Fluid Power Society

The International Organization for Fluid Power and Motion Control Specialists

Hydraulic Specialist Study Manual

Including Study Guide, Solutions & Pretests

Manual # 410—12/15/04

FLUID POWER SOCIETY INTERNATIONAL

P.O. Box 1420 ❖ Cherry Hill ❖ New Jersey ❖ 08030 ❖ phone: 800-303-8520 ❖fax: 856-424-9248 ❖ email: askus@ifps.org

Safety is Everyone's Responsibility

As equipment is improved and its capabilities are expanded, the knowledge to operate and maintain it increases. Individuals must constantly upgrade their skills and concerns for safety.

This review manual suggests several safety guidelines and rules. **It obviously cannot cover every situation and is not intended to do so.** Safety must be of primary consideration at all times. Each job, regardless of the type of work involved, presents problems that require special alertness and good judgment on your part. In addition, you must comply with the rules and requirements established by the particular site where the work is being performed.

In all cases, each individual and employer shall review the safety hazards present and establish additional practices as needed to minimize them. It is your obligation to work safely and to correct unsafe acts, practices, and/or conditions for the protection of yourself and others. **It is extremely important that you understand how each task is to be performed in a safe manner, and if you don not know how to do so, stop and ask how to perform the task before you being the work.**

Training, including the information in this document, for operators, maintenance personnel, assemblers, foreman, engineers, and other individual working with pressurized fluids, is highly recommended.

Fluid Injections - Fine streams of escaping pressurized fluid can penetrate skin and thus enter the human body. These fluid injections may cause severe tissue damage, gangrene, and loss of limb.

Consider various means to reduce the risk of fluid injections, particularly in areas normally occupied by operators and/or service personnel. Consider careful routing, adjacent components, warnings, guards, shields, and training programs.

Learn about the dynamic forces that act on equipment as it is operated. Relieve pressure before disconnecting hydraulic and other lines. Tighten all connectors before pressurizing the system. Avoid contact with escaping fluids. Treat all leaks as though they are pressurized and hot enough to burn human skin. Never use any part of your body to check for leaks. Use cardboard instead.

If a fluid injection accident occurs, see a doctor immediately. **Do not delay treatment or treat the injection as a simple cut.**

Any fluid injected into the skin must be surgically removed within a few hours or gangrene may result. Doctors unfamiliar with fluid injection injuries should consult a knowledgeable medical source.

Whipping Hoses - If a pressurized hose/tube assembly blows apart, the fittings can be thrown off at high speed, and the loose hose can flail or whip with great force. This is particularly true with compressible fluid systems, such as compressed air systems. Where the risk exists, consider the use of guards and restraints to protect against injury.

Burns from Conveyed Fluids - Fluid power media may reach temperatures that can burn human skin. If there is a risk of burns from escaping fluid, consider the use of guards and shields to prevent injury, particularly in areas occupied by operators.

Fire and Explosion from Conveyed Fluids - Most fluid power media, including fire resistant hydraulic fluids, will burn under certain conditions. As fluid escapes from a pressurized system, a mist or fine spray may be formed, The fluid may then flash or explode upon contact with an ignition force. Consider the use of guarding and routing of fluid conductors to minimize the risk of combustion.

Fire and Explosion from Static-Electric Discharge - Fluid passing through fluid conductors can generate static electricity, resulting in a static electricity discharge. This may create sparks that can ignite system fluids or gases in the surrounding atmosphere. When the potential of this hazard exists, select fluid conductors specifically designed to carry the static electricity charge to ground, thereby reducing the risk of injury or damage.

Electric Shock and High Amperage Discharge - Electrocution could occur if hydraulic tubing conducts electricity to a person. In the case of high amperage, tubing could short the electricity to ground, which in turn could create very high fluid temperatures (up to 5000° C, 9000° F). Routing electrical wires in contact with tubing or hoses is not recommended; electrical wiring and hydraulic lines should be isolated by being separated and securely fastened to avoid contact.

Mechanisms Controlled by Fluid Power - Mechanisms controlled by fluid in tubing and hoses may become hazardous when the tube or hose fails. For example, when a tube or a hose fails or has a catastrophic failure, objects supported by the pressurized fluid may fail. Vehicles or machines may loose their breaks and/or steering, which causes loss of control.

FLUID POWER SOCIETY

Mailing: P.O. Box 1420, Cherry Hill, NJ 08034-0054
Shipping: 1930 East Marlton Pike, Suite A-2, Cherry Hill, NJ 08003-2142
Phone: 856-424-8998 • Fax: 856-424-9248
E-mail: askus@ifps.org • Web: http://www.ifps.org

 Manual #401 - 7/8/04

Hydraulic Specialist Certification Study Guide

Foreword

This study guide has been written for candidates who wish to prepare for the Hydraulic Specialist Certification exam. It contains numbered outcomes, from which test items on the exam were written, a discussion of the related subject matter with illustrations, references for additional study, and review questions. While the study guide covers the basics of the exam, additional reading of the references is recommended.

The outcomes and review questions are intended to focus attention on a representative sample of the subject matter addressed by the exam. This does not mean that the study guide will teach the test. Rather, the study guide is to be used as a self-study course, or an instructional course if a Review Training Seminar is available, to address representative subject matter covered by the exam. Both the exam questions and review questions have been written from the same outcomes. To this extent, if the candidate understands the subject matter given here and can answer the review questions correctly, he or she should be prepared to take the Hydraulic Specialist Certification exam.

The U.S. Government Federal Occupational Code defines the special skills and knowledge required by Fluid Power Specialists as follows:

"Fluid Power applications engineer, Fluid Power sales representative, Fluid Power consultant. Analyzes power transmission and motion control situations and designs appropriate hydraulic or pneumatic systems to perform required tasks. Selects appropriate components, designs or modifies complete circuit, prepares Bills of Materials, and specifies fluids, prime movers and appropriate fluid conductors. Troubleshoots non-performing systems to determine necessary repairs. Also, designs appropriate instrumentation and control systems (hydraulic, pneumatic, and electronic) for industrial and mobile machinery. May become involved in sales activities, product warranty claims, and evaluation of prototype machinery. May become involved in instructional activities for Fluid Power Technicians and Fluid Power Mechanics concerning principles of hydraulics and pneumatics as well as the operation and maintenance of particular machines."

Based upon this description, the Hydraulic Specialist must demonstrate expertise in the skill areas, as well as knowledge, comprehension and application of various principles addressed in this study guide. The study guide follows a simple format that uses outcomes and review questions to focus attention on what is important. If a candidate can master the outcomes by understanding the technical information and answering the review questions correctly, he or she should be able to achieve a passing score on the examination, and the honor of becoming an Hydraulic Specialist.

Disclaimer

The Fluid Power Society (FPS) has attempted to verify the formulas, calculations, and information contained in this publication. However, the FPS disclaims any warranty, expressed or implied,of the fitness of any circuit, data, or information discussed in this publication for a particular application. Whenever the reader intends to use any of the information contained in this publication, the reader should independently verify that the information is valid and applicable to the application. The FPS neither endorses/recommends, nor disapproves of, any brand name or particular product use by virtue of its inclusion in this publication. The FPS has obtained the data contained within this publication from generally accepted engineering texts, catalog data from various manufacturers, and other sources. The FPS does not warrant any of this information in its application to a particular application. The FPS welcomes additional data for use in future revisions to this and other FPS publications.

Credits

The Fluid Power Society wishes to thank several professionals who developed and reviewed this manuscript. The guidance and support of this project has helped advance our industry and provided another benchmark for Fluid Power.

Manual Development Team

Probir Chatterjea	Prodesigner
Don DeRose	Fluid Power Society
Jim Fischer	Minnesota Com. & Tech. College
Ray Hanley	Fluid Power Society
Brad Jensen	Industrial Training Zone
Ernie Parker	Hennepin Technical College
Tim Petrishen	Altec Industries
John Seim	Alexandria Technical College
Bob Sheaf	CFC-Solar
Neil Skoog	CFC-Solar
Jim Sullivan	Southern Illinois University
Bob Yund	Hennepin Technical College

Certification Review Committee

Don DeRose, Fluid Power Society
Jim Fischer, Minnesota West Comm. & Tech. College
Clayton Fryer, Norgren
John Groot, The Knotts Company
Ray Hanley, Fluid Power Society
Ernie Parker, Hennepin Technical College
Paul Prass, Fluid Power Society
John Seim, Alexandria Technical College
Bob Sheaf, Certified Fluid Consultants
Jim Sullivan, Southern Illinois University
Tom Wanke, Milwaukee School of Engineering
Terry Ward, The Knotts Company
Robert Yund, Hennipin Technical College

Design and Layout

Kevin Bailey, Innovative Designs & Publishing, Inc.

Contents

Job Responsibility 1.0: Apply Hydraulic Circuits to Perform Desired Tasks

Job Responsibility 2.0: Analyze Loads and Motion

Job Responsibility 3.0: Select Components for Hydraulic Applications

Contents

Contents

Contents

List of Figures

Apply Hydraulic Circuits to Perform Desired Tasks

List of Figures-

Job Responsibility 2.0: Analyze Loads and Motion

Job Responsibility 3.0: Select Components for Hydraulic Applications

List of Figures

Prepare Bills of Material and Schematics

Recommend Fluid, Fluid Conductors, and Fluid Filtration

Analyze and Troubleshoot Hydraulic Systems

Interface Hydraulic Instrumentation and Control Systems

Note: *Many of the circuits shown do not illustrate full and complete circuits. The circuits shown are to be used for illustrative purposes only.*

List of Tables

Reference Equations

FPS Hydraulic Specialist Certification			
Eq. No.	**Page**	**Long Form**	**Short Form**
1-1	1-35	$Velocity_{in/min} = (Q_{gpm} \times 231 \text{ cu-in/gal}) / Area_{sq\text{-}in}$	$V = (Q \times 231) / A$
1-2	1-36	$Force_{lbs} = Pressure_{psi} \times Area_{sq\text{-}in}$	$F = P \times A$
1-3	1-36	$Area_{sq\text{-}in} = Diameter_{in}^2 \times 0.7854$	$A = D^2 \times 0.7854$
2-1	2-7	$Velocity_{in/min} = Stroke_{in} \times (60_{sec/min} / Time_{sec})$	$V = S \times (60 / T)$
2-2	2-7	$Q_{gpm} = (Velocity_{in/min} \times Area_{sq\text{-}in}) / 231_{cu\text{-}in/gal}$	$Q = (V \times A) / 231$
2-3	2-10	$Effort_{lbs} = Load_{lbs}$ / # of Load Bearing Cables	E = L / #LBC
2-4	2-12	$Load_{lbs} \times Load\ Distance_{ft} = Effort_{lbs} \times Effort\ Distance_{ft}$	$L \times LD = E \times ED$
2-5	2-13	Mechanical Advantage =	
		$Total\ Rod\ Length_{in} / Supported\ Rod\ Length_{in}$	MA = TRL / SRL
2-6	2-13	$Reaction\ Force_{lbs} = Mechanical\ Advantage \times Side\ Load_{lbs}$	$RF = MA \times SL$
3-1	3-3	$Force_{lbs} = Coefficient\ of\ Friction \times Load_{lbs}$	$F = C_f \times L$
3-2	3-5	$Total\ Incline\ Force_{lbs} =$	TIF =
		$(Coefficient\ of\ Friction \times Load_{lbs} \times \cos\theta) \pm (\sin\theta \times Load_{lbs})$	$(C_f \times L \times \cos\theta) \pm (\sin\theta \times L)$
3-3	3-7	$Toggle\ Force_{lbs} = Cylinder\ Force_{lbs} \times \tan\theta$	$TF = CF \times \tan\theta$
		$(Cylinder\ Force_{lbs} \times Remaining\ Stroke\ Length_{in}) \times \tan\theta$	
3-4	3-12	Speed Ratio = Output Shaft / Input Shaft	SR = OS / IS
3-5	3-14	$Torque_{lb\text{-}in} = Force_{lbs} \times Radius_{in}$	$T = F \times R$
3-6	3-15	$Work_{ft\text{-}lbs\ or\ in\text{-}lbs} = Force_{lbs} \times Distance_{ft\ or\ in}$	$W = F \times D$
3-7	3-15	Power = Work / Time	P = W / T
3-8	3-15	$HP = (T_{lb\text{-}ft} \times N_{rpm}) / 5252$	
		$HP = (T_{lb\text{-}in} \times N_{rpm}) / 63025$	
		$HP = (T_{N\text{-}m} \times N_{rpm}) / 7124$	
3-9	3-15	$Hydraulic\ Motor\ Torque_{lb\text{-}in} = (Displacement_{cipr} \times Pressure_{psi}) / 2\pi$	$HMT = (D \times P) / 6.28$

Reference Equations

FPS Hydraulic Specialist Certification			
Eq. No.	**Page**	**Long Form**	**Short Form**
3-10	3-17	$Q_{gpm} = (\text{Displacement}_{cipr} \times N_{rpm}) / 231_{cu\text{-}in/gal}$	$Q = (D \times N) / 231$
3-11	3-19	$Q_{gpm} = C_V \sqrt{\text{PressureDrop}_{psi} / \text{FluidSpecificGravity}}$	$Q = C_V \sqrt{PSID / Sg}$
3-12	3-20	$\text{Spring Force}_{lbs} = \text{Pressure}_{psi} \times \text{Area}_{sq\text{-}in}$	$SF = P \times A$
3-13	3-22	$(P_A \times A_A) + (P_B \times A_B) = (P_F \times A_F) + F_S$	
3-14	3-32	$\text{Volume}_{gal} =$ $(\text{Length}_{in} \times \text{Width}_{in} \times \text{Height}_{in} \times \text{Percent Full}_{decimal}) / 231$ cu-in/gal	$V = (L \times W \times H \times \%) / 231$
3-15	3-32	$HP = 0.001 \times \Delta T_{°F} \times \text{Vertical Reservoir Area}_{sq\text{-}ft}$	$HP = 0.001 \times \Delta T \times A$
3-16	3-34	$P_1 \times V_1 \times T_2 = P_2 \times V_2 \times T_1$ (The Perfect Gas Law)	
3-17	3-35	$P_1 \times V_1 = P_2 \times V_2 = P_3 \times V_3$	
3-18	3-37	$\text{Energy Loss}_{Btu/hr} = 2545 \times T_{hours} \times (HP_{input} - HP_{output})$ $\text{Energy Loss}_{Joules} = 746 \times T_{seconds} \times (HP_{input} - HP_{output})$	
3-19	3-37	$HP_{fluid} = (\text{Pressure}_{psi} \times \text{Flow}_{gpm}) / 1714$	$HP = (P \times Q) / 1714$
3-20	3-37	$HP_{input} = (\text{Pressure}_{psi} \times \text{Flow}_{gpm}) / (1714 \times \text{Efficiency}_{decimal})$	$HP = (P \times Q) / (1714 \times E)$
3-21	3-39	$V_{ft/sec} = (Q_{gpm} \times 0.3208) / \text{Area}_{sq\text{-}in}$	$V = (Q \times 0.3208) / A$
3-22	3-40	$\text{Burst Pressure}_{psi} = \text{Working Pressure}_{psi} \times \text{Safety Factor}$	$BP = WP \times SF$
3-23	3-40	$\text{Burst Pressure}_{psi} =$ $(2 \times \text{Wall Thickness}_{inches} \times \text{Material Tensile Strength}_{psi}) / \text{Tube OD}_{inches}$	$BP = (2 \times WT \times TS) / OD$
4-1	4-3	$\text{Motor Speed}_{rpm} =$ $\text{Pump Speed}_{rpm} \times (\text{Pump Displacement}_{Cipr} /$ $\text{Motor Displacement}_{cipr})$	$MN = PN \times (PD / MD)$
5-1	5-8	Beta Ratio = (# of particles up stream / # of particles down stream of filter)	
5-2	5-8	Efficiency = (# of particles removed / # of particles introduced) x 100	
6-1	6-3	$N_{rpm} = (Q_{gpm} \times 231 \text{ cu-in/gal}) / \text{Displacement}_{cipr}$	$N = (Q \times 231) / D$

Job Responsibility 1.0: Apply Hydraulic Circuits to Perform Desired Tasks

Task 1.1: Read hydraulic symbols.

Outcome 1.1.1:

Recognize basic hydraulic symbols.

The easiest method to learn hydraulic symbols is to keep a reference handy, and refer to it from time to time. The accepted reference to symbols is ISO 1219-1, but companies worldwide use a number of other standards, though all of them share fundamental similarities. To compound the problem of recognizing symbols, some manufacturers market proprietary components, some of which are assembled in manifolds that have slightly different configurations from those shown in standard references. Thus, having a number of standards is also worthwhile, and learning the symbols that major manufacturers use would be handy as well. Furthermore, symbols have evolved through the years. No matter which symbols that are used, what must be remembered is that it is the specialist's responsibility to identify the function of various components from the symbols that are shown on the circuit diagram without regard to their origin.

The most basic symbol is the line. Lines are drawn solid, solid with an arrow, dashed, and as center lines. Solid lines are used to indicate flow. The width of all flow lines should be equal, but the width of the line does not alter the meaning of the symbol. If there is an arrow in the solid line it indicates the direction of flow.

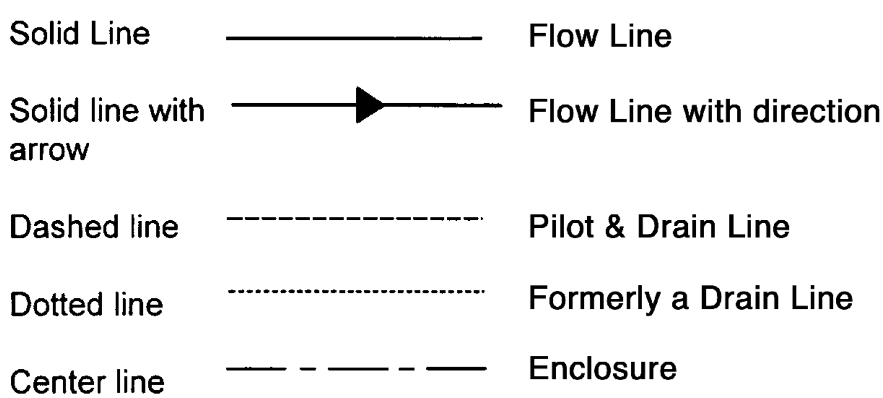


Figure 1-1: Lines Used in Schematics

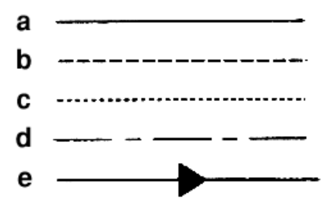
A dashed line is used for pilot pressure lines to indicate a pilot source used to actuate a pressure control or a directional control valve. A hollow triangle along a flow line denotes pneumatic flow while a solid triangle denotes liquid flow. The dotted line was used within the ANSI specification to indicate a drain line from a valve or a case drain line. The ISO 1219-1 standard does not use dotted lines for drain lines. Instead, the same dashed line used for pilot pressure lines is now also used for drain lines.

A center line is used to circumscribe an enclosure. An enclosure combines two or more other symbols into an assembly.

Job Responsibility 1.0: Apply Hydraulic Circuits to Perform Desired Tasks

Review 1.1.1.1: Which of the following symbols illustrates a pilot pressure line?

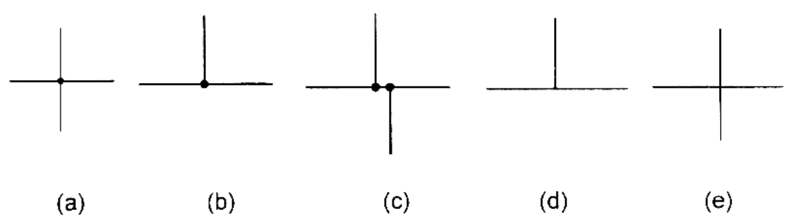
a. a
b. b
c. c
d. d
e. e



It is important to know when hydraulic lines join and when they are crossing. Per ISO 1219-1, lines that join are shown with a dot at the juncture and lines that are not joined cross each other without having a dot at the intersection. However, through the years other methods of showing juncture have been used. On some schematics, lines that joined didn't always have a dot at the juncture while lines that crossed were drawn with a semicircular loop where one line crossed the other line. It is also customary, though not required, that lines run horizontally or vertically.

Review 1.1.1.2: Per ISO 1219-1, which one of the following symbols represents one hydraulic line crossing another hydraulic line?

a. a
b. b
c. c
d. d
e. e



An arrow drawn through a symbol at approximately a 45° angle indicates that the component can be adjusted or varied. This is true for components such as pressure control valves, flow control valves, pumps, motors, and proportional solenoids. In the case of pressure control valves, the 45° line is drawn through the spring that brings the valve to the normal, at rest, position. For flow control valves, the 45° line indicates the flow through the valve can be adjusted. In the case of a 45° line through a pump or motor symbol, two pieces of information are given by the angled line. First, it indicates the displacement can be adjusted. Given this information, it also means the pump or motor must be a piston or vane type, since the displacement of gear pumps and gear motors cannot be changed.

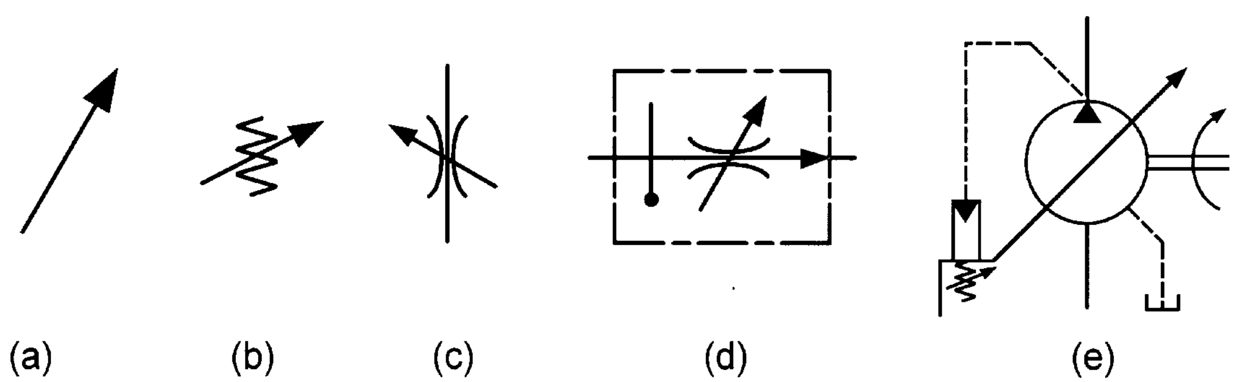


Figure 1-2: Adjustable Symbols and Components

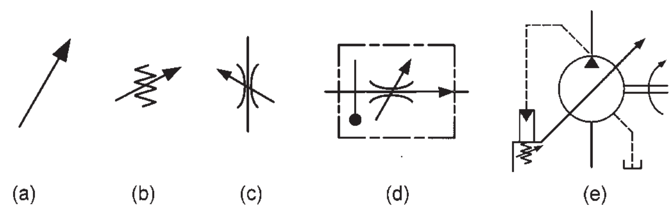
Figure 1-2 illustrates the symbols for five adjustable components. Item (a), an arrow at a 45° angle, is the general symbol indicating adjustability. Item (b) is the simplified symbol for an adjustable spring, which is typically found in a pressure control valve. Item (c) is the symbol for an adjustable orifice. Item (d) is the symbol for a pressure and temperature compensated flow control valve. Finally, item (e) is the ISO symbol for a pressure compensated pump.

In the pressure compensated flow control valve, the vertical arrow and the vertical thermometer symbols indicate the valve is both pressure and temperature compensated. This means that as the fluid warms up and becomes less viscous (thinner), or the pressure upstream or downstream of the valve changes, the flow control valve will continue to meter fluid through the valve at a constant flow rate.

In the case of the pressure compensated pump, the pump compensator will change the displacement of the pump as a function of the system pressure. When the pressure setting of the compensator is reached, the pump will destroke, thereby reducing its delivery. The pump will deliver only enough flow to maintain the desired system pressure. It is extremely important to understand that pumps do not produce pressure. Pumps deliver flow. Pressure is resistance to flow; pressure is a function of the load resistance.

Review 1.1.1.3: Which symbol indicates the component is varied only as a function of pressure?

a. a
b. b
c. c
d. d
e. e



Pumps and Hydraulic Motors

The basic symbol for pumps and motors is a circle with two lines (ports) extending outward from opposite sides of the circle. In Figure 1-3a, the component is a unidirectional, fixed displacement pump. The solid triangle pointed outward symbolizes "energy out." At the 3 o'clock position is the input shaft and a curved arrow indicating the shaft rotates in a clockwise direction. This is often termed to be right hand rotation. When facing the shaft the inlet line is drawn at the 6 o'clock position and the outlet line is drawn at the 12 o'clock position.

Figure 1-3b shows the symbol for a bidirectional, manually adjustable variable volume pump. The arrow intersecting the pump at a 45° angle indicates the displacemnet of the pump is variable. The rectangle attached to the lower left end of the arrow indicates the pump is manually adjusted. The two solid triangles indicate that this pump can provide flow out of either port. This implies then that this is a piston pump. By adjusting the swashplate position, the pump will provide flow out of either port. Variable volume vane pumps are not generally constructed to provide "two sides of center" flow. Two side of center pumps are most commonly used in hydrostatic transmissions (pump-motor combinations). Also shown is a dashed line that is connected to the symbol for a reservoir, indicating this pump has a case drain line.

Figure 1-3c shows the symbol for a pressure compensated pump. The dashed line connected at the outlet port of the pump indicates system pressure is directed to the pump compensator which is symbolized by the rectangular box that contains a solid triangle. The triangle is pointed inward to the rectangle indicating “energy in.” The adjustable spring symbolizes the adjustment mechanism of the compensator.

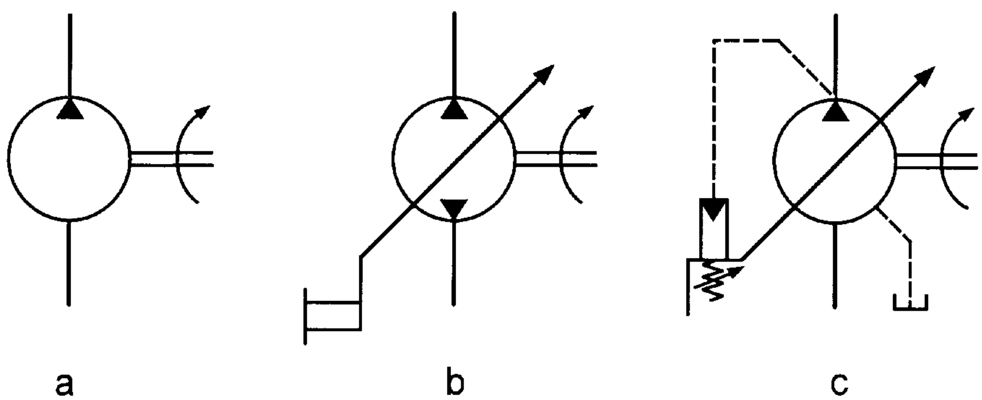


Figure 1-3: Hydraulic Pump Symbols

The basic symbol for a motor is a circle with two port lines on opposite sides and a solid triangle pointing inward at one or both of the ports. The triangle(s), pointing inward, symbolizes “energy in.” In Figure 1-4a shows the symbol for a unidirectional, fixed displacement motor, as there is only one triangle in the circle. Figure 1-4b shows a manually adjustable, variable volume, unidirectional motor. Figure 1-4c shows the symbol for a fixed displacement bidirectional motor, while Figure 1-4d shows the symbol for a manually adjustable, variable volume, bidirectional motor. The two variable volume motors include case drain lines, which are drawn the in the same way case drains for pumps are drawn. Motors usually are of the fixed displacement type, but some applications use variable displacement motors, in which case there will be a line at 45° through the symbol. The bidirectional motors have shaft rotation arrows with arrowheads on both ends of the arrow shaft. Thus, both the shaft rotation arrow and the double triangles indicate bidirectional capability.

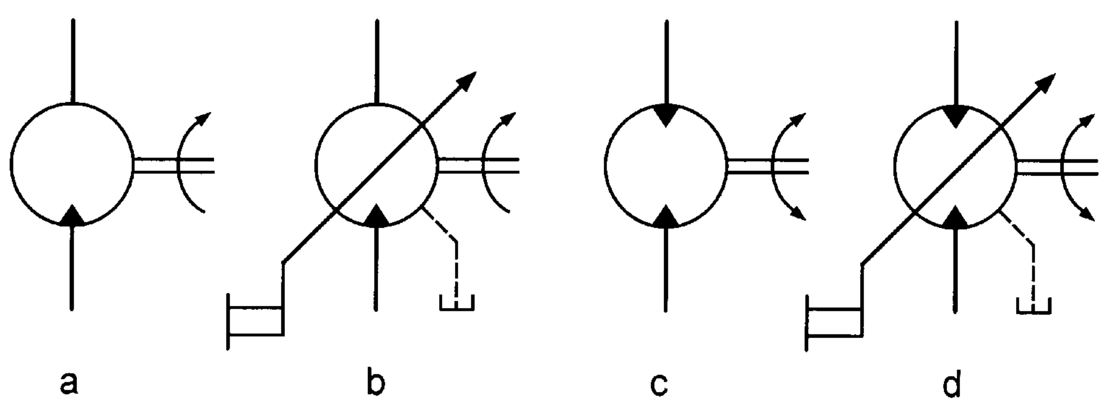


Figure 1-4: Hydraulic Motor Symbols

Hydraulic pump-motor combination symbols are shown with arrows at the port lines pointing inward as well as outward. In Figure 1-5(a), notice that the top arrow points outward, indicting the component is a pump, while a second arrow at the bottom points inward, indicating the component also operates as a motor. With the addition the rotation arc at the shaft with one arrow, the complete symbol represents a variable displacement component that operates both as a pump and motor with one direction of flow. Figure 1-5(b) shows a similar symbol, except that the arc at the shaft has two arrows, and both arrows inside the circle are at the top. This means that the variable displacement component operates in one direction as a pump, and in the other as a motor.

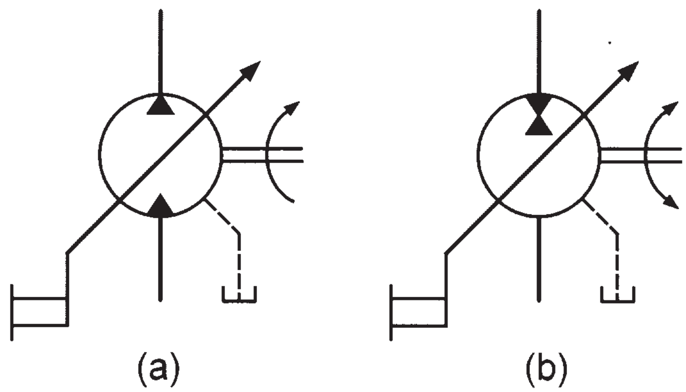
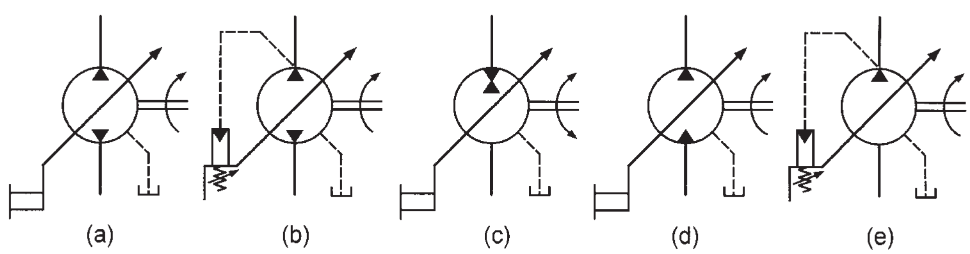


Figure 1-5: Hydraulic Pump-Motor Combination Symbols

Review 1.1.1.4: Which symbol indicates the hydraulic component operates in one direction as a pump, and in the other as a motor?

a. a
b. b
c. c
d. d
e. e



Linear Actuators

Linear output devices include hydraulic cylinders, rams, cushioning devices, servo positioners, and pressure intensifiers. Hydraulic cylinders are available in single acting and double acting types. Single acting cylinders normally have a single rod. Double acting cylinders come with a single rod (SR) or double rod (DR). Double acting, double rod, cylinders provide equal force and velocity for a given pressure and flow rate, if both rods are the same diameter, because the annular areas on both sides of the piston are equal. Double rod cylinders used for steering applications provide equal movement for the same steering wheel rotation in either direction.

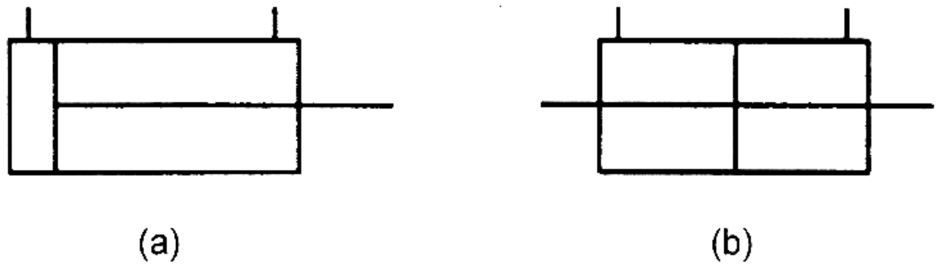


Figure 1-6: Single and Double Rod Cylinder Symbols

Cushions are used in cylinders to decelerate the movement of the piston and rod as a means to limit mechanical shock. Cushions may be installed in either or both ends of a cylinder. Fixed cushions consist of metering orifices in the head and cap ends of the cylinder to decelerate the cylinder rod. Adjustable cushions commonly consist of variable orifices that are adjusted by turning a threaded screw. When a cylinder cap is equipped with a cushion, a free flow check valve is installed in order to allow unrestricted incoming flow to the piston. The cylinder symbolized by Figure 1-7b has an adjustable rod end cushion which is operative only at the end of the extension stroke. High mass and/or high velocity loads can cause severe pressure intensification to occur in cushioned cylinders. Means other than cushions should be used to decelerate these types of loads.

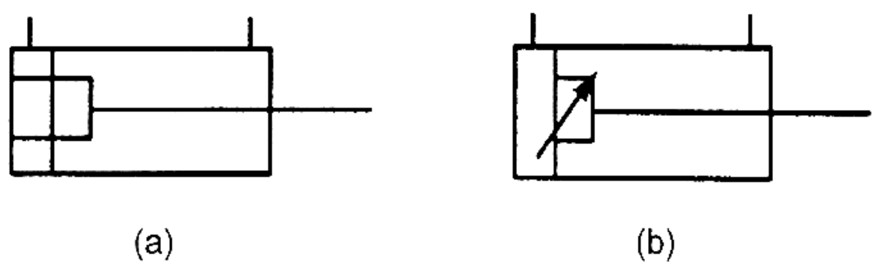


Figure 1-7: Fixed and Adjustable Cushions

A number of linear device symbols are sometimes misidentified because they are infrequently used. Figure 1-8a illustrates a hydraulic ram symbol while Figure 1-8b shows the symbol for a pressure intensifier symbol. A discrete positioner, shown in Figure 1-8c, consists of two or more double acting cylinders connected in series.

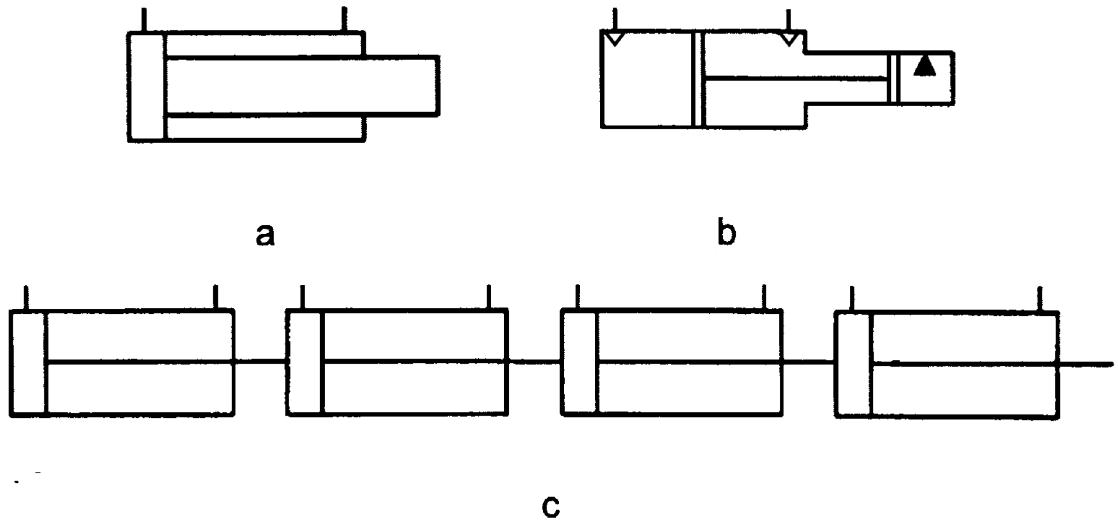


Figure 1-8: Hydraulic Ram, Pressure Intensifier, and Discrete Positioner Symbols

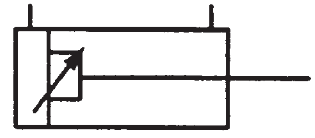
A hydraulic ram is a cylinder in which the rod diameter is equal to or greater than 50% of the bore diameter. Hydraulic rams may be single or double acting, but their main application is to exert the working force while extending. Gravity or some other means are used to retract the rod of a single acting ram. If the ram is double acting, hydraulic oil is used to retract the rod, but retraction force is small in comparison to extension force.

The symbol for a hydraulic intensifier shows two different bore diameter cylinders in a single housing. The outlet at the end of the smaller bore cylinder provides high pressure oil as the cylinder rod extends. A double check valve arrangement is used to fill the cylinder at the rod end of the smaller bore cylinder as the cylinder rod is retracted. Cycling the intensifier provides high pressure low volume oil at the outlet, with outlet pressure determined by multiplying the pressure at the cap end of the larger bore cylinder by the ratio of the area of the larger bore cylinder to the area of the smaller bore cylinder. For example, a pressure of 2000 psi and an area ratio of 3 to 1 would theoretically generate an intensified pressure of 6,000 psi (2000 psi x 3:1 ratio).

The symbol for a discrete positioner consists of several cylinder symbols coupled together in series. For example, the discrete positioner shown in Figure 1-8c consists of four double acting hydraulic cylinders coupled in series. The combination of the four symbols has five positions. These are: all cylinders fully retracted, plus another position when each cylinder is extended, one at a time.

Review 1.1.1.5: How is the cylinder shown by the graphic symbol equipped?

a. with a servo positioner
b. with a fixed cushioning device
c. with a cushioning device retracting
d. with an adjustable cushioning device extending
e. with an adjustable cushioning device extending and retracting



Pressure Control Valves

Pressure relief, pressure reducing, unloading, sequence, counterbalance, and brake valves control the pressure in systems. One of these valves can serve multiple purposes, depending upon where it is located in the circuit, how it is plumbed, how the pilot circuit operates, and whether or not the valve drains internally into the reservoir return line or has an external drain.

