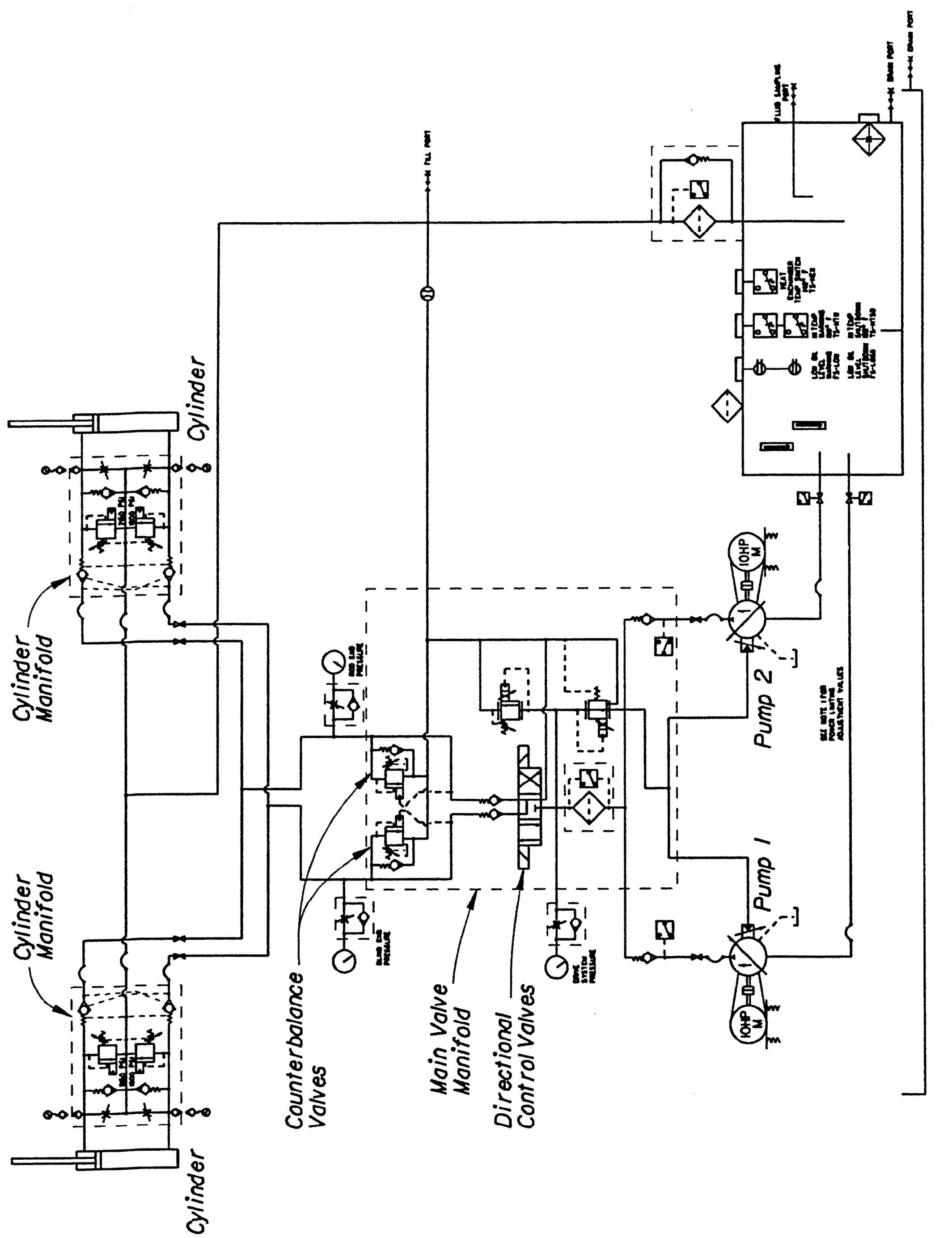
ADVANTAGE: EQUAL FLOW IN EITHER DIRECTION, MINIMIZES NET FORCE ON SUPPORTS



HYDRAULIC CYLINDER GEOMETRY

2. BASIC CYLINDER OPEN CIRCUIT

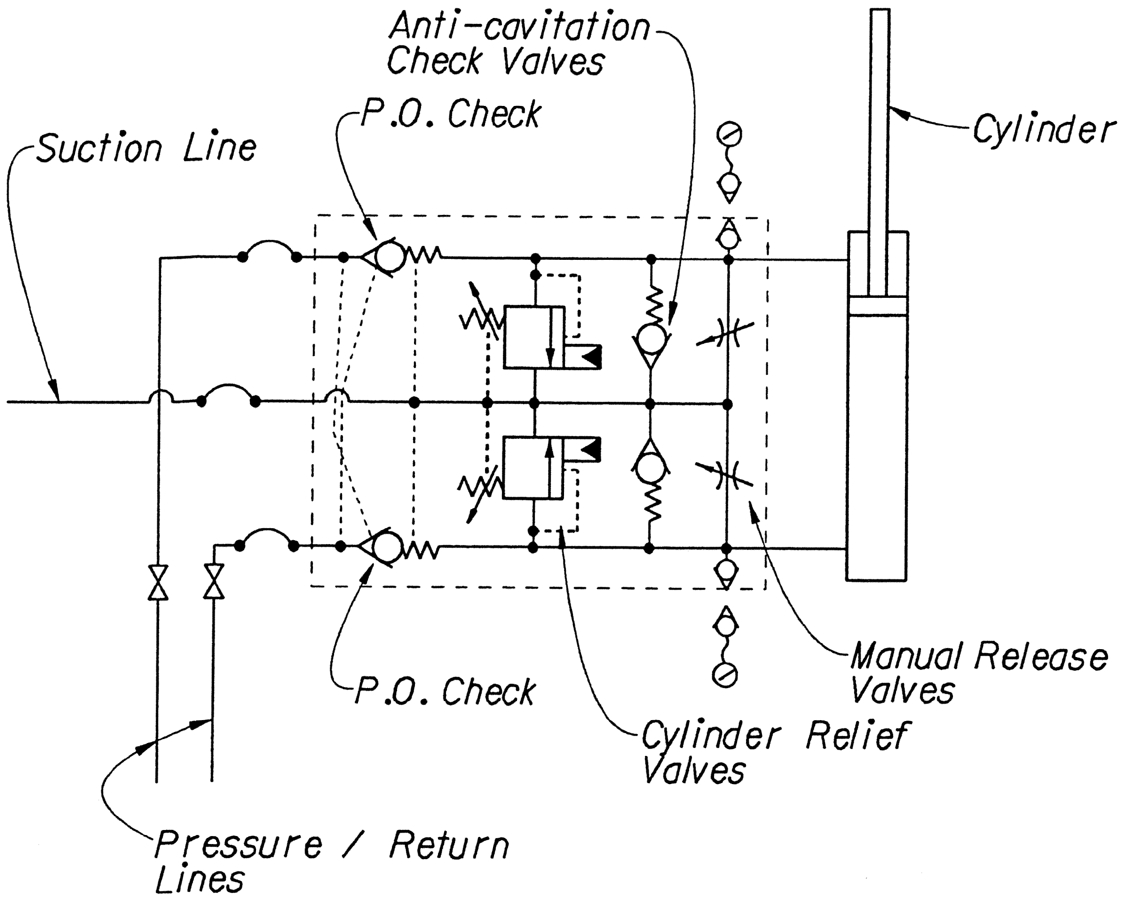
COMPONENT	FUNCTION
MOTOR / PUMP:	Provides fluid power via flow, Q, and pressure, P Horsepower Limiting option varies flow with load Pump Control options control flow at the pump
RELIEF VALVE:	Establishes the maximum pressure in the system Pressure control limits seating pressure
FLOW CONTROL VALVE:	Controls the direction of flow and/or the rate of flow
COUNTERBALANCE VALVE:	Provides resisting pressure to resist overhauling loads
ANTI-CAVITATION PLUMBING:	Provides make-up fluid if cylinders are overrun by extreme wind loads



TYPICAL CLYINDER BRIDGE DRIVE SYSTEM

BASIC CYLINDER MANIFOLD CIRCUIT

COMPONENT	FUNCTION
PILOT OPERATED CHECK VALVES (PO CHECKS):	Holds fluid in the cylinders unless bridge is actuated
CYLINDER RELIEF VALVES:	Limits maximum pressure in Rod and Blind end of cylinder
ANTI-CAVITATION CHECK VALVES	Allows make-up fluid to enter system under overrun conditions
MANUAL RELEASE VALVES:	Allows cylinder to be manually released in an emergency. Allows for free flow during cylinder isolation.



CYLINDER MANIFOLD

4. CYLINDER LOADS

a. OPERATING LOADS

- AASHTO Condition A, B, and C operating loads including wind, ice, unbalance, inertia and friction result in a total moment, M, resisting motion of the span. The cylinder force multiplied by the effective moment arm of the connection must overcome this resistance:

$$\sum M_{wind\ unbalance\ ice\ friction\ inertia} = F \times R_{eff}$$

- The maximum loads must be determined based upon variables for $Mw, Munb$, and $Reff$.

- The system relief valve(s) must be set high enough to account for the resulting pressure plus system losses and an allowance for intangibles (10 percent is recommended).

$$System\ Relief\ Setting = 1.10 \times [\frac{F_{\max}}{A} + \sum \Delta P_{Losses}]$$

CYLINDER LOADS

b. HOLDING LOADS

- AASHTO Condition E holding loads including wind and unbalanced loads less holding friction result in a total moment, M applied to the span. The cylinder force multiplied by the effective moment arm of the connection must resist this moment:

$$M_w \pm M_{unb} - M_f = F_{hold} \times R_{eff}$$

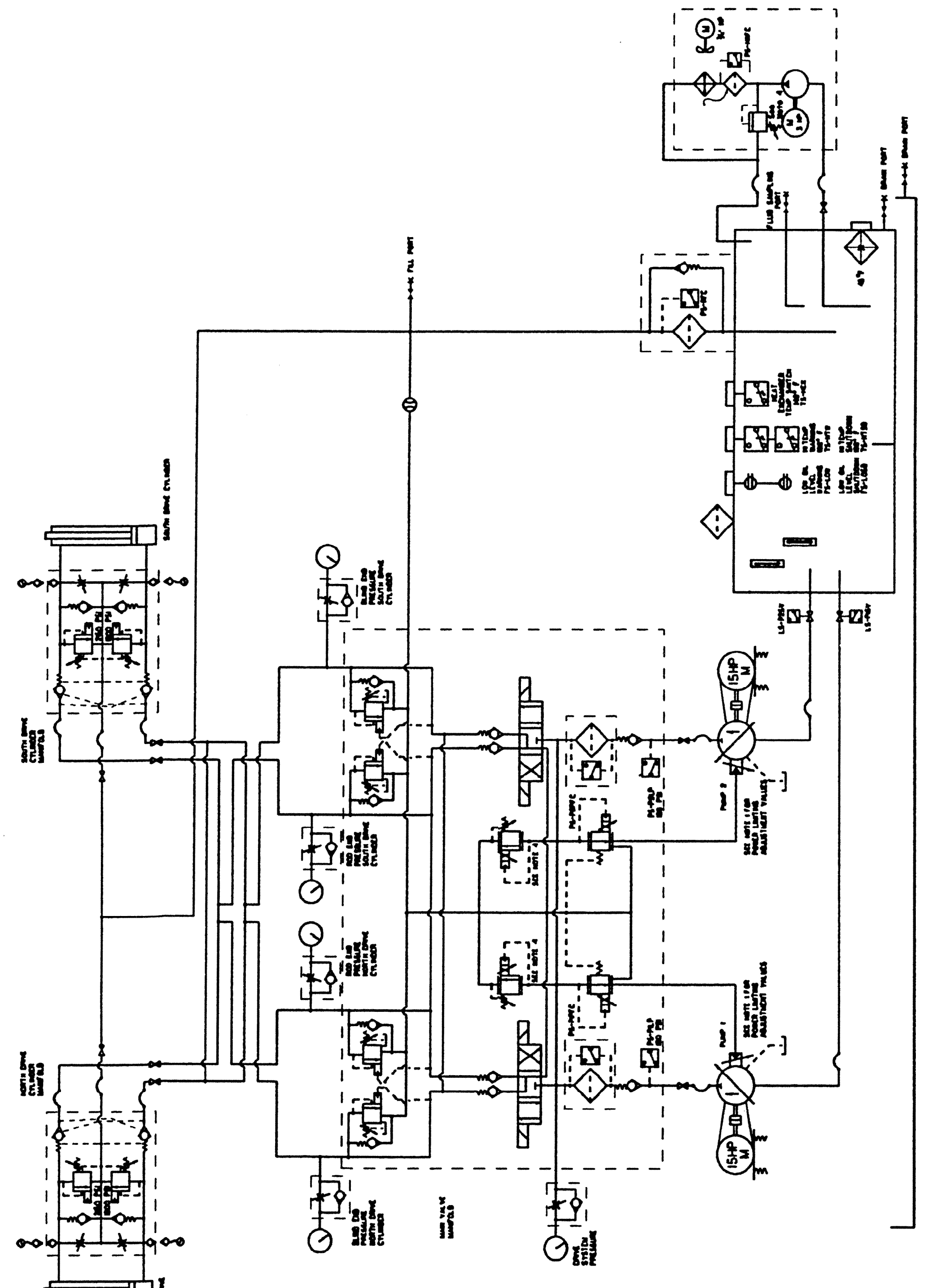
- The maximum loads must be determined based upon variables for ***Mw, Munb***, and ***Reff***.

- The pressure in the cylinder required to resist this load is the force, F divided by the effective area of the cylinder, A:

$$P_{hold} = \frac{F_{hold}}{A_{eff}}$$

- The cylinder relief valve(s) must be set high enough to restrain movement (recommended 10 percent above):

$$\textit{Cylinder Relief Setting} = 1.10 \times P_{hold}$$



REDUNDANT CYLINDER SYSTEM

HYDRAULIC SYSTEMS FOR MOVABLE BRIDGES

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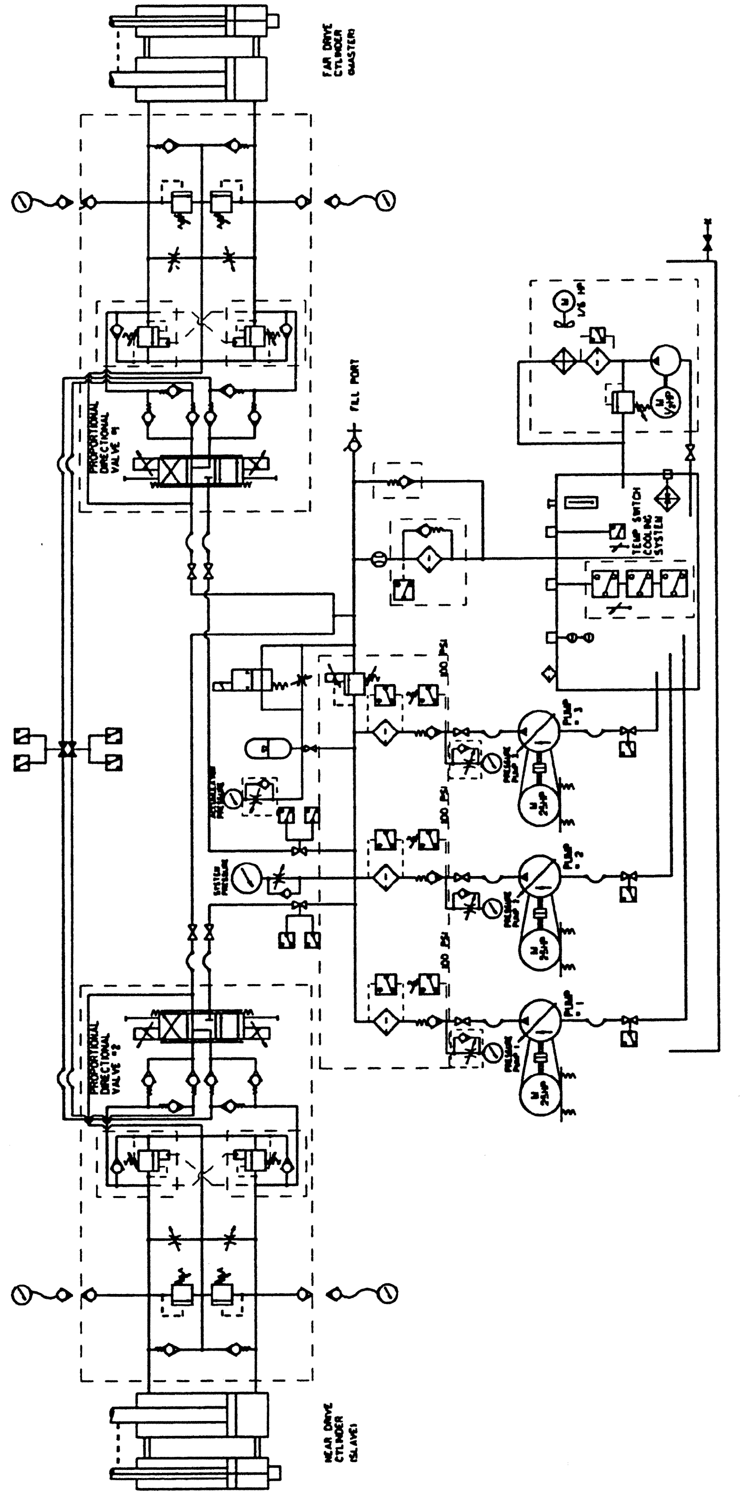

SYNCHRONIZED CYLINDER SYSTEM

HYDRAULIC SYSTEMS FOR MOVABLE BRIDGES

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Clearwater Beach, Florida, November, 1996

CYLINDER LOADS

c. POWER LOSS LOADS

UNLESS OTHER PROVISIONS ARE MADE IN THE HYDRAULIC CIRCUIT, AN ABRUPT POWER LOSS WILL RESULT IN NEARLY INSTANTANEOUS CLOSING OF THE PO CHECK VALVES, EFFECTIVELY LOCKING THE FLUID IN THE CYLINDER.

IF THE VALVES CLOSE, THE CYLINDER RELIEF VALVES WILL BE THE ONLY PATH FOR THE FLUID TO EXIT THE CYLINDER. IN A ABRUPT STOP THE PRESSURE WILL SPIKE WELL ABOVE THE RELIEF SETTING.

AN ALTERNATIVE IS TO ADD AN ACCUMULATOR TO THE CIRCUIT TO PROVIDE A FEW SECONDS OF FLUID POWER TO DECELERATE BEFORE THE VALVES CLOSE COMPLETELY.

5. POWER REQUIREMENTS

a. POWER WILL TYPICALLY BE CONTROLLED BY THE AVERAGE FLOW AND PRESSURE FOR AASHTO CONDITION A.

- AVERAGE FLOW CAN BE COMPUTED BASED UPON THE TOTAL STROKE, TIME OF OPERATION, t, AND RAMP TIMES, ta_1 and ta_2:

$$Q_{avg}(GPM) = \frac{Stroke(inch)\ x\ Net\ Area(inch^2)\ x\ 60}{(t - \frac{ta_1 + ta_2}{2})(seconds)\ x\ 231}$$

- AVERAGE PRESSURE SHOULD BE DETERMINED FROM AN EVALUATION OF THE PRESSURES THROUGHOUT THE STROKE, EXCLUDING INERTIA.

- THE POWER TO OPERATE THE SPAN CAN BE COMPUTED FROM THE AVERAGE FLOW AND PRESSURE PLUS AN EVALUATION OF THE PRESSURE DROPS THROUGH THE SYSTEM AT AVERAGE FLOW.

$$\Delta P_{avg} = \sum \Delta p_{avg}$$

$$HP = \frac{Q_{avg}\ x\ P_{avg}}{1714\ x\ \eta_t\ pump}$$

- THE MAXIMUM FLOW WILL OCCUR WHEN THE SPAN IS AT MAXIMUM SPEED AND THE EFFECTIVE STROKE IS AT A MAXIMUM, PRODUCING THE MAXIMUM CYLINDER VELOCITY, V_{max}:

$$V_{max}(ips) = MaxR_{eff}(inch)\ x\ Span\ Speed(Radians\ per\ second)$$

$$Q_{max}\ (GPM) = \frac{V_{max}(ips)\ x\ Net\ Area(inch^2)\ x\ 60}{231}$$

Note that the net area may be the blind or rod end area.

AASHTO ALLOWABLE PRESSURES (2.5.11)

HOLDING LOADS = 3,000 psi

OPERATING LOADS

NORMAL= 2,000 psi

MAXIMUM 1,000 psi

NOTE: COMMENTARY ALLOWS VARIATIONS BASED UPON ENGINEERING JUDGEMENT

CYLINDER BUCKLING IS OFTEN A CONSTRAINT AGAINST USE OF HIGHER PRESSURES FOR CYLINDER SYSTEMS.

b. SYSTEM EFFICIENCY

CALCULATION OF PRIME MOVER AND PUMP SIZE MUST CONSIDER TOTAL VOLUMETRIC AND MECHANICAL EFFICIENCIES AS WELL AS ALL SYSTEM PRESSURE DROPS. CONSULT MANUFACTURER'S DATA FOR EFFICIENCIES. USE FLUID POWER HANDBOOK FOR LOSSES IN PIPING.

GENERAL RULES:

CYLINDER EFFICIENCY = 90 TO 95 %

PUMP EFFICIENCY = 75 TO 90 %

COUNTERBALANCE VALVES AND PROPORTIONAL CONTROL VALVES ARE A SOURCE OF LARGE LOSSES

FOR SMALL CYLINDER DRIVES TOTAL EFFICIENCY MAY BE BETWEEN 75 AND 80 PERCENT.