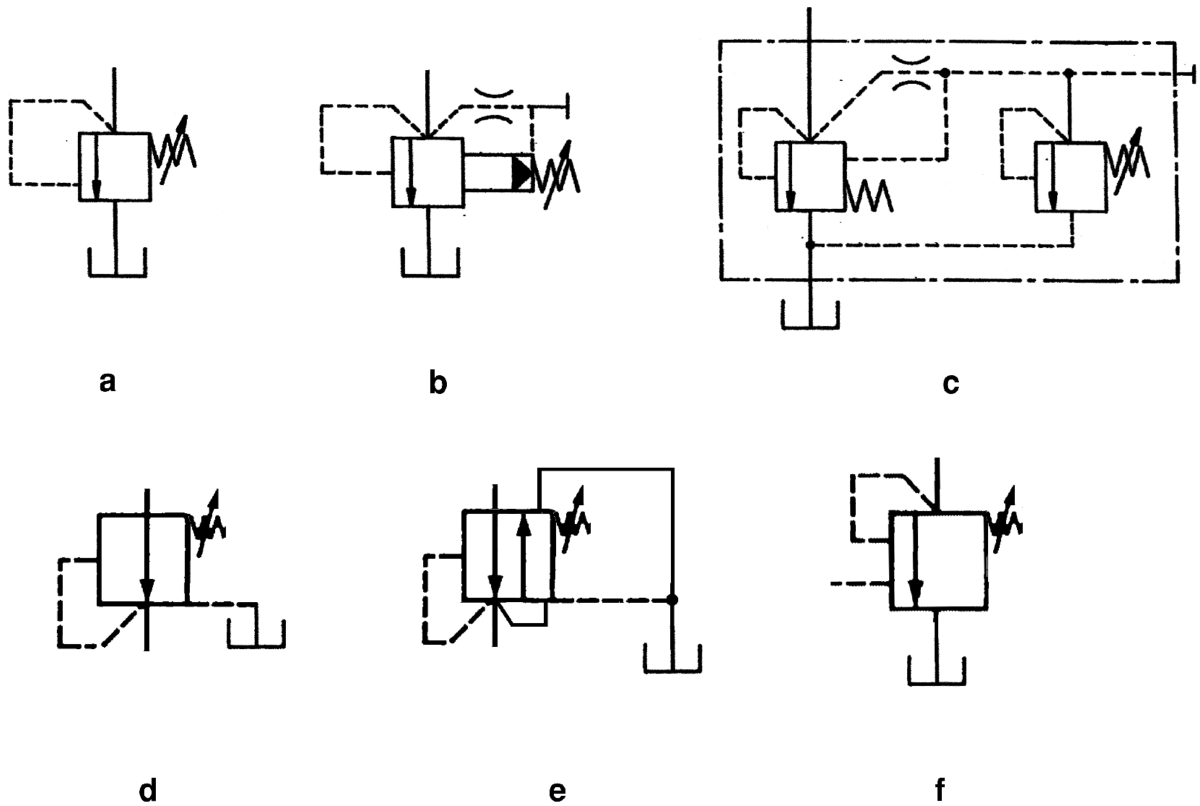


Figure 1-9: (a) Direct Acting Pressure Relief, (b) Pilot Operated Pressure Relief (Simple), and (c) Pilot Operated Pressure Relief (Detailed) Valve Symbols (d) Pressure Reducing, (e) Pressure Reducing Relieving, (f) Unloading Relief

Pressure relief valves, illustrated in Figure 1-9, limit maximum system pressure. Relief valves are normally closed valves which sense pressure upstream of the pressure relief valve. When the pressure reaches the setting of the valve, the valve opens to relieve the over pressure fluid to the reservoir. Figure 1-9a shows a direct acting, or single stage, relief valve. The dashed pilot line connected to the valve envelope at the point at which the inlet line meets the envelope indicates the pilot pressure in sensed internally to the body of the valve. The spring chamber in relief valves are internally drained to the outlet, or secondary port, though that feature is not shown by current ISO 1219-1 symbols. Backpressure in the outlet line of a relief valve acts on the spring side of the poppet or spool, and thus is additive to the pressure setting of the valve. What this means is that if the tank line backpressure increases by 100 psi, the valve will open at 100 psi more than it was set to open, though the differential pressure across the valve does not change.

Figure 1-9b shows the simplified symbol for a pilot operated, or two stage, relief valve, while Figure 1-9c shows the detailed symbol for a pilot operated relief valve. Pilot operated relief valves may be remote piloted, sometimes from the operator's station. The detailed symbol shown includes a vent port connection allowing a second direct acting relief valve to be connected to this port, thus allowing remote control of the main relief valve. In addition to a remote pilot relief valve, or as an alternative, a solenoid valve may be connected to the main relief valve in order to vent the main relief valve down to low pressure.

Circuits using fixed displacement pumps must have a pressure relief valve. Not all variable volume pumps are pressure compensated. Therefore, these pumps also require relief valves. Many pressure compensated pumps have compensators that can fail in an “on stroke” condition, therefore requiring a relief valve as well. The main relief valve in a circuit is generally termed the system relief valve. However, relief valves are also used in branch circuits in order to protect an actuator. These circuit relief valves are usually called crossport relief valves when used with a motor and cylinder relief valves when used to protect a cylinder.

Pressure reducing valves, shown in Figure 1-9d, are normally open valves used to limit the maximum force of actuators in branch circuits. Pressure reducing valves control the force by sensing the pressure at the secondary (outlet) port of the valve. When downstream pressure reaches the pressure setting of the valve, the spool begins to meter flow into the circuit, limiting the downstream pressure to the pressure setting of the valve. Since pressure is defined as resistance to flow, pressure can be controlled by regulating the flow into the circuit. In a typical application, the pressure reducing valve comes into operation when a cylinder in a branch circuit deadheads against the load resistance. The pressure then rises to the pressure setting of the reducing valve. By controlling the downstream pressure, the valve limits the maximum output force of the actuator in the branch circuit. Because pressure reducing valves sense pressure at the outlet port, they are externally drained. Obstructing the drain of a pressure reducing valve will prevent the valve from operating from the normally open to the closed position.

Figure 1-9e shows the symbol for a pressure reducing-relieving valve, which in addition to reducing downstream pressure, will relieve downstream pressure.

Unloading valves are used with high-low pump circuits and with accumulator circuits to save power when fixed displacement pumps are used. Some manufacturers market an unloading-relief valve which, in addition to the external pilot that is connected downstream of the check valve, includes an internal pilot connection. This version is shown by the symbol in Figure 1-9f. The main characteristic of an unloading valve is the external pilot line that allows the valve to sense pressure downstream of the check valve used in applications for unloading valves. Several manufacturers offer the unloading valve and check valve in the same body assembly.

In a typical accumulator application, shown in Figure 1-10, hydraulic oil from the fixed displacement pump will pass through an isolating check valve to fill the accumulator. This type of circuit uses a 3-position directional control valve that has a blocked pressure port in the center envelope. When the accumulator becomes filled, pressure on the accumulator side of the check valve pilots the unloading valve open, unloading the pump to the reservoir at low pressure. The unloading valve will remain open as long as the accumulator can supply pilot pressure above the setting of the valve. When the pressure downstream of the check valve drops below the pressure setting of the unloading valve, the unloading valve closes, allowing the pump to refill the accumulator. Unloading valves are normally closed, externally piloted, and may be internally or externally drained. An external drain is required if there is backpressure at the outlet port, for example if the fluid is unloaded through a heat exchanger or circuit that creates backpressure that would upset the pressure differential of the valve. An unloading valve has a low pressure drop across the valve when it is in the open state. The valve is held fully open by the pilot signal to unload the pump at low pressure.

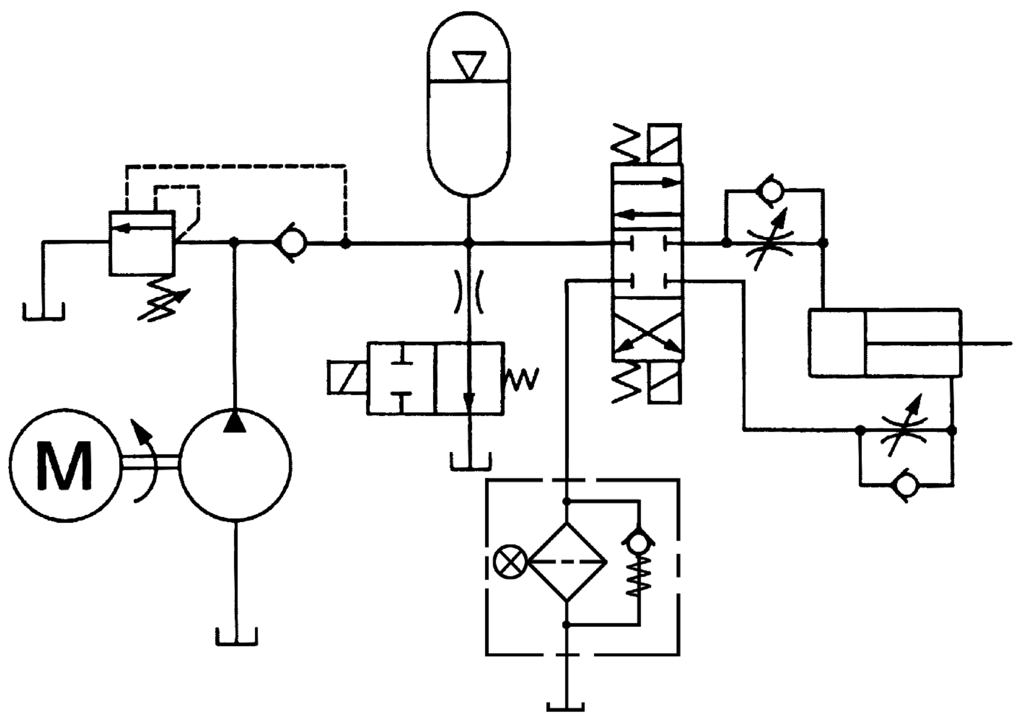


Figure 1-10: An Unloading Valve in an Accumulator Circuit

The unloading valve shown in Figure 10 is a variation on a standard unloading valve as it includes an internal as well as an external pilot, making the valve illustrated an unloading-relief valve. The valve will open upon sensing adequate pilot pressure from either pilot source. There are differential pressure unloading valves which are specifically used in accummulater circuits to open at a higher pressure than they close.

Figure 1-11 shows a typical high-low pump circuit. The unloading valve is actuated by rising pressure downstream from the check valve, unloading the high volume pump at low pressure. When an unloading valve is piloted open by the external pilot, there is a low pressure drop across the valve, as it is being held open by the pilot pressure. If an unloading valve is subject to backpressure, it should be externally drained.

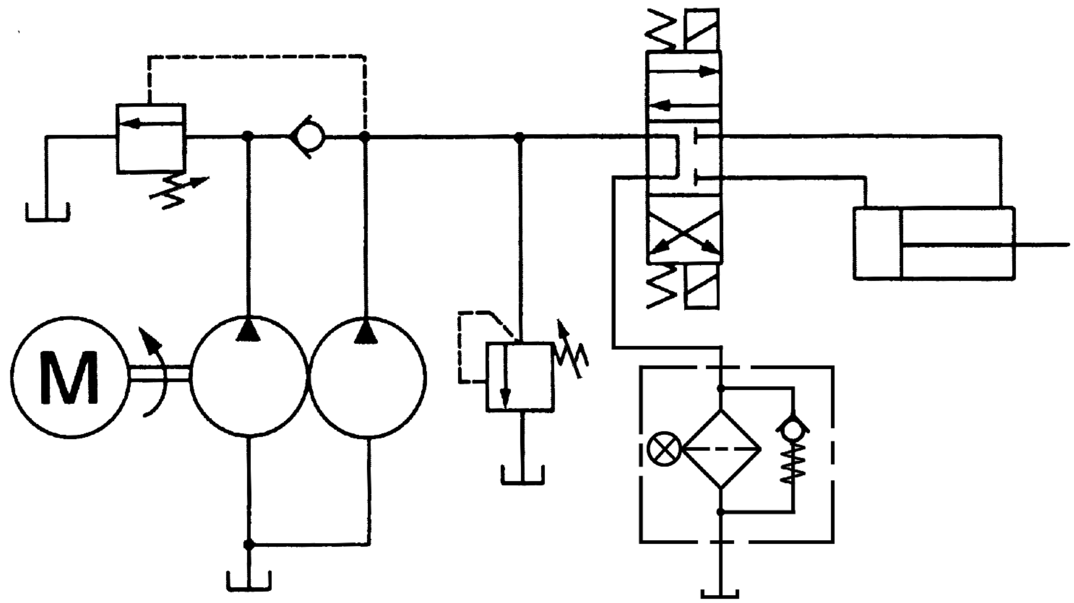


Figure 1-11: An Unloading Valve in a High-Low Pump Circuit

Sequence valves, shown in Figure 1-12, are used on clamp and work circuits to assure required clamping force is reached in the clamp cylinder before the work portion of the cycle begins. Sequence valves may be internally or externally pilot operated, but they must have an external drain because the outlet port is pressurized. Sequence valves may be equipped with integral reverse free-flow check valves. Sequence valves are normally closed and are pilot operated to open to allow full flow to the actuator. In a typical application, fluid is directed to extend both the clamp and drill cylinders at the same time. The sequence valve is installed in series with the drill cylinder. The clamp cylinder receives fluid first, with its minimum force determined by the pressure required to open the sequence valve at the drill cylinder, and the area of the clamp cylinder. When the minimum clamping cylinder pressure is reached, the sequence valve opens and the drill cylinder will advance. The maximum extension force of both cylinders is determined by the pressure setting of the system relief valve, the areas of the cylinders, or by pressure reducing valves, if any are used. When the directional control valve is reversed to retract the cylinders, some means must be employed to prevent both cylinders from retracting at the same time. This would cause the clamp to relax while the drill was still in the work piece. The proper sequence would be first to retract the drill, and then to retract the clamp. One method to accomplish the reverse sequence would be to install a second sequence valve at the rod end of the clamp cylinder. This would route flow first to the rod end of the drill cylinder, causing it to retract, followed by the opening of the sequence valve when the pressure rises, allowing flow to the rod side of the clamp cylinder. Both cylinders will now operate in the proper sequence. It should be noticed that the clamp cylinder loses pressure to hold the clamp closed when the directional control valve is shifted to retract both cylinders.

A sequence valve is essentially an externally drained relief valve. As such, it may be used as a relief valve in applications where the backpressure that acts on the tank port of the relief valve varies, causing changes in the opening pressure of the relief valve.

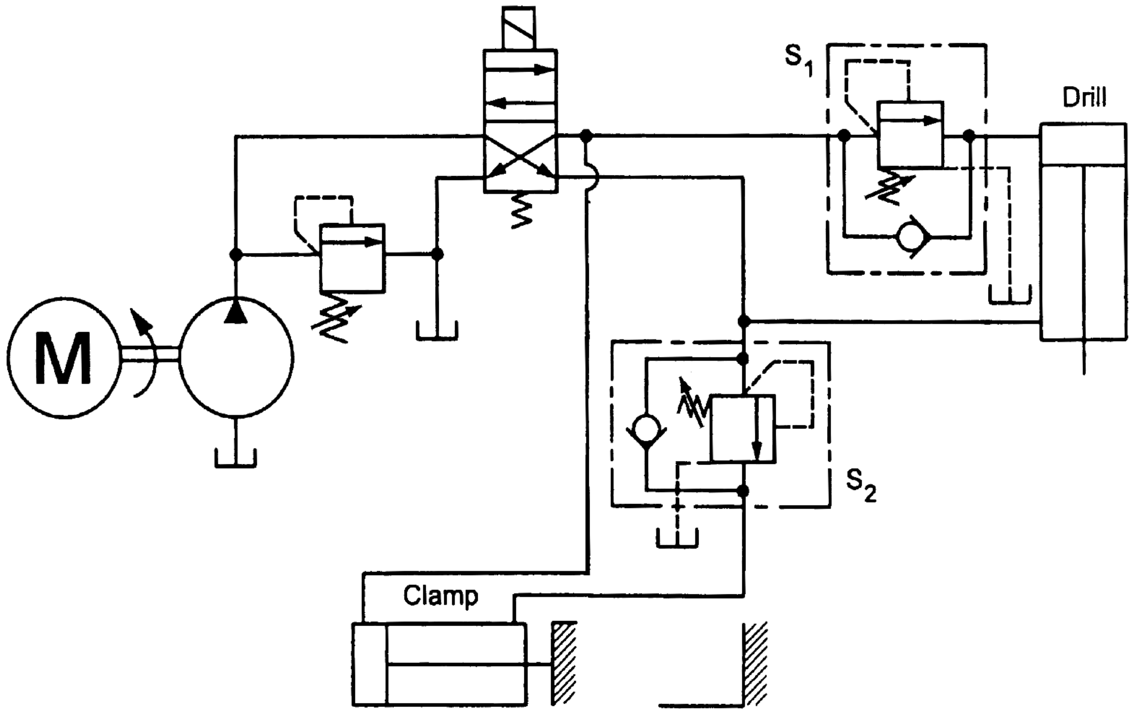


Figure 1-12: Sequence Valve Circuit

The purpose of a counterbalance valve is to prevent a loaded cylinder, having potential energy, from falling (extending or retracting). Counterbalance valves may be internally piloted, externally piloted, or piloted internally as well as externally, and they may be internally or externally drained. If conditions exist that would interfere with internal draining the valve, it should be externally drained, but usually this is not necessary. Counterbalance valves are equipped with a free reverse flow check valve to allow for retraction of the cylinder.

The simplest counterbalance valve application is to support a constant induced load. In a down acting press application, the counterbalance valve would be installed at the rod end of the cylinder to control return oil flow. This would prevent the press platen from dropping. Pilot pressure to open an internally piloted counterbalance valve would be set approximately 100 psi above the pressure of the rod end of the cylinder caused by the weight of the platen. In order for the platen to be lowered (and powered down), the pressure at the cap end of the cylinder would have to be sufficient to generate 100 additional psi at the rod end of the cylinder. Thus, 100 psi added to the pressure generated by the weight of the platen would open the counterbalance valve and allow the platen to lower smoothly.

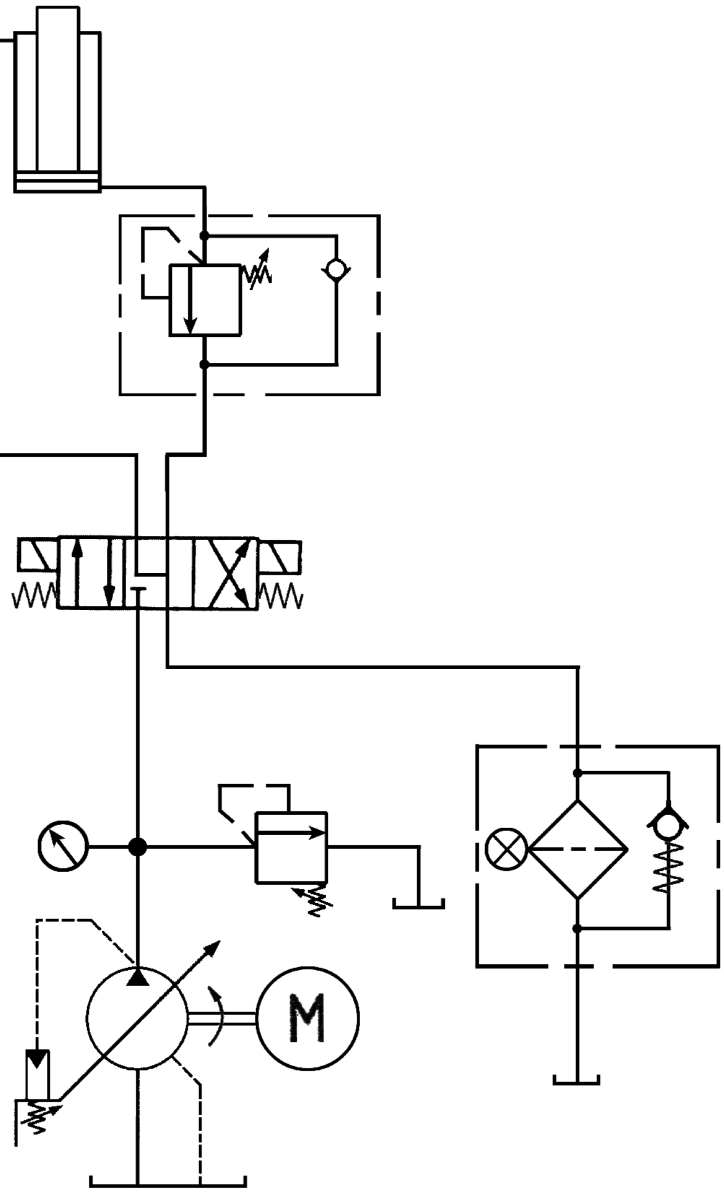


Figure 1-13: Counterbalance Valve in a Press Circuit

One disadvantage of the counterbalance valve shown in the circuit in Figure 1-13, is that backpressure on the cap side of the cylinder limits the effective force developed by the cylinder. In order to achieve full force from the cylinder, the backpressure must be relieved from the cap side of the cylinder. This is easily achieved by using a counterbalance valve that includes an external pilot. Counterbalance valves that include an external pilot in addition to the internal pilot are called holding valves, overcenter valves, load control valves, or motion control valves by some manufacturers. After the cylinder has stalled against the load, the external pilot will fully open the counterbalance valve, allowing the pressure in the cap end of the cylinder to fall to virtually zero psi. This configuration is shown in Figure 1-14.

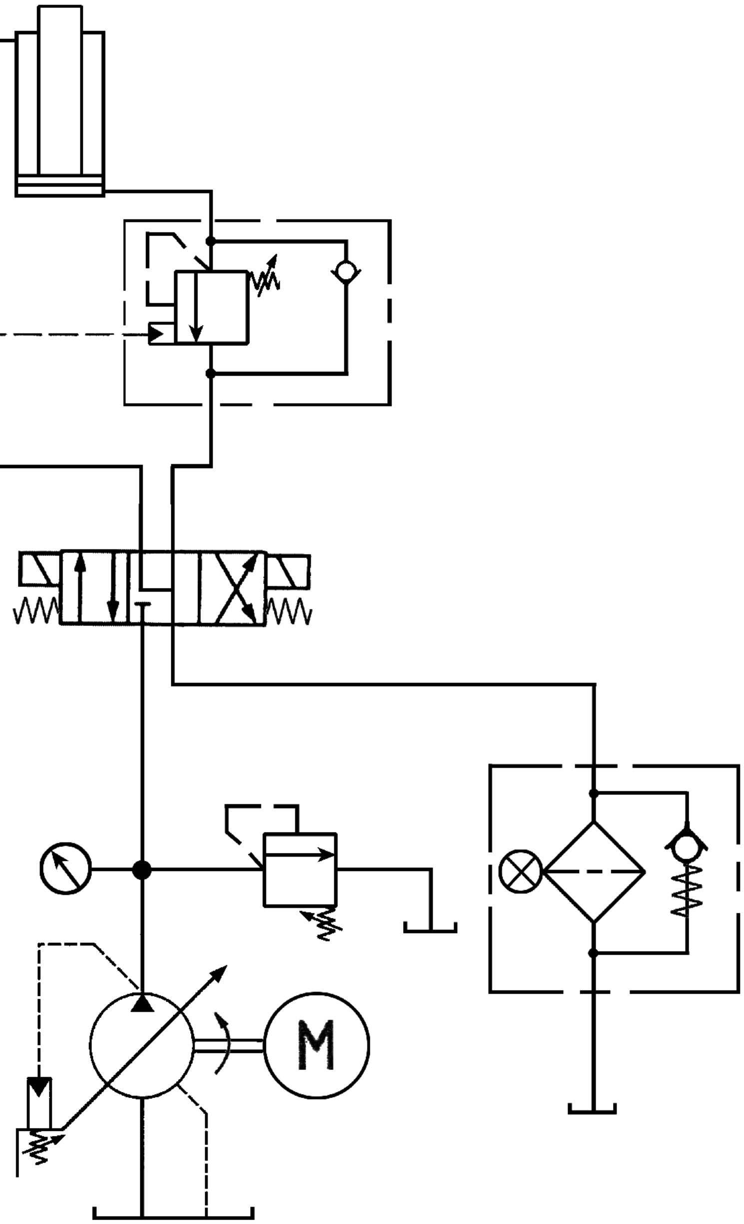


Figure 1-14: Overcenter Valve in a Press Circuit

Some applications require control of a varying overrunning load. In some applications, such as with a boom, the load may also move overcenter. In these applications, a counterbalance valve which includes both internal and external pilots is very useful. The term “overcenter valve” implies the valve has both internal and external pilots.

If a cylinder is subjected to a tractive (overrunning) load, the cylinder will typically chatter as it operates. This scenario is similar to that of a boom application where the induced pressure varies as the position of the boom changes. Overcenter valves should be used in these applications.

In the case of a boom application, the greatest induced load will occur when the boom is horizontal and the least induced load will occur when the boom is vertical. The internal pilot pressure setting must be set to hold the boom in the worst case situation, which is when the boom is horizontal. If the valve had only an internal pilot section, whenever the boom cylinder was moved, a very high pressure drop would be occur as the fluid passed through the valve. By adding an external pilot, the overcenter valve is piloted open by the pressure developed in the opposite cylinder line. If the valve opens too much, allowing the cylinder to begin to overrun the pump flow, pilot pressure in the opposite cylinder line will drop and the valve will begin to close. In actual operation, the poppet or spool in the overcenter valve modulates and provides for smooth control of the load, even as the induced load pressure rises, preventing the cylinder from running ahead of flow from the pump. Various pilot ratios are available for the external pilot section of the valve, allowing the application engineer to fine tune the response of the circuit.

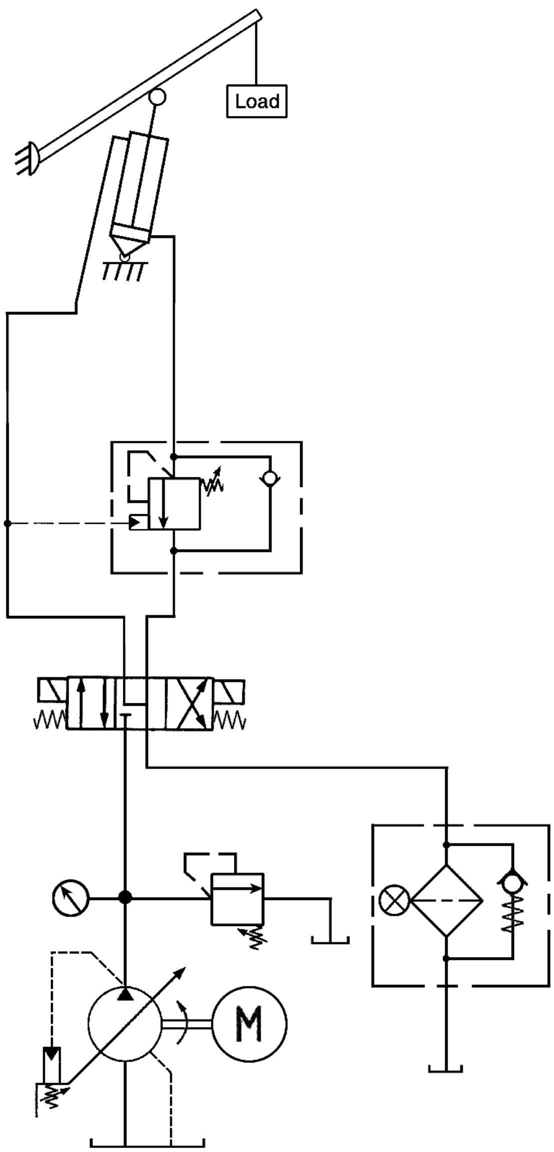


Figure 1-15: Overcenter Valve in a Boom Circuit

Motion control valves are manufactured with both poppet type as well as with spool type pressure control sections. Spool type motion control valves have inherently more leakage than do poppet type motion control valves. If the motion control valve must hold the load for an extended amount of time, a valve with a poppet type pressure element should be used. Additional means of locking the load in place, such as with shot pins or wedges, operated by additional cylinders, may be necessary.

Brake valves are installed in the return lines of hydraulic motors to prevent over speeding when the motor is under an overrunning load, and to prevent excessive pressure buildup when accelerating and stopping motor loads. If the motor is bidirectional, the brake valve is equipped with a reverse free flow check valve to reverse the direction of rotation. A brake valve application is shown in Figure 1-16. For the most part, a brake valve is simply an overcenter valve used with a motor.

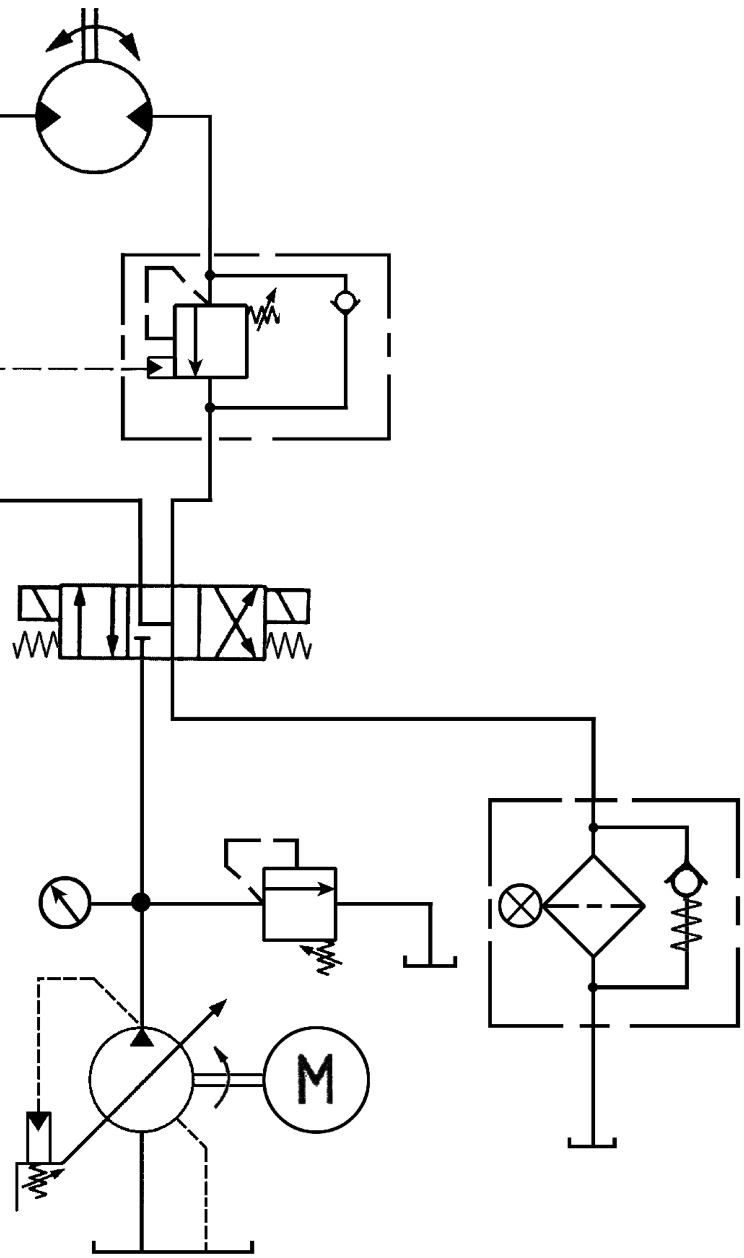


Figure 1-16: Brake Valve Circuit

When the valve is used for braking, it is internally piloted by pressure from the motor outlet line. When the motor is operating under load, pilot pressure from the inlet side of the motor pilots the brake valve open, thereby providing minimum resistance to flow at the motor outlet. An overrunning load will cause a loss of pressure at the motor inlet port, and therefore in the external pilot line that opens the valve. In response to this loss in pressure the brake valve will resist the outlet flow from the motor resulting in a braking action. When the directional control valve is shifted to the neutral position pressure at the inlet pressure is lost and the brake valve closes, raising the pressure in the motor outlet line. Therefore, when either an overrunning load is present when the motor is operating, or the directional control valve is centered, the motor essentially changes to a pump, attempting to pump fluid across the now closed brake valve. The internal pilot line of the brake valve senses the pressure rise caused by the resistance of the closed pressure section, piloting the brake valve open, braking the load applied to the motor. As the pressure drops across the brake valve, energy from the overrunning load is thus absorbed and dissipated. In order to avoid cavitating the motor, a makeup check valve or some other arrangement must be included in the circuit to provide makeup fluid to the motor port opposite the brake valve.

Since a motor inherently has internal leakage, even a poppet type of brake valve will not prevent a motor from drifting. Some sort of a brake must be used to prevent the motor shaft from turning. While an external brake may be designed into the machine, many hydraulic motor manufacturers offer brakes which are integral to the motor. Oft times, these are spring applied, pressure released brakes. A shuttle valve is often incorporated in the valve circuit. Upon sensing pressure in either operating line of the motor, pressure is directed to the brake assembly, releasing the brake, thereby allowing the motor shaft to rotate.

As discussed, counterbalance valves are used to prevent a load from falling. Their purpose is to prevent injury to personnel and damage to equipment. Press circuits are obvious examples of industrial applications. In order to provide protection should a hose or tube rupture, the valve must be located at the cylinder. Pilot check valves, such as the one symbolized in Figure 1-17, also will hold a load in position, but their operation is not as smooth as that of counterbalance valves in throttling the flow as the load is lowered. Pilot operated to-open check valves do have the advantage of locking the load in position because the check valve has a positive seat, where spool type counterbalance valves will have some leakage. Both counterbalance valves and pilot operated to-open check valves are mounted in or close to the cylinder port with rigid plumbing.

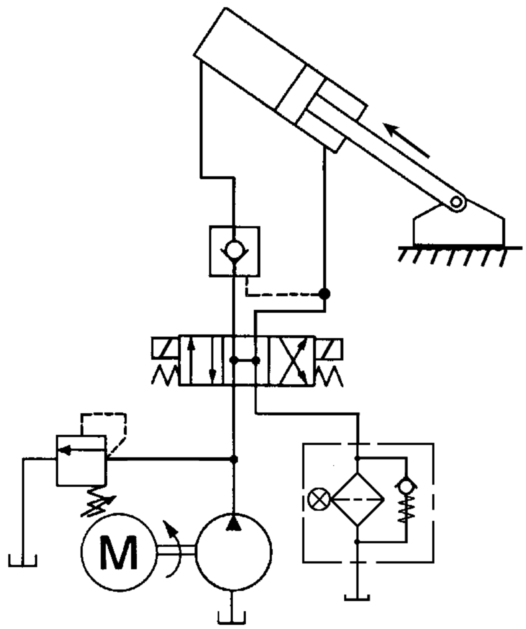


Figure 1-17: Pilot Operated Check Valve Application

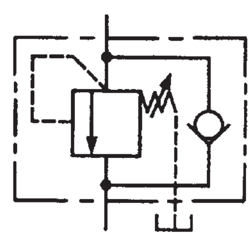
The common practice is to use pilot operated to-open check valves to hold static loads and to use counter-balance valves to hold dynamic loads. For example, aerial lifts include both the manlift boom as well as outriggers which steady the lift frame. The outriggers, such as the one shown in Figure 1-17, are held in position by pilot operated check valves while the manlift boom is commonly held in position by an over-center valve. The boom is a dynamic load in that it is moved frequently, while the outriggers are set and remain stationary during operation of the aerial lift.

Though pilot operated control valves have been included with motion control pressure control valves in this manual, they are sometimes considered to be a type of directional control valve.

Review 1.1.1.6: Which circuit would require an unloading valve?

a. boom circuit
b. press circuit
c. motor circuit
d. sequence circuit
e. high-low pump circuit

Review 1.1.1.7: The valve symbol shown is a:



a. sequence valve
b. counterbalance valve
c. unloading valve
d. pressure reducing valve
e. brake valve

Directional Control Valves

The accepted designation is to identify the number of flow ports, excluding pilot ports, when referring to "ways." Thus, a 2-way valve has two ports, a 3-way valve has 3 ports, a 4-way valve has four ports, and so on.

Hydraulic directional control valves stop, start, and direct the flow of oil. "Positions" refers to the number of active states of a directional control valve. For example, a 2-position valve has two states, "on" and "off," or "actuated" and "deactuated." A square envelope is drawn for each state. Thus, a 2-position valve is drawn with two square envelopes. The lines and arrows inside the envelopes describe what is happening with the flow paths for each state.

The number of ports and the number of positions are then combined, in that order, separated with a slash. Therefore, 2/2 describes a 2-way 2-position valve and 4/3 describes a 4-way 3-position valve.

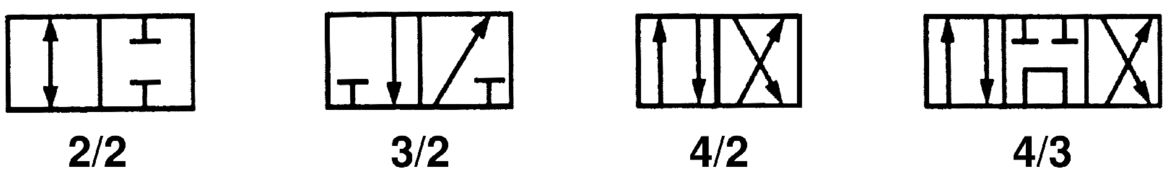


Figure 1-18: 2/2, 3/2, 4/2, and 4/3 Directional Control Valve Symbols

The center position on three position directional control valves is used most often to provide an idle or at-rest function for the circuit. How the center ports are connected depends upon which components are used and whether the circuit has an open or closed center. For example, fixed displacement pumps are normally connected to open center, tandem center, or through center circuits, relieving fluid to the reservoir at low pressure in the center position. This arrangement reduces the power waste and heat generation that would be caused by fluid flowing over the pressure relief valve had a blocked pressure center directional valve been used.

Fixed displacement pumps used in accumulator circuits use a directional valve spool that blocks the P-port in the center position. This type of circuit was shown in Figure 1-10. The pump is unloaded by an unloading valve. The blocked pressure port allows the accumulator to fill in preparation for use by the circuit. Variable displacement, pressure compensated pumps also use valve center configurations that have a blocked P-port. The compensator reduces pump flow to a rate that only provides enough flow to maintain pressure, compensating for system leakage. Thus, an unloading valve is not required. The pressure setting is determined by the adjustment of the compensator spring.

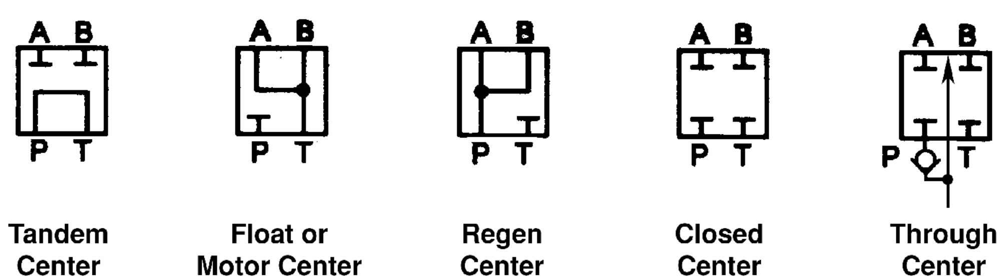


Figure 1-19: Center Positions for Three Position Directional Control Valves

Float center valves, sometimes termed motor center valves, connect the A, B, and T ports to reservoir in the center position. When this center configuration is used hydraulic cylinders, it is termed a float center, as the cylinder rod/piston position may float in response to forces external to the cylinder, and is termed a motor center when used with hydraulic motors, allowing the motor to coast to a stop when the directional valve is centered.

If the speed of the cylinder is to be controlled in the center position of a float center circuit, the A and B passages will have built in flow control orifices. This would limit the maximum rate of travel for a cylinder subjected to external loading.

The most common use of a float center spool with a cylinder is to allow the pilot lines in pilot operated check valves to drain to tank, allowing the check valve poppets to fully seat, thereby holding the cylinder in place. This type of circuit is shown in Figure 1-20. Due to bypass leakage, cylinders equipped with cast iron piston rings would drift if an external load is applied. However, cylinders with elastomer seals will not drift if the seals and the cylinder tube are in good condition.

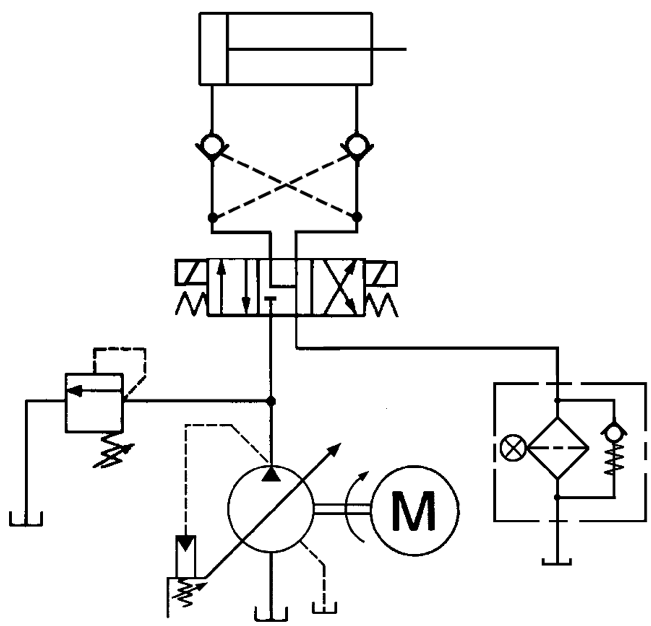


Figure 1-20: A Float Center Spool Used with Pilot Operated Check Valves in a Cylinder Circuit

Review 1.1.1.8: How many ports does the directional control valve shown have?

a. two
b. three
c. four
d. five
e. six

Directional control valves are operated manually (by a person), mechanically (typically by a cam), by solenoids, by hydraulic or pneumatic pilot pressure, or by a combination of these operators. Each of these operators has a unique symbol. The operator symbols are shown attached to the end of the valve envelope. Manual operators have the shape of a rectangle with the type of operator shown symbolically. Manual operators include levers, treadles, and push pins used to troubleshoot valves. Solenoid pilots are shown as a slanted line within the operator rectangle. The slanted line denotes the coil winding. Hydraulic pilots are shown as a solid triangle within the rectangle and pneumatic pilots are shown as a hollow triangle. Centering springs and return springs are attached to the ends of the envelope(s) and are denoted by a symbol that looks like a “w.”

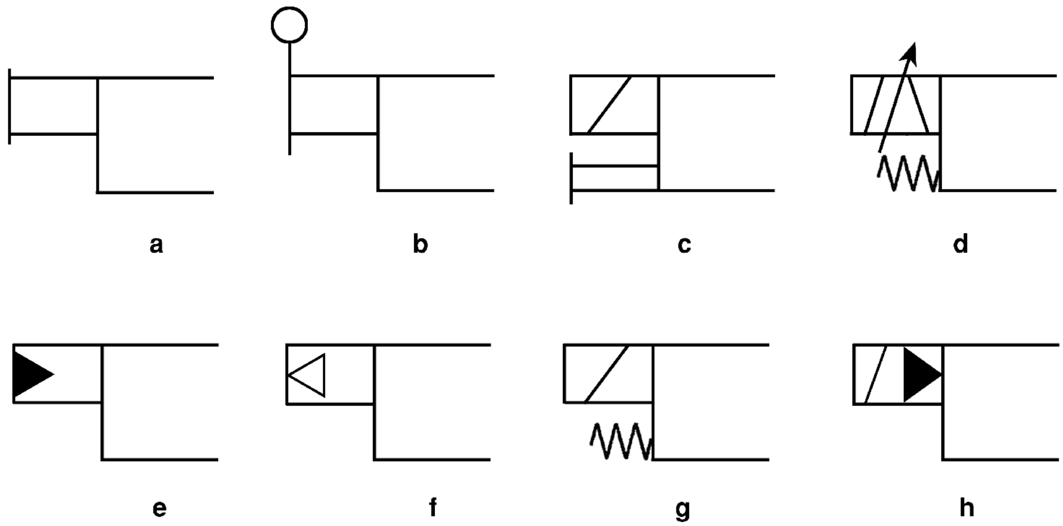
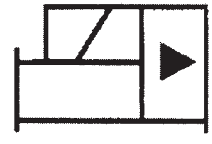


Figure 1-21: Valve Operator Symbols

Valve operators may operate in parallel or in series. For example, pilot operated valves generally have a solenoid operator in series with the pilot to shift the valve. Valve operators may also operate in parallel with each other. ISO 1219-1 allows valve operators to be placed anywhere along the end of an envelope. Figure 1-21a shows the general symbol for a manual operator. Figure 1-21b shows the symbol for a lever operator. A solenoid with a manual operator in parallel, most commonly a push pin, is shown by Figure 1-21c. A proportional solenoid with opposing windings is shown in Figure 1-21d. Figure 1-21e shows the application of hydraulic pressure as an operator while Figure 1-21f shows the release of pneumatic pilot pressure as an operator. A centering spring, in the case of three position valves, or a return spring, in the case of two position valves, is shown by Figure 1-21g, along with a solenoid. Either the spring or the solenoid could operate the valve. Finally, a solenoid controlled, hydraulic pilot operator is shown in Figure 1-21h.

Review 1.1.1.9: Based on the directional control valve symbol shown, in order for the valve to be actuated, the valve must receive:



a. a pilot signal or manual actuation
b. solenoid actuation and manual actuation
c. solenoid actuation or a pilot signal, and manual actuation
d. solenoid actuation and a pilot signal, and manual actuation
e. solenoid actuation and a pilot signal, or manual actuation and a pilot signal

Task 1.2: Read circuit diagrams, size components, and recognize functions of components.

Outcome 1.2.1:

Read basic circuit diagrams.

Basic hydraulic circuits are constructed to accomplish common tasks. Being familiar with them is useful in designing and troubleshooting circuits that are applied to specific tasks. Basic circuits include those identified as unloading, hi-low, accumulator, pressure reducing, sequence, regenerative, synchronous, counter-balance and pilot operated check valve.

A hydraulic circuit that uses a fixed displacement pump must have a pressure relief valve or an unloading valve to protect the system from over pressure. In Figure 1-22 the circuit operates a double acting cylinder through a three position tandem center directional control valve. The pump will unload to the reservoir in the center position at low pressure, and operate the relief valve at maximum system pressure when the cylinder stalls extending or retracting. During the time the cylinder is stalled the full power of the system, full flow at maximum pressure drop, is wasted across the relief valve. For this reason, open pressure center systems are used where the idle portion of the cycle is greater than the work portion of the cycle when the cylinder is stalled. If additional valving located between the directional valve and the actuator exists, such as pilot operated check valves, an open center (all ports connected to each other) directional control valve would be required in order to both unload the pump and drain the pilot operated check valve pilot lines. Care would need to be taken to ensure that minimal backpressure exists between the tank port of the directional control valve and the reservoir to avoid enough pressure to build up, acting on the pilot operated check valve pilots. Backpressure acting on the pilot operated check valves could pilot them open, allowing the cylinder to drift.

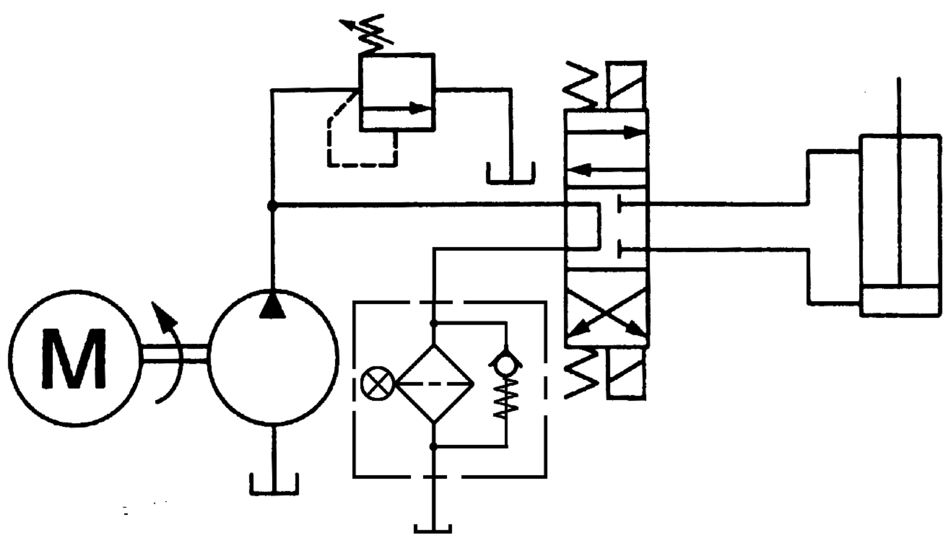


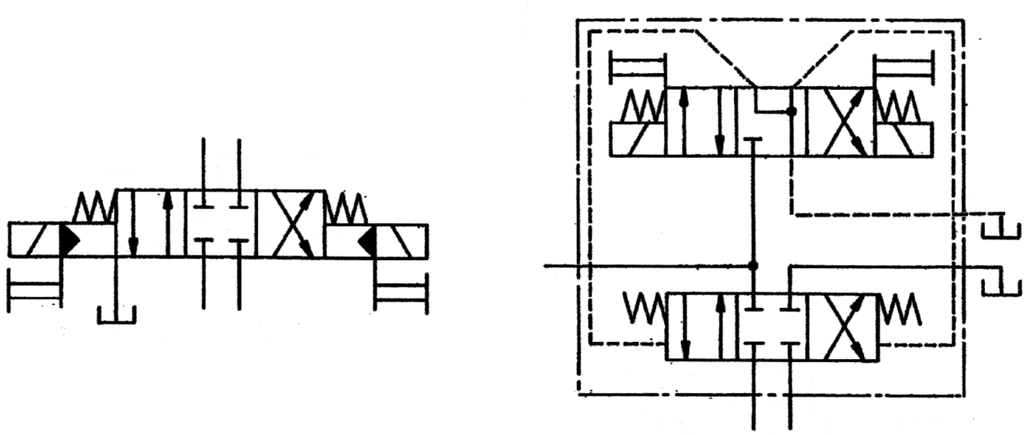
Figure 1-22: Tandem Center Circuit Equipped with a Relief Valve

While the fixed displacement pump circuit illustrated in Figure 1-22 works well with only one actuator circuit, it is very common to use fixed displacement pumps to power circuits with two or more actuator circuits. If two open center directional control valves are connected in parallel with the pump, when one of the valves is actuated, the actuator will not move because the pump flow will pass to tank at low pressure through the open center of the other directional control valve. However, if the directional control valves are connected in series with each other, flowing from the tank port of the first valve to the pressure port of the second valve in what is called a series circuit, the actuators will function properly.

Another option would be to use blocked pressure center directional control valves plumbed in parallel with each other. This would solve several problems encountered when using directional valves in series, but would not unload the pump during idle portions of the cycle. However, if three position, solenoid operated directional control valves are being used, a solenoid vented pressure relief valve, tied electrically into the directional control solenoids, could be used to unload the pump during idle time periods. Whenever a directional control valve is actuated, the solenoid on the relief valve is also energized, closing the low pressure path to tank.

One disadvantage of an open center valve circuit is that pressure is lost in the center position. This means the system is required to build pressure each time the valve is shifted to move the load. Pilot operated directional control valves require a minimum of approximately 65 psi in order to function. Therefore, there must be some means of developing this minimum pilot pressure when the directional control valve is in the center position. One solution is to place a check valve with a 65 psi spring somewhere in the circuit. If the check valve is located between the pump and the P port of the directional control valve, a pilot pressure line can be connected upstream of the check valve to provide external pilot pressure to the directional control valve. If the check valve is located between the tank port of the directional and the reservoir, an external drain line is connected to the directional control valve.

Figure 1-23a shows the simplified symbol for a pilot operated 4-way directional control valve and Figure 1-23b shows the detailed symbol for the same valve. Detailed symbols show both external pilot pressure and external drain connections. Internal pilot pressure and internal drain connections are not shown in simplified ISO 1219-1 compliant symbols.



a b

Figure 1-23: Simple and Detailed Pilot Operated Directional Control Valve Symbols

High-low circuits are used to provide rapid advance of cylinder rods at low pressure, followed by a slower advance at high pressure to perform the work. High-low circuits provide high flow from two pumps at low pressure and low flow at high pressure from one pump. This has the effect of speeding up the circuit without increasing power requirements. In Figure 1-24, both pumps are connected to one motor. In operation, shifting the directional control valve from the center position to extend the cylinder rod applies full flow from both pumps to the cap end of the cylinder at low pressure. The cylinder rod will now advance until it meets an obstruction that will cause system pressure to rise sufficiently to pilot open the unloading valve. This will relieve the low pressure pump to reservoir. The system is equipped with a relief valve to protect the high pressure pump when the cylinder stalls. Pump flow ratios do not have to be equal to each other.

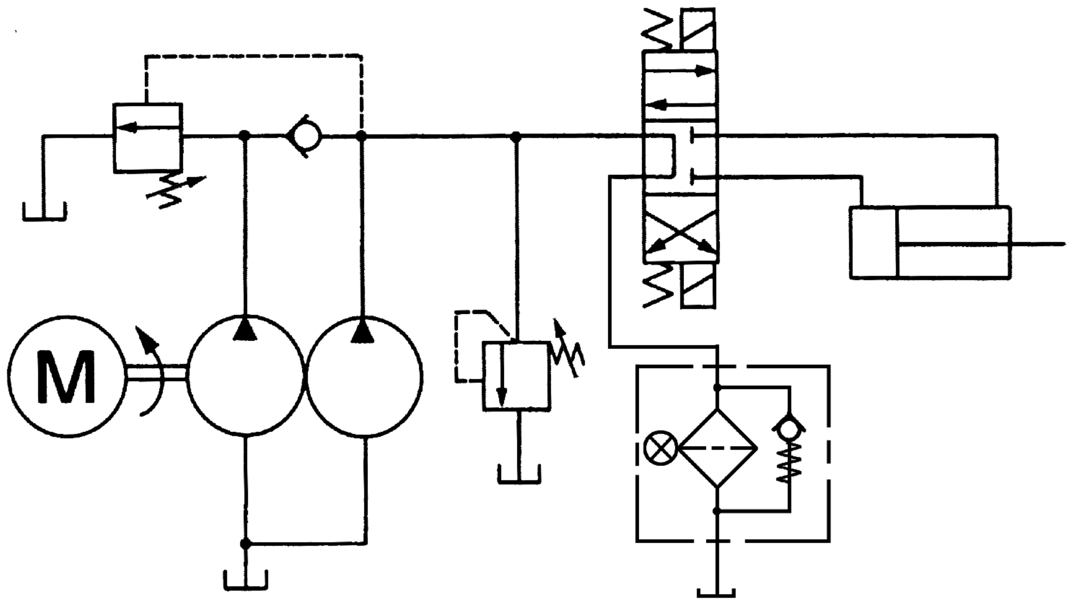


Figure 1-24: High-Low Circuit

Accumulator circuits store pressurized fluid. They are used for several purposes:

1. Maintain system pressure.
2. Absorb hydraulic shocks.
3. Supplement pump flow.
4. Provide auxiliary power.
5. Act as a barrier between dissimilar fluids.

The closed center accumulator circuit shown in Figure 1-25 helps to maintain system pressure. It could also supplement pump flow to operate the cylinder. Maintaining system pressure with the accumulator and closed center valve makes the circuit more responsive. The accumulator will also supplement pump flow to supply more fluid than the pump alone could during brief periods of high usage. When the cylinder deadheads and the directional valve is actuated, the pump refills the accumulator. Pressure will pilot open the unloading valve, unloading the pump, while the accumulator makes up fluid lost due to system leakage. When the fluid in the accumulator has been depleted and pressure falls below the setting of the unloading valve, the unloading valve will close, directing pump flow into the circuit, refilling the accumulator. Theoretically, very little flow will ever pass over the relief valve. The line to the accumulator is equipped with a free flow check valve allowing unrestricted flow into the accumulator and an adjustable orifice in parallel with the check valve to control flow from the accumulator to the circuit. Without the needle valve, the cylinder speed would be dependent on the discharge rate of the accumulator, which can be much greater than the flow rate required by the application.

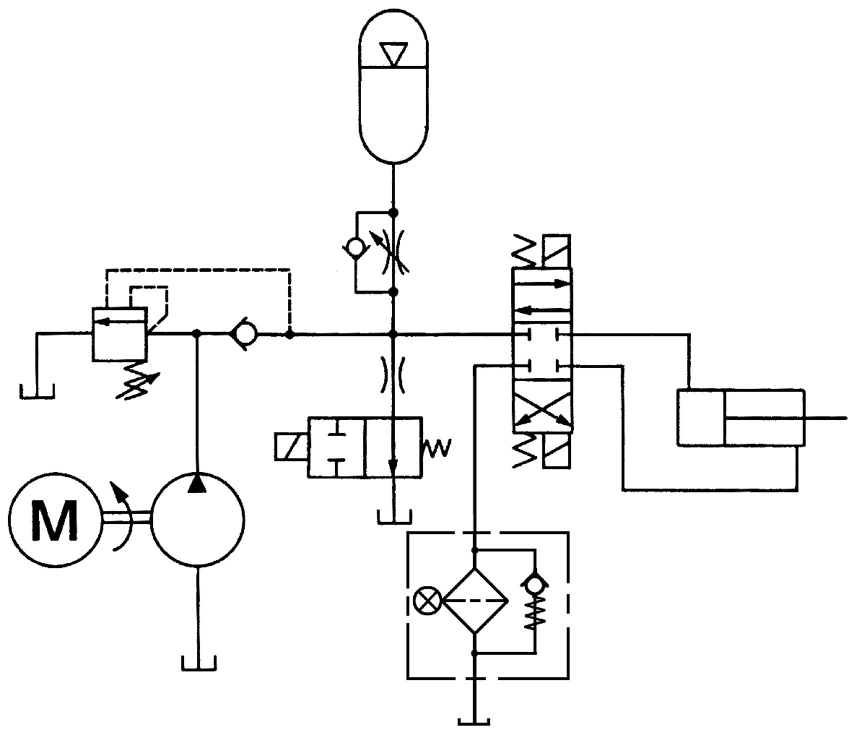


Figure 1-25: Accumulator Circuit

The fixed displacement pump fills the accumulator while the directional control valve is in the center position. Once the set point of the unloading valve has been reached, the unloading valve directs unneeded flow from the pump to the reservoir. Shifting the directional control valve releases the fluid in the accumulator and routes it to the cylinder. The pump remains unloaded as long as the accumulator can supply fluid to the cylinder at a pressure above the setting of the unloading valve. As the pressure drops, the unloading valve closes and the pump powers the cylinder, and given time, refills the accumulator. Maximum system pressure is controlled by the setting of the unloading-relief valve. The internal pilot would operate the relief valve should the external pilot become inoperative. As a safety measure, the 2/2 normally open solenoid valve releases pressurized fluid, through a small orifice, when the system is turned off. The check valve prevents downstream fluid from passing to tank when the unloading valve is piloted open.

The circuit in Figure 1-26 uses a fixed displacement pump to power two cylinders. The circuit is designed to stall each cylinder at a different pressure. Ordinarily, system pressure is controlled by the magnitude of the load on the cylinders. Assuming both cylinders have the same bore, this would mean the cylinder with the smallest load would extend first, followed by the cylinder with the larger load. Installing a pressure reducing valve in the line leading to one of the circuits permits limiting the pressure applied to that actuator to some value below maximum system pressure. For example, if the pressure relief valve were set at 2000 psi, the cylinder with the pressure reducing valve could be set to stall at 500 psi, even though the other cylinder would not operate the relief valve until 2000 psi was reached. When pressure downstream of the pressure reducing valve reaches the set point, 500 psi, the reducing valve closes, allowing only enough fluid to pass to maintain the downstream pressure at 500 psi. If the pressure reducing valve were to be replaced with a relief valve which was set at 500 psi, the pressure in the entire system would then be limited to 500 psi.

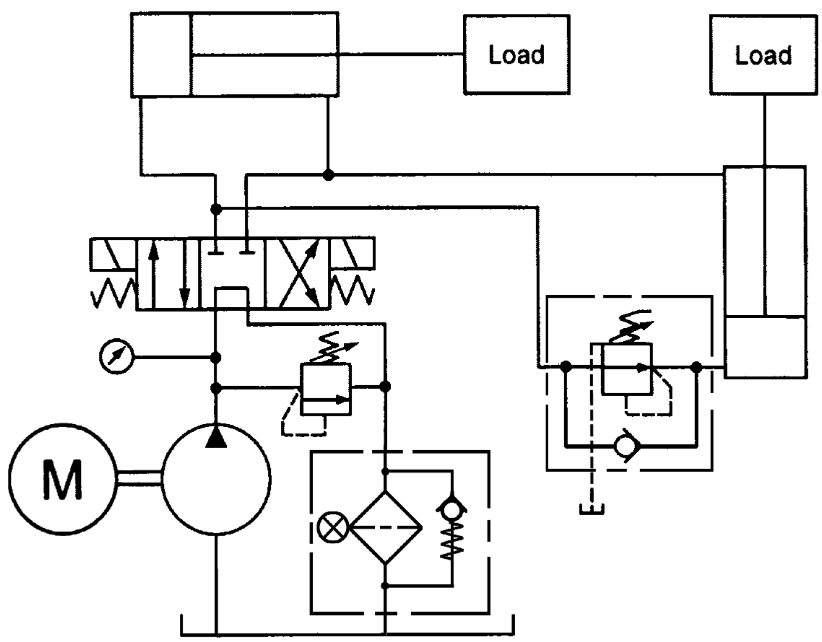


Figure 1-26: Pressure Reducing Valve Circuit

Pressure reducing valves are normally open valves which are pilot operated from the downstream side (secondary port) of the valve. As the pressure increases to the setting of the pressure reducing valve, the valve spool closes, limiting flow of fluid to the outlet (secondary port) until the pressure drops below the setting of the valve, regardless of the pressure at the inlet of the reducing valve. Pressure downstream of the pressure reducing valve will be limited to the pressure setting of the valve, though the pressure reducing valve will not relieve load induced pressure increases. Fluid cannot flow backwards through a pressure reducing valve. If the downstream load pressure, caused by load forces, must not rise above the pressure setting of the valve, a pressure reducing-relieving valve, mentioned earlier, should be used. In order to allow flow around the pressure section of the valve, in order to allow a cylinder or motor to reverse direction, a reverse free flow check valve, such as the one included in this valve assembly, is needed.

Sequence circuits are used to control the order of operation of two parallel branch circuits. The discussion of pressure reducing valves indicated that the force from a cylinder could be controlled by limiting the pressure, but this would not necessarily sequence cylinder operation, nor would it provide maximum pressure to both cylinders. A two cylinder circuit with two sequence valves is shown in Figure 1-27. A 4/2 directional control valve is used to extend and retract both cylinders. Shifting the directional control valve to extend the cylinders will cause the clamp cylinder to extend first because sequence valve S1 blocks the flow to the drill cylinder until pressure setting of the sequence valve has been reached. When the clamp cylinder stalls, the system pressure rises to the pressure setting of the sequence valve, which is then opened by an internal pilot. This allows fluid to pass through sequence valve to extend the drill cylinder. De-energizing the directional valve to retract the drill and clamp cylinders causes the drill cylinder to retract first. When the drill cylinder rod fully retracts, the pressure rises to the setting of sequence valve S2 at the rod end of the clamp cylinder. This allows fluid to enter the rod end of the clamp cylinder and it retracts.

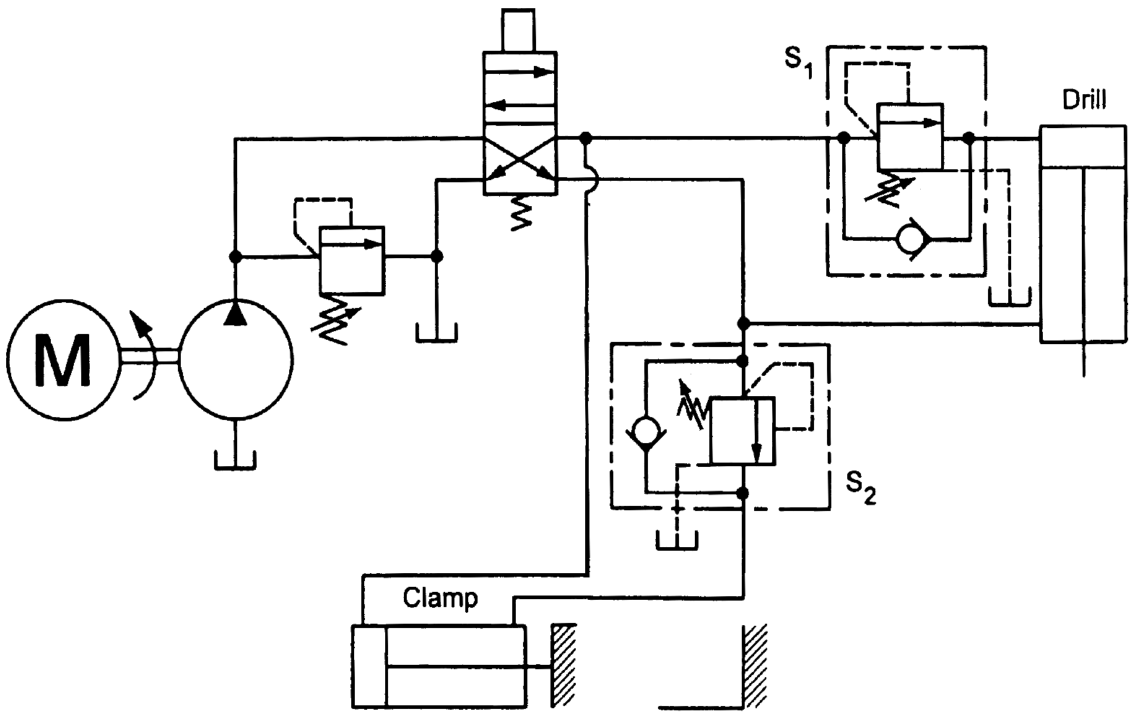


Figure 1-27: Sequence Valve Circuit

In Figure 1-27, both sequence valves have external drains on the spring chambers because the secondary (outlet) ports of the sequence valves are pressurized. If the sequence valves were not externally drained, backpressure on the secondary ports would act on the spring side of the poppet or spool, hydraulically locking the valves in a closed position.

Typical synchronous circuits allow two or more hydraulic cylinders to operate in unison regardless of differences in the load. There are a number of way's to achieve synchronous movement. In Figure 1-28, pump flow is directed through a gear-type flow divider, which is essentially two hydraulic motors with the same displacement connected to a common shaft. Input flow, which is equally divided, is then directed to two cylinders, having equal bore, rod, and stroke dimensions, plumbed in parallel. Assuming no leakage in the flow divider or past the piston seals, the cylinders will operate in unison.

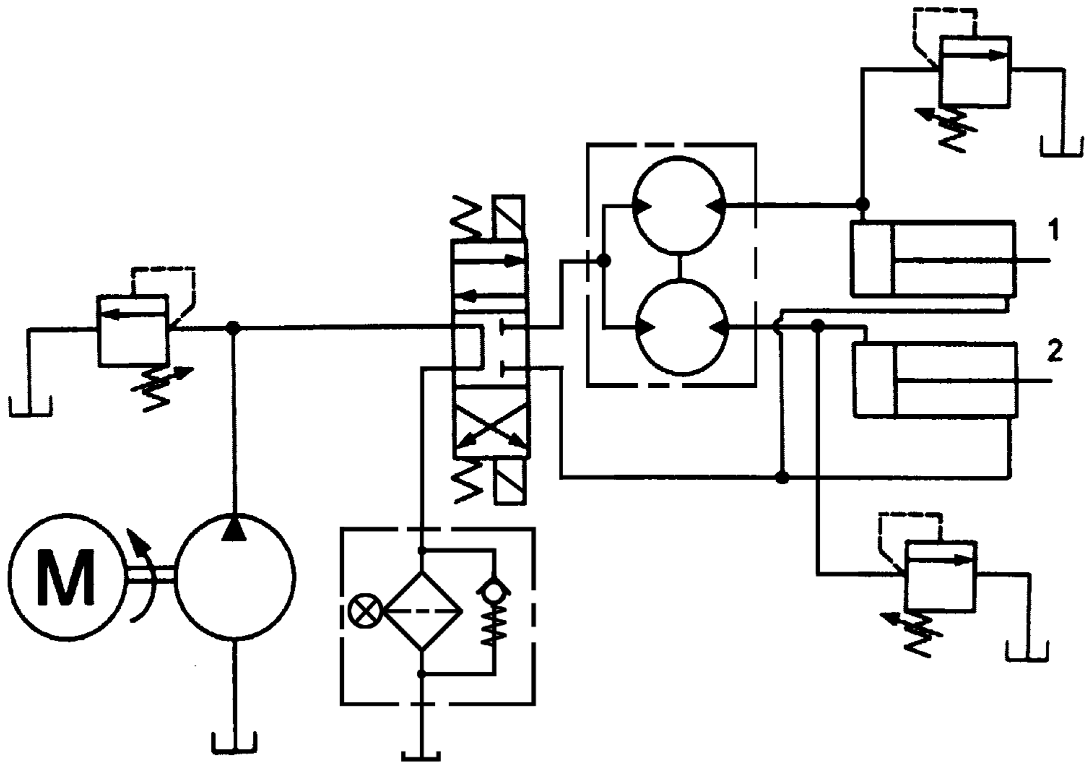


Figure 1-28: Synchronous Circuit with a Gear-type Flow Divider

The circuit in Figure 1-28 includes relief valves located between the flow divider and the cylinders. These relief valves protect the cylinders from over pressure. If one cylinder were to fully extend before the other cylinder, the second cylinder would still be able to accept flow from the flow divider. As the flow divider rotated, metering flow to the second cylinder, fluid would also be pumped by the flow divider to the first cylinder, causing a potentially catastrophic rise in pressure. Most commercially available gear-type flow dividers contain integral relief valves. These relief valves also allow rephasing (end of stroke synchronization) of the cylinders.

A second synchronous circuit is shown in Figure 1-29. Here, two double rod cylinders are connected in series so that fluid returning from the extension stroke of one cylinder is used to extend the second cylinder. Reversing the directional control valve directs fluid to retract the second cylinder, with return fluid from the second cylinder used to retract the first cylinder. One of the problems associated with synchronous cylinder circuits is that unequal loads can cause slippage in one or both of the cylinders, which results in loss of synchronization. Unequal loads, for example cutting metal on one end of a shear that has hydraulic cylinders at each end, can cause the cylinder with the greatest load to lag behind the other.

Correction for minor variation in synchronization at the end of each cycle can be made by using a replenishing circuit to supply makeup fluid for the fluid lost due to leakage. The circuit in Figure 1-29 uses a 4/3 valve to extend and retract cylinders 1 and 2, which are connected in series. Solenoid valve A replenishes the circuit. On extension, if cylinder 1 bottoms first, it contacts a limit switch energizing solenoid valve A that supplies additional fluid to fully extend cylinder 2. On retraction, if cylinder 2 bottoms first, it contacts a limit switch energizing solenoid valve A that supplies additional fluid to retract cylinder 1. In this manner, the strokes of both cylinders can be expected to begin and end when both cylinders are in the same position.

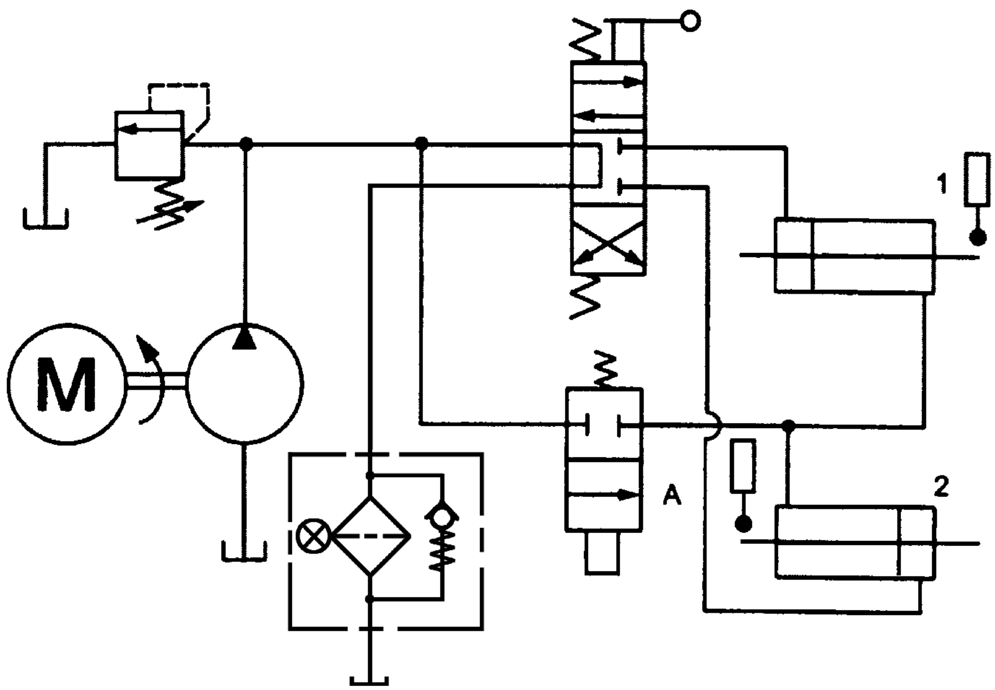


Figure 1-29: Synchronous Circuit with Cylinders Connected in Series

Valve-type flow dividers are available both as flow dividers and as flow dividers/combiners. Valve type flow dividers divide flow in only one direction, while flow dividers/combiners provide synchronization in both directions. Graphical symbols do not differentiate between these dividers and dividers/combiners. A common characteristic of valve-type flow dividers/combiners is that they provide for automatic cylinder rephasing.

Outcome 1.2.2:

Recognize the function of components in regenerative circuits.

Regenerative, also termed regen, cylinder circuits have a common characteristic, which is plumbing that directs rod end flow to the cap end of the cylinder to combine with pump flow as the cylinder rod extends. When the cylinder rod is extending, the effective area of the piston equals the cross-sectional area of the cylinder rod. Beyond this common characteristic, regenerative circuits can be plumbed to achieve a number of unique functions.

A full-time regenerative circuit that extends and retracts a cylinder is shown in Figure 1-30. “Full-time” means that the circuit is always in the regenerative mode of operation. A circuit of this type would extend and retract the cylinder rod of a 2:1 area ratio with the same velocity and force in each direction. However, a review of standard cylinder bore and rod sizes reveals that there are not any standard cylinder bore-rod combinations that provide exactly a 2:1 area ratio, though some combinations are very close. For example a cylinder with a 2.5 inch bore and 1.75 inch diameter rod is approximately 2:1 in piston area. The effective area extending is 2.41 sq-in, while the effective area retracting is 2.50 sq-in ($A_{1.75in}$= 2.41 sq-in; $A_{2.5in}$ = 4.91 sq-in; 4.91 sq-in - 2.41 sq-in = 2.50 sq-in).

In the neutral position, the 4/3 valve directs pump flow across the tandem center directional control valve spool to the reservoir. Shifting the directional control valve to extend the cylinder directs pump flow through the directional control valve from P to A to the cap end of the cylinder. Because the B port of the directional control valve is blocked, the fluid from the rod end of the cylinder cannot flow from B to T through the directional control valve. Instead, the fluid from the rod end of the cylinder flows around the directional control valve to join flow from the pump, and then through the directional control valve, from P to A, to the cap end of the cylinder. Reversing the directional control valve directs fluid from the pump from the P port to the B port. However, since this path is blocked, fluid can flow to the rod end of the cylinder by flowing through the connection between the pump, the relief valve, the directional control valve, and the rod end of the cylinder. Fluid from the cap end of the cylinder flows from A to T through the directional control valve and on to the reservoir.

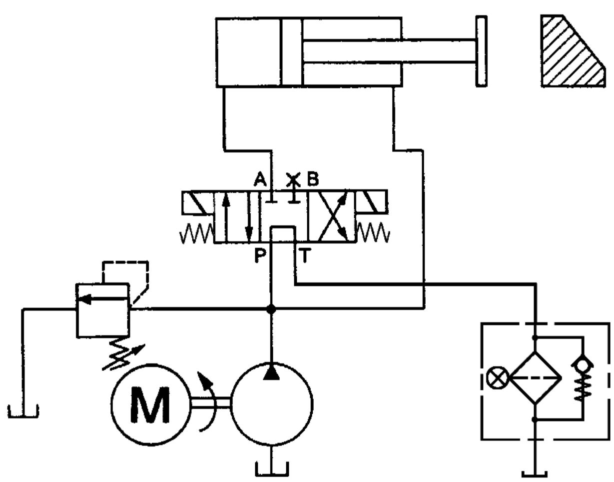


Figure 1-30: Full-time Regenerative Circuit with the “B” Port Blocked

The circuit in Figure 1-31 achieves the same function as the circuit in Figure 1-30 except the B port of the directional control valve is used. This requires that check valves be used to direct return flow from the rod end of the cylinder to the cap end of the cylinder as the rod extends. When the cylinder rod extends, return flow is directed around the directional control valve to join pump flow. When the directional control valve is reversed, flow is through the B port to the rod end of the cylinder.

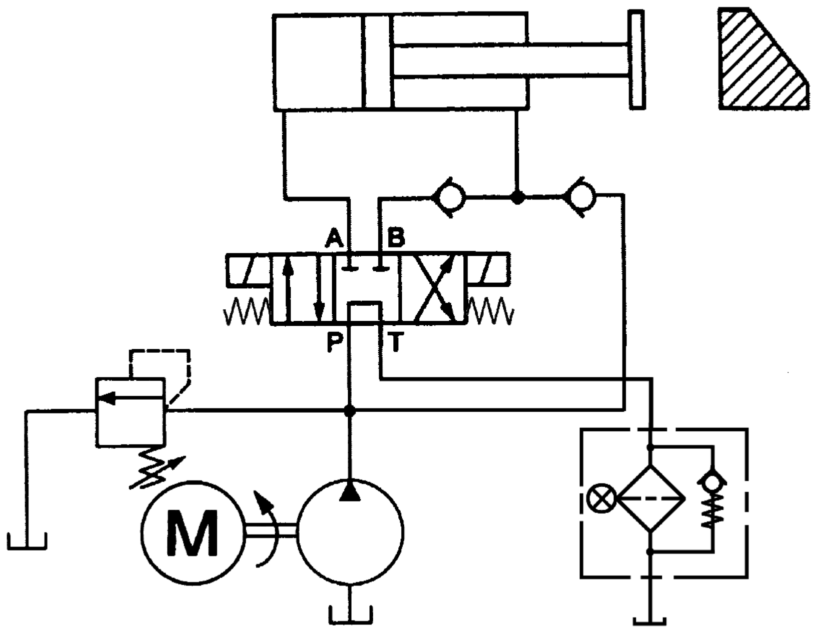


Figure 1-31: Full-time Regenerative Circuit with Two Check Valves

Normally, a cylinder rod will extend with more force than when it retracts because the area at the cap end of the cylinder is greater than the annular area around the rod. Of course, for the same flow, the rod extends slower than it retracts. In a regenerative circuit with a 2:1 cylinder, the force, as well as velocity, is the same extending as retracting. This can be a problem if the work portion of the cycle is at the end of the stroke extending and requires full cylinder force. There are a number of ways to achieve regeneration and still retain the capacity to apply full force at the end of the extension stroke. A circuit to accomplish this function is called a "part-time" regenerative circuit.

The circuit in Figure 1-32 achieves this with a bleed-off orifice at the rod end of the cylinder. On extension, the cylinder regenerates at low force until the rod contacts the work piece. When rod extension stops momentarily, the bleed-off valve allows a small amount of flow to return to reservoir, allowing the pressure at the rod end of the cylinder to fall. This allows full pressure to act against the cap side of the cylinder piston, thus developing full force.