FOR LARGE CYLINDER DRIVES TOTAL EFFICIENCY CAN BE LESS THAN 50% UNLESS EXTRA STEPS ARE TAKEN.

B. DESIGN OF SUPPORTS FOR HYDRAULIC CYLINDERS

1. AASHTO CRITERIA FOR CONNECTIONS

a. 150% OF RELIEF VALVE SETTING

AASHTO SPECIFICATIONS 2.2.14.

"THE STRESSES ON THE STRUCTURAL CONNECTIONS TO THE CYLINDERS SHALL BE BASED ON A CYLINDER PRESSURE OF 150 PERCENT OF THE SETTING OF THE PRESSURE-RELIEF VALVE CONTROLLING THE MAXIMUM PRESSURE AVAILABLE AT THE CYLINDER."

b. OPERATING LOADS

AASHTO SPECIFICATIONS 2.5.3.A.

c. HOLDING LOADS

AASHTO SPECIFICATIONS 2.5.3.E

2. STRENGTH CONSIDERATIONS FOR HYDRAULIC CYLINDER CONNECTIONS AND SUPPORTS

DON'T MIX MACHINERY AND STRUCTURAL LOADS AND ALLOWABLE STRESSES - REMAIN CONSISTENT

AASHTO SPECIFICATIONS

2.1.11. IMPACT

"STRESSES IN STRUCTURAL PARTS CAUSED BY THE MACHINERY, OR BY FORCES APPLIED FOR MOVING OR STOPPING THE SPAN, SHALL BE INCREASED 100 PERCENT AS AN ALLOWANCE FOR IMPACT."

"ALLOWANCE HAS BEEN MADE FOR IMPACT IN TRUNNIONS, WIRE ROPES, WIRE ROPE ATTACHMENTS, AND MACHINERY PARTS IN THE BASIC ALLOWABLE UNIT STRESSES SPECIFIED FOR SUCH PARTS."

STRENGTH CONSIDERATIONS FOR HYDRAULIC CYLINDER CONNECTIONS AND SUPPORTS

a. STRUCTURAL ELEMENTS

DESIGN PER AASHTO SPECIFICATIONS FOR HIGHWAY BRIDGES

APPLY 100% IMPACT FOR FORCES CAUSED BY THE CYLINDER (AASHTO 2.1.11)

DESIGN FOR GREATER OF:

1. 150% RELIEF VALVE
2. ACTUAL OPERATING LOADS PLUS 100% IMPACT
3. 100% SYSTEM RELIEF VALVE (SPAN RESTRAINED BY LOCKS LOAD SHOE OR BUMPER BLOCK)

USE APPROPRIATE LOAD COMBINATIONS AND ALLOWABLE STRESSES PER AASHTO (i.e. 125% ALLOWABLE FOR WIND LOADS)

STRENGTH CONSIDERATIONS FOR HYDRAULIC CYLINDER CONNECTIONS AND SUPPORTS

b. MACHINERY ELEMENTS

DESIGN PER AASHTO MOVABLE

MACHINERY ALLOWABLE STRESSES ALLOW FOR IMPACT (AASHTO 2.1.11)

INCREASE ALLOWABLE STRESS BY 50% FOR HOLDING LOADS IN CYLINDERS (AASHTO 2.5.3.E)

C. HYDRAULIC CYLINDER CUSHIONS

CYLINDER CUSHIONS ARE NOT ADDRESSED IN CURRENT AASHTO SPECIFICATIONS

RECOMMENDED CRITERIA:

1. CYLINDERS SHALL HAVE CUSHIONS AT BOTH ENDS (FIXED OR ADJUSTABLE WITH TAMPER PROOF LOCKING DEVICE)

2. SIZE CUSHIONS TO ABSORB THE ENERGY OF THE SPAN, MOVING AT FULL SPEED, PLUS RESIST CONDITION “A” WIND LOADS AND SPAN UNBALANCE. TOTAL ENERGY MUST BE DISIPATED BEFORE THE CYLINDER BOTTOMS OUT AT EITHER END.

3. CUSHIONS SHOULD BE WITHIN THE CREEP SPEED PORTION OF THE STROKE SO AS NOT TO RETARD NORMAL OPERATION UNLESS THE SPAN APPROACHES THE END OF THE STOKE AT A SPEED IN EXCESS OF THE CREEP SPEED.

C. HYDRAULIC CYLINDER CUSHIONS (cont.)

CUSHION CALCULATIONS:

1. COMPUTE AN EQUIVALENT LINEAR MODEL MASS, $\mathbf{M_L}$, FOR THE MASS MOMENT OF INERTIA OF THE SPAN ABOUT THE AXIS OF ROTATION, I_p.

$$M_L\left(\frac{kip\cdot ft}{\sec^2}\right)=\frac{I_p(kip\cdot ft\cdot \sec^2)}{R_{eff}(ft)^2}$$

2. COMPUTE THE REQUIRED CHANGE IN ROD VELOCITY FROM FULL SPEED TO CREEP SPEED.

3. COMPUTE THE EQUIVALENT LINEAR FORCE (AND RESULTING CYLINDER PRESSURE) TO DECELERATE THE EQUIVALENT MASS WITHIN THE WORKING LENGTH OF THE CUSHION.

4. ADD THE PRESSURE RESULTING FROM WIND AND UNBALANCE LOADS TO THE PRESSURE FROM 3 ABOVE.

5. SIZE THE CUSHION ORIFICE TO PRODUCE THE REQUIRED RESISTING PRESSURE AT THE FLOW CORRESPONDING TO ½ FULL SPEED.

6. VERIFY THAT CUSHIONING PRESSURE IS WITHIN ALLOWABLE LIMITS AND WORKING CUSHION LENGTH IS REASONABLE.

IV. 10:15 am - 10:45 am
Hydraulic Motor Systems
Presented by: Lou Wendel

A. Types of Hydraulic Motors

1. Hydraulic motors are manufactured using the gear, gerotor, vane, radial, and axial piston displacement principles.

2. This presentation will discuss the piston design types.
   a. High speed low torque axial piston motors.
   b. Low speed high torque radial piston motors.

3. Relative efficiencies of each design.

B. Rotary Motion

1. Relationship between horsepower, torque and speed.
   a. Torque defined.
      - Fluid motor torque.
      - Starting torque.
      - Stall torque.
   b. Fluid horsepower formula.
      - Input horsepower.
      - Transposing the horsepower formula.

C. Hydrostatic Circuits

1. Building block principle of the hydrostatic circuit.
   a. The hydrostatic pump, transmission valve, torque limiting.
   b. Dynamic braking.
   c. Horsepower limiting

HYDRAULIC MOTORS

HMS 6th BIENNIAL SYMPOSIUM
NOVEMBER 1996

The purpose of this paper is to familiarize the reader with the various types of hydraulic motors available in the marketplace and to dispel some of the myths, mystique and misunderstanding surrounding hydraulic motors.

We start off with a basic comparison between electric motors and hydraulic motors, from there we discuss the two major categories of hydraulic motors and then the different types of hydraulic motors within each of these two categories.

ELECTRIC AND HYDRAULIC MOTORS COMPARED

General Considerations

A three phase squirrel cage electric motor has for many years been a dependable and highly efficient means of producing rotary power, however, any piece of machinery no matter how good has applications for which it is not as well suited as it is for others. The electric motor has been deficient in performance on certain types of applications. Most notably those which involve frequent starting and stoping under load, frequent or extremely rapid reversal in the direction of rotation, variable speed control over a wide range, operation under hostile environmental conditions, small size requirements for mounting in limited space, high horsepower per pound ratio, and where the prime mover is obtained by a mechanical source, such as an engine.

The invention, development and application of hydraulic motors have made it possible to efficiently accomplish many of the jobs of the types listed above which are difficult or impossible with standard electric motors. This is not to say that electric motors in general should be replaced with hydraulic motors on all jobs, only that hydraulic motors have certain features that make them more suitable for some applications.

Hydraulic motors can be built with far more horsepower per pound of weight and with a correspondingly smaller physical size. This may make them a better choice where there are severe limitations on weight and space at the point of usage. To be specific, electric motors average from 30 lbs. per horsepower in the smaller frames to 12 lbs. per horsepower in the larger frames according to their catalog listing. On the other hand, hydraulic motors have a range from 1/2 to 5 lbs. per horsepower. Of course this varies widely according to brand type and size.

EFFICIENCY

There can be no argument that electric motors convert a greater protion of their input power into mechanical output. For instance, a 3 phase squirrel cage electric motor operates in the overall efficiency range of 90% ti 95%. Hydraulic motors cannot match this efficient utilization of input power.

Better quality, gear, vane and piston motors can operate up to 95 to 90 percent overall efficiency. Some precision built motors may be 90% or slightly higher.

UNFAVORABLE ENVIRONMENTS

Hydraulic motors can easily handle environmental jobs that would be difficult, dangerous, or impossible with ordinary electric motors. Hydraulic motors can operate completely submerged in water or various other fluids such as sewage or sludge at pumping stations or chemical plants. Standard hydraulic motors can operate safely in explosive or corrosive atmospheres for which special electric motors must be utilized. Hydraulic motors can also operate in hot environments that would be difficult for electric motors.

RUNNING TORQUE

Hydraulic motors are limited in running torque to the pressure of the system relief valve. To some applications this has the advantage of precise and continuous variable torque control. On other applications it may be a disadvantage since the motor has no torque reserve on which to draw to prevent stall during momentary or accidental high peak loads.

The three phase squirrel cage electric motor can develop double or more of full load torque rating before it stalls. This can be advantageous on some applications and objectional in others. The torque level can be easily varied, however, with electric motors there is no way of limiting the maximum torque and hydraulic motors can be frequently started and stopped as often as desired without overheating.

INSTANT REVERSABILITY

Since the rotor of a hydraulic motor usually contains much less mass than the rotor of an electric motor of the same horsepower, less energy is needed to reverse directions, especially at high speeds. Standard electric motors draw a very high peak current during reversal and therefore are subject to overheating and burnout on rapid cycling operations. If a electric motor is to be used in such a manner it should be a special design intended for high cycling service.

REPLACEMENT OF AN ELECTRIC MOTOR WITH A HYDRAULIC MOTOR

We sometimes consider replacing an electric motor with a hydraulic motor to gain a certain advantage. The usual method in selecting an electric motor is on the basis of its speed and horsepower, both of which appear on the motor nameplate. Torque ratings are not often mentioned unless the motor happens to be specially designed for high torque/high slip service.

In trying to estimate the size of hydraulic motor needed on a new design we sometimes compare applications with another similiar application which sucessfully used a certain size electric motor. This may be a valid approach but caution must be exercised because of different operating characteristics for the two kinds of motors.

In the selection of a hydraulic motor we are primarily interested in two requirements:

1) Speed in rpm
2) Torque in lb.ft. or lb.in.

Therefore, the usual staring point in the design of a fluid power system using a hydraulic motor is to clearly define the maximum speed and maximum torque needed.