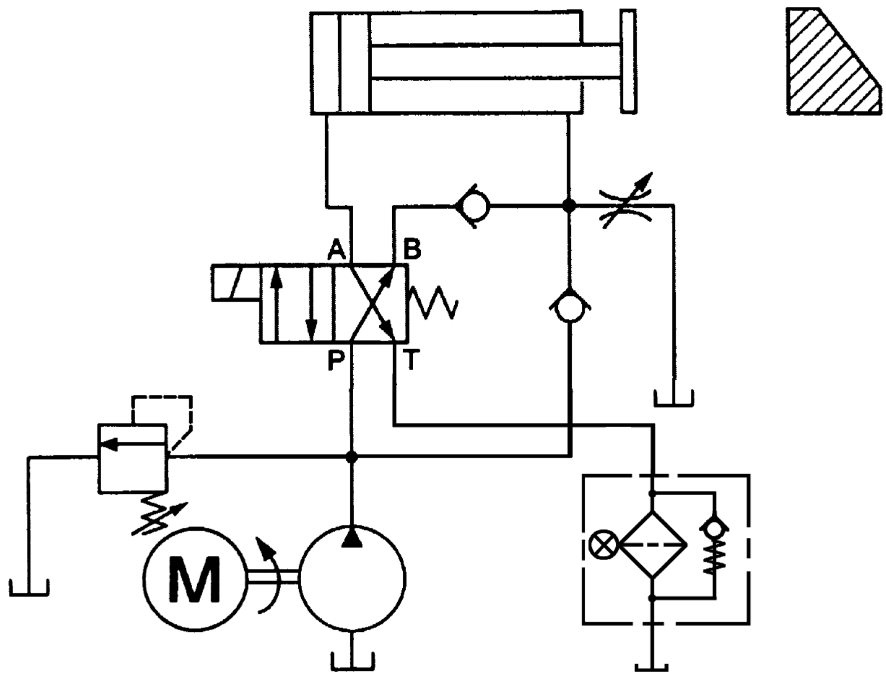


Figure 1-32: Part-time Regenerative Circuit with Bleed-off

When the directional control valve is shifted to retract the cylinder rod in Figure 1-32, pressurized fluid is routed through the B port to the rod end of the cylinder. Some leakage occurs through the bleed-off valve to reservoir, but the amount is small compared to total pump flow and the return stroke usually is not subject to an external load, so high force is generally unnecessary.

Another means to increase the extension force of a cylinder under regeneration is to install a counterbalance valve in the line between the rod end of the cylinder and one of the actuator ports of the directional control valve, and a check valve between the two cylinder ports allowing free flow from the rod end port to the cap end port. Shifting the directional control valve to extend the cylinder directs fluid to the cap end of the piston. Return flow from the rod end of the cylinder regenerates through the check valve to join with pump flow as the counterbalance valve prevents fluid from the rod end of the cylinder from returning to tank through the directional control valve. When the load pressure rises above the pressure setting of the counterbalance valve, the pressure rise is sensed through the external pilot line which is connected to the cap end of the cylinder, the counterbalance valve opens, venting the rod end of the cylinder. As the load pressure rises, the check valve seats, allowing pressure to continue to rise in the cap end of the cylinder. Thus, full force extending is developed after the load pressure rises. When the directional valve is actuated to retract the cylinder, the pilot-operated-to-close check valve is piloted closed, preventing the flow of fluid from the rod end of the cap end of the cylinder. This allows retraction of the cylinder.

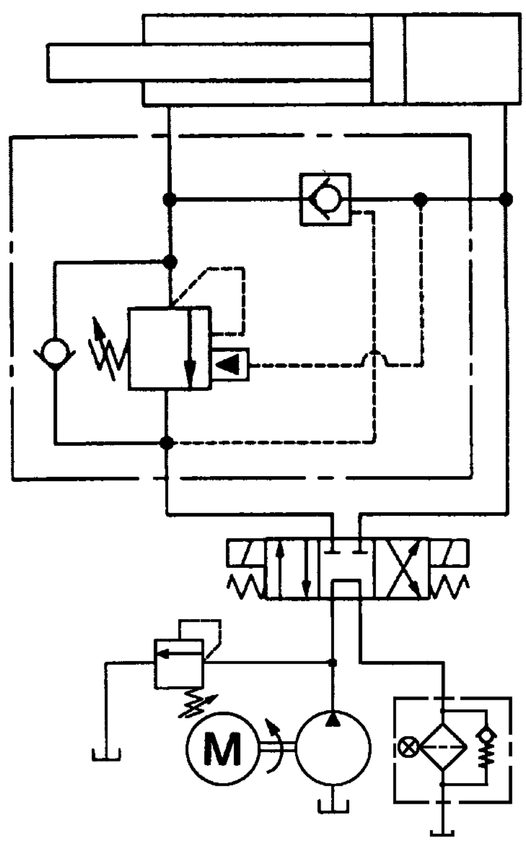


Figure 1-33: Part-time Regenerative Circuit with a Counterbalance Valve

In the circuit shown in Figure 1-33, the assumption is made that the load will not overrun the cylinder. For example, when the control valve is in the position shown, a tractive load would pull the rod out of the cylinder, directing flow through the check valve. However, these circuits are generally used on compacting machinery where the horizontally oriented cylinder is not subjected to an overrunning load.

Figure 1-34 illustrates a circuit that allows selection of a regeneration option with a fourth position in a 4-way directional control valve. The fourth position connects both ends of the cylinder with the pump while blocking flow to tank. For a 2:1 cylinder this gives the options of fast extension at half maximum cylinder force, and slow extension with maximum cylinder rod force. The regenerative position in the directional control valve looks similar to a float position except that both cylinder ports are connected to the pressure port rather than to the tank port.

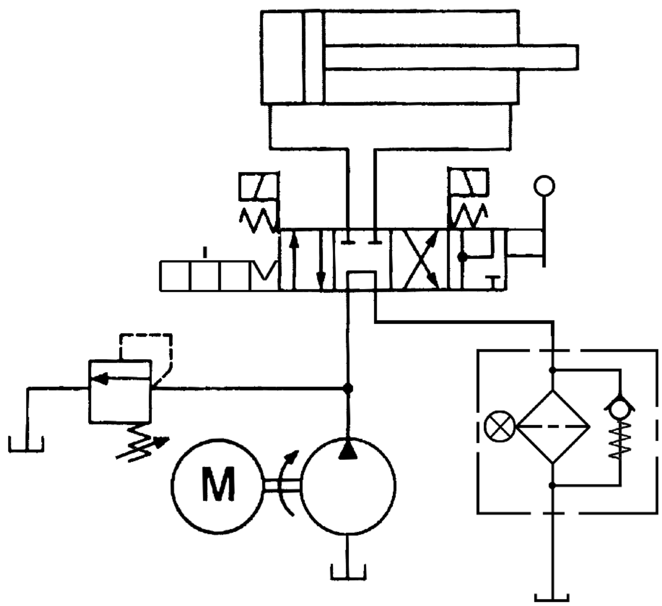


Figure 1-34: Regenerative Circuit with a Regenerative Position in the Directional Control Valve

A point to note regarding the 4/4 directional control valve in Figure 1-34 concerns the placement of the centering springs and solenoids in the symbol. ISO 1219-1 allows for envelope lines to be extended and for alternate locations of operators in an attempt to clarify the operation of the valve. The centering springs are shown to center the valve to a certain center position, with the fourth position controlled by the operation of the lever operator.

The force of the cylinder rod extending during regeneration equals the load pressure multiplied by the rod area. Velocity calculations during regeneration are also based on using the rod area.

Velocity calculations are made for cylinders, including those in regenerative circuits, using the formula:

$\text{Velocity}_{in/min} = (Q_{gpm} \text{ x } 231 \text{ cu-in/gal}) / A_{sq\text{-}in}$ $\qquad V = (Q \text{ x } 231) / A$ $\qquad$ Eq. 1-1

What must be determined first is whether the cylinder rod is extending under regeneration. Then compute the effective area against which hydraulic fluid is acting. If the cylinder rod is extending under regeneration the effective area equals the cross-sectional area of the cylinder rod. If the cylinder is not extending in regeneration, the full bore area is used in the calculation. When calculating the retraction velocity, the annular area must be used. Finally, enter the numbers into Eq. 1-1 to determine extension or retraction velocity in inches/minute. Lines should be sized to accommodate the flows present in the various branches of a regen circuit.

Force is calculated using Pascal's Law:

$Force_{lbs} = Pressure_{psi} \times Area_{sq\text{-}in}$ $F = P \times A$ Eq. 1-2

Area is calculated from the equation:

$Area_{sq\text{-}in} = Diameter_{in}^2 \times 0.7854$ $A = D^2 \times 0.7854$ Eq. 1-3

Review 1.2.2.1: In a full-time regenerative cylinder circuit, what approximate bore to rod diameter ratio would be required to extend and retract the cylinder with the same velocity?

a. 1.414 to 1.000
b. 1.375 to 2.125
c. 2.750 to 1.750
d. 3.250 to 1.750
e. 4.000 to 1.375

Review 1.2.2.2: The regenerative circuit in Figure 1-32 has a 2.5 inch bore and a 1.750 inch diameter cylinder rod. If maximum cylinder pressure is set at 1500 psi, what force will the cylinder rod exert after the cylinder stalls?

a. 1178 Ibs
b. 3608 lbs
c. 3755 lbs
d. 6378 lbs
e. 7365 lbs

Outcome 1.2.3:

Recognize the placement of pressure relief, reducing, sequence, and counterbalance valves in a circuit.

The circuit in Figure 1-35 shows the placement of pressure relief, pressure reducing, sequence, and counterbalance valves in a circuit. The pressure relief valve, located near the pump, limits maximum pressure in the system, protecting the pump, the lines and the actuators from over pressure. The pressure reducing valve, located near the clamp cylinder, limits the maximum clamping pressure, and thus the clamping force. The sequence valve, located at the cap end of the work cylinder, responds to pressure rise in the system to extend the rod of the clamp cylinder first, followed by extension of the work cylinder. The counterbalance valve at the cap end of the work cylinder prevents the load on the work cylinder from falling. The sequence valve located at the rod end of the clamp cylinder responds to pressure rise in the system to lower the work cylinder first, followed by retraction of the clamp cylinder.

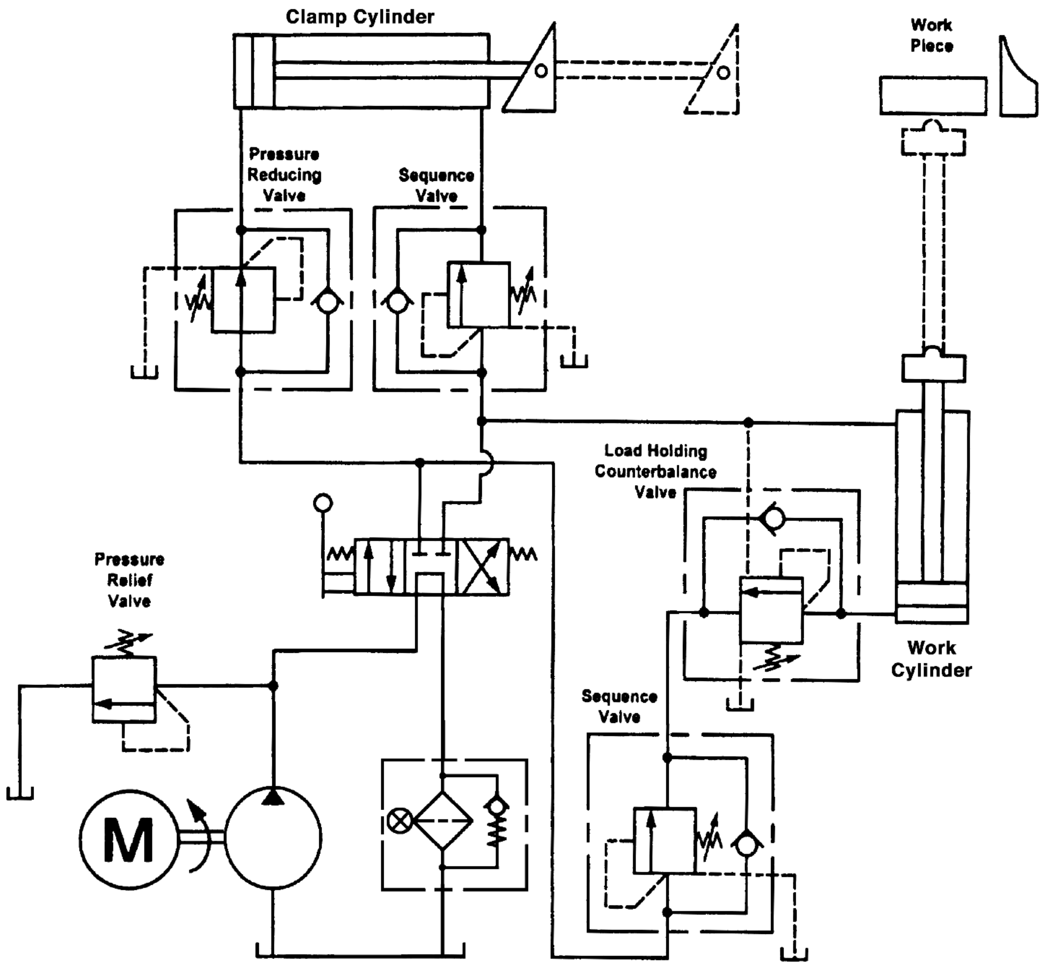


Figure 1-35: Pressure Relief, Pressure Reducing, Sequence, and Counterbalance Valves in a Circuit

The arrangement of the valves in the circuit shown in Figure 1-35 results in a number of pressure levels that can be identified and calculated during operation of the circuit:

1. Low pressure standby with the pump unloading through the open center directional control valve to the reservoir.

2. Intermediate clamp cylinder pressure while the cylinder rod is extending.

3. Maximum, but reduced, clamp cylinder pressure on the cap side of the piston, controlled by the pressure reducing valve when the clamp cylinder stalls extending.

4. Increased pressure at the pump when the sequence valve at the base of the work cylinder opens after the clamp cylinder stalls extending, to begin extension of the work cylinder.

5. Load pressure while the work cylinder rod is extending.

6. Maximum system pressure when the work cylinder stalls extending and the pressure relief valve opens to direct excess fluid to the reservoir.

7. Load pressure at the rod end of the clamp cylinder when the sequence valve opens.

The pressure relief valve protects the pump, plumbing, actuators, and the work piece from over pressure. Maximum pressure occurs at the pump when the work cylinder stalls extending and the clamp cylinder stalls retracting. The system is least efficient when fluid is dumped over the relief valve because the flow over the relief valve at maximum flow and pressure is converted to heat, as the input horsepower is not being converted to useful work. The choice of loads, pressure settings, and components will affect system efficiency.

The purpose and function of the sequence, pressure reducing, and counterbalance valves have already been discussed; further discussion of their purpose in this circuit is would be redundant.

Review 1.2.3.1: Which valve shown in Figure 1-35 would require the highest pressure setting?

a. counterbalance
b. pressure relief
c. pressure reducing
d. work cylinder sequence valve
e. clamp cylinder sequence valve

Job Responsibility 2.0: Analyze Loads and Motion

Job Responsibility 2.0: Analyze Loads and Motion

Load and motion analysis requires a basic understanding of cylinder pressures, areas, and forces, torque, speed, power, horsepower, friction, pulleys, ratios, and trigonometry.

Task 2.1: Identify system parameters (linear and rotary).

Outcome 2.1.1:

Recognize factors associated with hydraulic power systems.

Hydraulic forces are applied to extend and retract cylinders, to continuously rotate the shafts of hydraulic motors, and to rotate the shafts of rotary actuators clockwise and counterclockwise. Forces to be considered consist of breakaway forces, forces to accelerate and decelerate loads, force to maintain motion, and holding forces.

Force results from sufficient pressure applied against a movable area. In a cylinder, the moveable area is the piston. Pressure, force, and area are computed from the following formula, which is generally referred to as Pascal's Law:

$\text{Force}_{lbs} = \text{Pressure}_{psi} \text{ x } \text{Area}_{sq\text{-}in}$ $\qquad$ $F = P \text{ x } A$ $\qquad$ Eq. 2-1

Several commonly used conversion factors for pressure are shown in Table 2-1. Standard reference manuals list many more conversion factors. For example, to convert 1000 psig to kilo Pascals, (kPa) multiply 1000 psig by 6.895: 1000 psig x 6.895 = 6895 kPa.

To convert to: From:	psi	kPa Multiply by	MPa	bar
psi	1	6.895	0.006 895	0.069
kPa	0.145	1	0.001	0.010
MPa	0.000 145	0.001	1	0.000 010
Bar	14.5	100	0.100	1

Note: 1 Pa (Pascal) = 1 N/m^2 (Newton/meter^2)

Table 2-1: Pressure Unit Conversion Factors

Job Responsibility 2.0: Analyze Loads and Motion

Hydraulic energy in industrial plants is produced by electric motors that drive pumps, with the motor horsepower determined by the pressure and flow requirements of the system. The hydraulic power unit does not need to be at the point of use. It can be located many feet away with the fluid transported between the power unit and the machine through fluid conductors. The distance between the power unit and the machine creates a pressure loss that increases in proportion to the distance. This pressure loss can be minimized by properly sizing the fluid conductors in an effort to minimize the pressure drop. Hydraulic systems, consisting of reservoirs, pumps, valves, actuators, and fluid conditioning components, together with fluid conductors, are prone to leakage. Leakage presents an environmental hazard that must be solved by by applying good design practices. Hydraulic systems are also sensitive to variations in temperature since fluid components and the fluid expand and contract as the operating and ambient temperatures rise and fall. Resistance of the fluid to fire is an important factor, especially around heat sources and open flames. Hydraulic systems are particularly well adapted to moving and controlling heavy loads at slow speeds in linear, angular, and rotary motion vectors. For example, hydraulic cylinders, with large bores, can extend and retract heavy loads at slow speeds with a high degree of accuracy. However, smaller cylinders, such as those used on ride simulators, are capable of generating significant force with high accuracy at fairly fast speeds. Rotary actuators are used to rotate loads through arcs of up to 1440 degrees. Hydraulic motors offer a wide range of speed capability, ranging from less than one rpm to over 6000 rpm, operating large equipment such as drills, winches, and wheel motors, as well as smaller items such as machine spindles.

The thrust from cylinders, torque from hydraulic motors, and angular force from rotary actuators, are determined by a wide range of available pressures. Hydraulic systems incorporate pressure control valves that limit linear or angular force to predetermined values. Pressure control valves also protect the system from exceeding a maximum force that would otherwise damage the system. In this respect, an important advantage of fluid power systems, over mechanical and electrical systems, is that force or torque can be safely and indefinitely maintained against a load. Stalling an actuator against the load will not cause damage to the hydraulic system components.

Review 2.1.1.1: Which one of the following would be reduced to lower the extension force of a hydraulic cylinder?

a. Flow
b. Rod diameter
c. Cylinder stroke
d. System pressure
e. Displacement of the pump

Review 2.1.1.2: A pressure gauge indicates 1750 psi. What is the approximate pressure in kPa?

a. 12 kPa
b. 120 kPa
c. 1200 kPa
d. 12000 kPa
e. 120000 kPa

Review 2.1.1.3: A pressure gauge indicates 150 bar. What is the approximate pressure in psi?

a. 10.34 psi
b. 21.75 psi
c. 217.5 psi
d. 1034 psi
e. 2175 psi

Outcome 2.1.2:

Recognize basic design considerations.

Systems are made up of one or more circuits. A system is used to complete the work cycle, while a circuit is used to complete one or more specific tasks in the work cycle. For example, a hydraulic power unit supplying a table grinder with a hydraulic table feed cylinder, a hydraulic motor driven grinder head, control valves, plumbing and accessories, would constitute a grinding system. The accessory package could include a filter circuit to keep the fluid clean and a heat exchanger to cool the fluid. Systems are more complex than circuits because circuits must work together in the total package.

There are a number of ways to design a system. Most include the following steps:

1. The work requirements must be defined first.

2. System pressure and fluid are selected next.

3. Actuators are sized to move the load at less than the maximum system pressure.

4. The work cycle is plotted to established flow rates, to size the component and fluid conductors, and to determine the control functions.

5. The machine design and control logic is analyzed for safe operation and compliance with Occupational Safety and Health Administration Standards.

6. The system is constructed and the performance is checked against the predetermined performance specifications.

In general, system pressures are classified as low (under 1000 psi) medium (1000 psi to 3000 psi), or high (above 3000 psi). However, there is no universal agreement on what low, medium, and high pressure means.

System pressure also influences component size. For example, for a given load, operating at lower pressures requires larger bore cylinders, while higher operating pressures allow smaller bore cylinders to be used. The adoption of higher pressures in mobile equipment was influenced by space and weight limitations of the hydraulic package.

Job Responsibility 2.0: Analyze Loads and Motion

Task 2.2: Determine the work profile to move loads (force, distance, work, torque, speed, velocity, and power).

Outcome 2.2.1:

Apply linear load and motion parameters.

Figure 2-1 illustrates two cylinders being used in a hydraulic system. Cylinder 1 moves the load horizontally and cylinder 2 lifts the load vertically. The pressure-time plot of both cylinders extending is shown in Figure 2-2. When cylinder 1 extends to move the load horizontally, the pressure curve spikes up at the beginning as breakaway forces, an effect of static friction (sometimes called "stiction"), are overcome and the load begins to move. An additional force to be considered is the force of acceleration. The initial pressure curve is followed by a straight horizontal line as, representing a constant state force, as the load is pushed over to cylinder 2. The horizontal pressure line drops back to zero when the cylinder stops extending. When cylinder 2 begins to extend, the pressure line spikes slightly, as stiction in the cylinder and the force of acceleration are overcome. Thereafter the pressure line becomes horizontal, remaining remains slightly below the initial pressure level when the cylinder comes to rest at the end of the stroke, because the cylinder is still supporting the load. In the pressure-time plot in Figure 2-2, cylinder 1 extends under load for 10 seconds, while cylinder 2 extends under load for 5 seconds.

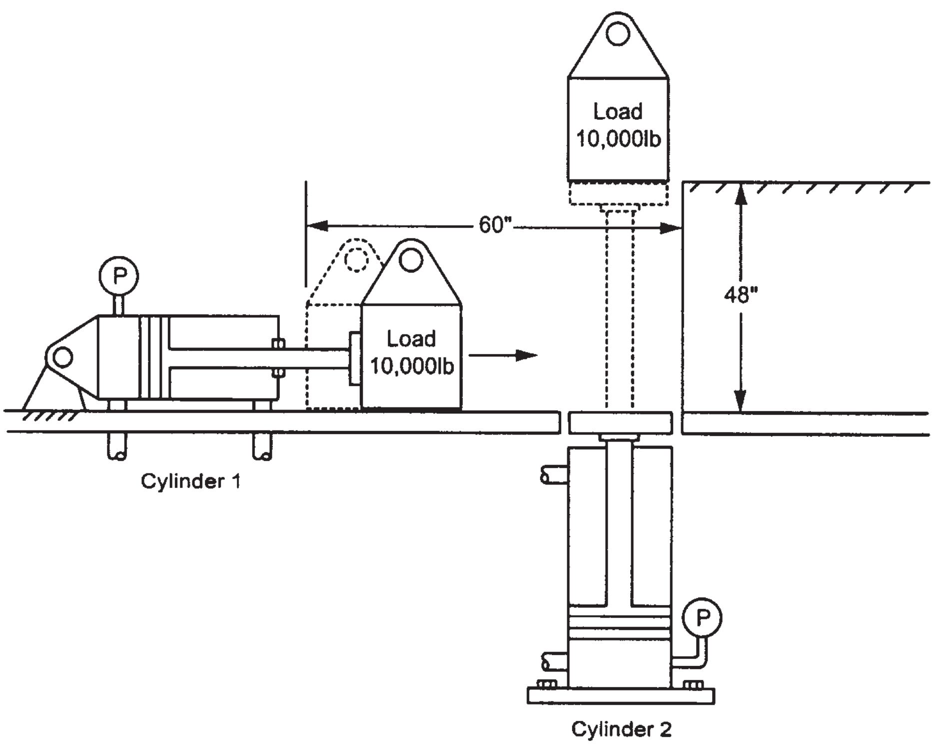


Figure 2-1: Circuit to Illustrate Outcome 2.2.1

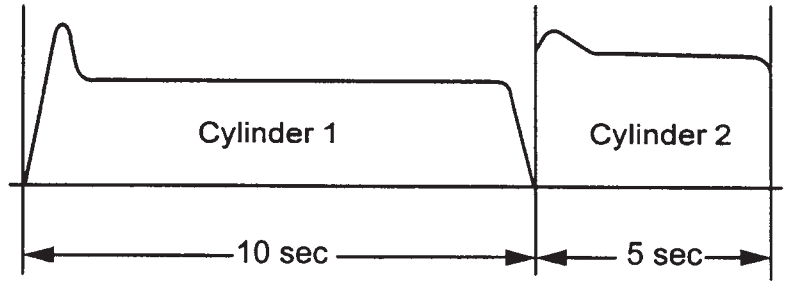


Figure 2-2: Pressure-Time Plot for the Circuit in Figure 2-1

Design calculations follow for the extension stroke of the cylinders in Figure 2-1.

Step 1: Select the system pressure and fluid.

The relief valve pressure will be set at 2500 psi. Petroleum base fluid will be used.

Step 2: Size the actuators to move the load at less than system pressure.

The weight of the load on cylinder 1 is 10,000 lbs. The load will be moved through a stroke of 60 inches. With moderate friction, it would be safe to estimate a cylinder force requirement of 4000 lbs to extend cylinder 1. (Calculation of the force with a friction factor will be explained later in this section.)

In order to keep the pressure in cylinder 1 below 2500 psi, a 2 inch bore cylinder is selected. It is assumed that the rod would have sufficient rigidity (column strength) such that rod sagging, buckling, and jack-knifing will not be a factor through the 60 inch stroke.

Solve for the pressure in the cylinder extending using Eq. 1-2 and Eq. 1-3:

$\text{Force}_{\text{lbs}} = \text{Pressure}_{\text{psi}} \text{ x Area}_{\text{sq-in}}$ $\quad F = P \text{ x } A$ $\quad$ Eq. 1-2

$\text{Area}_{\text{sq-in}} = \text{Diameter}_{\text{in}}^2 \text{ x } 0.7854$ $\quad A = D^2 \text{ x } 0.7854$ $\quad$ Eq. 1-3

Area = (2 inches)^2 x 0.7854 = 3.14 sq-in

4000 lbs = Pressure x 3.14 sq-in

Pressure = 4,000 lbs. / 3.14 sq-in = 1273 psi

Next, solve for the static pressure required by Cylinder 2 to balance the load using a 2.5 inch bore cylinder. Again, it is assumed the cylinder applies only the lifting force and that friction is not a factor. In actual practice, a margin of safety is added in order to allow for system losses, friction, and the force of acceleration. Since these factors can vary greatly depending on the system dynamics, the static pressure required to balance the load shall be used in the calculations in this manual and on the exam.

Area = (2.5 inches)^2 x 0.7854 = 4.91 sq-in

10000 lbs = Pressure x 4.91 sq-in

Pressure = 2037 psi

The load pressure, or “working” pressure, should be 150 to 1000 psi below the relief valve or pump compensator setting.

Cylinders with smaller bores could have been selected without exceeding 2500 psi, but friction and inertia will cause the pressure to increase to higher levels. The friction that must be overcome is caused by several factors. One factor is the mechanical friction between the dynamic seals of the cylinder and the cylinder walls and rod. In addition, as the fluid flows through the fluid conductors, pressure drops as a function of both the internal friction of the fluid as well as the friction between the fluid and the walls of the conductors. As fluid velocity increases, and as the amount of bends in the conductors increase, pressure drop increases. This pressure drop is a measure of the amount of fluid friction evident in the system. Good system design strives to minimize this pressure drop. For the calculations in this manual and on the exam, losses due to mechanical and fluid friction shall be ignored unless they are specifically included in the problem. An example of the inclusion of friction as a factor is when the coefficient of friction is discussed.

Step 3: Plot the work cycle, calculate flow rates, and size components.

The cylinder extension portion of the work cycle is shown in Figure 2-2. Before calculating the flow rate required to extend the cylinder, the velocity of the cylinder must be calculated. For cylinder 1, solve Equation 2-1 for this value:

$\text{Velocity}_{in/min} = \text{Stroke}_{in} \text{ x } (60_{sec/min} / \text{Time}_{sec})$ $V = S \text{ x } (60 / T)$ Eq. 2-1

Velocity = 60 inches x (60 / 10 seconds) = 360 in/min

Calculation of the flow rate to a cylinder is given by Eq. 2-2 as:

$Q_{gpm} = (\text{Velocity}_{in/min} \text{ x Area}_{sq\text{-}in}) / 231_{cu\text{-}in/gal}$ $Q = (V \text{ x } A) / 231$ Eq. 2-2

The symbol “Q” is an engineering term used to denote “flow.”

Solve Eq. 2-2 for flow rate to extend Cylinder 1:

Q = (360 in/min x 3.14 sq-in) / 231 = 4.89 gpm

Solve Equations 2-1 and 2-2 for cylinder 2:

Velocity = 48 inches x (60 / 5 seconds) = 576 in/min

Q = (576 in/min x 4.91 sq-in) / 231) = 12.24 gpm

Thus the pump would have to supply approximately 5 gpm to cylinder 1, and 13 gpm to cylinder 2. If the idle portion of the cycle were longer than the work portion of the cycle, a 13 gpm fixed displacement pump would be efficient at the larger flow rate, but would dump approximately half its flow over the relief valve while extending cylinder 1. One way to solve this problem would be to use a double pump arrangement with a 5 gpm and 13 gpm pump segments. This would be similar to the application of steering and implement pump segments used on field tractors.

Figure 2-3 shows a circuit for the two cylinders that uses a fixed displacement double pump, two manually operated 4/3 (4-way, 3-position) directional control valves, and a counterbalance valve for cylinder 2 to prevent the load from dropping. No attempt is made here to size fluid conductors or to put interlocks in place, to make the cylinders sequence in a prescribed order.

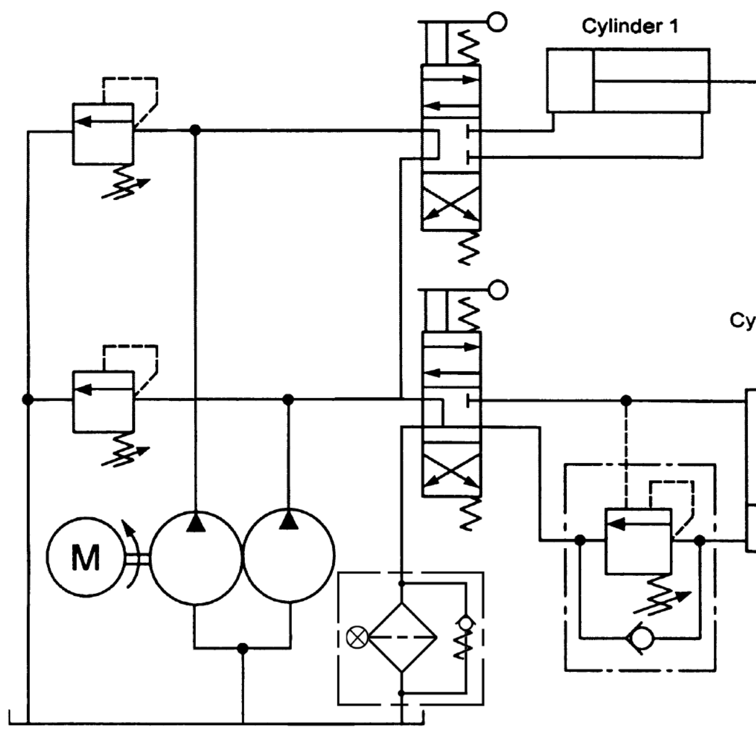


Figure 2-3: Circuit for the Two Cylinder Application in Figure 2-1

Outcome 2.2.2:

Solve pulley systems for force at the point of application.

Cylinders are often used with cables and pulleys to reverse direction, to increase lifting force, to provide tension force for feed-through systems such as conveyor belts, and to shorten or lengthen travel mechanisms.

The simplest use of a pulley is to reverse direction. In Figure 2-4a, effort is applied to pull the cable down in order to lift the load. There is one sheave (pulley) and two cables. The pulley axle is stationary, which is important to notice, because unless the pulley itself moves linearly there is no multiplication of force. Thus if friction is neglected, an effort of 1000 lb applied downward in one cable results in a force of approximately 1000 lb upward in the other cable. The direction is reversed, but there is no multiplication of force.

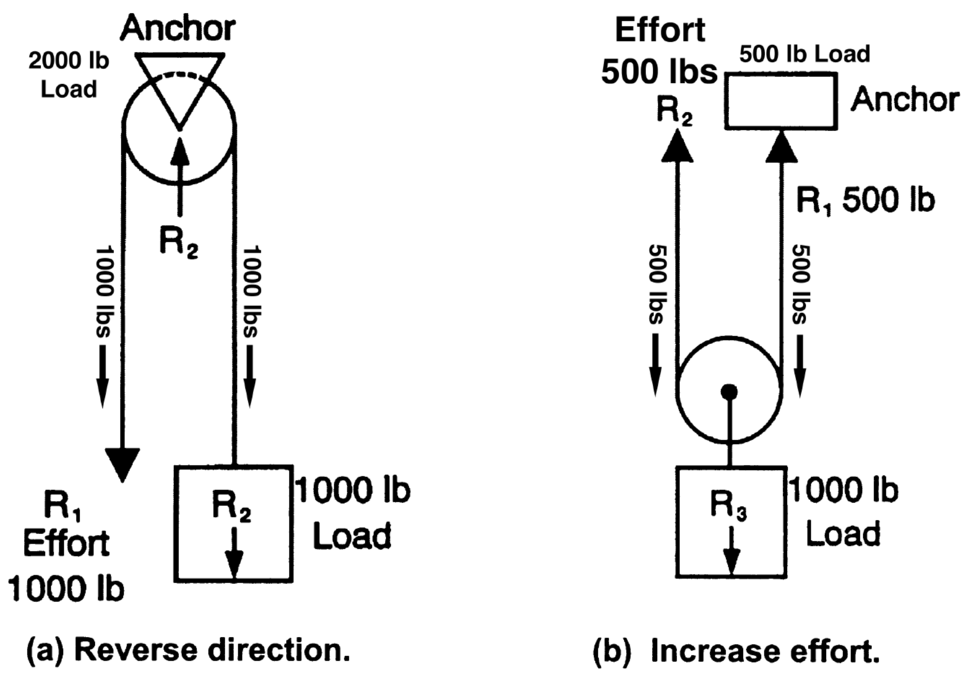


Figure 2-4: Pulley Systems to: a) Reverse Direction, and: b) Decrease Effort.

In Figure 2-4b, there also is one sheave and two cables, but one of the cables is anchored while the other cable is pulled upward. Pulling the left cable up two units of length will pull the sheave's axle up one unit of length. In this example, there is a multiplication of force at the point of application which is at the axle of the sheave, so that an effort of 500 lb in the left cable will result in a 500 lb force in the right cable, resulting in a 1000 lb force at the point of application. This leads to an important rule for pulleys:

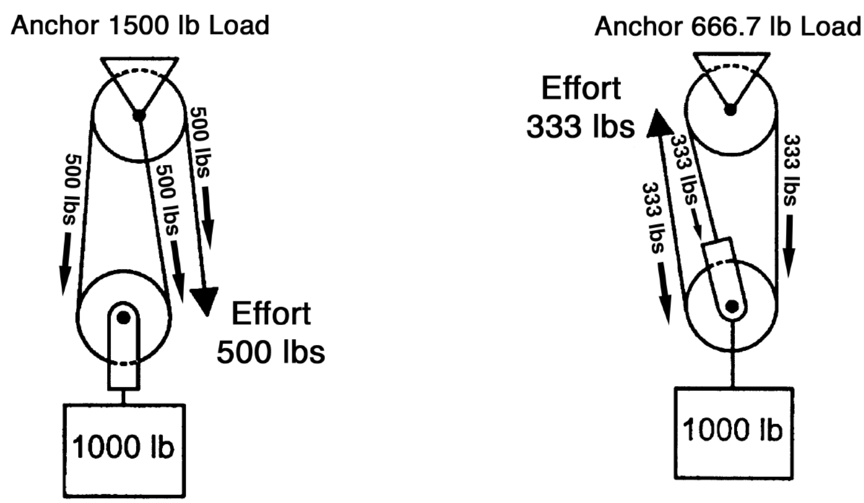
Pulley Rule: **The effort at the point of application equals the load supported by the pulley divided by the number of cables supporting the load.**

An equation representing this relationship is:

$Effort_{lbs} = Load_{lbs}$ / # of Load Bearing Cables   E = L / #LBC   Eq. 2-3

Force at the point of application can be achieved by adding more sheaves to the arrangement. Figure 2-5 illustrates the effort required to lift 1000 lbs at the point of application. Figure 2-5a is a combination of Figures 2-4a and 2-4b. Multiplication of force is achieved by a movable pulley and two cables, while the change of direction is made with the stationary pulley. Multiplication of effort is achieved at the expense of the distance the load is moved. The cable in Figure 2-5a, at the place where the effort is applied, moves twice as far as the load at the point of application.

Figure 2-5b also illustrates the multiplication of force with a pulley system, but it differs from Figure 2-5a in that there are three cables between the movable pulley and the effort that is applied. Two of the three cables between the movable pulley and the effort are looped around the lower pulley, while the third strand attaches to the axle of the pulley. The three cables between the movable pulley and the effort multiplies the effort by a factor of three. Thus, the application of 333 lbs of effort results in a lifting capacity of 1000 lbs at the point of application.



(a) 2x multiplication.   (b) 3x multiplication.

Figure 2-5: Multiplication of Effort

Hydraulic cylinders are applied to pulley systems in a number of ways. For example, in Figure 2-6 the cylinder is multiplying the travel distance where the cable on one side of the sheave is attached to the load at the point of application, and the cable on the other is attached to the anchor.

In Figure 2-6, if the load at the point of application is 2000 lb, the applied force at the axle of the pulley would be 4000 lbs, while the distance the load is moved would be twice the stroke of the cylinder. In applications like fork lifts, this reduces the stroke of the cylinder while still providing significant lifting force.

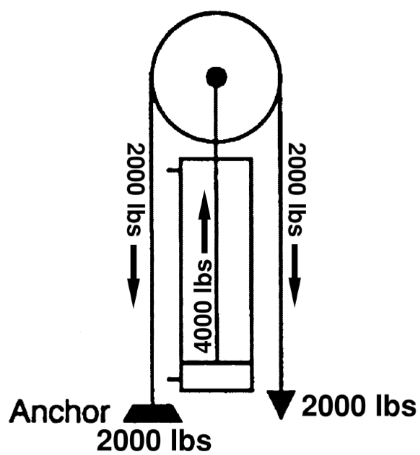


Figure 2-6: Application of a Hydraulic Cylinder and a Pulley

Review 2.2.2.1: Using Figure 2-6, what minimum pressure would be required in a hydraulic cylinder with a 3 inch bore to raise a 2000 lb load? Assume 100% efficiency.

a. 566 psi
b. 650 psi
c. 2000 psi
d. 4000 psi
e. 8000 psi

Task 2.3: Perform load calculations using the principle of moments.

Outcome 2.3.1:

Understand the principles of levers.

Several applications in hydraulics can be simplified and explained with levers and lever terminology. In addition, solving equations for levers is required to calculate the mechanical advantage of cylinders in certain applications. One of these applications is the reaction force applied to the rod bearing by a cylinder rod subjected to a side load. Another application involves cylinder and boom arrangements. In the following discussions, the weight of the boom is neglected.

A first class lever applies a force at one end of the lever, which acts across a fulcrum to lift a load at the other end of the lever. The lever pivots about the fulcrum somewhat like a teeter-totter.

In a second class lever, the fulcrum is moved to one end. The effort is applied to the other and the load acts at a point somewhere between the two ends. A wheelbarrow is an example of a second class lever.

In a third class lever, the fulcrum is at one end and the load is applied at the other end. The effort is applied at a point somewhere between the two ends. A boom is an example of a third class lever.

It must be remembered that a discussion of levers holds true only when the lever is at a 90° angle to the fulcrum. As soon as the lever begins to move, the change in the angle will require trigonometric calculations of angles in order to solve for the direction and the magnitude of the forces involved.

The relationship between force and distance is given by the equation:

$\text{Load}_{lbs} \text{ x Load Distance}_{ft} = \text{Effort}_{lbs} \text{ x Effort Distance}_{ft}$

$L \text{ x } LD = E \text{ x } ED$ Eq. 2-4

This equation can be used regardless of the class of lever being analyzed.

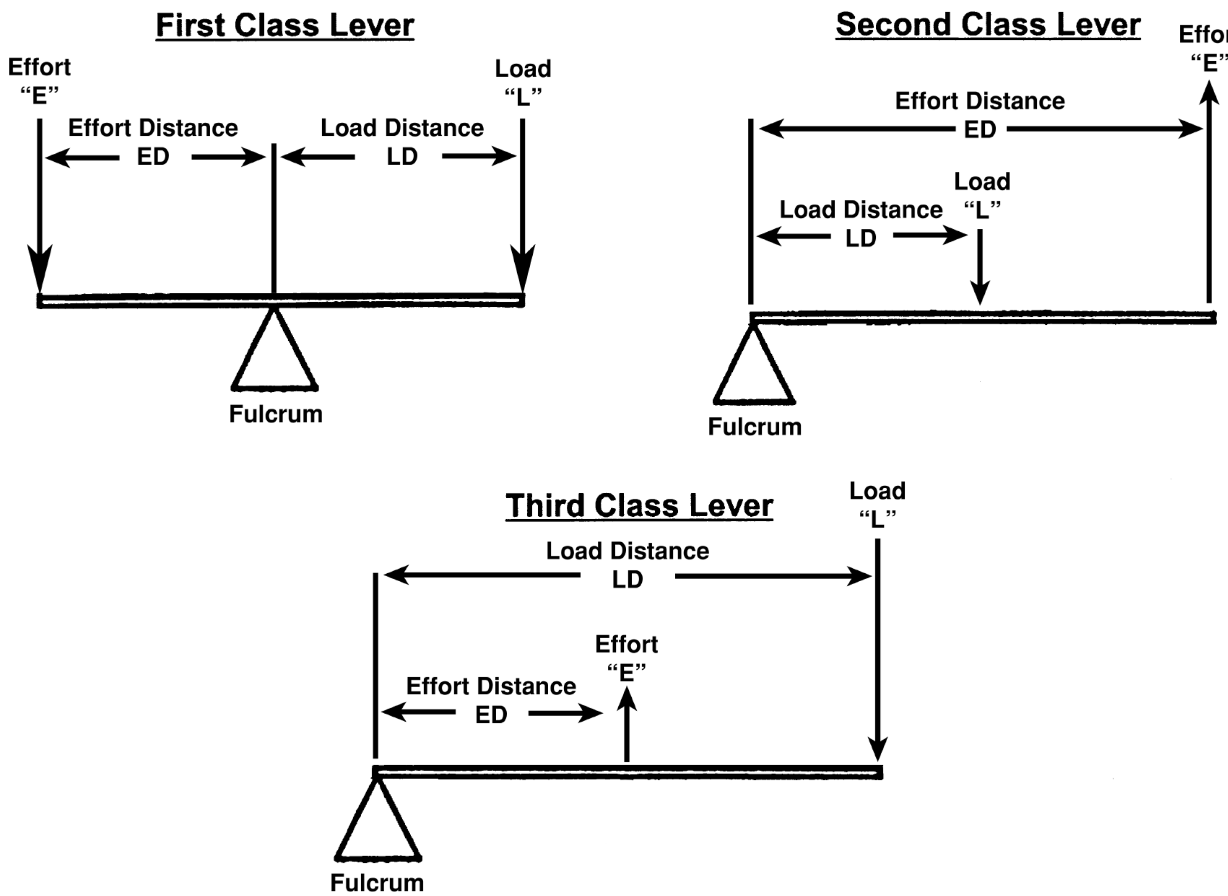


Figure 2-7: First, Second, and Third Class Levers

Job Responsibility 2.0: Analyze Loads and Motion

Outcome 2.3.2:

Solve for the reaction force on a cylinder rod bearing from the stroke, the mechanical advantage, and the side load on the rod.

Side loads on cylinder rods apply a reaction force to the rod bearing of a cylinder. The cylinder rod is a third class lever with the piston acting as a fulcrum at one end, and the side load (side force) acting on the rod at the other end. The reaction force is applied to the rod bushing which is located somewhere in between the fulcrum and the side load. The side load acting on the rod end has a tendency to rotate the rod about the piston. If the side load on the half-extended cylinder rod shown in Figure 2-8 puts a 100 lb force in a downward direction on the rod bushing, then the piston would exert a 100 lb force against the rod bearing. Overloading the cylinder in this manner will wear the rod bearing and rod seals on one side, and the piston and the cylinder bore on the other side. If the lubrication film ruptures, galling can be expected.

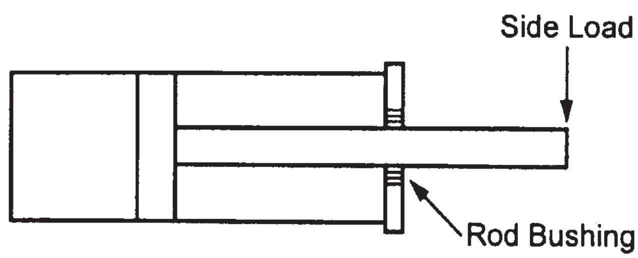


Figure 2-8: Reaction Force on a Cylinder Rod

The mechanical advantage of the side load acting against the rod bearing can be calculated from the equation:

Mechanical Advantage = Total Rod Length$_{in}$ / Supported Rod Length$_{in}$

MA = TRL / SRL Eq. 2-5

Using the Third Class lever equation: MA = ED / LD

The reaction force acting against the rod bearing is the mechanical advantage times the magnitude of the side force acting against the end of the rod:

Reaction Force$_{lbs}$ = Mechanical Advantage x Side Load$_{lbs}$ RF = MA x SL Eq. 2-6

Figure 2-9 illustrates rod buckling and rod cocking caused by side loading on a cylinder rod. As the rod extends from the cylinder, a constant side loading force acting against the end of the rod will increase the force against the rod bearing. If side loading is great enough, especially as the cylinder becomes fully extended, the rod will cock, causing premature wear at the rod bearing. This wear can damage the rod bearing, the rod, and then the rod seals. This cocking occurs because as the rod extends, the leverage against the rod bearing increases.

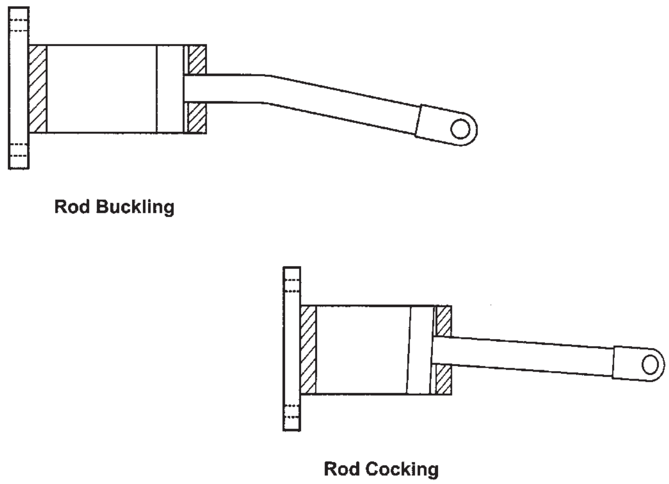


Figure 2-9: Rod Buckling and Rod Cocking Caused by Side Loading

One method used to reduce the effect of side loading is to specify the cylinder with a longer stroke than is required and then limiting the stroke with a stop tube such as that shown in Figure 2-10. The stop tube prevents the piston from fully extending, reducing the magnitude of the reaction force by reducing the mechanical advantage of the side load.

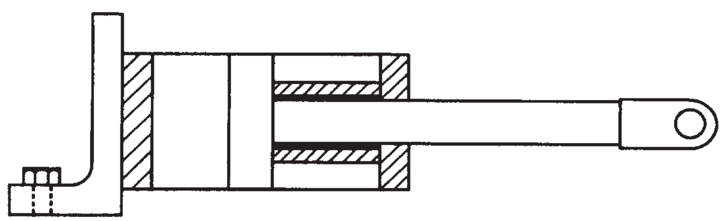


Figure 2-10: Cylinder Equipped with a Stop Tube in Order to Reduce the Reaction Force

Review 2.3.1.1: If a hydraulic cylinder, similar to the one shown in the figure 2-9, has a 20 inch stroke, is extended 8 inches, and has a 90 lb side load acting against the end of the rod, what would be the reaction force acting against the rod bushing?

a. 36 lbs
b. 54 lbs
c. 90 lbs
d. 150 lbs
e. 225 lbs

Job Responsibility 3.0: Select Components for Hydraulic Applications

Task 3.1: Specify hydraulic cylinders.

Outcome 3.1.1:

Size cylinders.

The bore and stroke of cylinders are sized to move loads, within given pressure and flow parameters, a given distance. Oversizing the bore of a cylinder results in the needless expense of an oversized pump, valving, and fluid conductors. Undersizing the bore of the cylinder will result either in stalling the cylinder or the need for higher system pressures to move the load. The bore of the cylinder is initially sized to move the load, neglecting friction. Then the bore is oversized in order to allow for the force of acceleration and to accommodate expected overloading caused by friction and load factors. The stroke is given by the mechanical dimensions of the machinery.

Sizing the bore and establishing the working stroke for a cylinder is only part of cylinder the specification. The rod end style and the cylinder mounting must also be selected. The rod diameter must be determined with rod sag, column strength, and possible cylinder jackknifing taken into consideration. Port type, size, and location, whether or not cushions are necessary, whether a stop tube is required in the case of a long stroke, and seals are among the factors which need to be analyzed.

The bore size is determined from the load and the working pressure of the system using Eq.1-2 and Eq. 1-3:

$\text{Force}_{\text{lbs}} = \text{Pressure}_{\text{psi}} \text{ x } \text{Area}_{\text{sq-in}}$ $\qquad F = P \text{ x } A$ $\qquad$ Eq. 1-2

$\text{Area}_{\text{sq-in}} = \text{Diameter}_{\text{in}}^2 \text{ x } 0.7854$ $\qquad A = D^2 \text{ x } 0.7854$ $\qquad$ Eq. 1-3

Review 3.1.1.1: What minimum bore diameter cylinder is required to lift a load of 8000 lbs on the extension stroke at an operating pressure of 1500 psi?

a. 2.61 inches
b. 5.33 inches
c. 6.78 inches
d. 8.42 inches
e. 9.10 inches

Outcome 3.1.2:

Compute the bore diameter and pressure for a cylinder to move loads while allowing for a coefficient of friction (friction factor).

Friction increases the force required to move a given load along a surface. If a load resting on a horizontal surface is lifted there is no friction between the surface and the load. Only the dead weight of the load needs to be considered. If the load is pushed horizontally, friction between the load and the surface impacts the force required to move the load. How much the force is changed is determined by the coefficient of friction. The force required to move a load along a horizontal surface is computed as the product of the coefficient of friction and the load:

$\text{Force}_{lbs} = \text{Coefficient of Friction x Load}_{lbs}$

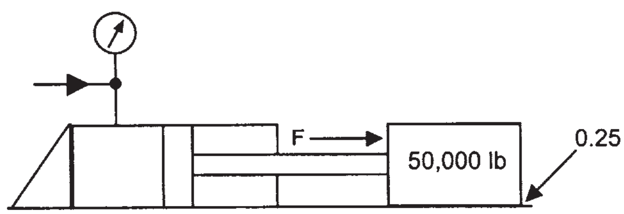
$\text{Force}_{lbs} = C_f \text{ x Load}_{lbs}$ $F = C_f \text{ x } L$ Eq. 3-1

The coefficient of friction may be lower or higher than "1." If the load is resting on a smooth bearing surface, or is on wheels that are rolling on a firm surface, the C_f will be less than 1. If the surface is rough, the C_f would likely be greater than 1.

If the load is to be moved by a hydraulic cylinder, pressure and bore size are computed after determining the force from the load and coefficient of friction.

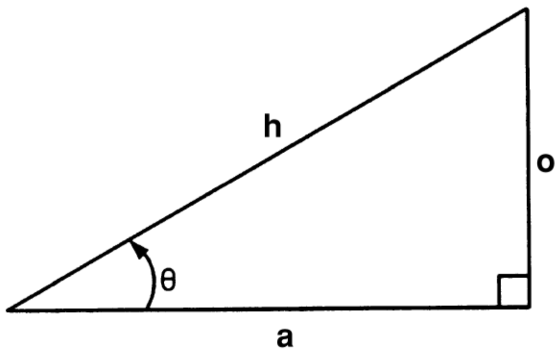
Review 3.1.2.1: A 50000 lb load is pushed horizontally by a hydraulic cylinder. If the coefficient of friction between the surface and the load is 0.25 and the cylinder shown has a 3 inch bore diameter, what minimum pressure given would be required to move the load?

a. 360 psi
b. 520 psi
c. 640 psi
d. 1800 psi
e. 2000 psi



If a load is resting on an incline, the total force required to move the load is determined from the sum of the force necessary to move the load along the surface plus the incline force which results from raising or lowering the load as it moves along the incline. Here, surface force and incline force are determined from the angle of the incline. Of course loads being pushed up an incline require more force than loads being pushed down an incline. In fact, some loads moving down an incline may generate negative (tractive) force. That is, the load has a tendency to overrun, causing the cylinder to extend.

The solution to problems where a load is being pushed along an incline requires solving a right triangle for the relative lengths of the sides. This can be done with the trigonometric functions, or for common angles, by inspection, or by applying the Pythagorean Theorem: $a^2 + b^2 = c^2$; (side 1^2 + side 2^2 = hypoteneuse^2). For example, the length of the sides of a right triangle where $\theta = 30°$ are o = 1, h = 2, and $a = \sqrt{3}$. Similarly, the lengths of the sides of a right triangle where $\theta = 45°$ are o = 1, a = 1, and $h = \sqrt{2}$. Note that "a" represents the side adjacent to the angle, "o" represents the side opposite the angle, and "h" represents the hypoteneus.



$$\text{Sin } \theta = \frac{\text{Opposite}}{\text{Hypotenuse}} = \frac{o}{h}$$

$$\text{Cos } \theta = \frac{\text{Adjacent}}{\text{Hypotenuse}} = \frac{a}{h}$$

$$\text{Tan } \theta = \frac{\text{Opposite}}{\text{Adjacent}} = \frac{o}{a}$$

Figure 3-1: Trigonometric Solutions to Right Triangles

The force required to move a load up or down an incline consists of two components. The first component is the friction force required to move the load along the surface already defined as the surface friction force (SFF). The second component may be defined as the incline force (IF). The total force required to move a load up an incline is expressed by:

Total Incline Force = Surface Friction Force + Incline Force

The surface friction force is a function of the coefficient of friction of the surface, the cosine of the angle of the incline and the magnitude of the load. As the angle of the incline increases, the surface friction force decreases. This is because the drag between the load and the surface of the incline decreases as the angle of incline increases. Note that at an incline angle of 0°, the cosine equals 1.0. At an incline angle of 90°, the cosine equals 0.0, effectively eliminating the addition of the surface friction force component of the final force required to move the load along the incline. The surface force works at a right angle to the surface of the incline.

The incline force is a function of the coefficient of the surface friction, the sine of the angle of the incline and the magnitude of the load. The incline force rises as the angle increases. For example, the sine of 0° is zero, reflecting the fact that the load is not being lifted as it travels across the surface of the incline. If the incline has an angle of 30°, the sine is 0.50. If the angle of the incline is increased to 45° degrees, the value of the sine increases to 0.71, reflecting the fact that as the load moves along the incline, the load also rises vertically a greater amount as it moves along the incline. Taken to its natural conclusion, if the angle increases to 90°, the sine increases to a value of 1.0 and the movement along the incline turns into a pure lift of the load.

Therefore, as the angle of the incline increases, the surface force component of the equation decreases as the incline force component increases.

If the load is being moved up the incline, the incline force is positive. If the load is being moved down the incline, the incline force is negative. Integrating the formulas for the surface friction force and the incline force results in the following formulas:

Total Incline Force$_{lbs}$ = (Coefficient of Friction x Load$_{lbs}$ x Cos θ) $\pm$ (Sin θ x Load$_{lbs}$)

TIF = (C_f x L x Cos θ) $\pm$ (Sin θ x L) Eq. 3-2

Note that you must add when the load is being moved up an incline, and subtract when the load is being moved down an incline.

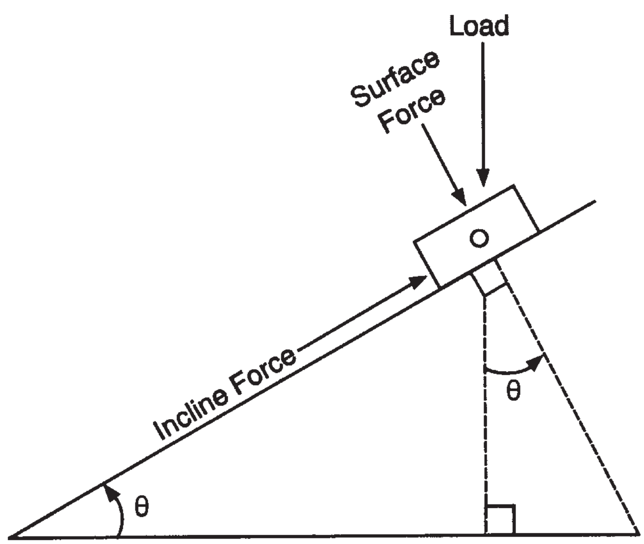
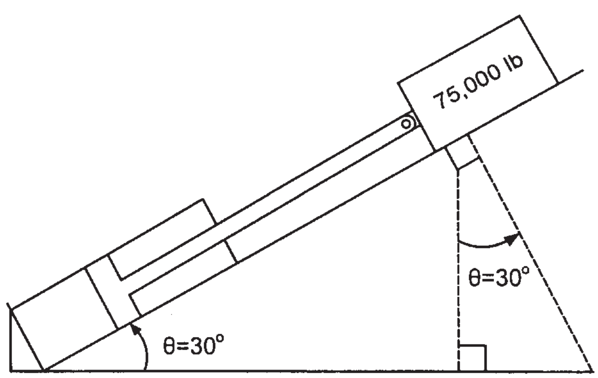


Figure 3-2: Forces of a Load on an Incline

Review 3.1.2.2: A 75000 lb load is to be pushed up a 30° incline by a hydraulic cylinder with a 4 in. bore. If the coefficient of friction between the surface and the load is 0.35, what minimum pressure will be required to extend the cylinder to move the load?

a. 2400 psi
b. 3500 psi
c. 4800 psi
d. 8500 psi
e. 10000 psi



Outcome 3.1.3:

Compute the thrust for a toggle mechanism.

Toggle mechanisms are used where high locking forces are required. Die casting machines, injection molding machines, and coining machines make use of toggle mechanisms. Figure 3-3 illustrates a single lever toggle force mechanism with a moving pivot at the rod eye that allows the cylinder rod to extend and retract in the horizontal plane. Toggle force is exerted upward in the vertical plane as the rod extends. The cap end of the cylinder is often mounted to a pivot.

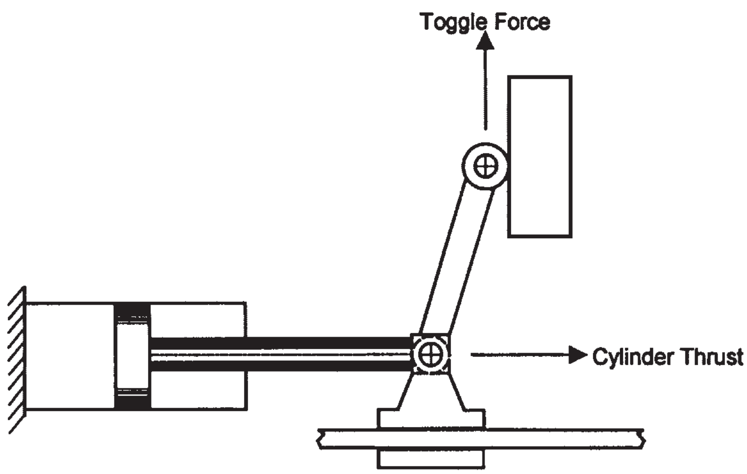


Figure 3-3: A Cylinder Used to Operate a Toggle Mechanism

For a single lever toggle, the upward force of the toggle can be calculated from the tangent of the angle between the lever arms and the horizontal:

Tan θ = opposite / adjacent

Where "o" (opposite) = the vertical direction and "a" (adjacent) = the horizontal direction.

The hypotenuse is equal to the length of the lever arm. Knowing remaining stroke length, the adjacent side (a) and applying the Pythagorean Theorem, the opposite side of the triangle (o) may be calculated:

hypoteneuse^2 = adjacent^2 + opposite^2

Or: Lever Arm^2 = Vertical Axis^2 + Horizontal Axis^2

The Pythagorean Theorem is usually written as $c^2 = a^2 + b^2$, though it has been rewritten here in an attempt to clarify the concept of a toggle mechanism.

The equation representing the force developed by a single arm toggle is:

SA Toggle Force$_{lbs}$ = Cylinder Force$_{lbs}$ x (Opposite / Adjacent)

Since (Opposite / Adjacent) = Tan θ, this maybe rewritten as:

SA Toggle Force$_{lbs}$ = Cylinder Force$_{lbs}$ x Tan θ

SATF = CF x Tan θ Eq. 3-3

The action of a cylinder rod extending horizontally to exert a toggle force vertically is shown by the triangle in Figure 3-4.

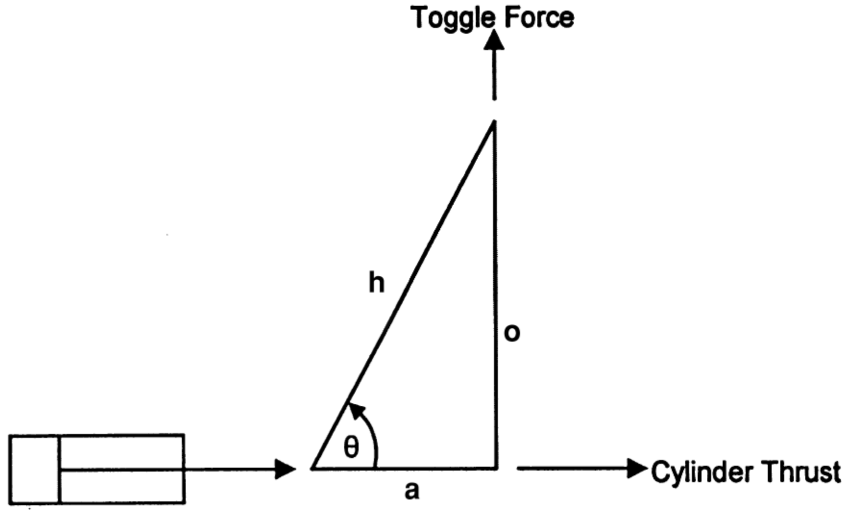


Figure 3-4: Force Exerted by a Toggle

For lever angles less than 45° from the horizontal plane, single arm toggle force is less than the cylinder force, in extension; for lever angles greater than 45° from the horizontal plane toggle force is greater than the cylinder rod force in extension. As the cylinder nears the end of its stroke as it extends, the toggle force is many times the magnitude of the rod force.

Review 3.1.3.1: If a cylinder rod exerts 5000 lb of force extending, how much force would a single lever toggle exert when the 12 inch lever arm (h) is 1 inch (a) from the vertical axis (o)?

a. 8000 lbs
b. 45312 lbs
c. 59790 lbs
d. 60210 lbs
e. 62312 lbs

Figure 3-5 shows a toggle mechanism with two equal length lever arms. The bottom lever arm is anchored to the machine at a pivot. The cylinder must be free to pivot as the toggle moves. In order to input full thrust into the toggle when the toggle reaches the overcenter position, the cylinder must be mounted so that its axis is perpendicular to the toggle when the toggle is overcenter. If the cylinder is not perpendicular when the toggle is overcenter, it will be acting at an angle to the load and its thrust into the toggle will be reduced (See Outcome 3.1.4). The upper arm (sliding pivot) of the double arm toggle shown in figure 3-5 will develop the same amount of thrust (toggle force) as the single arm toggle shown in figure 3-3. Though it is true that the lower (fixed pivot) lever arm will produce a force equal to that of the upper arm, the force developed by the lower arm will be countered by the reaction force developed by the machine frame. Therefore, the same equation is used to calculate the force from the double arm toggle shown as was used for the single arm toggle.

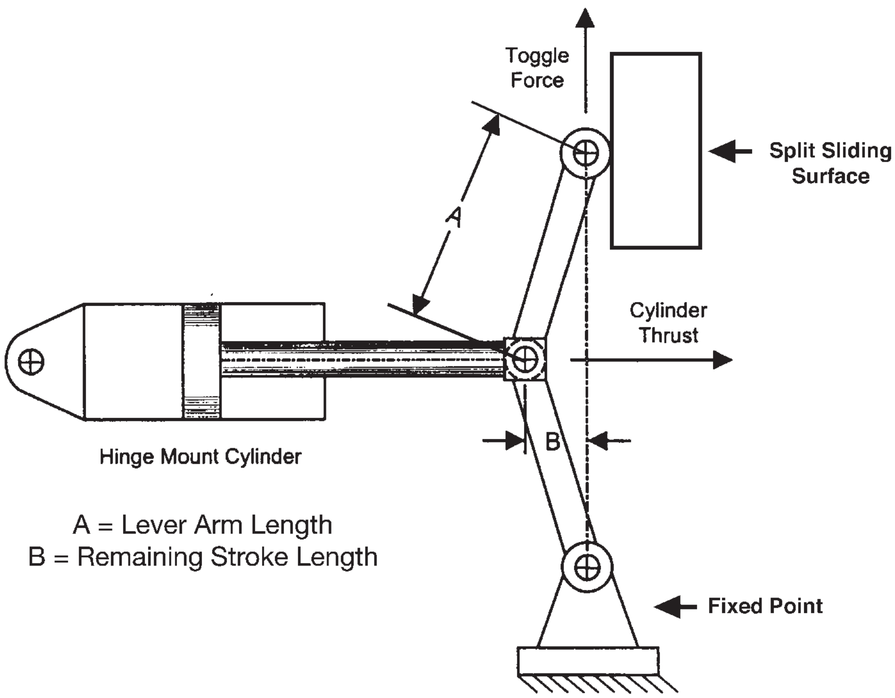


Figure 3-5: A Toggle Mechanism with Two Equal Length Lever Arms

Review 3.1.3.2: A toggle mechanism with two 15 inch lever arms, such as the mechanism shown in Figure 3-5, is closed by a hydraulic cylinder exerting 7200 lbs of force. What would be the toggle force when the position of the cylinder rod eye is 1/2 inch from the vertical?

a. 107,880 lbs
b. 107,935 lbs
c. 215,750 lbs
d. 215,870 lbs
e. 216,000 lbs

Outcome 3.1.4:

Compute the hydraulic pressure to support jib-boom loads.

The jib boom is an example of a third class lever. That is, one end of the lever rotates about a stationary pivot, the cylinder acts against the lever somewhere between the two ends, and the free end of the lever is used to move the load.

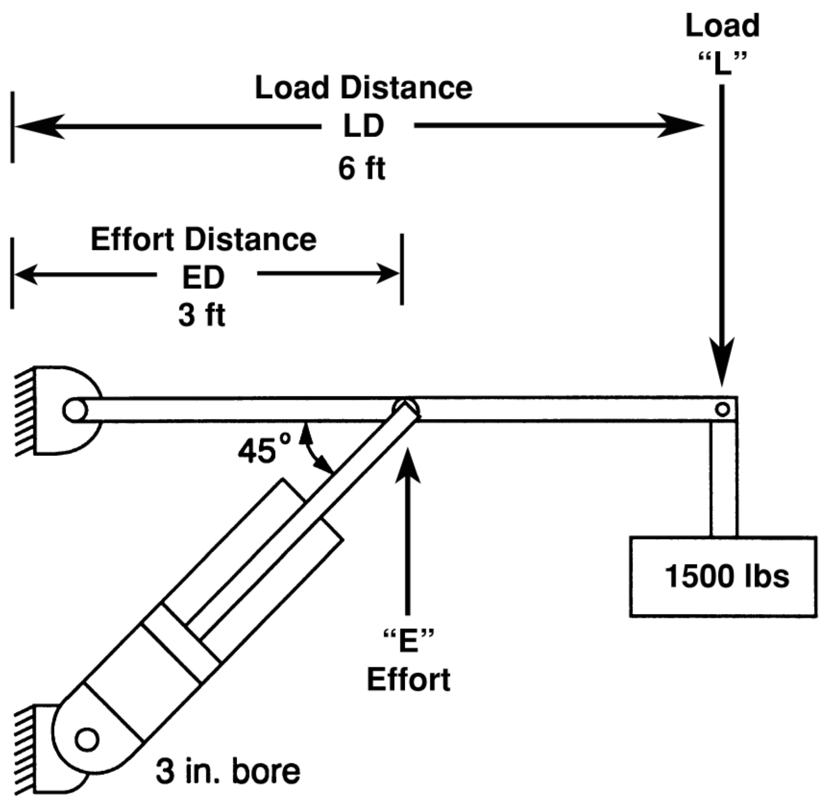


Figure 3-6: Jib Boom

Solving jib boom crane problems, such as the example illustrated in Figure 3-6, for the force acting against the cylinder rod, requires one to first solve for the vertical load at the fulcrum. Then, solve the right triangle for the force that acts against the cylinder on a line through the center of the rod.

For example, if the load at the end of the jib boom cylinder rod is 1500 lbs, the vertical load at the end of the cylinder rod can be determined from the balance of forces on the lever.

Load_{lbs} x $\text{Load Distance}_{\text{ft}}$ = $\text{Effort}_{\text{lbs}}$ x $\text{Effort Distance}_{\text{ft}}$

L x LD = E x ED Eq. 2-4

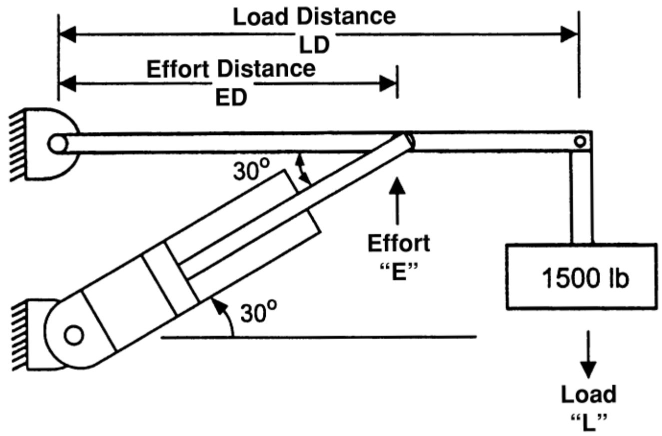
1500 lbs x 6 feet = EF x 3 feet

EF = 3000 lbs

The force acting against the end of the cylinder rod can determined by solving the right triangle formed by the cylinder, the beam, and the wall using trigonometric functions. However, by inspection it can be seen that for a right triangle where the other two angles are 45°, the relative side lengths are 1, 1, and $\sqrt{2}$ which equals 1.414. Therefore the force on the cylinder rod is 1.414 x 3000 lb = 4242 lb.

Review 3.1.4.1: In the figure shown, the angle between the cylinder rod and boom is 30° and LD = 2 x ED, what minimum theoretical pressure would there be in the cap end of a 3 inch bore cylinder?

a. 350 psi
b. 500 psi
c. 850 psi
d. 1700 psi
e. 1850 psi



In Figure 3-6 that the beam is horizontal, the support is vertical, and the clevis mounted cylinder is mounted at one of the standard angles, 45°, 30° or 60°. This is usually not the case. As soon as the cylinder rod moves the beam, the angles change. For example, in Figure 3-7 the beam is shown 45° from the horizontal while the cylinder is mounted to a horizontal ground support at a 30° angle to the beam. Moreover, as the cylinder rod extends and retracts the angles will constantly change. What will not change is the direction of the force to move the beam, which always acts perpendicular (at 90°) to the beam.

The dotted lines shown on Figure 3-7 identify right triangles that can be solved to calculate the magnitude of the forces that act perpendicular to the beam. To solve problems of this type:

1. Use the vertical angle between the load and the boom to compute the load that acts perpendicular to the end of the beam.

2. Calculate the mechanical advantage of the load at the end of the beam to the load that acts perpendicular to the beam at the cylinder rod end pin.

3. Calculate the load that acts perpendicular to the beam at the rod end pin.

4. Calculate the load supported by the cylinder and minimum pressure required to hold the load at the given angle.

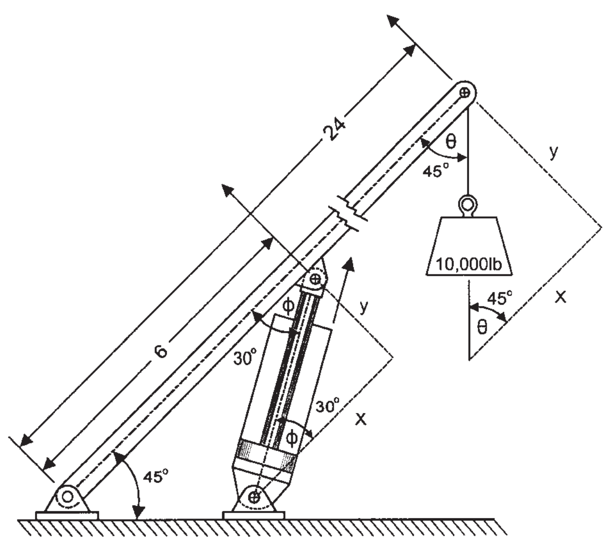


Figure 3-7: Perpendicular Forces Acting Against a Beam

Review 3.1.4.2: The hydraulic cylinder in Figure 3-7 has a bore diameter of 5 inches. What minimum pressure in the hydraulic cylinder would be required to hold a load of 10,000 lb at the angles shown?

a. 1426 psi
b. 1440 psi
c. 2161 psi
d. 2882 psi
e. 4321 psi

Task 3.2: Determine the motor characteristics required to move a load.

Outcome 3.2.1:

Solve formulas for torque, speed, and horsepower of hydraulic motors.

Low speed, high torque motors are used for winch and wheel motor applications. When a hydraulic motor is “geared down,” the speed of the output shaft is decreased as the torque is increased.

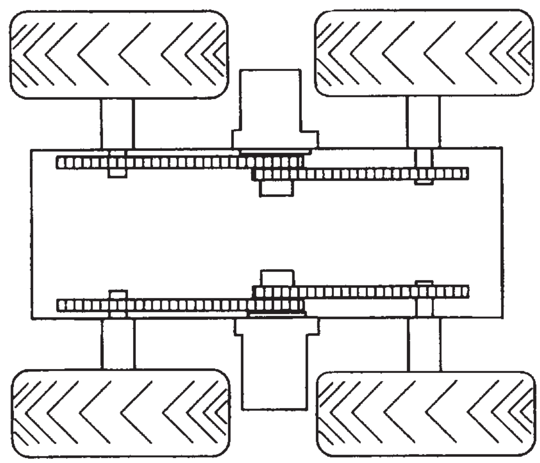


Figure 3-8: A Reduction System Driven by a Hydraulic Motor

The gear drive in Figure 3-8 uses a chain and sprocket reduction system between the hydraulic motor drive shaft and the skid steer loader drive wheels. The speed ratio can be computed in several ways. 1) as the quotient of the pitch diameter or number of teeth of the driven wheel shaft to the pitch diameter or number of teeth of the hydraulic motor drive shaft, 2) as the quotient of the number of teeth on the driven gear by the number of teeth on the drive gear, or 3) as the quotient of the diameter of the driven pulley by the diameter of the drive pulley. This relationship is expressed by the equation:

Speed Ratio = Output Shaft / Input Shaft SR = OS / IS Eq. 3-4

A close-up illustration of what is meant by pitch diameter for a sprocket is shown in Figure 3-9. Essentially, the pitch diameter is the distance between the chain pins on opposite sides of the sprocket measured on a line through the center of the sprocket shaft. The pitch diameter is not the same as chain pitch, which is the distance between the pins measured along the chain.

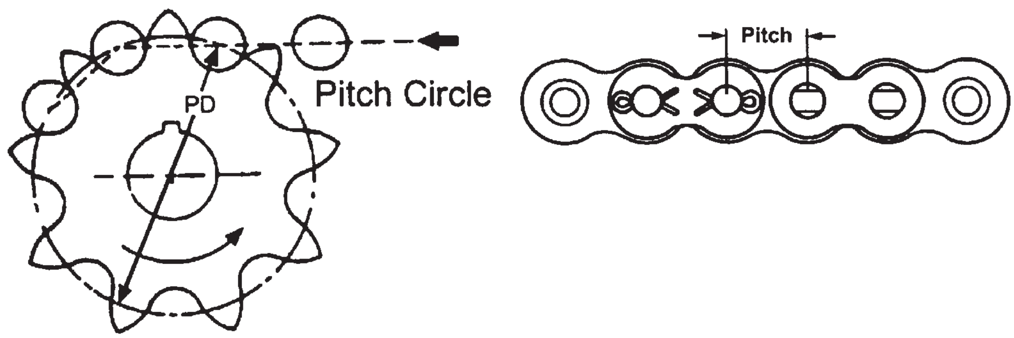


Figure 3-9: Pitch Diameter (PD) of a Sprocket vs. Chain Pitch

For example if the pitch diameter of the hydraulic motor sprocket is 3 inches and the pitch diameter of the wheel sprocket is 9 inches, the speed ratio would be:

Speed Ratio = Output Shaft / Input Shaft = 9 inches / 3 inches = 3:1 reduction

This means the hydraulic motor drive shaft turns three times every time the wheel shaft turns once.

If a compound drive is used, that is a double or triple reduction through belts, chains or gears, the final speed ratio is the product of the individual ratios.

If the diameter of the sprockets is difficult to measure, the ratio of the number of teeth on each sprocket can be used to compute the speed ratio.

Review 3.2.1.1: A hydraulic motor drives a shaft through a double reduction chain drive. The hydraulic motor has a sprocket with 8 teeth driving an intermediate shaft with 40 teeth. The second sprocket on the intermediate shaft has 7 teeth connected to a final drive sprocket with 28 teeth. What is the final speed ratio between the hydraulic motor and final drive sprocket?

a. 4 :1
b. 5 :1
c. 9 :1
d. 15 :1
e. 20 :1

Hydraulic motors impart a twisting, or turning motion, to the output shaft. This is called torque. Torque simply means to make an effort to twist or turn the shaft. It is measured as a force applied at a tangent to the circumference, some distance from the center of the shaft. Torque can be measured with the shaft stationary or rotating. Both starting and running torque calculations are made for machines driven by hydraulic motors. Numerically, the torque ratio is the inverse of the speed ratio. In the example shown in Figure 3-8, if the hydraulic motor sprocket turns three times for every one turn of the skid loader wheel shaft sprockets, the torque output at the wheel sprockets is three times the torque input of the hydraulic motor sprocket. Thus, gearing the hydraulic motor down increases the torque at the output, but at the expense of a decrease in speed.