The duty cycle of the application should also be carefully considered throughout the entire cycle to see if there are any unusual or peak torque requirements such as start up under heavy load or peaking out at an unusually high torque at the end of the cycle. The most accurate way of doing this is to use an amp meter to take amp readings on the electric motor at all parts of the duty cycle to find the maximum peak current drawn by the motor. Then convert this amperage to torque by comparing it to a torque/current graph for that particular motor or use the following rule of thumb: at 440 volts, a 3 phase motor draws 1.25 amps/hp.

The hydraulic motor should be selected which will deliver an equal amount of torque. If sized for less than peak torque, the hydraulic motor will surely stall at that part of the cycle. Lacking any measuring equipment we can only assume that the electric motor has been sized correctly and is working at or close to its nameplate rating for most of the cycles and that its peaking out at 1 to 1 1/2 times its full load torque. This often results in oversizing the hydraulic motor and it is much better to take measurements of the torque if it is at all possible.

HYDRAULIC MOTORS

Hydraulic motors come in two basic categories:

1) HIGH SPEED LOW TORQUE (HSLT)
2) LOW SPEED HIGH TORQUE (LSHT)

The most obvious difference between the two types is that the HSLT motor normally requires a speed reducer and the LSHT normally does not.

Hydraulic motors are built using the gear, gerotor, vane, radial and axial piston principles. By far the most common type of HSLT hydraulic motor is the axial piston type. This type of motor is further divided into inline and bent axis designs.

Listed below are some general statements associated with axial piston hydraulic motors.

AXIAL PISTON MOTORS

A) Axial piston motors are very efficient.

B) Axial piston motors cost more than vane or gear motors of comparable horse power but they generally have a much longer life.

C) In general, axial piston motors have excellent high speed capabilities but unlike low speed motors they are limited in low speed operations. Most axial piston motors cannot operate below 100 rpm.

D) Axial piston motors are available with displacements ranging from a fraction to more than 200 cubic inches per revolution.

E) Axial piston motors are available in both in line and bent axis design and in fixed and variable displacements.

F) Typically high speed hydraulic motors do not develop appreciable torque until rotational speeds reach 300 to 400 rpm and ideal torque is not produced until speeds are 1000 or 2000 rpm.

BENT AXIS PISTON MOTORS

With bent axis motors the speed and torque change with changes in the swashplate angle from a predetermined minimum speed with the maximum displacement and torque at an angle of approximately 30 degrees to a maximum speed with minimum displacement and torque at about 7 1/2 degrees. Both fixed and variable displacement models are available.

LOW SPEED HIGH TORQUE HYDRAULIC MOTORS

LSHT hydraulic motors are also available in two major classifications; axial piston and radial piston designs. However, the vast majority of all LSHT hydraulic motors are of the radial piston design. Axial piston LSHT hydraulic motors have not underwent serious development and have not gained wide spread customer acceptance. Some advantages to these motors are a fairly small working diameter and large fluid volume displacement due to multiple piston strokes per revolution. In fact, for motors with displacements above 20 cubic inches per revolution approximately 95% of todays market uses the radial piston design,

Historically, the radial piston design has been in the forefront of large low speed high torque motor technology. Because of advances in seal design, piston bypass leakage is low, even at low speeds. In some designs the pistons can make several strokes per revolution, producing high displacements. This is important because motor torque is proportional to displacement. Also because the pistons are mounted radially, they can be relatively large. Taking advantage of these factors, motors with capacities as large as 10 to 15 gallons per revolution have been produced.

Listed below are some general statements associated with radial piston hydraulic motors.

RADIAL PISTON MOTORS

A) Their efficiency characteristics are similiar to those of axial piston motors.

B) The high degree of precision required in the manufacture of radial piston motors increases the initial cost, however they generally have a long life.

C) They provide high torque at relatively low speeds and excellent low speed operation with high efficiency.

D) They have limited high speed capabilities.

E) Radial piston motors have displacements up to 4000 cubic inches per revolution.

F) They normally do not require speed reducers.

Radial piston motors are further defined into multi-cam lobe or excenter shaft motors.

Excenter shaft hydraulic motors generally have 5 and sometimes 7 cylinders in a single bank. Larger displacement motors combine 2 banks of cylinders.

In the multi-cam lobe motor concept, the pistons are part of a rotor assembly that is connected to the output shaft. Piston power strokes work out against an inner cam ring that is attached to the side covers. Piston of multi-cam lobe motors

have considerable smaller bores than in excenter shaft hydraulic motors. Up to 15 pistons and 8 cam lobes accounting for 120 strokes per revolution, are incorporated in some designs of multi-cam lobe motors.
Motors of this multi cam lobe design usually use roller bearings mounted to a crosshead shaft that follow the cam surface. Because of the roller action these motors have very low start up friction with typical starting torque from 90 to 97% of theoretical. Also, due to the multiple piston design they operate smoothly at very low speeds and generate good low speed torque. Multi-cam lobe motors have been used in continuous duty applications at speeds as low as .02 rpm. In addition the cam profile has been refined into a shape that produces nearly constant angular velocity, regardless of the position of the motor output shaft. This is an advantage in applications that require fine velocity control at fractional speed.

Low speed high torque hydraulic motors, developed primarily in Europe, have only slowly penetrated the U.S. market. Two factors inhibiting their widespread use have been a relatively high initial cost and the lack of a rational method for analyzing and selecting the motors.

Recent studies have shown that overall systems cost for low speed high torque motors are comparable to those of other hydraulic motors. The selection procedure for low speed hydraulic motors is complicated by the fact that all the information needed may not be readily available in manufacturers catalogs. In some cases, the required data must be requested directly from the builder or supplier.

The overall efficiency of a low speed high torque motor is the product of its mechanical efficiency and its volumetric efficiency. These efficiencies are not constant but will vary with operating speeds and pressure. Normally, mechanical and volumetric efficiency can be determined from data provided in the manufacturer's catalog. Occassionally published data is not available for the operating conditions of interest. In such cases, efficiency values can be requested directly from the manufacturer.

The factors reducing mechanical efficiency are friction and flow losses. Friction losses are most prevalent at low speeds and can be attributed to bearings, shaft seals, valve plates, swash plates and piston rings.

For drives using high speed motors and gear reducers, the mechanical efficiency losses also include the losses within the gear reducer which have to be taken into account. Flow losses on the other hand occur at higher speed when a volume of oil is forced through restricted flow passages.
The following are some general statements pertaining to hydraulic motors:

Torque output is expressed in pound inches or pound feet and is a function of system pressure and motor displacement. Motor torque figures are usually given for specific pressure drop across the motor.

Theoretical figures indicate the torque available at the motor shaft assuming no mechanical losses. A hydraulic motor is a positive displacement device. That is, as it receives a constant flow of fluid the motor speed will remain relatively constant regardless of the pressure. A piston motor is a positive displacement motor which develops an output torque at its shaft by allowing hydraulic pressure to act on its pistons.

A) Piston motors, operated under proper conditions, wear at extremely slow rates. The wear in a piston motor can be detected by monitoring the volume of case drain leakage and contamination in the case drain line.

B) High speed hydraulic motors are basically the same as hydraulic pumps except they are used to convert hydraulic energy back to mechanical energy.

C) With hydraulic motors, the pressure level is determined by the torque load and the size of the hydraulic motor.

D) The size of a hydraulic motor is determined by its geometric displacement in cubic inches per revolution. The discplacement is the amount of oil the motor consumes in making one complete revolution.

E) Volumetric efficiency is higher at low pressures and falls off as you approach maximum pressure capabilities.

F) Volumetric efficiency is highest at maximum speed. This is true because leakage is mostly pressure dependent and consequently is a small percentage of the larger flow rate supplied at higher speeds.

G) Mechanical efficiency is higher at low speeds because of less friction and flow losses through the motor.

H) Depending on motor design, starting torque can be as low as 60% and as high as 97% of the motor's running torque capability. The starting torque capabilities must always be considered when the motor must start under load.

Hydraulic motors can provide a constant torque over a wide state range with reasonable efficiency.

The amount of torque that can be produced by a LSHT hydraulic motor does not depend on its speed directly, although its speed may have a slight effect due to porting loss. The torque depends on two factors. Number 1, the pressure level in the fluid and 2, the number of square inches of working elements (piston, vanes, etc.) exposed to the pressure. With fixed displacement motors the amount of torque produced is in direct proportion to the pressure in the fluid so if you double the pressure the torque doubles.

If a LSHT hydraulic motor is operated at a constant pressure its torque output should be theoretically flat from zero to maximum speed. In practice however, torque output measured on the motor shaft will show a decrease as speed is increased. In well designed motors this loss of torque will be reasonably small. It can be accounted for as follows:

> Since speed is a direct function of gallons per minute supplied to a motor, at high speeds there is a high fluid flow through the motor and consequently a higher fluid frictinal loss of the fluid passing through the inlet and outlet ports and internal passages. This is called porting loss or pressure drop. Whatever portion of the fluid pressure is used in overcoming pressure drop is unavailable for producing useful output torque. Instead, this loss for the most part is converted into heat in the fluid.

Hydraulic oil does not contain energy unless it has both pressure and flow. The pressure produces torque and the flow produces speed. It takes both of these conditions to produce horsepower.

Volumetric efficiency losses are caused by leakages that consist of so called case drain leakage and cross port leakage. Case drain leakago results from the flow of oil through part clearances such as between pistons and cylinders or between a valve plate and distribution ports.

This flow drains into the motor case and is relatively easy to measure. Cross port leakage is the fluid that flows from the input directly to the output without doing useful work. Because it is internal this leakage is difficult to measure.

Both mechanical and volumetric losses can show up as torque and speed pulsations that rob a motor of power. Flow losses can be minimized by installing a larger pump to off-set leakage losses and by using a larger reservoir or heat exchanger to dissipate the heat generated by the losses. These last considerations point up the importance of having of accurate efficiency data. Without the proper data a motor must be oversized to insure adequate output torque. This in turn means that larger more expensive lines, valves, pumps and reservoirs must be used to accommodate the flow. Such an oversized system is not only more expensivve initially it is more expensive to operate and to maintain.

Motor displacement refers to the amount of fluid required to turn the motor output shaft one revolution. Manufacturers of hydraulic motors often rate their motors by cubic inches per revolution. This can be converted to gallons per minute by using the equivalent: 231 cubic inches equal 1 gallon.

Displacement of hydraulic motors may be fixed or variable. A fixed displacement motor provides constant torque. Speed is varied by controlling the amount of input flow into the motor.

A variable displacement motor produces variable torque and variable speed. With input flow and pressures remaining constant, the ratio between the output torque and speed can be varied to meet load requirements by varying the displacement.

With a fixed displacement hydraulic motor, pressure is the only factor that can be selected by the designer or varied by the operator to vary the output torque, and flow is the only factor which can be selected or varied by the designer or the operator of the machine to control the output speed. Speed is directly proportional to the flow.