In the English system of units, torque is measured in pound-feet (lb-ft) or pound-inches (lb-in). In the SI Metric system torque is measured in Newton-meters (N-m). The force unit, lbs or Newtons, is applied at right angles to the radius of the shaft, while the length unit, in, ft or m is the distance from the center of the shaft to the place on the circumference of the shaft where the force is applied at a tangent (right angle to the radius) to the circle.

The basic formula for torque is:

$Torque_{lb\text{-}in} = Force_{lbs} \times Radius_{in}$ $\qquad$ $T = F \times R$ $\qquad$ Eq. 3-5

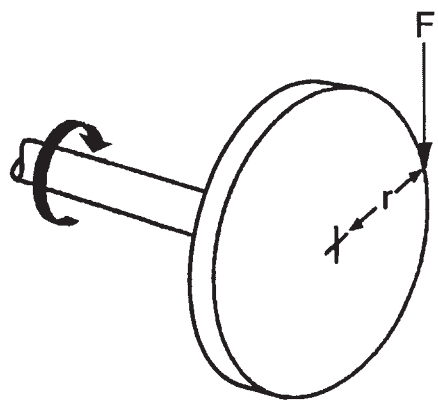


Figure 3-10: Torque

Torque has the same units as work, except that the units are reversed by convention to reduce confusion. Also, remember that torque is a rotary or turning "effort." This means the shaft can exert a torque whether turning or not turning, as long as there is a turning effort. For example, if the shaft of a hydraulic motor driving a chain saw is turning at 1000 rpm, it is exerting a torque. The same is true if the chain saw is stalled. A stalled hydraulic motor is still exerting a torque, even though there is no rotation of the output shaft. Thus, the torque available from a hydraulic motor can be computed at speed or independently of speed.

Review 3.2.1.2: The torque specification for a nut 65 is lb-ft. If the torque wrench handle is 20 inches long, how much force must the technician apply at the end of the handle to tighten the nut?

a. 26 lbs
b. 39 lbs
c. 45 lbs
d. 52 lbs
e. 65 lbs

Torque is measured as a rotating or turning effort, while work is measured as a force "F" exerted through a distance "D."

$\text{Work}_{\text{ft-lbs or in-lbs}} = \text{Distance}_{\text{ft or in}} \times \text{Force}_{\text{lbs}}$ $\qquad W = D \times F$ $\qquad$ Eq. 3-6

The rate at which work is performed is called power, which is work divided by time:

Power = Work / Time $\qquad P = W / T$ $\qquad$ Eq. 3-7

There are two common ways to rate hydraulic motors. One is by the torque output at a given horsepower and RPM. In the absence of data from the manufacturer, the horsepower for all rotating machinery can be computed from Watt's formula:

$HP = (T_{\text{lb-ft}} \times N_{\text{rpm}}) / 5252$ $\qquad$ Eq. 3-8

$HP = (T_{\text{lb-in}} \times N_{\text{rpm}}) / 63025$

$HP = (T_{\text{N-m}} \times N_{\text{rpm}}) / 7124$

"N" is an engineering term denoting "speed."

Review: 3.2.1.3: What torque could be expected from a hydraulic motor that is rated at 2.5 hp at 2250 rpm?

a. 0.93 lb-ft
b. 1.07 lb-ft
c. 1.97 lb-ft
d. 5.84 lb-ft
e. 8.45 lb-ft

A second formula for hydraulic motor torque is:

$\text{Hydraulic Motor Torque}_{\text{lb-in}} = (\text{Displacement}_{\text{cipr}} \times \text{Pressure}_{\text{psi}}) / 2\pi$

$HMT = (D \times P) / 6.28$ $\qquad$ Eq. 3-9

Review 3.2.1.4: A pressure gauge at the inlet of a hydraulic motor with a displacement of 0.66 cipr (cubic-inches per revolution) reads 1250 psi as the motor rotates the load at 100 RPM. What is the theoretical output torque of the hydraulic motor? Assume the pressure at the outlet of the motor is 0 psi.

a. 4.4 lb-ft
b. 6.6 lb-ft
c. 8.3 lb-ft
d. 10.9 lb-ft
e. 12.5 lb-ft

While torque values can be computed from Eq. 3-9 and Eq. 3-10, there are tables available that solve for any one of the three variables if the other two are known. Table 3-1 shown below solves Eq. 3-9 while Table 3-2 solves Eq. 3-10. Values in the bodies of tables 3.1 and 3.2 are for torque in lb-in.

	rpm									
HP	**100**	**500**	**750**	**1000**	**1200**	**1500**	**1800**	**2400**	**3000**	**3600**
0.25	158	31.5	21.0	15.8	13.1	10.5	8.8	6.6	5.3	4.4
0.33	210	42.0	28.0	21.0	17.5	14.0	11.7	8.8	7.0	5.8
0.50	315	63.0	42.0	31.5	26.3	21.0	17.5	13.1	10.5	8.8
0.75	473	94.5	63.0	47.3	39.4	31.5	26.3	19.7	15.8	13.1
1	630	126	84.0	63.0	52.5	42.0	35.0	26.3	21.0	17.5
1.50	945	189	126	94.5	78.8	63.0	52.5	39.4	31.5	26.3
2	1261	252	168	126	105	84.0	70.0	52.5	42.0	35.0
3	1891	378	252	189	158	126	105	78.8	63.0	52.5
5	3151	630	420	315	263	210	175	131	105	87.5
7.5	4727	945	630	473	394	315	263	197	158	131
10	6303	1261	840	630	525	420	350	263	210	175
15	9454	1891	1261	945	788	630	525	394	315	263
20	12605	2521	1681	1261	1050	840	700	525	420	350
25	15756	3151	2101	1576	1313	1050	875	657	525	438
30	18908	3782	2521	1891	1576	1261	1050	788	630	525
40	25210	5042	3361	2521	2101	1681	1401	1050	840	700
50	31513	6303	4202	3151	2626	2101	1751	1313	1050	875
60	37815	7563	5042	3782	3151	2521	2101	1576	1261	1050
75	47269	9454	6303	4727	3939	3151	2626	1970	1576	1313
100	63025	12605	8403	6303	5252	4202	3501	2626	2101	1751
125	78781	15756	10504	7878	6565	5252	4377	3283	2626	2188
150	94538	18908	12605	9454	7878	6303	5252	3939	3151	2626
200	126050	25210	16807	12605	10504	8403	7003	5252	4202	3501
250	157563	31513	21008	15756	13130	10504	8753	6565	5252	4377

Table 3-1: Hydraulic Motor Torque Values from the Formula: $T_{lb\text{-}in} = (HP \times 63025) / N$

gpm @		**psi**								
1200	**cipr**	**250**	**500**	**750**	**1000**	**1250**	**1500**	**2000**	**2500**	**3000**
2.6	0.50	19.9	39.8	59.7	79.6	99.5	119.4	159.2	199.0	238.9
5.2	1.00	39.8	79.6	119	159	199	239	318	398	478
7.8	1.50	59.7	119	179	239	299	358	478	597	717
15.6	3.00	119	239	358	478	597	717	955	1194	1433
31.2	6.00	239	478	717	955	1194	1433	1911	2389	2866
46.8	9.00	358	717	1075	1433	1791	2150	2866	3583	4299
62.3	12.00	478	955	1433	1911	2389	2866	3822	4777	5732
77.9	15.00	597	1194	1791	2389	2986	3583	4777	5971	7166
93.5	18.00	717	1433	2150	2866	3583	4299	5732	7166	8599
109.1	21.00	836	1672	2508	3344	4180	5016	6688	8360	10032
124.7	24.00	955	1911	2866	3822	4777	5732	7643	9554	11465

Table 3-2: Hydraulic Motor Torque Values From the Formula: $T_{lb\text{-}in} = (D \times P) / 2\pi$

Review 3.2.1.5: Using Table 3-2, what would be the expected torque from hydraulic motor with a displacement of 10.5 cipr at 2500 psi?

a. 3583 lb-in
b. 3981 lb-in
c. 4180 lb-in
d. 4379 lb-in
e. 4777 lb-in

Review 3.2.1.6: Using Table 3-1, what would be the theoretical output torque from a 1 HP hydraulic motor rotating at 1100 rpm?

a. 52.50 lb-in
b. 55.30 lb-in
c. 57.75 lb-in
d. 59.75 lb-in
e. 63.00 lb-in

Task 3.3: Specify hydraulic pumps and motors for system applications.

Outcome 3.3.1:

Size pump circuits.

Pumps are sized to meet the delivery requirements of the system within the range of pressures and shaft speeds (rpm) specified for the pump. Pump delivery is a function of the displacement and the shaft speed such that:

$Q_{gpm} = (\text{Displacement}_{cipr} \text{ x } N_{rpm}) / 231_{cu\text{-}in/gal}$ Q = (D x N) / 231 Eq. 3-10

For example, a pump with a displacement of 1.65 cipr (cubic inches per revolution) rotating at a maximum speed of 2400 rpm will deliver:

Q = (1.65 cipr x 2400 rpm) / 231 = 17.14 gpm

Review 3.3.1.1: What is the theoretical delivery in gpm of a gear pump with a displacement of 1.69 cipr turning at 1200 rpm?

a. 5.22 gpm
b. 6.54 gpm
c. 7.23 gpm
d. 8.78 gpm
e. 9.88 gpm

Most industrial pumps operate at one of the nominal industrial electric motor speeds: 1200, 1800, or 3600 rpm. Mobile pumps, on the other hand, operate at a range of speeds. The delivery of the pump will decrease slightly as the back pressure at the pump outlet increases. This is due to volumetric slippage in the close fitting parts of the pump, which reduces the output flow. The delivery of most pumps is given at zero pressure.

Pumps are sized to meet delivery demands. For cylinders, this is computed from the bore area in square inches times the velocity requirements in inches per minute. This generates a number with units of cubic inches per minute which can be converted to gallons per minute (gpm) by dividing by 231, the number of cubic inches in a gallon.

Recall that operating pressure was established to match prime movers and actuators used in the system.

Review 3.3.1.2: A pump operating at 1750 rpm is used to extend a 4 inch bore x 2 inch diameter rod x 24 inch stroke cylinder at an average velocity of 540 inches/minute. What is the theoretical displacement of the pump?

a. 0.62 cipr
b. 0.97 cipr
c. 1.23 cipr
d. 2.91 cipr
e. 3.88 cipr

Task 3.4: Size valves for various hydraulic circuits.

Outcome 3.4.1:

Calculate the flow coefficient for directional control valves.

Directional control valves route the flow of fluid in a circuit. Four-way, three-way, two-way and even check valves are common examples of directional control valves. When fluid flows through a valve from the inlet to the outlet port there is a pressure loss, termed “pressure drop,” caused by flow friction, resulting in a loss of efficiency. In the past, valves have been sized by their port opening but more recently valves are sized in relation to a flow coefficient based upon a C_v factor. All manufacturers provide flow friction data in the form of gpm flow vs. pressure drop performance curves. Using C_v factors allows valves from various manufacturers to be compared on this common flow characteristic.

The volumetric flow rate for hydraulic fluid flowing through a control valve using the flow coefficient can be determined from:

$$Q_{gpm} = C_v \sqrt{\text{Pressure Drop}_{psi} / \text{Fluid Specific Gravity}}$$

$$Q = C_v \sqrt{PSID/SG} \qquad \text{Eq. 3-11}$$

When calculations are made using Eq. 3-13, the specific gravity of the hydraulic fluid is given at operating temperature, because the specific gravity changes as the temperature changes.

Review 3.4.1.1: The pressure drop across a directional control valve flowing 20 gpm is 150 psid. If the fluid has an Sg of 0.90 at 140° F, what is the flow coefficient for the valve?

a. 1.101
b. 1.286
c. 1.443
d. 1.549
e. 1.728

Review 3.4.1.2: The literature from a manufacturer indicates a control valve has a C_v of 2.85 at a flow rate of 20 gpm. If the fluid has a Sg of 0.92 at 170° F, what would be the pressure drop across the valve?

a. 8.21 psid
b. 45.31 psid
c. 52.44 psid
d. 61.96 psid
e. 142.60 psid

Outcome 3.4.2:

Recognize the characteristics of DIN valves.

DIN valve systems consist of a poppet or spool type valve with two flow paths, A and B and one or two control ports. A DIN valve is made up the sleeve, the poppet, oft times a bias spring, and the cover assembly. The working ports are labeled "A" and "B," while the control ports are labeled "X" (pilot pressure) and "Y" (reservoir). By convention, the A port is located at the nose of the valve and the B port is located at the side of the valve. Most DIN valves have a spring that biases the poppet to a closed position. The value assigned to the spring force is F_A. Cartridge valves are configured to perform as simple check valves, piloted check valves, pressure relief valves, pressure reducing valves, sequence valves, flow control valves, and directional control valves. DIN valves may be installed into a block which contains just the one DIN valve, or into a manifold that contains one or more DIN valves. Such a manifold may include screw-in cartridge valves and surface mounted valves. The DIN valves are held in place by a cover plate.

The present discussion does not cover all configurations or cartridge type valves or circuits that apply them. It does, however, cover basic principles and applications that are required to understand how cartridge valves function and how they are applied in hydraulic circuits.

The two common configurations for DIN valves are unequal area ($A_F = A_A + A_B$), and equal area ($A_F = A_A$). One standard for differential area DIN valves gives the area of the bore, A_F, as 1.6 times the area of working port A_A. This makes the differential annular area around the poppet connected to the control port A_B equal to 0.6 times the area of A_F. Area ratios other than 1.6:1 are available. Depending upon the application, the force of the spring against the poppet generates a pressure of 3 to 60 psi (0.2 to 4 bar) to make the valve non-passing (closed) in the normally off position. A differential area DIN valve is shown in Figure 3-11a. A second variant of the DIN valve is shown in Figure 3-11b where the seat diameter is equal to the bore of the valve body. Here, the area of $A_A = A_F$. Where there is no differential area A_B, flow through the valve is from A to B and for an area of A equal to 3/4 inch and a port pressure of 6 psi, the closing force of the spring F_A may be computed from:

Spring Force$_{lbs}$ = Pressure$_{psi}$ x Area$_{sq\text{-}in}$ SF = P x A Eq. 3-12

SF = 6 psi x 0.442 sq-in = 2.65 lbs

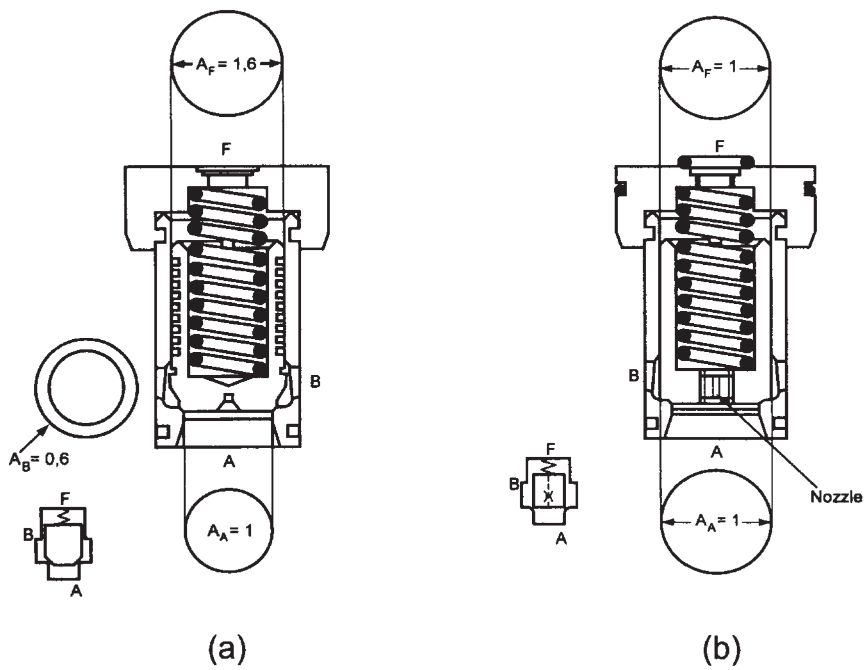


Figure 3-11: Cross-sections of Differential Area and Equal Area DIN Valves

The configuration of working and pilot ports is shown in Figure 3-12. The F pilot port is used to pressurize the poppet closed or vent the poppet open. Depending upon the configuration of the valve, the X pilot port is a pressure source to pilot the poppet closed, The X port can be plugged, connected externally to a pressure source, or connected to the A port or B port. The Y pilot port connects to the reservoir to vent port F. The pressure in port F is controlled by various valves, depending on the function of the DIN valve.

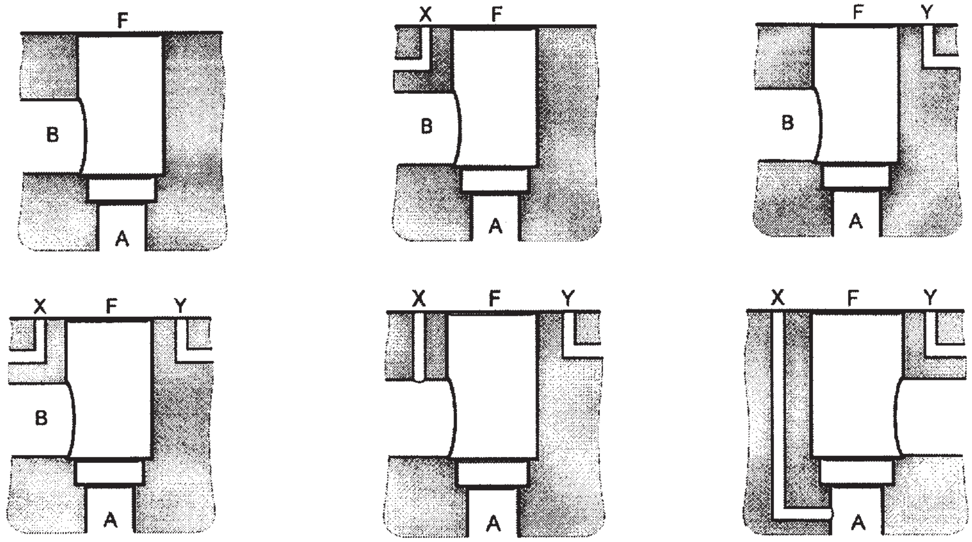


Figure 3-12: Configuration of Working and Pilot Ports in DIN Valves

Pressure control valves typically use a 1:1 ratio poppet. Fluid flows from the A port to the B port. Fluid entering port A opens the poppet by applying pressure to the area A_A at the bottom of the poppet or spool. For the valve to open, the poppet or spool chamber must be drained, either to the B port, internally or externally, or through the Y port to drain. Also, because area $F_A = A_A$, the valve is held closed by the force of only the spring.

Check valves and directional control valves typically use a 1.6:1 or 2:1 ratio poppet. In the normally non-passing position the spring holds the poppet closed, but because area F_A is greater than the area of working port A_A, equal pressure to both ends of the poppet will exert an additional closing force on the poppet of P x B_A. In addition, the differential area B_A provides the means for piloting the valve poppet open from the X port.

Review 3.4.2.1: What pressure would be required to open a poppet valve with pressure at port A_A which has a bore diameter A_F of 0.5 inch, a 1.6:1 area differential, and a closing spring force of 7 Ibs? (Assume that the force holding the poppet closed comes from only the spring).

a. 57 psi
b. 122 psi
c. 245 psi
d. 1100 psi
e. 1220 psi

Directional control DIN valves are controlled with an external pilot valve. Three examples are shown in Figure 3-13. The DIN valves shown are differential area valves. This provides three areas against which pressure can be applied: Area F_A (area above the poppet) area A_B (annular area around the poppet), and area A_A (area below the poppet). In Figure 3-13a, pilot pressure taken from supply pressure at port A, keeps the valve closed because F_A is larger than A_A. Shifting the pilot valve drains the chamber at the top of the poppet, allowing fluid to flow from port A to port B. In Figure 3-13b, pilot pressure is taken from port B in the same manner, except that draining the chamber at the top of the poppet directs flow from port B to port A. In Figure 3-13c, pilot pressure is taken from the X port to keep the valve closed. When pilot pressure is relieved to drain, the poppet opens, allowing flow from port A to port B (because of pressure acting at port A to open the poppet), or from port B to port A (because pressure at port B acts against the annular area around the poppet).

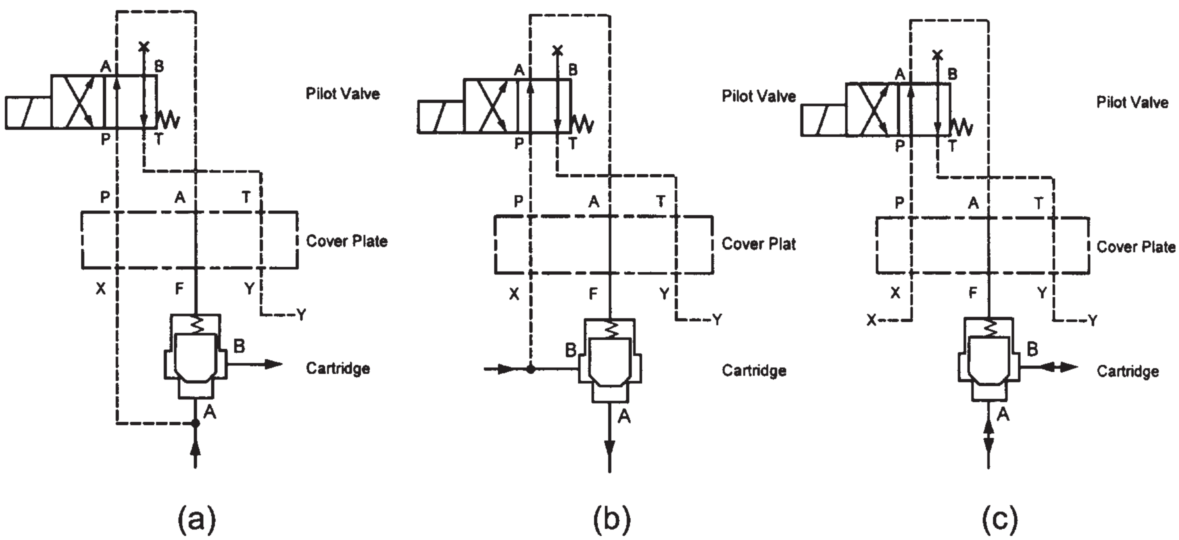


Figure 3-13: Pilot Operation of Differential Area DIN Valves

When the X port is used to operate a DIN valve, the balance of forces acting on the poppet may be calculated from:

$$(P_A \text{ x } A_A) + (P_B \text{ x } A_B) = (P_F \text{ x } A_F) + F_S \qquad \text{Eq. 3-13}$$

Where P_A, P_B, and P_F are pressures at ports A, B, and F. A_A, A_B, and A_F are the areas of ports A, B, and F. F_S is the closing force of the valve spring.

Review 3.4.2.2: In a cartridge valve with a differential area of 1.6 to 1 and valve closing spring force equal to F_S, what is the pressure at the F port to balance the poppet?

a. $P_F = (0.6\ P_A + P_B - F_S) / 1.6$
b. $P_F = (P_A + 0.6P_B - F_S) / 1.6$
c. $P_F = (P_A + P_B - F_S) / 0.6$
d. $P_F = (P_A + 1.6\ P_B - F_S) / 0.6$
e. $P_F = (P_A + 0.6\ P_B + F_S) /1.6$

Figure 3-14 illustrates the application of differential area cartridge type valves to extend and retract a hydraulic cylinder. The cartridge valves are operated by a 4-way directional control pilot valve. Pilot pressure to operate the cartridge valves is applied or relieved at the F ports. When the directional control pilot valve is in the center position, pilot pressure is applied to the F port of all four cartridge valves, locking the cylinder rod in position.

The cylinder rod is extended by shifting the directional control pilot valve to the right (b) envelope. This relieves pilot pressure from valve 1, allowing flow from the pressure line to flow through the cartridge valve from the A port to B port, and then to the cap end of the cylinder. Pilot pressure also is relieved from cartridge valve 3, allowing return oil from the rod end of the cylinder to flow through the valve from the A port to B port, and then to the reservoir. Pilot pressure is applied to the F ports of cartridge valves 2 and 4, keeping them closed.

The cylinder rod is retracted in a similar way by shifting the directional control valve to the left (a) envelope. This relieves pilot pressure from cartridge valves 2 and 4, providing a path through valve 2 for pressurized oil to the rod end of the cylinder, and through valve 4 for return oil flowing to the reservoir.

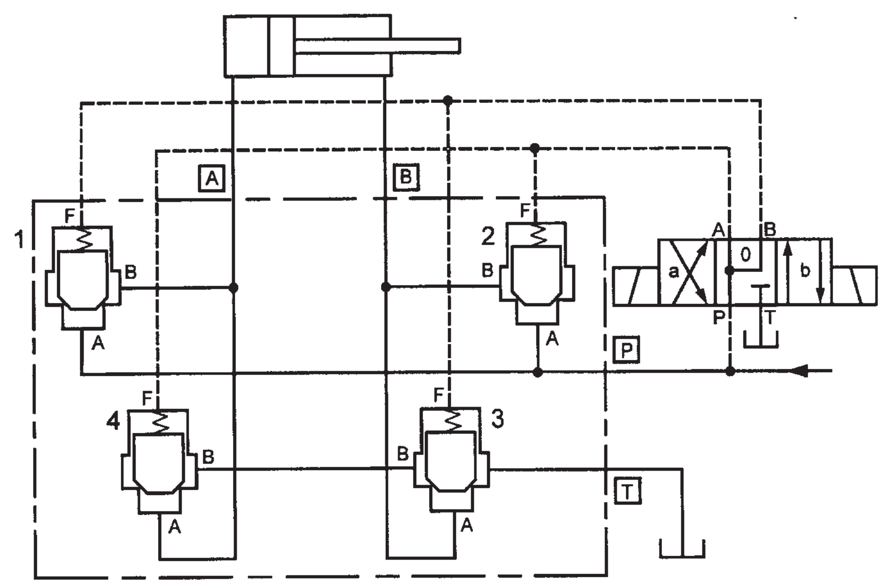


Figure 3-14: Differential Area Directional Control DIN Valve Cylinder Circuit

Review 3.4.2.3: What would occur if an over-running load were applied to extend and retract the cylinder rod when the directional control pilot valve in Figure 3-14 is in the center position? Assume the induced pressure of the over-running load exceeds the pressure at the P port.

a. The cylinder rod will drift in when the pilot valve is in the center position.
b. The cylinder rod will drift out when the pilot valve is in the center position.
c. The cylinder rod is free to float because the pilot valve has a float center.
d. Pilot pressure to all spring chambers locks the cylinder in position.
e. Valve springs in the cartridge valves lock the cylinder rod in place.

Outcome 3.4.3:

Recognize the characteristics of proportional control valves.

Proportional valves are used to control the output through the full range of operation of the valve. They are applied as pressure control valves, flow control valves, and directional control valves for cylinders and motors. The two most common types of actuators used to power the valves are solenoids and torque motors. Solenoid controlled proportional valves use either force or position solenoids.

Proportional valves are distinguished from servo valves by the use of proportional force solenoids instead of torque motors or recently, by high force solenoids. Mechanical feedback (MFB) servo valves utilize the traditional torque motor and internal feedback. Electronic feedback servo valves (EFB) use a linear force motor and electronic feedback. Proportional valves utilize proportional solenoids that have lower force than the linear force motors used in EFB servo valves. In addition, proportional valves may or may not have electronic feedback. The performance differences (frequency response capability) between proportional valves have narrowed greatly in recent years. In the case of single stage valves, proportional solenoids and linear force motors act directly on the valve spool, while torque motors control a hydraulic amplifier which in turn controls the position of the valve spool.

While an in-depth analysis of static and dynamic design calculations are beyond the scope of the following discussion, the hydraulic specialist should be familiar with operating curves used to select spools, cycle times, and components on the amplifier card that can be adjusted to tailor operation of the valve to the specific application.

Proportional force solenoids are modified versions of the DC powered solenoids used on directional control valves. Where a standard DC solenoid travels full stroke when it is energized, a proportional force solenoid generates a fairly linear force in proportion to the current input over a stroke of about 1.5 mm (0.060 in.). Theoretically, a current setting of 800 milliamps would generate twice the force of a 400 milliamp setting through full travel of the solenoid. The input/output relationship is not linear at the ends of the range of the coil, though the spring against which the coil acts is very linear. Higher quality solenoids are more linear than are lesser quality solenoids, and are linear over a greater range of stroke. Proportional solenoids produce forces on the order of 14 lbs whereas linear force motors produce forces as great as 45 lbs.

Pressure relief and pressure reducing valves use force solenoids to actuate single stage valves or the pilot stage of two stage relief valves. The use of a proportional valve allows the operator to electrically change relief pressures at will from the operator's station, both between and within operating cycles. The same is true for pressure reducing valves that use proportional force solenoids to actuate the pilot stage.

Review 3.4.3.1: If a force solenoid that actuates a proportional valve exerts a maximum force of 14 lbs at 800 milliamps, approximately how much force will the solenoid exert at 300 milliamps?

a. 3.25 lbs
b. 4.50 lbs
c. 5.25 lbs
d. 6.00 lbs
e. 7.50 lbs

When used to pilot the second stage of a directional control valve, spool movement in the second stage is proportional to the pilot pressure signal, which ranges from about 20 psi to 365 psi.

A simplified illustration of a solenoid operated sliding spool four-way, three position proportional directional control valve is shown in Figure 3-15. What must be understood at the outset is that for a proportional control valve to maintain control of an actuator over the full range of the valve, the input and output flow to the actuator must be metered. This results in a neutral leakage flow that sets up a bridge circuit with the pressure drop split between P_S to C_1 and C_2 and from ports C_1 and C_2 to P_T.

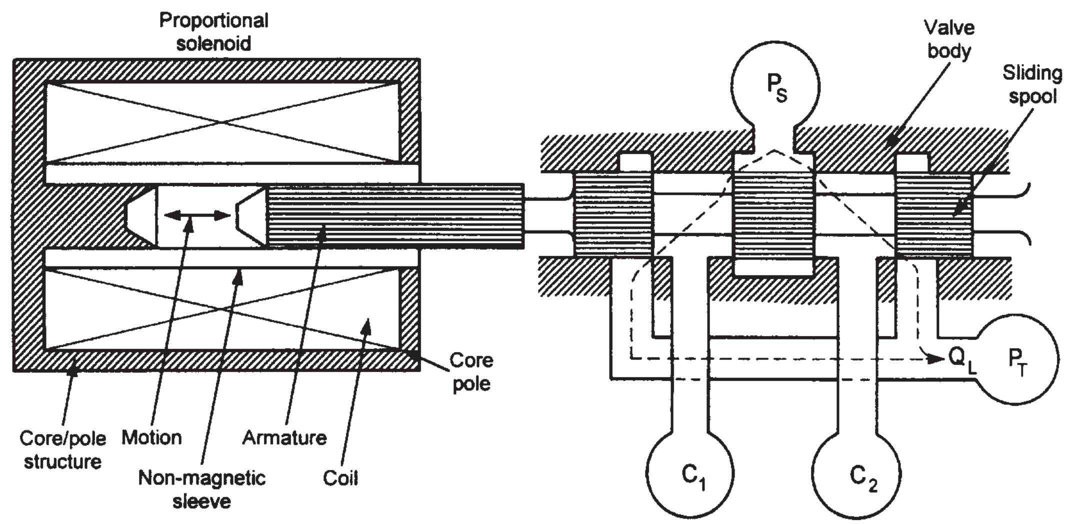


Figure 3-15: Sliding Spool Pilot Stage for a Proportional Valve

In the neutral position quiescent flow across the meter-in and meter-out edges of the spool balances the pressure to the actuator ports. When an electrical signal is sent to the solenoid, the spool is displaced, imbalancing the bridge circuit by opening one set of ports and closing the other set.

A second type of proportional control valve is shown in Figure 3-16. This type of valve, which is commonly termed a mechanical feedback servovalve (MFB), uses a torque motor for actuation control. Essentially, the flapper-nozzle assembly is a hydraulic resistance bridge circuit with two parallel connecting paths. Pressurized fluid passes through both nozzles from a common pressure supply. Each flow path has an upstream orifice and downstream nozzle. The flapper is located equidistant between the two nozzles.

When the flapper is in the neutral position, the bridge circuit is balanced and the pressures at the nozzle ports are equal. Energizing the torque motor coils causes the armature to shift, in turn, moving the flapper toward one nozzle and away from the other nozzle. The movement of the flapper causes the pressure at each nozzle to change. This is how the flapper imbalances the bridge circuit and causes a differential pressure to act on the main spool, or in the case of two stage valves, to shift the boost valve spool. Flapper displacements as small as 0.005 inch will create a differential pressure greater than 80% of the supply pressure with maximum output flow rates of 50% of the quiescent flow rate. A jewel located at the end of the spring wire attached to the flapper rides in a groove cut into the spool. As the spool shifts, force is transmitted into the wire, which in turn, acts upon the flapper to bring it back to a neutral position, even though the torque motor is energized. At this point, the pressure at each nozzle, and therefore the ends of the spool, becomes balanced and the spool remains shifted. In this way, the valve achieves internal mechanical feedback.

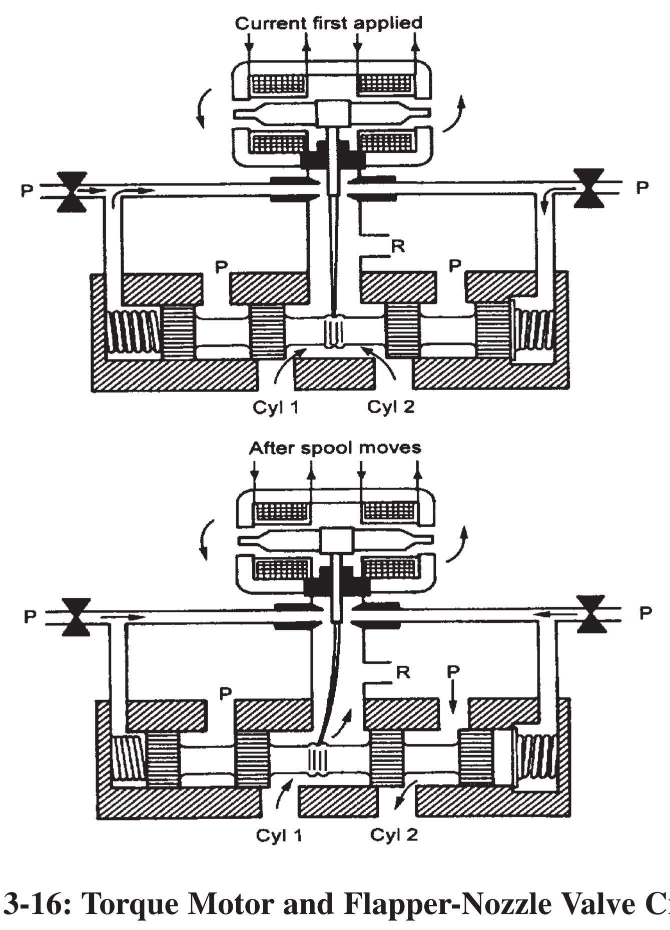


Figure 3-16: Torque Motor and Flapper-Nozzle Valve Cross-section

Application of a torque motor to the proportional control of a hydrostatic pump is shown in Figure 3-17. In this application, the position of the pump stroking mechanism is proportional to the electrical input, so a lever and spring is connected from the stroking mechanism to the torque motor. When an electrical signal is applied to the torque motor coils, the armature rocks either clockwise or counter-clockwise. The resulting torque displaces the flapper between the two nozzles. The differential nozzle pressure moves the spool to either the right or left. The spool continues to move until the feedback torque applied to the flapper from the spool valve counteracts the electromagnetic torque. At this point the armature and flapper are returned to center, the spool stops and remains stationary until the electrical input level signal changes to a new level. Thus, valve spool position is proportional to the electrical signal. Flow from the valve to the stroking mechanism will cause movement until a displacement is achieved where the feedback spring just counteracts the electrical input torque of the force motor.

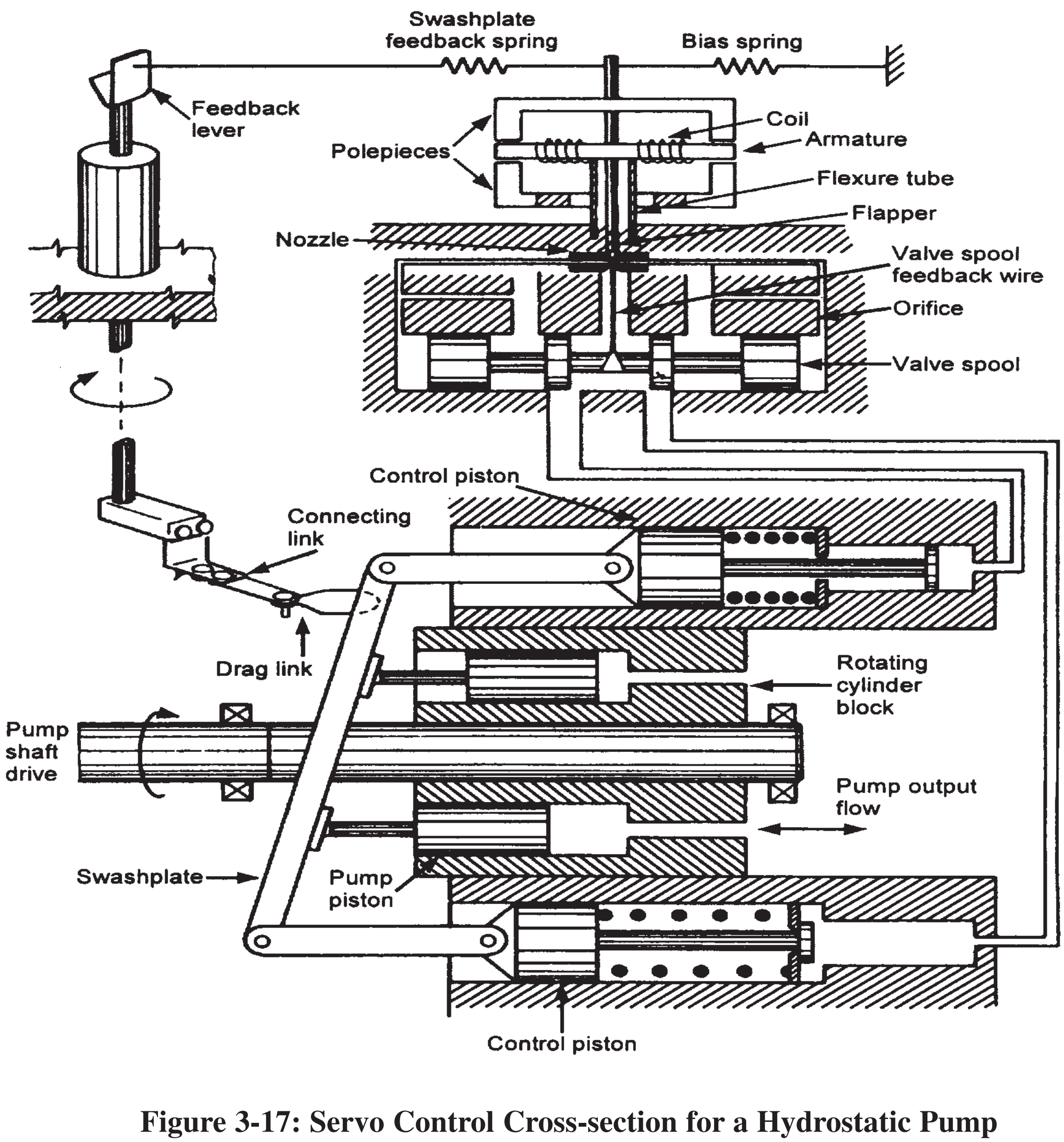


Figure 3-17: Servo Control Cross-section for a Hydrostatic Pump

Outcome 3.4.4:

Read operating curves for proportional directional control valves.

When selecting the proper valve spool for a circuit, it is important to make use of the control characteristics of proportional valves over the full range of their operation. Figure 3-18 shows a family of curves for control current vs. flow in gpm for a valve spool, illustrating the importance of pressure drop across the valve in maintaining control. Each curve shows the flow rate for a given input command based on the pressure drop (Δp) across the valve. As pressure drop across the spool increases, the flow rate also increases. The value of the pressure drop for each curve is given at the right of the graph. For this valve spool, at a pressure drop of 145 psi and an input command of 100%, the valve will flow approximately 26.4 gpm. Half of the pressure drop is from port P to port A, and the other half is from port B to port T. Should the circuit require only 20 gpm, the highest current would only control the valve up to approximately 90%, the remaining 10% being lost, because the full flow capacity of the valve is not used. Thus, oversizing a valve spool reduces control, whereas undersizing a spool, while it may provide 100% control, results in an unnecessarily high pressure drop across the valve. This situation is shown by curves 2 through 5. For example, at a pressure drop of 725 psi at a 100% input command, the same valve will flow 60 gpm. Thus, reading performance curves for proportional valves is necessary for proper control.

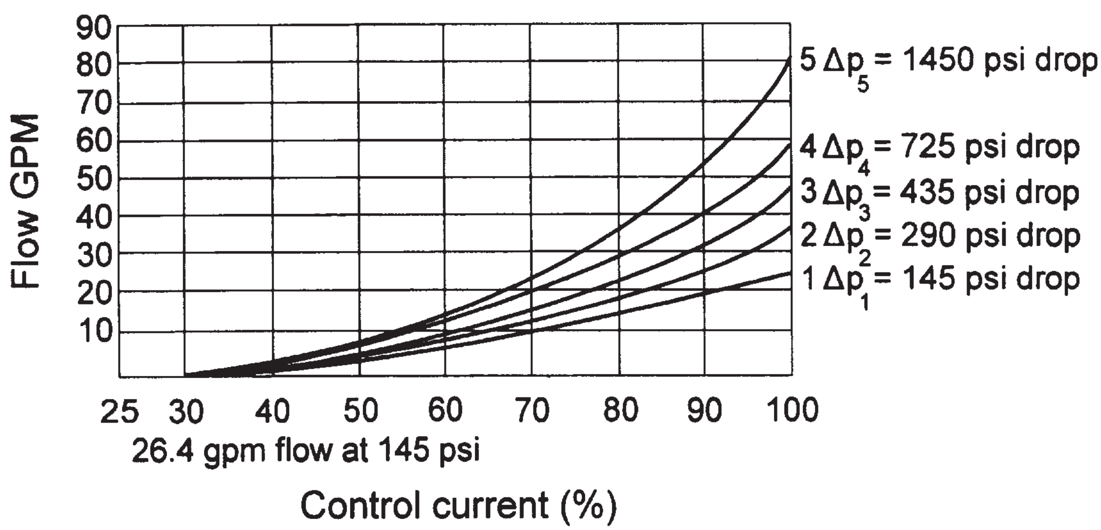


Figure 3-18: Curves Showing Flow (gpm) vs. Control Current (%) vs. Δp

Review 3.4.4.1: For the family of curves for control current vs. flow for the 26.4 gpm spool valve shown in Figure 3-18, what would be the approximate pressure drop across the valve for 100% control at a flow rate of 45 gpm? (Neglect load conditions)

a. 145 psi
b. 290 psi
c. 435 psi
d. 725 psi
e. 1450 psi

Outcome 3.4.5:

Compute the cycle time for a proportional control valve.

All valves, including proportional control valves, require time to cycle. Opening and closing is not instantaneous. Larger spools have more mass than smaller spools and therefore have more inertia.

Proportional valve spool stroke with varying input curves, such as the one in Figure 3-19 can be used to predict maximum cycle rate as a limiting factor. However, this does not figure into the effects of natural frequency calculations performed in engineering the circuit.

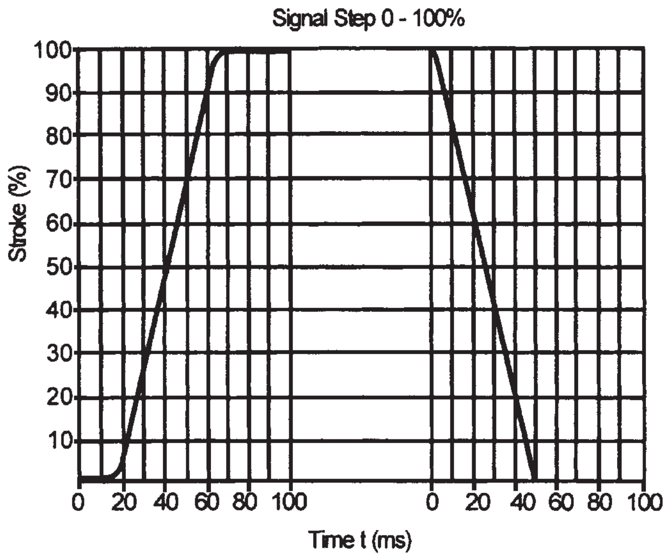


Figure 3-19: Command Signal and Stroke Control vs. Time for a Proportional Valve

The curve in Figure 3-19 plots stroke vs. time valve, as a percentage of the command signal, for opening and closing the where a 50% command signal would be expected to actuate the valve spool through 50% of spool stroke. In the figure, stroking the valve to 100% open is achieved in approximately 70 ms while stroking the valve 100% closed the valve is achieved in approximately 50 ms. Dwell time, during which the valve is actuated to some given command input, is added to the opening and closing times when a cycle is being analyzed.

The curve in Figure 3-19 profiles the spool stroke through 100% of the command signal. Since proportional control valves are designed for variable control, command signal and stroke percentages may be varied within the range of travel of the spool, for example, through 50%, 70%, or 90% of spool stroke.

Review 3.4.5.1: Assuming a cycle dwell time of 3 seconds, what is the maximum cycle rate in cycles per minute (cpm) for a proportional valve with response times given in the valve graph shown in Figure 3-19?

a. 19.23 cpm
b. 28.35 cpm
c. 42.88 cpm
d. 53.57 cpm
e. 120.00 cpm

Outcome 3.4.6:

Identify components on the amplifier card.

Proportional valves require a power supply and amplifier to drive the proportional solenoid or torque motor. The power supply converts 120 volts AC to 12 or 24 volts DC. The amplifier is part of the amplifier card circuitry.

A single solenoid amplifier card is shown in Figure 3-20 with the major components identified by numbers within circles.

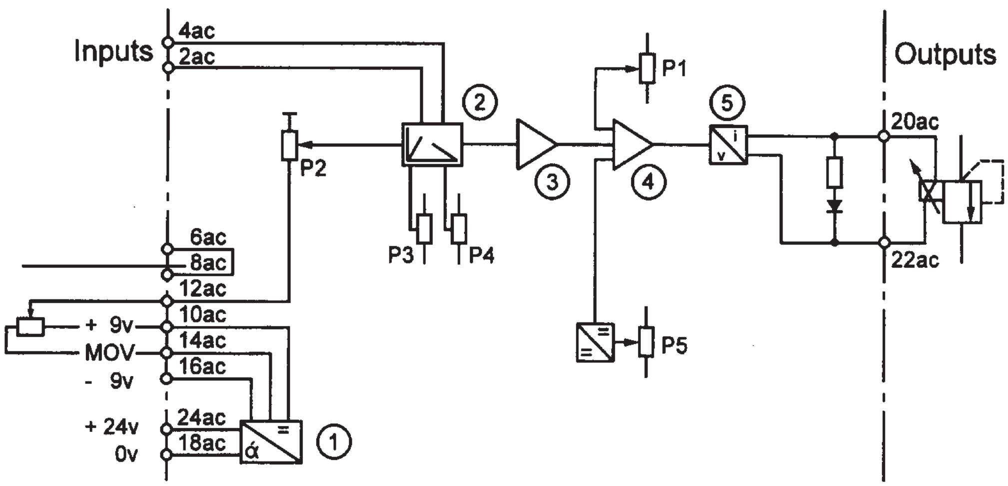


Figure 3-20: Single Solenoid Amplifier Card

Component 1 in the circuit is the voltage filter and regulator which provides the circuit with plus or minus 9 volts. Component 2 in the circuit is the ramp generator. Component 3 is an amplification stage. Component 4 is a summing amplifier, adding an input signal from P1 and a dither signal from P5. The power amplifier, component 5, raises the signal to the level necessary to drive the solenoid as well as providing current feedback to stabilize the effects of temperature changes and wiring losses.

A number of adjustments can be made to the amplifier card. The label “P” indicates a potentiometer. P1 controls the minimum current (I-min) going to amplifier 4. “I” denotes “current.” I-min can be used to set the minimum output of the valve and to reduce the deadband of the valve. For example, a given application may require that without an input signal, a pressure relief valve is biased to 500 psi. This minimum pressure would be set using P1.

P2 is a current limiting adjustment (I-max) for the command signal, and is used to set the maximum pressure of the valve. The value of P2 is added to the signal of P1. Thus, the bias current level P1 is set first, followed by maximum current adjustment with P2. Adjusting the settings in reverse will result in a setting of P2 that is too high. Continuing with the example used in the last paragraph, assume that at 100% input command, the relief valve would achieve 3000 psi, yet the system needed to be limited to 2500 psi. I-max could be set to limit the pressure to a maximum of 2500 psi.

P3 a controls the rate of the acceleration ramp while P4 controls the deceleration ramp.

P5 controls the adjustment for the dither oscillator to reduce valve "stiction" and shifting force. The dither signal is intended to minimize solenoid hysteresis.

Review 3.4.6.1: For the proportional valve amplifier card circuit shown in Figure 3-20, which setting would be adjusted to increase the power signal?

a. p1
b. p2
c. p3
d. p4
e. p5

Task 3.5: Specify hydraulic reservoirs to meet system requirements.

Outcome 3.5.1:

Size hydraulic reservoirs from volume requirements.

Hydraulic reservoirs hold fluid in reserve. Reservoirs serve several other functions. Reservoirs also condition the fluid by allowing contaminants to settle out and allow air bubbles and foam to rise to the surface and dissipate. Reservoirs can act as a heat exchanger to cool or to heat the fluid, depending on the temperature of the fluid and the ambient temperature. Reservoirs can also provide a mounting surface for system components.

When a cylinder extends, pump delivery to the cap end of the cylinder will reduce the volume of fluid in the reservoir. Single acting cylinders reduce the volume more than double-acting cylinders because there is no return flow from the rod end of the cylinders. Double acting rams and telescoping cylinders would act approximately the same as single acting cylinders in depleting fluid from the reservoir because the rod side of the piston volume is small compared to the cap end. Reservoirs must be sized to hold enough fluid to make up the lost rod volume as cylinders are extended.

It is accepted practice to size an industrial reservoir three to five times the output flow of the pump, while mobile reservoirs are typically sized smaller due to space and weight constraints. Regardless of the ratio used, the reservoir must be able to compensate for the differential volume of the system as the cylinders are actuated while maintaining an adequate minimum fluid level.

The volume of a reservoir, in gallons, is calculated from the product of its length, width, and height measurements, in inches. The product is then divided by 231, the number of cubic inches in a gallon, to arrive at the size of the reservoir, in gallons. A volume of approximately 20% is allowed above the fluid in order to compensate for the expansion of the fluid as it becomes heated.

$Volume_{gal} = (Length_{in} \text{ x } Width_{in} \text{ x } Height_{in} \text{ x Percent } Full_{decimal}) / 231$ cu-in/gal

$V = (L \text{ x } W \text{ x } H \text{ x } \%) / 231$ Eq. 3-14

Review 3.5.1.1: The inside dimensions of a hydraulic reservoir are 30 inches long x 15 inches wide x 15 inches high. How much fluid would the reservoir hold if 20% of the volume above the fluid is left for expansion and contraction of the fluid?

a. 5.8 gallons
b. 20.8 gallons
c. 23.4 gallons
d. 29.2 gallons
e. 36.5 gallons

Outcome 3.5.2:

Size hydraulic reservoirs from cooling requirements.

When oil warms up it will expand slightly, increasing in volume.

In order to allow for changes in reservoir volume caused by expansion of the fluid as well as extension and retraction flow from cylinders, the reservoir is equipped with a breather. The breather is usually, but not always, incorporated into the fill cap. Breathers also include filters to trap contaminants and moisture in the air entering and exiting the reservoir.

A detailed discussion for sizing heat exchangers based on cooling requirements is covered in Task 3.8, but the cooling capacity of the reservoir should be calculated first, so that an oversized heat exchanger is not specified. Oversizing a heat exchanger could result in a reduction of cooling efficiency. Turbulent flow through the heat exchanger is essential in order to achieve efficient heat transfer.

The following formula uses only the vertical area as the cooling capacity of the bottom and top of a reservoir is generally negligible:

$HP = 0.001 \text{ x } \Delta T_{\circ F} \text{ x Vertical Reservoir } Area_{sq\text{-}ft}$

$HP = 0.001 \text{ x } \Delta T \text{ x } A$ Eq. 3-15

Review 3.5.2.1: A raised hydraulic reservoir is 4 feet long x 2 feet high x 2 feet wide. If the reservoir is 3/4 full, and the operating temperature is expected to be 100°F above ambient temperature, how much cooling capacity will the reservoir provide in Btu/hr?

a. 4581 Btu/hr
b. 6108 Btu/hr
c. 7132 Btu/hr
d. 8102 Btu/hr
e. 9800 Btu/hr

Task 3.6: Specify hydraulic accumulators for system operation.

Outcome 3.6.1:

Size an accumulator using gas laws.

Hydraulic accumulators use weights, springs, or gas pressure to generate the precharge force against the fluid that is stored for use in the system. Gas charged accumulators use pistons, bladders, or diaphragms to separate the hydraulic fluid from the gas charge. Bladder type accumulators are available in sizes ranging from 1 cubic inch to 80 gallons, generally in pressure ranges of 3000 and 5000 psi.

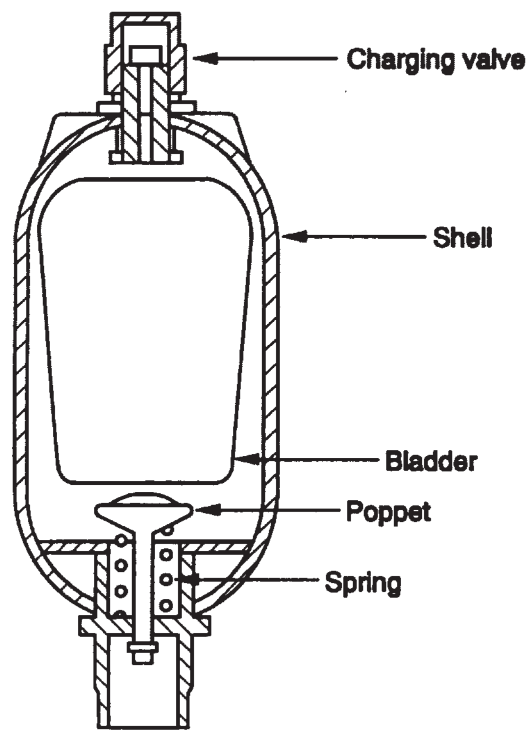


Figure 3-21: Bladder-type Accumulator

Gas charged accumulators operate by placing the compressible gas over the incompressible hydraulic fluid in a constant volume pressure vessel. The hydraulic pressure and volume of fluid available to the system are dependent upon the precharge pressure and the expansion characteristics of the gas. Dry nitrogen is used to precharge accumulators.

Caution: Because of the risk of spontaneous combustion, never use oxygen or air to precharge an accumulator.

The terms isothermal and adiabatic are used to describe the expansion characteristics of the gas. If the temperature of the gas is kept constant, for example as the volume is reduced slowly, the pressure in the gas will be inversely proportional to the volume. This is called isothermal contraction and expansion. When a gas is compressed and expanded quickly, heating and cooling cause pressure changes in addition to those occurring strictly as the result of volume changes. If the gas is insulated so that no heat is allowed to escape, the pressure of the gas will increase and decrease more than inversely to the reduction in volume. Under compression, heat added to the gas as it is compressed will raise the pressure, in addition to the pressure increase caused by reducing the volume. Under expansion, the pressure will decrease more than would be expected just by increasing the volume. This is called adiabatic contraction and expansion.

To accommodate changes in both pressure and temperature of the precharge gas the General Gas Law can be used to compute the volume available from an accumulator. Absolute values are used for temperature and pressure when making computations. Rankin is the absolute scale for Fahrenheit and Kelvin is the absolute scale for Celsius. Formulas for converting from Fahrenheit to Rankin and Celsius to Kelvin are as follows:

°F to °R: $°R = °F + 459.7$
°C to °K: $°K = °C + 273.7$

The Perfect Gas Law:

$$P_1 \times V_1 \times T_2 = P_2 \times V_2 \times T_1 \qquad \text{Eq. 3-16}$$

Review 3.6.1.1: A one gallon capacity accumulator supplies fluid to a hydraulic system between 1750 psi and its precharge pressure of 1000 psi. Using the General Gas Law, how many cubic inches of hydraulic fluid are available from the accumulator if the temperature changes from 80°F to 150°F as the accumulator fills? Assume adiabatic compression and expansion of the gas.

a. 17 cu-in
b. 81 cu-in
c. 150 cu-in
d. 180 cu-in
e. 231 cu-in

Outcome 3.6.2:

Size accumulators for isothermal conditions.

An accumulator cannot supply fluid below the precharge pressure. If the precharge pressure leaks off, for example to half the precharge pressure specified, the accumulator will supply less fluid than had the precharge pressure been correct. Low precharge pressures also cause the system to become sluggish.

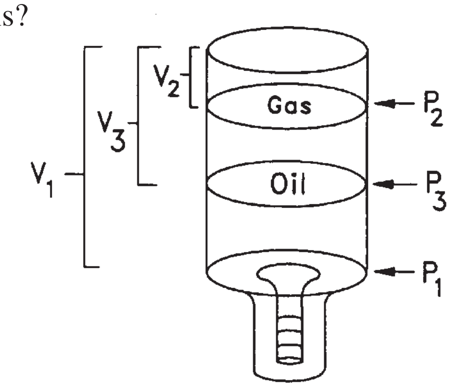
In Review example 3.6.1.1, the accumulator supplied fluid from 1750 psi to 1000 psi, suggesting that system pressure was just balanced by the precharge pressure when the accumulator was empty. This is not usually the case. A more likely scenario is that an accumulator is precharged to 1000 psi, the pressure relief valve is set to 3000 psi and the accumulator supplies fluid from 3000 psi to 2000 psi.

For isothermal conditions, the equation is:

$P_1 \times V_1 = P_2 \times V_2 = P_3 \times V_3$ Eq. 3-17

Review 3.6.1.2: A 2 gallon accumulator supplies fluid to a hydraulic system between 3000 psi and 2000 psi. If the precharge pressure is 1000 psi, how many cubic inches of hydraulic fluid is available from the accumulator if the process is isothermal as the accumulator fills?

a. 77.2 cu-in
b. 81.4 cu-in
c. 155.5 cu-in
d. 180.6 cu-in
e. 232.7 cu-in



Task 3.7: Specify hydraulic intensifiers and boosters.

Outcome 3.7.1:

Calculate pressure intensification from piston sizes.

Hydraulic intensifiers boost pressure by pumping hydraulic fluid with a larger bore hydraulic piston into a smaller bore hydraulic cylinder to extend the rod. The intensifier can be powered using either air or hydraulics.

Air over oil hydraulic intensifiers convert a high volume, low pressure air input source to a low volume, high pressure source of hydraulic fluid. Intensifiers are used for hydraulic presses, hose crimpers, and flaring tools in shops where the cost of a hydraulic pressure source would be prohibitive. Where an intensifier is used to intensify the pressure, the degree of intensification is equal to the ratio of the larger area to the smaller area.

The circuit shown in Figure 3-22 illustrates a two-pressure booster system with oil returns for low and high pressure oil sources. Valve A is actuated to extend and retract the press cylinder at the bottom of the circuit, while Valve B is used to actuate the high pressure booster piston. The degree of intensification equals the ratio of the areas of the larger booster piston to the area of the smaller booster piston.

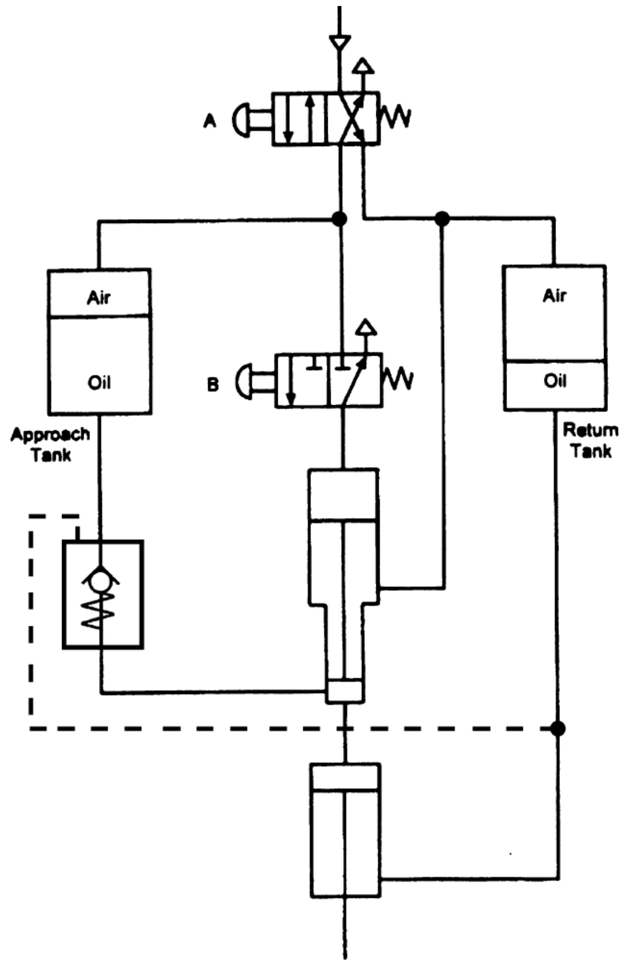


Figure 3-22: Intensifier System with an Air-Oil Return Tank

Review 3.7.1.1: A 10 inch bore hydraulic cylinder operates a press and develops a force of 100 tons. The fixed displacement hydraulic pump advances the cylinder. At 200 psi a sequence valve opens to provide flow to a hydraulically powered intensifier which boosts the pressure to the cylinder. The bore size of the intensifier output piston is 0.50 inches and the bore size of the intensifier's input piston is 1.5 inches. At what pressure should the air pressure regulator be set in order to achieve the correct output pressure from the intensifier?

a. 288 psi
b. 636 psi
c. 719 psi
d. 1273 psi
e. 2547 psi

Task 3.8: Specify heat exchangers.

Outcome 3.8.1:

Calculate the heat loss and temperature rise for a hydraulic pressure drop.

Heat exchangers relieve the fluid of excess heat to lower its operating temperature. In cold climates heaters are used to heat the fluid. In simple terms, the amount of heat to removed and transferred to the cooling medium equals the difference between the input horsepower to the pump and the output horsepower of all of the system actuators. This assumes environmental conditions are not adding or subtracting heat into or out of the fluid, which is generally the case. The system designer must take into account that fact that fluid conductors and machine surfaces also act to dissipate or absorb heat.

By analyzing the inefficiencies of the various components in a specific system, an experienced designer can often estimate the anticipated heat generation level with a good degree of accuracy.

In English units: 1 HP = 2545 Btu/hr = 42.4 Btu/min = 0.7069 Btu/sec

In metric units: 1 HP = 2,685,600 Joules/hr = 44,760 Joules/min = 746 Joules/sec

Energy loss therefore equals:

$\text{Energy Loss}_{Btu/hr} = 2545 \text{ x } T_{hours} \text{ x } (HP_{input} - HP_{output})$ Eq. 3-18

$\text{Energy Loss}_{Joules} = 746 \text{ x } T_{seconds} \text{ x } (HP_{input} - HP_{output})$

The equation for fluid horsepower is:

$HP_{fluid} = (\text{Pressure}_{psi} \text{ x Flow}_{gpm}) / 1714$ $HP = (P \text{ x } Q) / 1714$ Eq. 3-19

Fluid horsepower is the actual energy contained in the hydraulic fluid. Input horsepower takes into account the overall efficiency of the component:

$HP_{input} = (\text{Pressure}_{psi} \text{ x Flow}_{gpm}) / (1714 \text{ x EFF}_{decimal})$ $HP = (P \text{ x } Q) / (1714 \text{ x } E)$ Eq. 3-20

Review 3.8.1.1: An in-plant hydrostatic transmission delivers 7 hp at the output shaft. A flow meter and a pressure gauge indicate the motor portion of the transmission is receiving 10 gpm at 1500 psi. If losses through the motor are converted to heat, how many Btu/hr is the unit generating?

a. 3385 Btu/hr
b. 4015 Btu/hr
c. 4454 Btu/hr
d. 5255 Btu/hr
e. 6011 Btu/hr

Task 3.9: Size fluid conductors.

Outcome 3.9.1:

Size fluid conductors.

Hydraulic conductors include manifolds, pipes, tubing, hoses, fittings and flexible couplings. The two primary considerations are proper sizing to carry the flow, and wall thickness to withstand system pressures. Other important factors are leakage prevention and compatibility with the fluid.

Per standard industrial practice, fluid conductors are sized to limit fluid velocity to 3 to 4 ft/sec in inlet lines, to 10 ft/sec in return lines, and to 15 to 20 ft/sec in pressure lines. Proper sizing assures that the pump will not be starved for fluid at the inlet and that the system will operate quietly and efficiently. The efficiency of flow in the inlet line to the pump can be checked with a vacuum gauge. Installations with high vacuum readings should be checked for clogged inlet screens, kinked lines and other obstructions. Elbow shaped fittings in the inlet that are plumbed too close to the pump inlet also will increase vacuum readings.

As noted, inlet plumbing should be sized generally for a velocity of less than 3 to 4 feet per second. Laminar flow is more important than velocity in reducing cavitation. The inlet of the pump should be connected to a hose with a length that is 10 times its diameter to provide laminar flow. For example, a one inch diameter hose should be a least 10 inches in length before any turbulent connections. In this example, that means elbows, tees, shutoff valves, suction screens or filters, or anything else that would cause turbulent flow, should not be located within the 10 inch distance. A 90° sweeping bend on the end of a hose is acceptable. An adequately sized suction strainer and a shutoff ball valve may be used in the suction line, providing that they are at least 10 diameters away from the pump inlet. However, some pump manufacturers do not recommend the use of suction strainers or filters with their pumps. Pumps can still cavitate, even with a positive inlet pressure (positive head), if the flow isn't laminar. Flange fittings work better than SAE, JIC, or pipe fittings. Even if a pump is submerged below the surface of the fluid, one should still put a straight inlet tube on the pump that is 10 diameters in length in order to ensure laminar flow to the inlet port of the pump. Flaring the open end of the tube will give excellent results and will lower in pump noise by at least one half or more.

Flow velocity through a conductor is not uniform across the inside diameter. Instead, the cross-section of flow is parabolic, with the maximum fluid velocity found near the center of the conductor and near zero velocity near the wall of the conductor. Between these two regions the velocity varies. For this reason, values calculated using formulas, or arrived at using nomographs, are average velocities.

While using sizing rules is an obvious requirement to check pump inlet and pressure outlet lines, return lines that route fluid back through directional control valves should not be overlooked. For example, when a single rod double acting cylinder is retracted, more oil is returned to the reservoir than is delivered by the pump. This occurs because the cap end of the cylinder holds more oil than the rod end. Telescoping cylinders and single acting cylinders increase return line flow to an even greater extent. Furthermore, regenerative circuits require resizing the line to the cap end of the cylinder because fluid from the rod end of the cylinder joins with pump flow during regeneration.

The fluid velocity in a fluid conductor can be calculated from:

$\text{Velocity}_{ft/sec} = (Q_{gpm} \times 0.3208) / \text{Area}_{sq\text{-}in}$ $\quad\quad$ $V = (Q \times 0.3208) / A$ $\quad\quad$ Eq. 3 - 21

There are a number of ways to solve for fluid velocity or conductor size. Perhaps the easiest way is to use flow nomographs or tables that are given in most hydraulic handbooks.

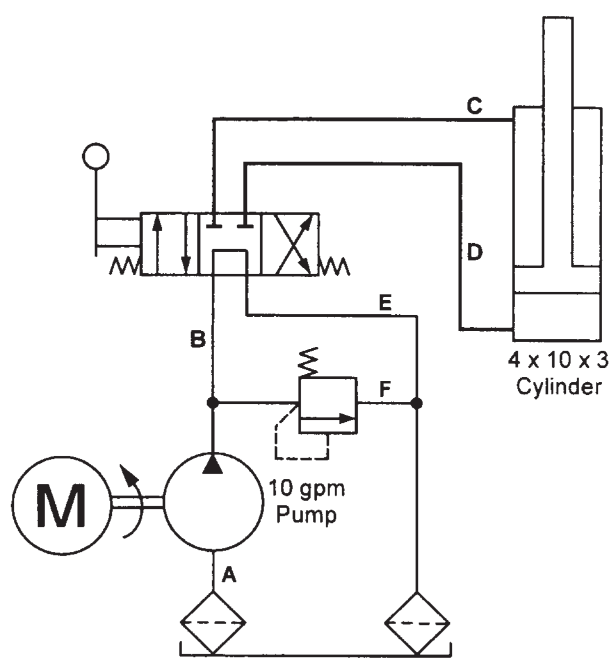
Tube, hose and pipe diameters are measured differently. The diameter of tubing is measured to the outside of the conductor. The inside diameter is then determined by subtracting twice the wall thickness. Except for 100R5 and 100R15 hose, the diameter of hydraulic hose is given as the inside diameter, and therefore may be used directly. The inside diameter of pipe must be read from a pipe table.

Review 3.9.1.1: What is the average velocity of 20 gpm flowing through a 1 inch hydraulic tube if the wall thickness is 0.065 inch?

a. 6.4 ft/sec
b. 10.8 ft/sec
c. 12.7 ft/sec
d. 18.3 ft/sec
e. 21.6 ft/sec

Review 3.9.1.2: The circuit shown in the figure directs fluid from a 10 gpm pump to cycle a double acting cylinder. If the wall thickness is 0.065 inches, what minimum inch size tube would be required to prevent the average fluid velocity from exceeding 15 ft/sec in line E?

a. -8 inch
b. -10 inch
c. -12 inch
d. -16 inch
e. -20 inch



Outcome 3.9.2:

Use Barlow's Formula to calculate the wall thickness and safety factor for tubing.

The wall thickness of a conductor is determined by the system pressure, the tensile strength of the material, the outside diameter of the conductor, and the safety factor. Safety factors are determined by the application and by customer and government specifications. In the absence of more stringent requirements, a 4:1 safely factor is generally used. If the system is subjected to shock loading, a safety factor as high as 10:1 may be used.

The wall thickness of tubing and pipe can be determined from tables, or calculated directly using Barlow's Formula which calculates burst strength from the tensile strength of the material.

Before using Barlow's Formula, calculate the burst pressure by multiplying the working pressure times the safety factor:

Burst Pressure$_{psi}$ = Working Pressure$_{psi}$ x Safety Factor BP = WP x SF Eq. 3-22

Burst Pressure$_{psi}$ =

(2 x Wall Thickness$_{inches}$ x Material Tensile Strength$_{psi}$) / Tube OD$_{inches}$

BP = (2 x WT x TS) / OD Eq. 3-23

Review 3.9.2.1: Using a safety factor of 4:1, determine the working pressure of a of a -12 x 0.049" wall hydraulic tube if the tensile strength is 40,000 psi.

a. 871 psi
b. 1287 psi
c. 1307 psi
d. 5227 psi
e. 6667 psi

Job Responsibility 4.0: Prepare Bills of Material and Schematics

Job Responsibility 4.0: Prepare Bills of Material and Schematics

Task 4.1: Convert circuit elements into a working schematic.

Outcome 4.1.1:

Calculate circuit force, distance, and sequence times.

Circuit sequence charts are calculated from a number of factors that enter into the construction of a circuit. Among these factors are pressures, areas, volumes and flow rates.

For example, the circuit shown in Figure 4-1 shows two cylinders connected in series. Flow entering the cap end of cylinder A will cause flow from the rod side to extend cylinder B.

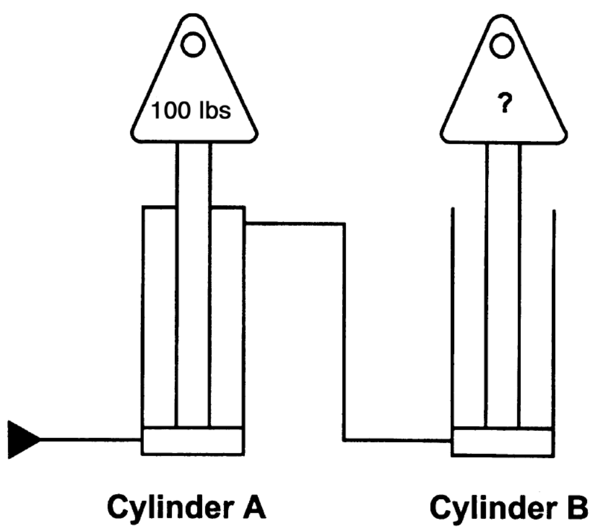


Figure 4-1: Two Cylinders Connected in Series (3 inch bore x 2 inch rod x 8 inch stroke)

Review 4.1.1.1: In Figure 4-1 fluid enters cylinder A at a pressure of 100 psi. Assuming 100% efficiency, what load on the rod of cylinder B would just stall cylinder B?

a. 100 lb
b. 154 lb
c. 607 lb
d. 707 lb
e. 1092 lb

Outcome 4.1.2:

Calculate motor rpm from the pump and motor displacement.

Pump - motor combinations are used in hydrostatic transmissions. The speed of the motor is determined by the rpm of the pump and the displacement ratio between the pump and motor.

Motor $\text{Speed}_{\text{rpm}}$ = Pump $\text{Speed}_{\text{rpm}}$ x (Pump $\text{Displacement}_{\text{cipr}}$ / Motor $\text{Displacement}_{\text{cipr}}$)

$$MN = PN \times (PD / MD) \qquad \text{Eq. 4-1}$$

Figure 4-2 illustrates a simplified hydrostatic transmission. The pump at the bottom of the schematic drives the motor at the top of the schematic. Make-up fluid needed to compensate for leakage in the pump and the motor is drawn from the reservoir through one of the check valves on the suction side of the pump, with the opposite check valve closed by pressure in the pressure leg of the circuit. The pressure relief valve prevents the system from over pressure.

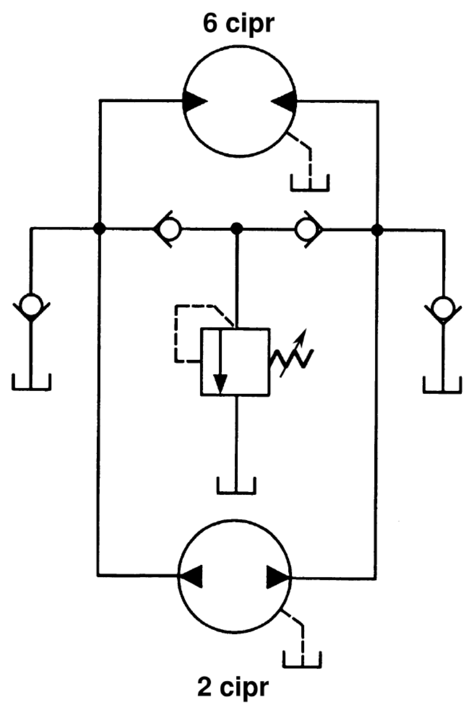


Figure 4-2: Simplified Hydrostatic Transmission

Review 4.1.2.1: In the hydrostatic transmission shown in Figure 4-2, the pump is driven at 1200 rpm. At what speed will the motor rotate?

a. 400 rpm
b. 800 rpm
c. 1200 rpm
d. 2400 rpm
e. 3600 rpm

Task 4.2: Construct state diagrams indicating the control sequence and machine operation.

Outcome 4.2.1:

Recognize machine operations described by the Time Cycle Chart.

Time cycle charts describe the actuation sequence of components such as control valves and actuators. The operation of each component is plotted on a grid against time and against the operation of other components.

Figure 4-3 illustrates a state diagram consisting of three 4/3 solenoid valves controlling two cylinders and a hydraulic motor. The states for clamp cylinder A are shown in section A in the chart. The operation of the 4/3 directional control valve which controls the clamp cylinder is shown in section B. The three states of operation of the valve are: 1) clamp; 2) hold; and 3) unclamp. The states of operation of this circuit are: 1) extend; 2) hold under pressure; 3) retract and remain retracted until the next cycle begins. The states for the spindle cylinder directional control valve (D) are: 1) advance; 2) hold; and 3) retract, which controls the spindle cylinder (C) as it extends, holds under pressure, retracts, and remains retracted until the next cycle. The states for the mill spindle directional control valve (F) are: 1) rotate clockwise; 2) stop; and 3) rotate counter-clockwise, which controls the mill spindle hydraulic motor (E) as it rotates clockwise, stops, rotates counter-clockwise, and remains stopped until the next cycle. The actuation of the directional control valves are shown as instantaneous events, whereas the movement of the two actuator cylinders and the mill spindle have ramp times in each direction.

Extension of clamp cylinder A is controlleded by valve B, advancing over two time periods, from the beginning of the cycle until the 0.3 second point, and then from 0.3 seconds until the 0.8 second point. The first portion of the cylinder extension is rapid, followed by a slower extension rate to clamp the work piece. At the end of the extension period (the 0.8 second point), with the work piece secure, valve D is energized, causing spindle cylinder C to extend. At the same time the spindle cylinder extends, the tap rotates as valve F causes hydraulic motor E to rotate. The tap contacts the work piece 1.3 seconds into the cycle and continues rotating until the 6.3 second mark. The rate of advance of the spindle cylinder after the tap contacts the work piece is controlled by the mill spindle rpm and the pitch of the tap's thread. For example, if the tap has a pitch of 13 (threads per inch), and the mill spindle is operated at two revolutions per second, in the five seconds of rotation between 1.3 seconds and 6.3 seconds, the tap would advance the thread 10 threads, or just over three quarters of an inch (10/13 = 0.76992 in.).