TERMS ASSOCIATED WITH HYDRAULIC MOTORS

A) Breakaway torque - this term refers to a requirement of a hydraulic motor. Breakaway torque is the torque required to get a non-moving load turning. More torque is normally required to start the load moving than to keep it moving.

B) Running torque - this term can refer to a motor's load or to the motor. When it is used with reference to a load it indicates the torque required to keep the load turning. When it refers to the motor, running torque indicates actual torque which a motor can develop to keep a load turning. Running torque takes into account a motor's inefficiency and is expressed as a percentage of its theoretical torque. Running torque of common LSHT piston motors is approximately 95% of theoretical.

C) Starting torque - this term refers to the capability of a hydraulic motor. It indicates the amount of torque which a motor can develop to start a load turning. In some cases it is much less than a motors running torque. Starting torque is expressed as a percentage of theoretical torque. Starting torque for common LSHT piston motors range between 70 and 97 percent of theoretical.

V. **10:45 am - 11:15 am**
Hydraulic Motor Bridge Drives
Presented by: Jim Phillips

A. Movable Bridge Systems

1. Rack/Pinion/Motor Geometry
    a. Motor Speeds
    b. Brake Considerations
    c. Differential Reducer
    d. Tandem Drives
    e. Frame Mounted Drives

2. Basic Motor Circuits
    a. Open Circuits
    b. Closed Circuits

B. Motor Loads
1. Operating
2. Braking
3. Holding
4. Power Loss

C. Power Requirements
1. Maximum Pressures (AASHTO 2.5.18)

D. Machinery Design Loads

A. MOVABLE BRIDGE SYSTEMS

1. RACK/PINION/MOTOR GEOMETRY

a. MOTOR SPEEDS

- HYDRAULIC MOTORS CAN OPERATE AT A VARIETY OF SPEEDS.

- HIGH SPEED HYDRAULIC MOTORS OPERATE AT SPEED COMPARABLE TO ELECTRIC MOTORS (I.E. 900, 1200 RPM ETC.)

  HIGH SPEED HYDRAULIC MOTORS CAN BE USED AS A SUBSTITUTE FOR AC OR DC MOTOR DRIVES IN CONVENTIONAL MOVABLE BRIDGE DRIVES

- LOW SPEED HIGH TORQUE (LSHT) HYDRAULIC MOTORS CAN OPERATE AT SPEEDS AS LOW AS 1 RPM

  LOW SPEED MOTORS ALLOW THE ELIMINATION OF MOST INTERMEDIATE GEARING BETWEEN THE MOTOR AND THE RACK PINION.

b. BRAKE CONSIDERATIONS

- ELIMINATION OF INTERMEDIATE GEARING RESULTS IN THE INCREASE IN REQUIRED BRAKE TORQUE
- HYDRAULICALLY RELEASED DISK BRAKE CAN PROVIDE LARGE BRAKING TORQUE AND MOUNT DIRECTLY TO THE LSHT MOTOR
- USING A PRIMARY REDUCER OF ABOUT 40:1 REDUCTION WILL REDUCE BRAKE SIZE TO AVAILABLE COMPONENTS

c. DIFFERENTIAL REDUCER

- LSHT MOTORS CAN BE MOUNTED ON A COMMON INPUT SHAFT OF A PRIMARY DIFFERENTIAL REDUCER IN AN ARRANGEMENT TYPICAL OF ELECTRIC MOTORS

- MOTORS CAN BE SYNCHRONIZED ELECTRONICALLY OR TORQUE BALANCED VIA A COMMON MANIFOLD

d. TANDEM DRIVES

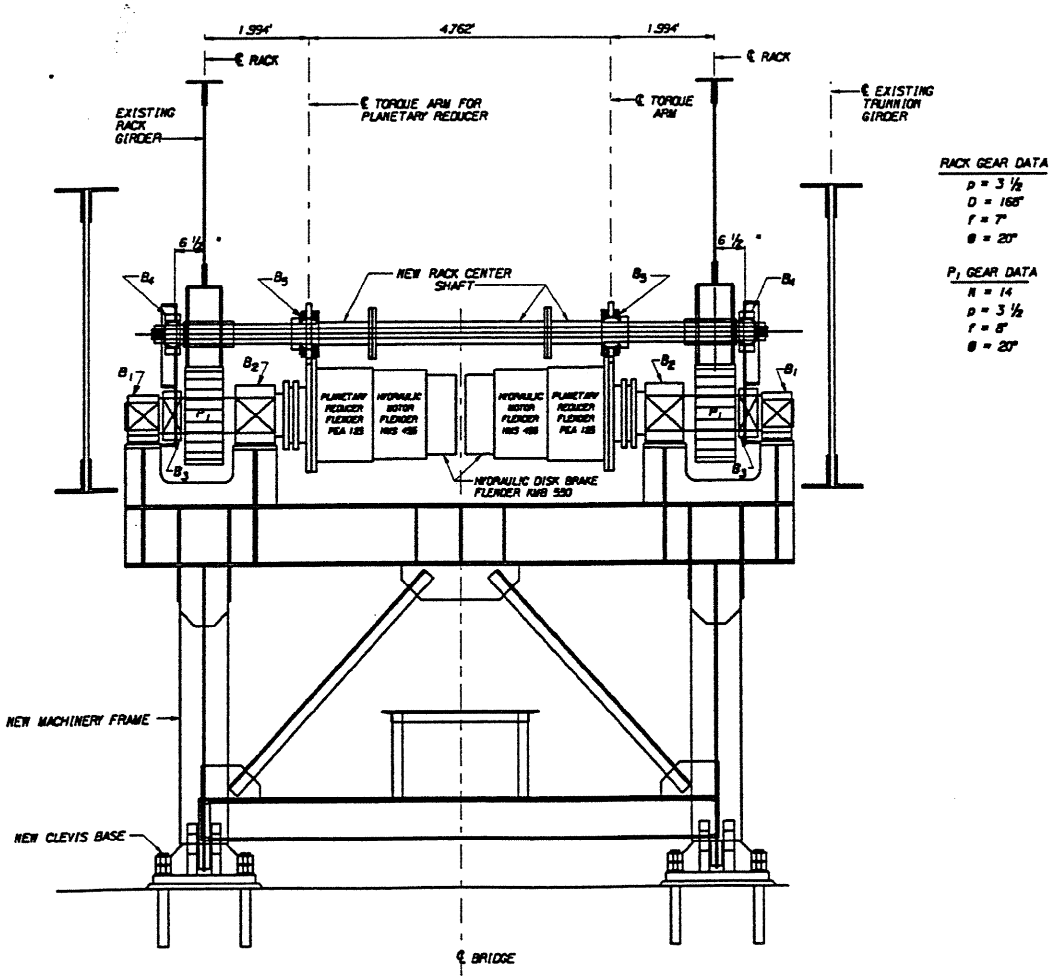
- LSHT MOTORS CAN BE MOUNTED ON INDEPENDENT SHAFTS, EACH DRIVING A SEPARATE PINION

- THE TORQUE APPLIED TO EACH PINION BY THE LSHT MOTORS CAN BE BALANCED VIA A COMMON MANIFOLD

e. FRAME MOUNTED DRIVES WITH LSHT MOTORS

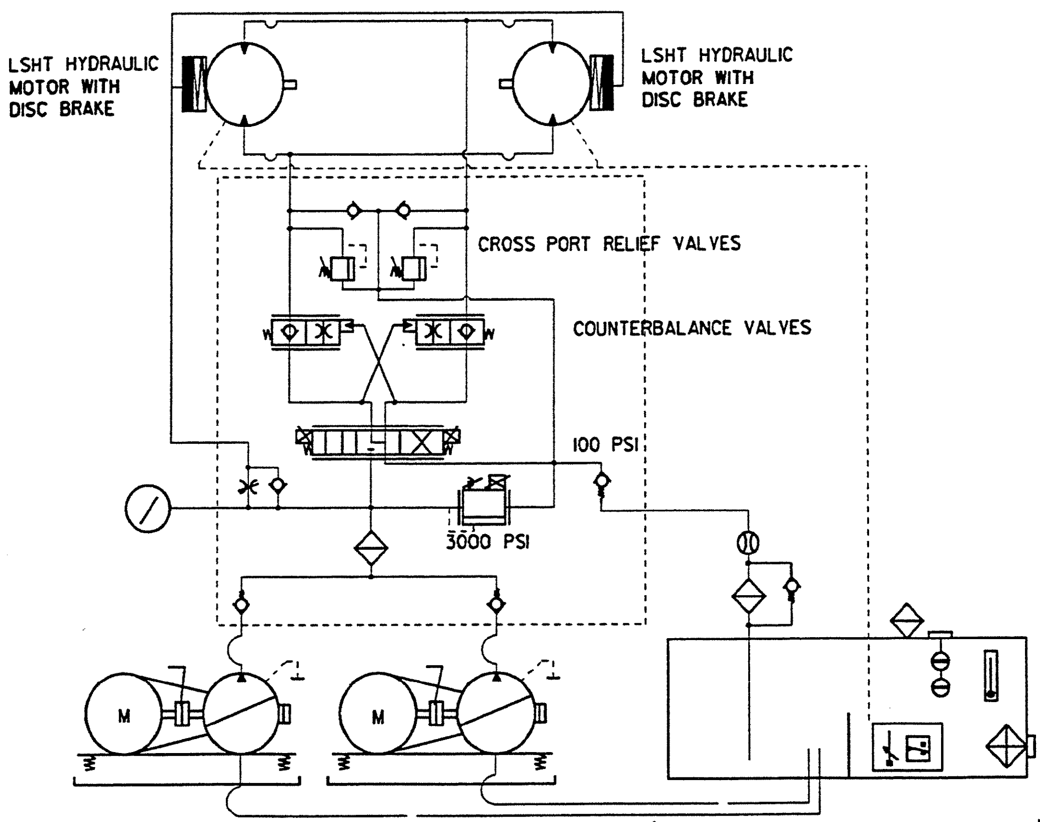
- LSHT MOTORS CAN BE MOUNTED ON A FRAME SIMILAR TO A HOPKINS FRAME WITH EACH MOTOR MOUNTED ON AN INDEPENDENT SHAFT AND DRIVING A SEPARATE PINION

- THE TORQUE APPLIED TO EACH PINION BY THE LSHT MOTORS CAN BE BALANCED VIA A COMMON MANIFOLD



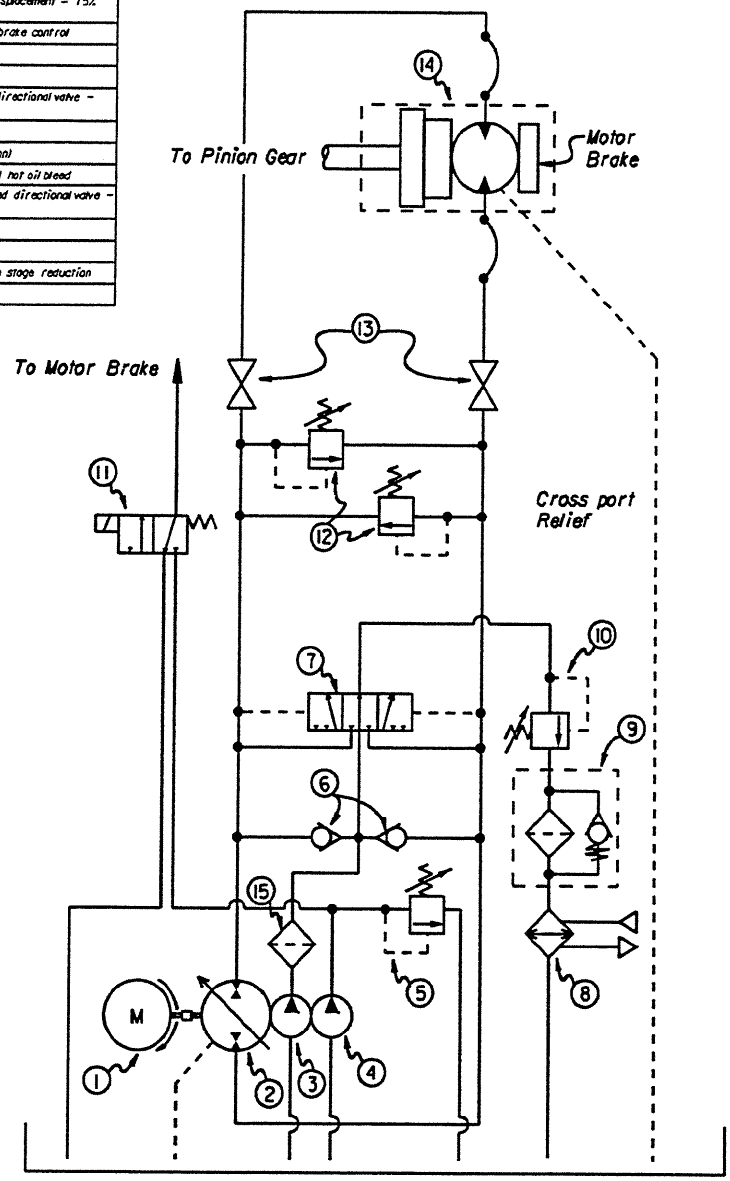
2. BASIC MOTOR CIRCUITS

a. OPEN CIRCUIT



HYDRAULIC MOTOR
BASIC OPEN LOOP CIRCUIT

	BILL OF MATERIALS
ITEM #	DESCRIPTION
1	Electric motor. 480VAC. 3Ø. Induction. TEFC. 1800RPM
2	Variable displacement hydraulic pump 3000 PSI max. with electronic proportional flow control
3	Fixed displacement charge pump-Displacement = 15% displacement of item #2
4	Fixed displacement pump - Motor brake control
5	Relief valve - Motor brake control
6	Check valve - Make up check
7	4 way. 3 position. pilot controlled directional valve - Hot oil bleed
8	Heat exchanger
9	Return filter with bypass (10 micron)
10	Relief valve - for charge pump and hot oil bleed
11	3 way. 2 position. solenoid controlled directional valve - Motor brake control
12	Relief valve - Cross port relief
13	Ball valve - System isolation
14	Hydraulic motor with integral double stage reduction
15	Pressure filter (10 micron)



CLOSED LOOP MOTOR CIRCUIT

B. MOTOR LOADS

1. OPERATING LOADS

LOADS ARE COMPUTED IN THE SAME MANNER AS FOR AN ELECTRIC MOTOR DRIVE. THE DIFFERENCE IN DESIGN IS THAT SEVERAL AREAS ARE NOT ADDRESSED BY AASHTO, INCLUDING STARTING TORQUE & SINGLE MOTOR OPERATION OF TWO MOTOR SYSTEM

COMPARISON AND RECOMMENDED DESIGN CRITERIA

	ELECTRIC MOTORS	HYDRAULIC MOTORS
STARTING TORQUE	180% FLRT	2500 PSI
SINGLE MOTOR OPERATION OF TWO MOTOR SYSTEM	150 % FLRT	3000 PSI

2. BRAKING LOADS

HYDRAULIC MOTOR MUST PROVIDE FOR DYNAMIC BRAKING TORQUE WHICH WILL APPROACH MAXIMUM DRIVING TORQUE

- CLOSED LOOP DRIVES CAN PROVIDE REGENERATIVE DYNAMIC BRAKING
- OPEN LOOP DRIVES REQUIRE A COUNTERBALANCE VALVE FOR DYNAMIC BRAKING
- HEAT GENERATION MUST BE EVALUATED UNDER DYNAMIC BRAKING CONDITIONS

3. HOLDING LOADS

BRAKES ARE TYPICALLY USED TO PROVIDE THE REQUIRED HOLDING TORQUE

4. POWER LOSS

HYDRAULIC MOTORS REACT DIFFERENTLY TO POWER LOSS DEPENDING ON THE MOTOR AND CIRCUIT DESIGN

- SOME MOTORS DO NOT PROVIDE TORQUE WHEN PRESSURE IS REMOVED. THESE SYSTEMS REQUIRE A FAILSAFE BRAKE SYSTEM

- OTHER MOTORS WILL STOP ABRUPTLY IF FLUID IS LOCKED IN THE MOTOR DURING A POWER LOSS. THIS CASE IS SIMILAR TO THE HYDRAULIC CYLINDER CASE.

AN ALTERNATIVE IS TO ADD AN ACCUMULATOR OR OTHER MEANS TO THE CIRCUIT TO PROVIDE A FEW SECONDS OF FLUID POWER TO ALLOW THE SPAN TO DECELERATE BEFORE THE VALVES CLOSE COMPLETELY.

C. POWER REQUIREMENTS

1. AASHTO MAXIMUM PRESSURES

AASHTO ALLOWABLE PRESSURES (2.5.11)

HOLDING LOADS = 3,000 psi

OPERATING LOADS

NORMAL= 2,000 psi

MAXIMUM 1,000 psi

NOTE: COMMENTARY ALLOWS VARIATIONS BASED UPON ENGINEERING JUDGEMENT

BECAUSE BUCKLING IS NOT AN ISSUE WITH MOTORS, AND MOST MOTORS ARE RATED AT 4600 PSI OR MORE, HIGHER OPERATING PRESSURES MAY BE UTILIZED.

D. MACHINERY DESIGN LOADS

AASHTO DOES NOT ADDRESS DESIGN LOADS FOR HYDRAULIC MOTOR SYSTEMS

AASHTO 2.5.4. MACHINERY SHALL BE DESIGNED FOR:

- 150% FLRT OF ELECTRIC MOTORS
- 50% GREATER THAN STARTING OR BREAKDOWN TORQUE
- BRAKING FORCES

RECOMMENDED EQUIVALENT FOR HYDRAULIC MOTORS

- RATED MOTOR TORQUE, T_{max} AT SYSTEM RELIEF VALVE PRESSURE, P_{relief}

$$T_{\max}\ (inch{\cdot}lbs) = \frac{P_{relief}\ (\frac{lbs}{inch^2}) \times DISPLACMENT\ \frac{inch^3}{rev}}{2\pi}$$

- BRAKING FORCES

VI. 11:15 am - 11:45 am
Electronic Control for Modern Hydraulic Systems
Presented by: David Fair

A. Control of Hydraulic Proportional Valves

B. PLC vs. Relay Systems

1. PLC as Main Control System
2. PLC as Monitoring System

C. Operator Interfaces

1. For Bridge Operator
2. For Maintenance

D. Remote Monitoring of Systems

Electronic Control for Bridge Systems

by David Fair

There are two issues on the subject of Electronic Control for Bridge Systems, the Control of Hydraulic Proportional Valves and PLC Systems for Bridge Control, which we will address. The first subject, , deals with the use of amplifier cards to control proportional valves. Why are they necessary and what features do they offer. The second subject, PLC Control for Bridge systems, deals with changes in the PLC field which should be considered when designing new bridge control systems.

Control of Hydraulic Proportional Valves

In hydraulic systems valves are used to control the fluid's flow path, pressure or flow rate. Directional valves control the flow path, pressure relief and regulating valves control pressure and flow control valves regulate the volume of flow. Each of these types of valves comes in two general types. The basic unit is a fixed variety which is mechanically adjusted and, once properly set, generally stays that way for its life. The second variety is an electrically operated unit which controls its parameter based on an 'electrical command'. Most people are very familiar with the industry standard electrically operated 'directional valve.' It is a valve which controls the flow path of hydraulic fluid. It is designed to be ON or OFF. It allows flow or it stops flow. In many cases this is adequate for the control of the a small system or a simple process. In some cases, however, the process requires that the flow be controlled more precisely or be time-variant, either to avoid shock or to provide finer more accurate control of the process. The hydraulic 'proportional' valve has been developed to provide this function. The directional proportional valve has a spool with a 'notched' or tapered opening. The notch acts as variable orifice. With no power on either of the coils, the spool is held in its centered position by springs. By applying power to a coil, a force, proportional to the ***current*** in the coil, is applied to the spool. This force is countered by the opposite spring. Since F=kx (where k is a constant) for a spring, the distance the spool moves is proportional to the current. The size of the orifice and the restriction to flow is controlled with valve shift. The current required to shift a proportional valve it's maximum spool travel is valve dependent but generally is in the range of 600 to 2000 ma with 800 to 1000 ma being the most common.

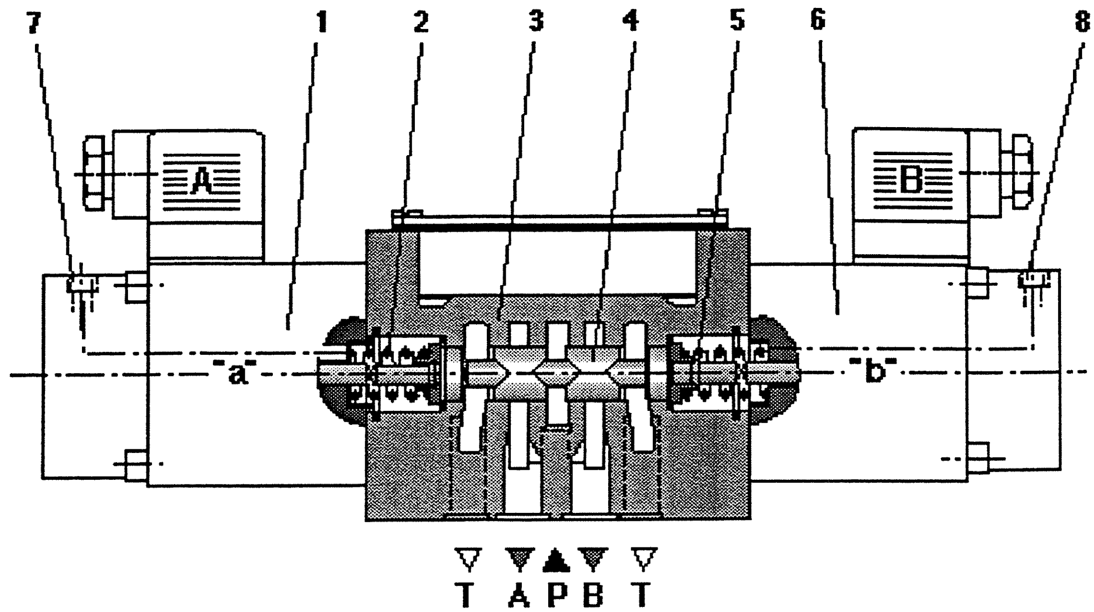


Figure 1 - Typical Proportional Valve

Figure 1 shows a typical proportional valve. The spool (4) is centered by two springs (2, 5) when neither coil (1, 6) is energized. When solenoid 'a' (1) is energized, it pushes on the spool. The 'V notches' are moved into the flow path of P→B and A→T. When the coil is de-energized the spring (5) moves the spool back to the center position.

For optimum performance proportional valves have matching electronic amplifier cards. The amplifier cards provide several basic functions and most also offer a few, additional functions which make a system designers life a little easier. The basic functions provided by amplifier cards are 1) a voltage to current converter, 2) a dither circuit, 3) a 'jump' circuit and 4), for higher performance valves, an LVDT demodulator and a 'closed-loop' spool position system. As was previously noted, the spool's position is proportional to the current in the coils. As a hydraulic system is used, the temperature of the system increases. As a proportional valve is used, the pressure drop across and the flow through the 'notch' in the spool also generates heat. This causes the temperature of the valve's coils to rise. As the temperature rises the coil's resistance changes. If we applied a voltage source to the coil the current would drop as the temperature rose and the force on the spool would decrease changing the spool position. The amplifier card provides a voltage to current converter for the proportional valve. For X volts in it gives Y milliamps out over the entire working temperature range of the valve. The force on the spool and, therefore, the spool position is then independent of temperature. The dither circuit provides the valve with a low amplitude oscillatory signal which keeps the spool oscillating very slightly around its position. This reduces hysteresis and makes it more responsive to command changes. Most proportional valve spools have blocked center positions with an overlap. This means that the spool must be moved 10-15% before the spool's 'notch' is moved into the flow path. If an amplifier card is properly matched to the valve a small input signal, typically 1%, will cause the spool to shift, or 'jump', the 10 - 15% immediately and then to move proportional to the input signal on the remaining 85-90% of its range. The hysteresis of a typical proportional valve is approximately 3%. In higher performance proportional valves this is improved by attaching LVDTs (Linear Variable Differential Transformers) to the spool. This provides position 'feedback' to the amplifier card. Instead of using current to control the valve, the amplifier card controls the position of the spool. Typical hysteresis of these valves is less than 1%.

The options which amplifier cards provide vary by card. Most amplifiers have ramp generators which allow the user to control the rate at which the current to the valve's coil is allowed to change. The ramp time is defined as the time it takes the current to the valve to reach its full output with a 100% step input. For step inputs of less than 100% the time to complete the change is proportionally shorter. Some cards have separate adjustments for both up and down ramps. Some cards have more than one ramp. The maximum ramp rate on most standard amplifier cards is either 1 or 5 seconds. Cards can be modified for longer ramps but since the ramp is generated with an RC timing circuit ramp rates of longer than 60 seconds should be avoided. Amplifier cards may have built in potentiometers and relays to provide 'setpoint' selection for the speed or pressure command. Some cards have a relay dedicated to switching the polarity of the source voltage to the potentiometers. This effectively doubles the setpoints available to you, as long as the same ones are used in both directions. The different commands and the different ramps are selected via a discrete input signal. Most cards will have a differential amplifier to allow the command signal to come from another device, such as a PLC. We prefer to use the differential inputs and connect the cards to the PLC via an analog output card. This allows us to generate the command signal in the PLC which will not vary as components age and is not subject to tampering by unauthorized individuals. There are no 'limits' on the number of setpoints or ramp rates. When an amplifier card is replaced there is no setup procedure to adjust the setpoints or ramp rates.

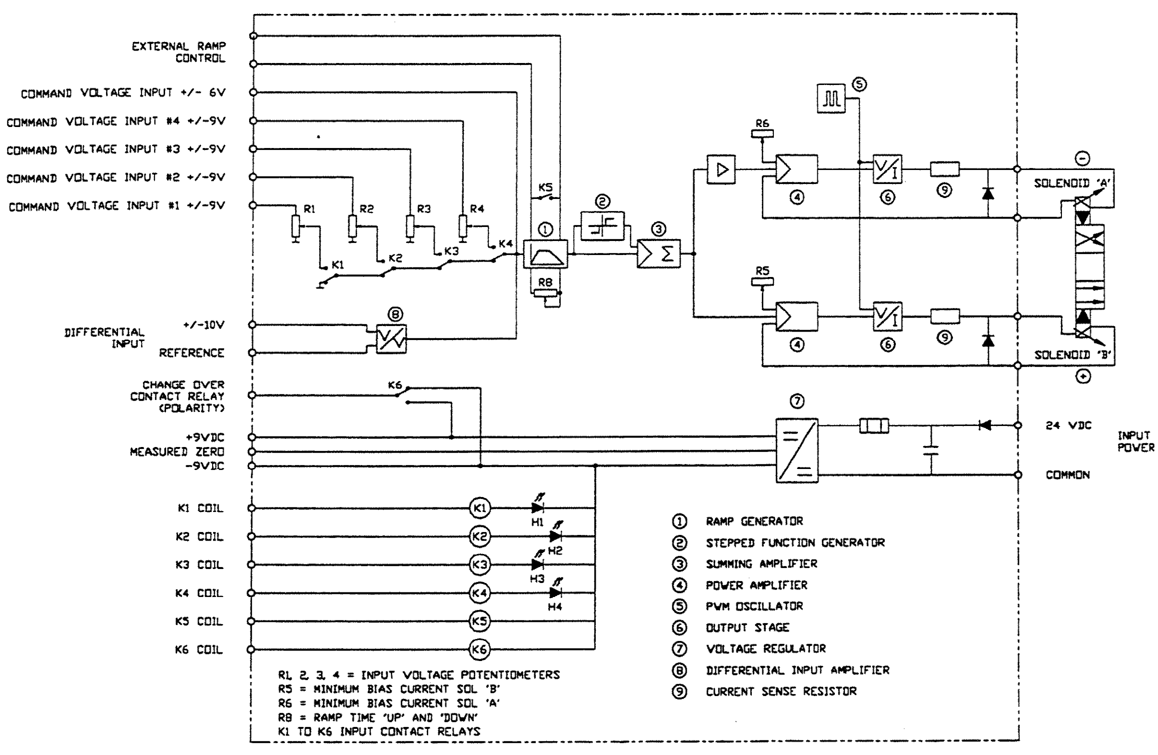


Figure 2 - Typical Amplifier Card

Figure 2 is a block diagram of a typical proportional valve amplifier card. It has all of the features mentioned above. The valve, which is similar to the one shown in Figure 1, does not have spool position feedback. The dither function is provided indirectly by the PWM output stages and is therefore not shown as a block in the diagram. On amplifiers with linear output regulators, there will be a separate oscillator for the dither. Potentiometers R1 through R4 may be used as 'setpoints'. Relays K1 through K4 are used to select the different setpoints. Relay K5 is used to turn the ramping function OFF. Relay K6 may be used to change the polarity of the voltage source to the potentiometers R1 through R4. This provides the same setpoints in both directions.

PLC SYSTEMS for BRIDGE CONTROL

The logic for control systems for industrial processes, including bridges, originally was entirely implemented with relays. These relays had many forms, standard control relays, motorized timing relays, , pneumatic timing relays, more recently, electronic timing relays, and sequencers. They were fixed with voltmeters and a good set of drawings. Pneumatic relays were vulnerable to dirt and temperature changes. Early electronic timing relays were vulnerable to noise and transients. Motorized timing relays were cumbersome and provided few contacts to use. Relays often did not have enough contacts to satisfy the designer, he always ended up a few short. Compromises were a way of life. The designer always had to worry about that 'sneak' circuit, where he forgot that despite the drawing electricity would flow from left to right when the contacts closed.

PLCs were the next step in control systems. They have had their growing pains and have their own unique problems. They are a 'black box' to some maintenance personnel. They require 'special devices or software packages' for programming. Gone, however, are the design compromises. Gone are the limits on relay contacts. Timers are plentiful and very precise. LSI circuitry has improved their reliability by a factor of 10 over the past 15 years. PLCs continually verify the integrity of their hardware and software. They have hardware watchdog timers which will shut down the system should the processor fail. We have found PLCs to be exceptionally reliable. Our standard supplier tests 100% of their cards in a temperature chamber before shipping.

Many of the systems we have supplied on bridge projects have been a mixture of PLC and relay controls. In some cases the PLC was the prime controller and the relays backup. In others the relay systems were the primary controller and the PLC was used for data logging and operator prompting. As a supplier of control systems I cringe when I see the requirement for relay based systems. I know that the system design will take longer than a PLC system. I know compromises will be made and points overlooked. I know the end result will not be what it should be. I strongly believe that the user is doing himself a disservice in requiring relay systems, even those in which a PLC 'monitors' operation. The fear of some users is that the PLC will fail at a critical time or that it is difficult to maintain. There are three areas where PLC hardware and/or software has advanced in recent years which I believe enhance the reliability and maintainability of PLCs and one area of common sense which should be addressed by all PLC system users.

'Redundant' PLC control - Recently 'small' PLCs have come into their own. These units are half the size and a third the price and have 90% of the processing power of their big brothers. Scan times are the same. The smaller PLCs provide an incredible amount of processing power and flexibility in small footprints. This has made a 'redundant' system easier and more cost effective to implement and, therefore, more attractive. I have read several articles from recent conferences on triple redundant systems. Triple redundant systems provide the maximum degree of protection for a failed component and continued operation. However, voting circuits and the hardware for three complete control systems including input devices increases the number of components and the overall complexity of the system tremendously. The increase in the number of components and the design complexity leads to systems which have the potential to be nightmares to implement and maintain. To my knowledge they have really only been implemented in large commercial aircraft and space vehicles. We have provided several systems recently which are redundant systems. They contain a 'backup' PLC system which except for one or two rungs is running the same program as the 'primary' PLC. The backup system monitors a 'heartbeat' signal from the primary PLC and if it fails the backup PLC takes over. While the transfer is not 'bumpless' it is relatively quick. This arrangement provides users with the security of a second, operating PLC. Input devices, which are not redundant in relay systems, are wired to both PLCs. Power is enabled to the outputs of only the controlling PLC.

Man Machine Interfaces (MMI) - Operator interfaces, or MMIs as they are referred to today, and properly designed systems can provide the operator/maintenance person with an incredible amount of information. In addition to the standard operator prompts of 'CLOSE TRAFFIC GATE' and 'START HYDRAULIC SYSTEM', messages can tell the operator 'WAITING FOR LS43, GATE OPEN LS' or 'PUMP/MOTOR 1 TRIPPED ON OVERLOAD'. PLC inputs can monitor outputs and power supplies voltages. Messages can say 'CB1 TRIPPED' or '24V FUSE FAILED'. A repair person can be prompted as to the potential source of the problem before he is dispatched. On more advanced MMI packages, a picture of the bridge or the hydraulic power unit can be graphically displayed. Devices can be shown as they physically exist. Displays can be animated or colors changed to highlight different conditions. More sophisticated packages allow AUTOCAD or similar drawings to be imported and animated. Data can be accumulated in the PLC and displayed on the screen for maintenance personnel. Motor currents, operating pressures and operating times can be monitored and displayed. Limits can be set, trends noted and problems alarmed.

Remote Monitoring/Programming - Present day PLC systems may be accessed from remote locations via modem. They may be accessed by programming packages or the same operator interface packages described in the previous section. Presently SCADA packages exist which will constantly pole up to 256 PLCs on a rotating basis and download information from them. Operational data can be downloaded from a remote location on a periodic basis. Trending functions can be monitored and logged at the central location. With programming software a centralized maintenance location can connect to a remote system and observe exactly what an operator or local maintenance person is seeing and aid them in repairing or fine tuning the system.

System software (common sense) - We have all heard of the KISS principle. I emphasize it in software writing. Programs should be organized in a structured, simple fashion. Interlocks should be placed in the minimum number of places. A combination of 'computer' type instructions for STEPping through the process is combined with the 'standard' relay type instructions. Programs are organized in the following fashion:

- **FAULTS** - Detection of improper operating conditions, temperatures, fluid levels, etc.
- **MODE SELECTION** - Selection of how the system is to work.
- **MOTOR CONTROL** - Arbitrarily placed here. Basically includes the operation of the HPU or any devices which run in support of the process.
- **MANUAL LOGIC** - Usually very simple. Operator devices are interlocked with other operator devices. Interlocks are not implemented here.
- **AUTOMATIC SEQUENCE LOGIC** - Process is broken up into logical STEPs and sequentially numbered. Completion of a STEP advances a STEP counter. The STEP number easily enables us to monitor certain functions for completion. A certain time can be allocated to complete a step. If it is not completed in that time, error messages are easily generated.
- (**ANY OTHER 'MODES' IF REQUIRED**) Same function as the AUTOMATIC sequence.

- **FUNCTION LOGIC AND INTERLOCKING** - Combination of the Manual , Automatic and 'Other' Modes. A 'function' bit, such as PULL NORTHWEST LOCK is generated. All interlocks are placed in these rungs. What is most probably an interlock for MANUAL mode is also an interlock for the AUTOMATIC mode(s). This minimizes the use of interlocks and assures that when we decide to interlock a GATE with a LEAF it is done in all of the appropriate modes. We don't have to scan the program to find where a GATE is used in each mode.
- **SOLENOID/PROCESS OUTPUTS** The FUNCTION bits are used to energize or de-energize the appropriate solenoid valves or process outputs. Outputs are not scattered throughout the program. A single FUNCTION bit is used to energize all of the outputs associated with that function. If two solenoids are required to OPEN a GATE, one FUNCTION bit is used to energize them. Since an interlock bit disables a FUNCTION, we are guaranteed that all of the appropriate outputs are de-energized when the interlock trips.

The above items simplify the job of Maintenance personnel by creating reliable, accessible, user friendly systems which provide diagnostic and operating information to local and remote users.

VII. 11:45 am - 12:15 pm
Power Unit Design and Construction
Presented by: Parveen Gupta

A. Design Criteria

B. Pump & Control Options

C. Thermal Considerations

D. Filtration

E. Manifolds

F. Piping

G. Noise and Vibration

H. Power Unit Layouts

I. Welding / NDE Testing

J. System Pressure

K. Safety Issues

L. Hydraulic Fluids

HYDRAULIC POWER UNIT DESIGN & CONSTRUCTION

by Parveen Gupta

When designing a hydraulic power unit, the following criteria must be considered:

I. Must meet all technical requirements.

II. Components and assembly items must meet life expectancy of approximately 50 years for bridge applications. Must use best available standard, industrial components for replacement and spares reason.

III. Layout should lend itself to easy operation and maintenance.

IV. The unit should be quiet.

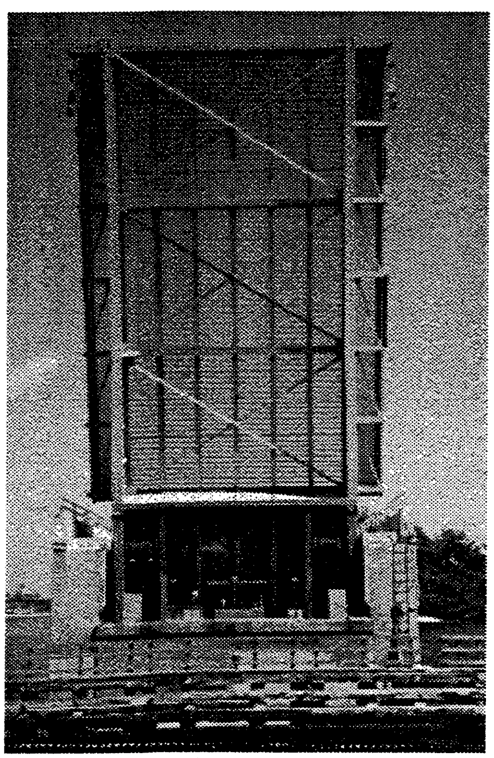
V. Trouble shooting and other diagnostic features should be included.

VI. Provisions for redundancy and power failure should be made.

VII. Since the bridge hydraulics are designed for a 50 year life, latest technologies at the design and construction time must be used.

VIII. Hydraulic Power Unit design must consider the initial power unit manufacturing cost and also the cost to operate and maintain it over its lifetime.

IX. Lastly but most importantly, the design should be as simple as possible.



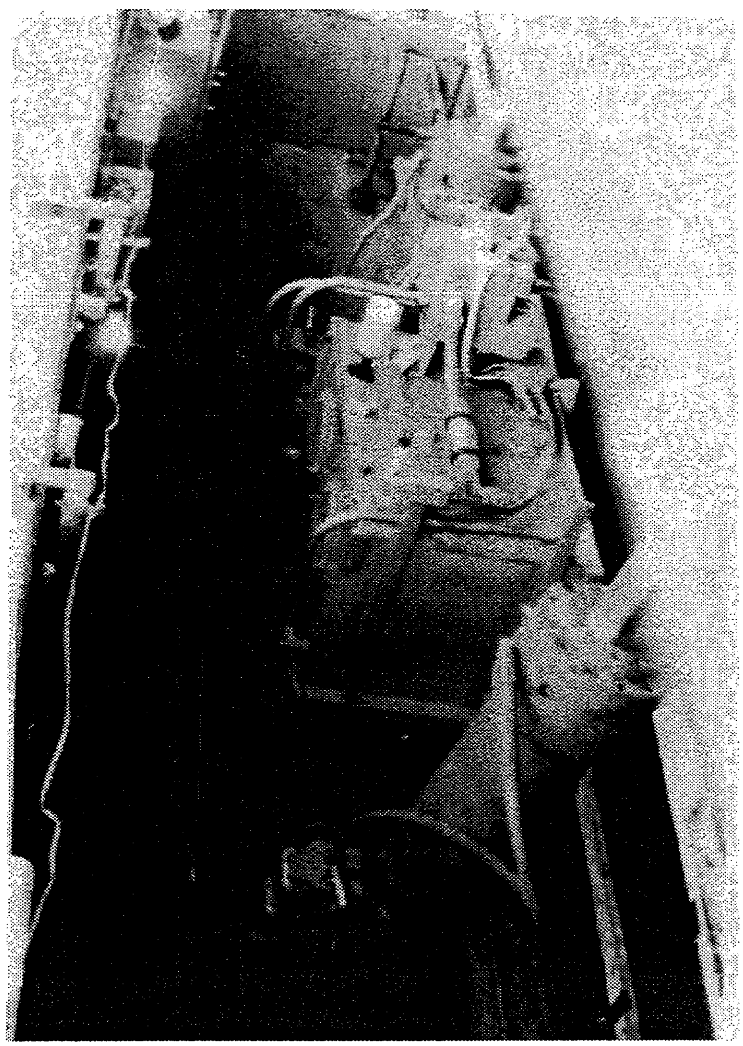
Movable Bridge with Hydraulic Drive

Hydraulic Cylinders on Bridge Bascule

Following are the design criteria for major components of the Hydraulic Power Unit:

Oil Reservoir:

- Tank size and material
- Tank location and power unit layout.
- Inside baffles for cooling, sedimentation and de-aeration
- Down pipe lengths, orientation and diffusion characteristics
- Clean out covers location, size and quantity
- Welding standards and NDT requirements
- Provision for temperature switch, thermometer, level switch, site level gauge (with guards).
- Need and location of drip pans



Hydraulic Power Unit

Pump / Motor Groups:

- Type of pump, pump control and pump pressure ratings
- Electrical motor type and sizing
- Bell housing and coupling vs. foot mounting
- Mounting rails and shock mounts
- Use of noise dampening flanged bell housing
- Use of hose on pump suction, discharge and case drain
- Use of case drain flow meter

Hydraulic System Filtration:

- Use, location, sizing and type of pressure filter
- Return line filtration, type, location and size
- Concept of separate off line filtration
- Problems with suction strainers
- Start with clean / filtered oil
- Regular oil quality checks

Control Valves and Manifolds:

- Use of standard and custom manifolds
- Leakage reduction
- Space savings
- Design features to include troubleshooting and diagnostic checks
- Pressure drop considerations

Piping and Tubing:

- Material and pressure rating
- Type of welding: Socket Weld or Butt Weld
- ASME Standards for welding and NDT requirements
- Chemical cleaning / pickling
- Flushing

ACCESSORIES:

Coolers:

- Normally not needed due to intermittent duty cycle
- If needed can be added in return line, case drain line or separate filter / cooling loop

Heaters:

- Select proper oil for the complete temperature range if possible
- Use relief valve as heater
- If separate heater has to be used, use low watt density heaters to avoid localized oil burning.

Accumulators:

- Size the accumulator properly for the intended purpose
- Should be mounted properly with clamps and rubber gaskets.
- Provision for pressure relief, drain and isolation must be made for each accumulator.

Air Breather:

- Use hygroscopic breathers and maintain them properly. This is for any incoming air.
- Use "kleen" vent type air breathers for changes in oil volume

Pressure Gauges:

- All pressure gauges should have shut off isolation valves and snubbers.
- All gauges should be panel mounted for maximum use. Gauge panels should be mounted at locations which are convenient for proper operation.

Key to Success:

- Hydraulic systems require very little maintenance.
- Normal maintenance is limited to checks of oil quality and visual check of oil level, filter indicators or any visual signs of leakage or abnormal noise etc.

Hydraulics provide a simple, long term, maintenance free solution for bridge operation.