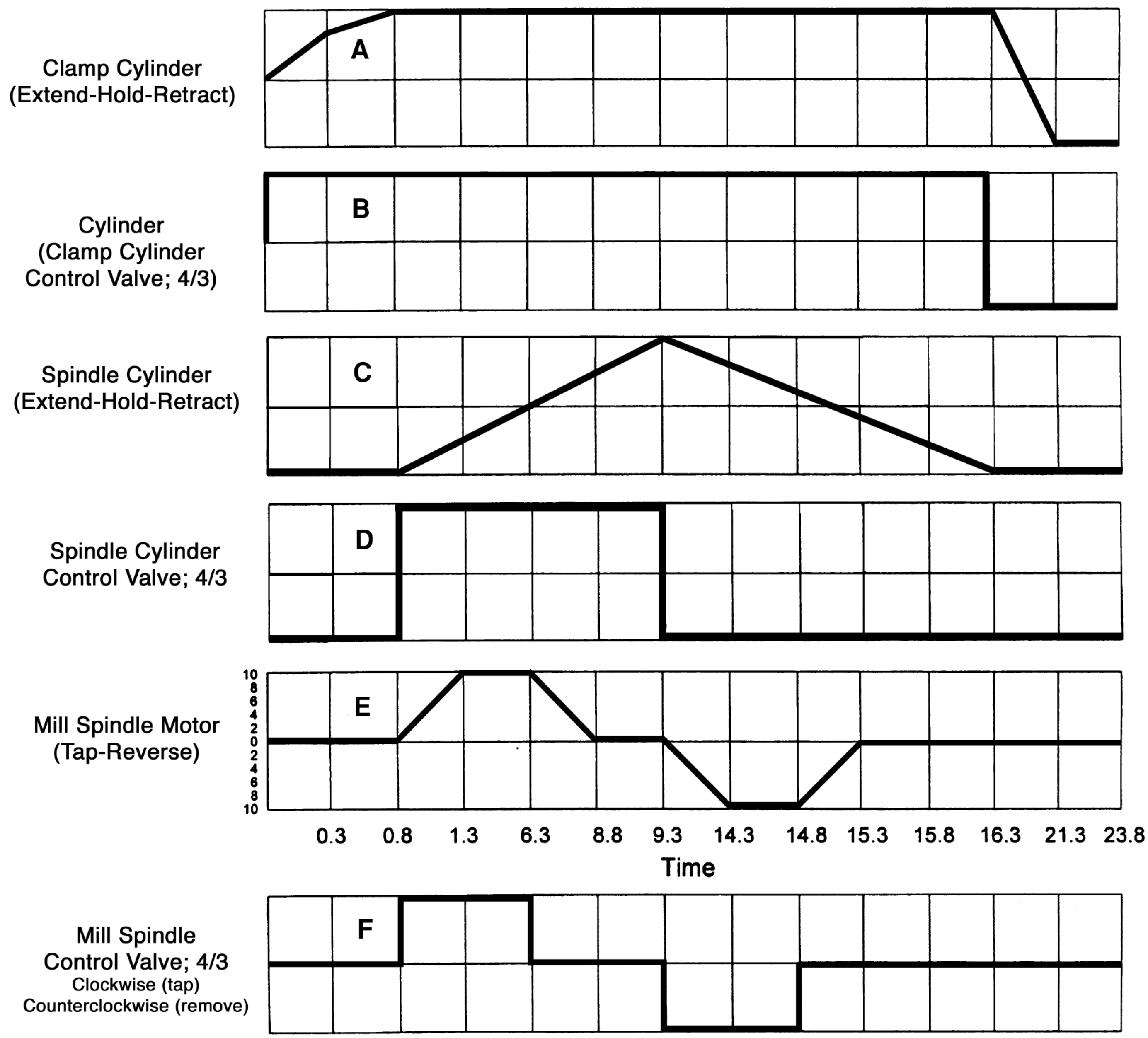


Figure 4-3: Time Cycle Chart

Review 4.2.1.1: In Figure 4-3, how long is the work piece clamped?

a. 13.5 seconds
b. 14.0 seconds
c. 14.5 seconds
d. 15.5 seconds
e. 16.3 seconds

Job Responsibility 5.0: Recommend Fluid, Fluid Conductors, and Fluid Filtration

Task 5.1: Select the fluid type and grade to meet the pump manufacturer's specifications.

Outcome 5.1.1:

Identify properties of hydraulic fluids.

Properties that affect the performance of a hydraulic fluid include viscosity, specific gravity, viscosity index, pour point, neutralization number, flash point, fire point, auto ignition temperature, anti-wear properties, resistance to oxidation and rust, anti-foaming, and detergent dispersing properties. Of these factors, viscosity is the most important operational property.

Selection of viscosity must be made with the normal operating temperature of the machine in mind. The temperature is usually measured at the pump. The correct viscosity at the expected operating temperature will allow the fluid to transmit the power most efficiently, while still lubricating and cooling the machine without exhibiting excessive leakage. If the viscosity is too high (the fluid is too thick), the operation of the machine will be sluggish and excessive heat will be generated as the fluid flows through the system. If the fluid is not viscous enough, the fluid will bypass the close fitting parts of components. This leakage is called slippage. As the fluid leaks internally from areas of high pressure to areas of lower pressure, heat will be generated. External leakage may occur as well, as the fluid may be thin enough to bypass seals.

Liquids used for hydraulic fluids include petroleum-base oils, synthetics, water glycols, oil-in-water and water-in-oil emulsions, and high water content fluids (HWCF). It is estimated that about 80% of the hydraulic fluid sold each year is petroleum-base oil, with the remaining 20% being the other four types, which are fire resistant. Industries that use fire resistant fluids include forging and extrusion, coal mining, chemical-petroleum power, die casting, foundries, metal fabrication, plastic injection molding, and primary metals manufacturing.

Petroleum-base hydraulic fluids, refined from selected crude oil, are formulated with an additive package to reduce friction, wear, rust, oxidation, and foaming. Acidity and viscosity improvers help to maintain the viscosity between narrow limits at the expected operating temperature. For most applications the viscosity of the hydraulic fluid is in the range of 100 SSU to 500 SSU. If the fluid thins out excessively as the machine comes to operating temperature, the volumetric efficiency of the pump and actuators decreases and the likelihood of internal leaks increases.

Petroleum oil has the natural ability to transmit power, has good lubrication properties, dissipates heat reasonably well under in-plant operating conditions, and is compatible with most sealing materials. As long as the temperature is held below 150° F (65.5° C) and the oil is kept free of contaminants, petroleum-base fluids are long-lasting and stable. If there is a catastrophic failure of the pump, the system must be properly flushed and new, filtered fluid must be added to prevent contaminating the new pump and other components. The condition of the fluid should be determined by periodically analyzing samples.

There is a misconception that fire resistant fluids will not burn. All hydraulic fluids will burn if the temperature is high enough. What fire resistance means is that the fluid will not continue to burn once the source of ignition is removed. Straight synthetics are used where a fire hazard exists and where the high cost of the fluid is not prohibitive. Water-glycols and water-in-oil emulsions reduce the possibility of ignition. High temperature will evaporate the water component of the fluid, removing heat from the reaction. High water content fluids are the most fire resistant because the water content to additive ratio is too high for the additive to ignite.

Oil-in-water emulsions surround the oil molecules with water, making water the continuous phase. Oil-in-water emulsions are used primarily for cooling, such as in cutting oils and coolants. Water in oil emulsions are mixtures of oil and water with the oil surrounding the water droplets. Thus, the oil is the continuous phase, which usually gives superior lubricating quality. Water-in-oil emulsions are called invert emulsions.

Water-glycol fluids are designed for use in hydraulic systems operating in areas with a source of heat, ignition, or where there is a potential fire hazard. They are a mixture of water, ethylene glycol, and high viscosity lubricating and thickening agents, along with additives to prevent corrosion. Most water glycol fluids are limited to use in low to medium pressure and non-critical applications because their water content limits their lubricating and wear resistance characteristics. Water glycol fluids are not recommended for use in radial piston pumps, though use in axial piston pumps is permissible at lower operating pressures and rotational speeds. Water glycol fluids attack zinc, cadmium and aluminum, so these elements must not be located anywhere in the system in order to prevent contamination and deterioration of the hydraulic system. No special hoses, seals or packing materials are required when changing over from hydraulic oil, although neoprene, Buna N, and Viton are the most common compounds used for seals. Commonly, nylon, butyl, or neoprene, are used in hose construction. The high water content in water glycol fluids excludes the use of leather, cork, untreated cotton and cellulose packing. Water glycol fluids will also dissolve certain pipe sealing compounds, as will phosphate ester fluids. Only compatible pipe sealing compounds and dopes may be used. Water glycol fluids have low toxicity and are not irritants to the skin.

High water content fluids (HWCF) are relatively inexpensive and have excellent fire-resistant qualities. Lubrication quality and viscosity stability under severe use and at elevated temperatures depends upon which fluid additive is used. Because 90% of the fluid is water, HWCF has excellent heat transfer qualities, but the temperature must be kept within the range of 40° F to 120° F to prevent freezing and evaporation. Because the viscosity is typically low, HWCF tends to scour the system and carry foreign particles. For this reason, fine filtration is recommended. Also, because the water content is very high, the fluid must be closely monitored to keep the additive level at the proper balance, and to control the microbe level. Due to the fact that the specific gravity of high water base fluids is much greater than that of petroleum base fluid, it is highly recommended that reservoirs should be mounted above the pump to prevent cavitation.

The change in viscosity as the temperature of the fluid changes is described by the viscosity index. The viscosity index is a measure of the stability of the viscosity of the fluid between two temperature extremes. As temperature increases, fluid becomes thinner and viscosity decreases. The proper viscosity index of a fluid for a particular application is determined from the fluid temperature change requirements of the machine. Industrial production machinery, for example, generally keeps the oil within a narrow temperature range, so a low viscosity index would be suitable. Mobile hydraulic equipment that must operate between freezing, or even sub-freezing temperatures, to nearly 160° F requires a fluid with a stable viscosity over the entire temperature range, calling for a viscosity index of 100 or more. Consult the pump manufacturer’s fluid specifications.

Review 5.1.1.1: What is the most important operational property of a hydraulic fluid?

a. viscosity
b. flash point
c. specific gravity
d. pour point
e. neutralization number

Task 5.2: Confirm that the fluid is compatible with the component seals and the environment.

Outcome 5.2.1:

Match the seal materials with compatible fluids.

Seals prevent internal and external leakage in a hydraulic system. Static seals are used for pump housings, valve bodies, and reservoir covers, and between cylinder ends and the cylinder tube, where there is not any relative movement between the parts to be sealed. Dynamic seals are used between moving parts, for example, between the piston and cylinder bore and between the cylinder rod and the rod bearing. About 80% of seal applications use static seals.

The starting and operating temperatures of the machine are also used to select hydraulic seals that retain their sealing properties in ambient and hot conditions. Neoprene, a general use sealing material, has a temperature range of -65° F to 225° F (-54° C to 107° C). Neoprene is satisfactory for static gasket seals and O-rings with petroleum base fluids, but is not as well suited as Nitrile rubber (Buna N), urethane, and even leather, for dynamic applications. Extremely cold and hot conditions require a sealing material that is more tolerant of heat. Viton and silicone compounds are the sealing material of choice for applications with temperature extremes.

The ability of a sealing material to prevent leakage is tested by measuring the durometer (indentation hardness), creep, and compression set after deformation. The sealing compound must be hard enough to resist wear and extrusion for the clearance gap and pressure that will be encountered, yet still be soft enough to maintain a seal during continued use. Seals have a tendency to swell in the presence of hydraulic fluid. Some swelling is permissible, and even somewhat desirable, in maintaining a fluid tight seal, but excessive swelling and accompanying softness of dynamic seals will increase friction and the tendency to abrade. Twenty percent swelling is considered to be the maximum allowable swell; swelling by a factor of 40% to 50% is allowable for confined static seals.

Seal Material	Compatible Fluids	Temperature Range
1. Metallic piston rings	Petroleum-base and synthetic fluids, phosphate esters - for high pressure and severe conditions	Low to 500° F (260° C)
2. Leather	Petroleum-base and some synthetics, phosphate esters - for medium to high pressure	-65° F to 225° F -54° C to 107° C
3. Neoprene rubber	General purpose industrial use, Freon 12; weather and salt water resistant	-65° F to 300° F -54° C to 149° C
4. Nitrile rubber (Buna N)	Petroleum-base fluids and mineral oils - used for some rotating seals, extrusion resistant	-65° F to 225° F -54° C to 107° C
5. Silicone rubber	Water and petroleum-base fluids, phosphate esters; low tensile strength and tear resistance-recommended for static seals only.	-80° F to 450° F -62° C to 232° C
6. Fluoro-Elastomers (Viton and Fluorel)	Petroleum-base, synthetic, diester, silicate ester, and halogenated hydrocarbon fluids - for high temperature fluid applications.	-20° F to 400° F -29° C to 204° C
7. Polyurethane	Petroleum-base fluids - high resistance to ozone, sunlight and weathering; low water resistance.	-65° F to 225° F -54° C to 107° C F to 300° F -54° C to 159° C

Table 5-1: Common Sealing Materials and Their Applications

The compatibility and temperature range for common hydraulic sealing materials is given in Table 5-1.

Review 5.2.1.1: Referring to Table 5-1, which seal material has the widest temperature range?

a. Polyurethane
b. Leather
c. Neoprene
d. Silicone rubber
e. Fluoro-elastomer

Task 5.3: Match the filter specifications to the requirements of a machine.

Outcome 5.3.1:

Match filter specifications to a machine.

The type of fluid to be used in a particular machine is determined by the manufacturer of the hydraulic components to match service related conditions. The manufacturer of a machine, such as a hydraulic press, may or may not manufacture the hydraulic pump and components used on the machine. Therefore, in order to back up the guarantee on the machine, the machine manufacturer generally uses the fluid specification published by the component manufacturer. Not using the specified fluid will usually void the machine warranty.

Most hydraulic component wear is caused by contaminants in the system. Contamination present in the system from the construction stage is termed built-in contamination. Contaminants generated by components in the system, such as pumps and motors, are called system generated contaminants. Contaminants added with the hydraulic fluid, or during service of the system, are called induced contamination. Other contaminants, such as moisture, are ingested during operation. Ingested contaminants typically enter the system through the reservoir breather or on the cylinder rod as the rod retracts past the rod seals.

Screens and filters are used to capture contamination before it damages the system. Screens are placed at the reservoir filler cap, at the end of the pump inlet line, and before contaminate sensitive components such as servo valves. Filters are placed in pressure lines downstream of components, such as pumps and accumulators, that generate contaminants, in return lines, and in off-line filtration circuits. The pore size, in microns, of various mesh screens is:

100 mesh = 149 microns = 0.005,811 inch
200 mesh = 74 microns = 0.002,886 inch
325 mesh = 44 microns = 0.001,716 inch

One micron (micrometer) = 1 millionth of a meter. This equals 39/1,000,000 = 0.000,039 inches. By comparison, the smallest particle visible to the unaided human eye is approximately 40 microns.

Filter performance is measured by the extent to which silt and particulate matter can be removed from the fluid. As fluid passes through the filter, filtration is proportional, which means that only a portion of the particles above a certain micron size are removed from the fluid.

There are several common ratings given for filters. Absolute filtration, $ß_X = 75$, refers to the size of the largest hard, spherical particle that can pass through the filter. The nominal filtration rating is given by some filter manufacturers, but it is not well standardized by the industry, and is therefore meaningless. The Beta Ratio is a comparison of the number of particles in the fluid greater than the micron rating of the filter on the upstream side of the filter, to the number of particles in the fluid greater than the micron rating on the downstream side of the filter. The Beta Ratio is derived from the ISO Standard 16889 (1999) Multipass Filter Test.

In typical contamination reports, the contamination level is measured by range numbers which give the actual number of contaminants above a specified micron size per milliliter of hydraulic fluid. When two range numbers are given the micron sizes are 5 and 15 microns. When three range numbers are given the micron sizes are 2, 5 and 15 microns.

The Beta Ratio compares the number of particles, of a certain size, upstream of the filter to the number of particles of the same size downstream of the filter. Efficiency represents the percentage of particles removed from the fluid as it passes through the filter. A rating of $ß_{10} = 12$ means that 92% of the particles larger than 10 microns were captured by the filter when the test was conducted. The Multipass Filter Test is ended when the pressure drop across the filter reaches a predetermined pressure differential, for example 20 psi, indicating the filter has become saturated with contamination. Particle counts are taken at one minute intervals during the test. After the completion of the test, the results are divided into ten equal time increments and the upstream and downstream particle counts for the three standard particle sizes are analyzed and ratios are calculated. Then, a final ratio is produced by averaging the ten calculated ratios.

Beta Ratio = (# of particles introduced / # of particles passed) Eq. 5-1

$ß_{10} = (100/25) = 4$

Efficiency = (# of particles removed / # of particles introduced) x 100 Eq. 5-2

Efficiency = (75/100) x 100 = 75%

Review 5.3.1.1: Assume that a hydraulic fluid being analyzed for 5 micron sized particles contains 100 particles of contaminant upstream of the filter and 50 particles of contaminant downstream after passing through the filter. What is the Beta Ratio of the filter?

a. $ß_5 = 2$
b. $ß_5 = 4$
c. $ß_5 = 5$
d. $ß_5 = 10$
e. $ß_5 = 50$

Review 5.3.1.2: What is the efficiency of a filter with a Beta Ratio of $\beta_{10} = 12$?

a. 76.8%
b. 88.2%
c. 89.5%
d. 91.7%
e. 92.7%

Table 5-2 gives the number of particles for range numbers 6 through 24. For a range number of 24, the number of particles will be between 80,000 and 160,000. The range number does not give the particle size, just the number of particles.

ISO 4406 Chart (Number of Particles per milliliter of hydraulic fluid)		
Range Number	**More than**	**Up to and including**
24	80,000	160,000
23	40,000	80,000
22	20,000	40,000
21	10,000	20,000
20	5,000	10,000
19	2,500	5,000
18	1,300	2,500
17	640	1,300
16	320	640
15	160	320
14	80	160
13	40	80
12	20	40
11	10	20
10	5	10
9	2.5	5
8	1.3	2.5
7	0.64	1.3
6	0.32	0.64

Table 5-2: Range Numbers and Particle Counts

The range numbers in Table 5-2 are part of a coherent system in that as the range number decreases from 24 to 6, the number of particles is half the preceding number. For example, a range number of 24 shows that the number of particles is between 80,000 and 160,000, whereas a range number of 23 shows the number of particles to be between 40,000 and 80,000. This means that if the particle count for one range number is known, the particle count for the remainder of the range numbers between 24 and 6 in Table 5-2 can be figured out.

Cleanliness levels for hydraulic fluid can now be specified by the ISO code number which identifies the micron size of the particles, and the actual count of the number of particles equal to or greater than the micron size given by the range number in 1 ml of hydraulic fluid. For example, an ISO code number of 13 means that the number of particles equal to or greater than the micron size given in a sample of 1 ml of hydraulic fluid will be between 40 and 80 particles. Of course, the sample size taken from the machine is larger than 1 ml, but the count is given per ml of hydraulic fluid. Also, the number of particles smaller than the micron size given by the range code cannot be determined. For example, in the range number 19/18/14, the number of particles greater than 15 microns in size is between 80 and 160 particles per milliliter (ml). The number of particles greater than 5 microns is size is between 1300 and 2500 particles per ml. The number of particles greater than 2 microns in size is between 2500 and 5000 particles per ml.

The current standard for the number of particles lists the count for three sizes of contaminants. The three sizes are given at the bottom of Table 5-3. Prior to 1999, the ISO tables listed only two micron sizes: >5µ and >15µ. The classification code gives the >2µ size particle range number on the left, the >5µ particle size range number in the center, and the >15µ size particle range number on the right. For example, a classification 18/16/13 indicates that the particle size is measured at three levels (>2µ / >5µ / >15µ). The maximum number of particles equal to or greater than 15 microns (>15µ) will lie between 40 and 80 particles. The maximum number of particles equal to or greater than 5 microns (>5µ) will be between 320 and 640. The maximum number of particles equal to or greater than 2 microns (>2µ) will be between 1300 and 2600. The quantities are given for a sample size of 1 ml of hydraulic fluid.

The ISO Classification Code is used to specify fluid cleanliness levels for various components in the hydraulic system, as well as for the entire system. Table 5-3 lists a number of mobile components together with the required cleanliness levels. The cleanliness level of new hydraulic fluid is satisfactory for cylinders and flow control valves, but the new fluid is not clean enough for the rest of the components listed in Table 5-3. At the very least, this indicates that new fluid must be tested prior to use to establish exactly what the cleanliness level is before it is added to the reservoir. If the fluid is not clean enough, it must be filtered to the required cleanliness level before it is added to the machine. A practice that is not recommended is to add the fluid to the machine, relying on the filtration system to adequately clean the fluid.

Fluid Cleanliness Codes Required for Mobile Hydraulic Components	
New hydraulic fluid	20/18/14
Gear pumps and motors	19/17/15
Piston pumps and motors	18/16/14
Solenoid control valves	20/18/15
Pressure control valves	19/17/14
Hydraulic cylinders	20/18/15
Flow control valves	19/17/14
Proportional control valves	7/15/12
Servo valves	16/14/11

Table 5-3: Typical ISO classification code cleanliness levels required for components operating at 2000 psi. (Source: Vickers Mobile Hydraulics Manual)

Task 5.4: Interpret contamination level reports for machines

Outcome 5.4.1:

Identify contaminants in a hydraulic fluid analysis.

Periodic fluid sampling and analysis are undertaken to establish a controlled maintenance program. Six basic steps in the procedure must be followed to realize the benefits of a fluid analysis program. One such task is labeling the sample for laboratory analysis. As simple as this may seem, errors in labeling causes samples to become separated from the owner, so the person who needs the information never receives it. The point is that, to be beneficial, each of the six steps must be completed in an organized and timely manner, as part of an ongoing program, to reduce maintenance and capital costs associated with fluid power machinery.

Job Responsibility 5.0: Recommend Fluid, Fluid Conductors, and Fluid Filtration

The six basic steps to establish a hydraulic fluid maintenance program are:

1. Each piece of equipment has an identification form with the history of the machine.

2. Representative oil samples are taken from the unit while it is warm using one of the accepted methods.

3. The sample is labeled correctly for return, and includes the necessary sample information.

4. The laboratory should be given information on any mechanical work or repairs made to the machine.

5. Promptly send the samples to the laboratory for analysis.

6. Laboratory reports are used to direct the maintenance program and are saved to monitor the condition of the fluid in the machine over time.

Oil analysis tests are grouped into three categories: 1) simple visual and physical testing, 2) standardized ASTM procedures, and 3) examination and identification of wear metal particles. A physical analysis includes tests for viscosity, water, total solids, fuel dilution, glycol, infrared analysis, total acid neutralization number (TAN), solids by weight, automated optical particle count, dielectric strength and ferrographic analysis. ASTM (American Society for Testing and Materials) follows established standards, for example density, wear, pour point, neutralization number, auto-ignition temperature, and contaminant analysis data reporting. Most tests that are conducted follow ISO procedures.

A particle count measures and counts the amount of individual particles present in a specific sample. The method does not identify the composition of each particle, though subsequent inspection can define the composition of the contaminates. Abnormal particle contamination levels are associated with increased wear, operational problems with close tolerance parts, fluid contamination, fluid degradation, and loss of filter life.

If the equipment supplier has not issued an ISO cleanliness level, the Fluid Power Society recommends selecting a level based upon the most critical component in the system. To ensure even greater reliability and service life from the system, go one level cleaner.

NAME AND ADDRESS INFORMATION

(Please Print or Type)

Your computer code from bottom line of last report: ____________________

COMPANY NAME: ____________________

P.O. BOX or STREET ADDRESS: ____________________

CITY/STATE/ZIP: ____________________

ATTN: ____________________

EQUIPMENT INFORMATION

To aid us in our analysis and evaluation, please complete the following with the maximum information. Complete the starred (★) questions if this is the first sample from this unit ***or*** if there have been any changes.

Unit Name / Registration / Veh. I.D.# ____________________

Sample from:
Engine / Transmission / Differential / Planetary / Torque Convertor
Final Drive / Reduction Gear / Air Compressor / Steam Turbine
Hydraulic System
Other: ____________________

★ Unit Make: ____________ ★ Unit Model: ____________

Fluid & Grade: ____________ Viscosity (SAE or ISO): ____________

★ System Capacity: ____________ ★ Oil Consumed or Added: ______ per _____
(Circle One) Qts / Gals.

★ Oil Filtration: Cannister / Strainer / Off Line ★ Coolant Type: ____________

Date Sampled: ____________________

Unit Previously Sampled? ☐ Yes ☐ No If yes, lab # of previous sample: __________

PLEASE INDICATE MILES, HOURS, DAYS, MONTHS, ETC ON THE FOLLOWING:
(Circle One)

Total Run Time: ____________ Time Since Overhaul: ____________

Run Time on Oil: ____________ Run Time on Filter: ____________

Recent Maintenance Comments on Unit Operation and/or Reason for Sampling:

____

____

____

Figure 5-1: Fluid Sample Label (adapted from Parker-Hannifin)

Ferrographic measurement identifies the quantity and composition of iron particles in the sample. A ferrogram slide is prepared in the presence of a strong magnetic field. After washing away the fluid, the ferrogram is examined to determine the composition and sources of particles and types of wear present. The report gives a description of the particles and names their most likely origin.

The sample should be labeled correctly. The form should include, at a minimum, the company name and address, the name and telephone number of the person to be notified in the event of a critical condition with the unit, the machine information and identification number, the date the sample was taken, and the type of fluid. Representative oil samples are taken from the unit while it is warm, preferably from a sampling valve in the pressure line.

Inform the laboratory of any recent mechanical work or repair done on the unit. This feedback may influence future interpretation of oil samples and recommendations. Overhauled machinery, for example, frequently generates more wear particles than worn machinery, even though a high incidence of wear particles for overhauled machinery is a normal condition.

Evaluate and save the laboratory report. The report file is the heart of the fluid analysis program. The reports record the condition of the fluid sample, plus maintenance recommendations. Typically, the reports will indicate the type and amount of wear metals and other contaminants, the water content, the viscosity, the acid neutralization number, and any other abnormal conditions found during analysis. Typical results for a report are shown in Figure 5-2.

SPECTROMETRIC ANALYSIS		
WEAR METALS AND ADDITIVES	PPM BY WEIGHT	STATUS
IRON	120.0	H
COPPER	510.0	H
CHROMIUM	< 1.0	N
LEAD	< 1.0	N
ALUMINUM	1.0	N
TIN	< 1.0	N
SILICON	< 1.0	N
ZINC	423.0	N
MAGNESIUM	< 1.0	N
CALCIUM	540.0	H
PHOSPHORUS	10.0	L
BARIUM	1.0	N
BORON	< 1.0	N
SODIUM	< 1.0	N
MOLYBDENUM	< 1.0	N
SILVER	< 1.0	N
NICKEL	< 1.0	N
TITANIUM	< 1.0	N
MANGANESE	< 1.0	N
ANTIMONY	< 1.0	N
L = LOW N = NORMAL H = HIGH		

Viscosity Analysis - ASTM D445	
SSU @ 100°F: 100.0	CST 40°c: 21.6

Water Analysis - ASTM D1744
Water Content (ppm): 101.0

Neutralization Analysis - ASTM D974
TAN: 0.1

REMARKS
1) Please check spectrometric analysis abnormal conditions

Figure 5-2: Fluid Analysis Report

The spectrometric analysis identifies many different wear metals and additives in parts per million (PPM). If a base line contamination level has not been established, a sample of new fluid also should be analyzed for comparison purposes with the machine fluid. The spectrometric test shown is limited to identifying particles below 5-7 microns in size.

To read the spectrometric analysis, look at the particle count, and then at the status section which tells if the number of contaminate metals is low (L), normal (N), or high (H).

Low (L) - Deterioration of the physical and lubricating properties of the fluid is low when compared with acceptable limits.

Normal (N) - Deterioration of the physical and lubricating properties of the fluid is within acceptable limits.

High (H) - Deterioration of the physical and lubricating properties of the fluid is clearly above the normal level, indicating significant component wear, but is not at the critical stage.

For example, the report for the sample shown indicates the fluid contains high levels of iron and copper. The iron probably originated at the pump, particularly if the sample was taken from a pressure line near the pump outlet.

The chart to the right of the spectrometric analysis in Figure 5-2 gives the SSU viscosity of the fluid as 100 SSU at 100°F of and the metric equivalent as 21.6 centistokes (CST) at 40 °C. This is in the mid range of viscosity.

Water may be thought of as the universal contaminant in hydraulic systems. Most hydraulic oils can absorb water up to 300 ppm in solution. This puts the water in a dissolved state. Amounts of water that exceed 300 ppm cannot be dissolved and will exist in a free state, giving the fluid a milky discoloration.

In Figure 5-2, the water content is given to be 101 parts per million (ppm). As a percentage this is calculated to be:

$\text{Water}_{\%} = (101.0/1{,}000{,}000) \text{ x}100 = 0.01\%$

This is read as one hundredth of one percent water, and is dissolved in the hydraulic oil. In this concentration the water cannot be seen in the fluid.

The TAN number is a designation that indicates to what degree the hydraulic fluid is acidic or alkaline. Petroleum base fluids have a tendency to become acidic after they are in use for some time, and this acid condition causes deterioration of the fluid, bearings, component parts, and seals. In water glycol fluids it is important to control the alkalinity to assure continued good performance. Any time the water content is adjusted, the fluid alkalinity must be checked after a run-in period of 24 hours. The TAN number refers to the number of milligrams (mg) of potassium hydroxide necessary to neutralize a 1 gram (gm) sample of hydraulic fluid when the ASTM D974 test procedure is used. Hydraulic fluids are fortified with additives to reduce the tendency to become acidic and to keep the neutralization number below 0.1 during normal service. The point at which the color changes is the indicator that the neutralization point has been reached. The neutralization number is then calculated from:

$\text{Neutralization Number} = (\text{Total Volume of Titrating Fluid}_{ml} \text{ x } 5.61) / \text{Weight of the Sample}$

The automatic particle count summary gives the size and particle count per milliliter of fluid, the cleanliness code, and whether there is free water present in the sample. Remember that a small amount of water can be dissolved in hydraulic oil and cannot be seen with the naked eye. The cleanliness code shown to the right in Figure 5-3 is derived from the number of particles larger than 2 microns, 5 microns, and 15 microns. Referring to Table 5-2, 353,242 particles larger than 2 microns is beyond the range numbers given. However, it is known that for an increase of 1 in range number, the particle count doubles. Extrapolating range number 25, which allows a particle count of more than 160,000 particle, up to and including 320,000 particles, to a range number of 26, would result in a particle count range of up to 320,000 particles up to and including 640,000 particles.. This would include the count of 353,242 given in Figure 5-3.

AUTOMATIC PARTICLE COUNT SUMMARY			
Size		Counts per mil.	Cleanliness Code
> 2	μm	353242.0	
> 5	μm	34434.0	
> 10	μm	2342.0	26/22/14
> 15	μm	154.0	
> 25	μm	17.0	
> 50	μm	1.0	

FREE WATER PRESENT
☐ YES
☒ NO

Figure 5-3: Automatic Particle Count Summary

The second number in the cleanliness code includes particles larger than 5 microns. The particle count given in Figure 5-3 is 34,434, which would indicate a range code number of 22 in Table 5-2. Finally, the count given for particles larger than 15 microns is 154, which fall within the range code 14 in Table 5-2. The range code number starts with the smallest particle size, in this case particles larger than 2 microns, and then proceeds to range code numbers for 5 and 15 micron particles.

Review 5.4.1.1: The TAN neutralization number of a hydraulic oil sample refers to the:

a. moisture content of the fluid
b. weight of the fluid sample
c. acidity of the fluid
d. contamination level
e. total analysis number

Job Responsibility 6.0: Analyze and Troubleshoot Hydraulic Systems

Task 6.1: Interpret diagnostic readings (pressure, flow, heat, vibration, noise, and cycle times).

Outcome 6.1.1:

Identify troubleshooting parameters.

Troubleshooting is the process of determining the causes of problems. Basic problems in hydraulics are categorized as pressure, flow, leakage, heat, noise, and vibration. Problems are reduced to one of these six types using a process like the one shown in Figure 6-1. The process consists of steps. The most efficient way to find a problem is to follow these steps.

1. Service Manual
2. Service History
3. Initial Inspection
4. List of Symptoms
5. Basic Calculations
6. General Problem Statements
7. Specific Problem Statements
8. Tests to Accept or Reject Specific Problem Statements
9. Teardown for Visual Inspection and Verification

Figure 6-1: Troubleshooting Procedure.

Critical to the troubleshooting process is the knowledge and experience of the person performing the troubleshooting. The ability to understand fluid power symbols and read schematics is very important. Also of great importance is an understanding of the function and operation of each of the components used in the circuit. As well as understanding how each component is constructed and how it "works," it is useful to know the typical failure modes of each component and the symptoms of each type of failure mode.

Problems are indicated by symptoms. Symptoms are visible evidence of a problem. The symptom lets the operator know that something is wrong.

Job Responsibility 6.0: Analyze and Troubleshoot Hydraulic Systems

A symptom, such as "the cylinder moves too slowly", could be a result of a number of problems: a) a restriction in the pressure line; b) low fluid level; c) cold oil or oil with too high a viscosity; d) a clogged inlet filter; e) an air leak in the suction line; f) oil bypassing the relief valve; g) fluid bypassing a cylinder piston; h) or a worn pump. The troubleshooting procedure is designed to simplify the process of identifying the cause of the problem.

When troubleshooting a system, there is a tendency to jump immediately to the suspected problem and to go for the quick fix. Though the quick fix may get the machine running again, this is no guarantee the machine will continue to run. Sometimes the quick fix addresses the symptom rather than the problem.

The troubleshooting process starts with the manufacturer's documentation which should list specifications for the system and describe how the system should operate. The documentation should also contain a Bill of Material (BOM) identifying the components in the system, preferably by manufacturer name and model number. This information may indicate that substitutions were made when the machine was repaired. Documentation should also include both electrical and hydraulic schematics.

The next step in identifying a problem is to review the service history and perform an inspection of the machine. Locate critical components, such as filters, pumps, valves, actuators, and controls, and try to determine if the machine has been worked on lately. Often, problems occur soon after a machine has been serviced or repaired. If components, including filter elements, have been replaced, check to see that they were correct for the application. Installing incorrect components can interfere with proper operation of the machine. Also, look for signs of abuse, hammer marks, damage from machinery such as fork lifts, and for evidence of operator errors. If a machine is being abused it will not operate properly very long.

Operating manuals often give machine cycle times and parameters, including system pressures, cylinder extension and retraction rates, pump flow rates, and hydraulic motor speeds. These parameters can be entered into formulas to calculate whether the system is operating as the operator and manual states it should operate. For example, the cylinder pressure required to move a specified load resistance is computed using Pascal's Law:

$\text{Force}_{lbs} = \text{Pressure}_{psi} \text{ x } \text{Area}_{sq\text{-}in}$ $\quad\quad$ $F = P \text{ x } A$ $\quad\quad$ Eq. 2-1

For a single rod cylinder moving a constant load, the pressure retracting a given load will be higher than the pressure extending because the cross-sectional area against which the fluid is acting is smaller. The basic pressure, force, area formula consists of variables used to verify the load capacities of cylinders.

The motor speed formula is used to relate the displacement, flow rate and speed of a hydraulic motor. The formula is useful in calculating wear and slippage in a motor. Knowing the actual flow rate supplied to a motor and its displacement, for example, someone could calculate the theoretical speed of the motor. Then, by measuring the actual speed of the motor, the amount of internal slippage can be determined.

$N_{rpm} = (Q_{gpm} \text{ x } 231 \text{ cu-in/gal}) / \text{Displacement}_{cipr}$ $\quad\quad$ $N = (Q \text{ x } 231) / D$ $\quad\quad$ Eq.6-1

Hydraulic motor torque can be derived the equation:

Hydraulic Motor $Torque_{lb\text{-}in} = (Pressure_{psig} \text{ x } Displacement_{cipr}) / 2\pi$

HMT = (P x D) / 6.28 Eq. 3-11

The cylinder formula shown below computes the velocity of a cylinder extending or retracting in gallons per minute (gpm) based using the piston area when extending and the annular area when retracting:

$Velocity_{in/min} = (Q_{gpm} \text{ x } 231 \text{ cu-in/gal}) / Area_{sq\text{-}in}$ V = (Q x 231) / A Eq. 1-1

Fluid horsepower is computed from the pressure and flow rate. Where the load is given, the formula is useful in determining whether the hydraulic system is capable of maintaining cycle times. Load and cycle times also can be used to determine if fluid horsepower is being delivered by the hydraulic system efficiently. This can be determined by calculating volumetric and overall efficiencies. Fluid horsepower differs from input horsepower in that fluid horsepower is the horsepower delivered to the circuit. Input horsepower is the horsepower that is input to the pump or pumps. The input horsepower formula includes a factor in the denominator for the overall efficiency of the pump(s).

$HP_{fluid} = (Pressure_{psig} \text{ x } Flow_{gpm}) / 1714$ HP = (P x Q) / 1714 Eq. 3-20

$HP_{input} = (Pressure_{psig} \text{ x } Flow_{gpm}) / (1714 \text{ x } Efficiency_{decimal})$ HP = (P x Q) / 1714 x E Eq. 3-21

General and specific problem statements can be taken from the operator's manual or be developed for the overall system from other sources. The easiest way to write problem statements is to classify them as one of six types (pressure, flow, leakage, heat, noise, and vibration) of major problems and then work backwards. For example, pressure problems can be listed as:

1. No pressure.
2. Low pressure.
3. Pressure fluctuates.
4. Pressure spikes as the control valve is shifted.
5. Pressure drops when the control valve is shifted.
6. Pressure drops after the machine warms up.

For each of the conditions listed above, specific problem statements can be written that would narrow the problem:

General Area		**Specific Problem Statemens**
No Pressure	-	A pressure gauge is faulty. A pump has failed. A circuit is bypassing fluid to the reservoir. A motor-pump coupling has failed. No fluid or a low fluid level in the reservoir. There is no load on the actuator.

Job Responsibility 6.0: Analyze and Troubleshoot Hydraulic Systems

Review 6.1.1.1: What type of problem would be indicated if a cylinder connected to a closed center directional control valve retracts under load in the center position?

a. pressure
b. leakage
c. flow
d. heat
e. noise and vibration

Review 6.1.1.2: Which one of the following calculations would determine if cylinder pressure is sufficient to lift a load resistance?

a. $N_{rpm} = (Q_{gpm} \text{ x } 231 \text{ cu-in/gal}) / \text{Displacement}_{cipr}$
b. $HP_{input} = (P \text{ x } Q)\ 1714$
c. $\text{Force}_{lbs} = \text{Pressure}_{psi} \text{ x } \text{Area}_{sq\text{-}in}$
d. $\text{Velocity}_{in/min} = (Q_{gpm} \text{ x } 231 \text{ cu-in/gal}) / \text{Area}_{sq\text{-}in}$
e. $\text{Hydraulic Motor Torque}_{lb\text{-}in} = \text{Pressure}_{psig} \text{ x } \text{Displacement}_{cipr} / 2\pi$

Review 6.1.1.3: What gpm flow rate would be required to power a hydraulic cylinder with a 4 inch bore x 2 inch rod x 20 inch stroke operating at a rate of 10 cycles per minute (cpm)? There are no dwell times nor any acceleration or deceleration profiles at the ends of stroke.

a. 8.16 gpm
b. 10.88 gpm
c. 16.32 gpm
d. 19.04 gpm
e. 21.77 gpm

Task 6.2: Verify the correct choice, installation, and operation of components.

Outcome 6.2.1:

Ensure the correct component has been specified for the application.

The ports on directional control valves are usually labeled “P” (pressure), “T” (tank), “A,” and “B.” The A and B ports are the actuator ports. In the case of servovalves used for directional control, the actuator ports are commonly labeled “C1” and “C2.”

A second concern with 4-way 3-position (4/3) directional control valves is the use of an incorrect center configuration (flow paths). The circuit designer must select a center configuration that will satisfy the needs of both the type of pump being used as well as the control requirements of the actuator and accessory valving for control of the actuator. It is assumed that the circuit designer understands which valve center configuration will provide proper circuit performance.

Figure 6-2 illustrates the five valve centers most commonly used in industrial systems. The five centers shown are: a) tandem center; b) open center; c) closed center; d) float or motor center; and e) regenerative (regen) center.

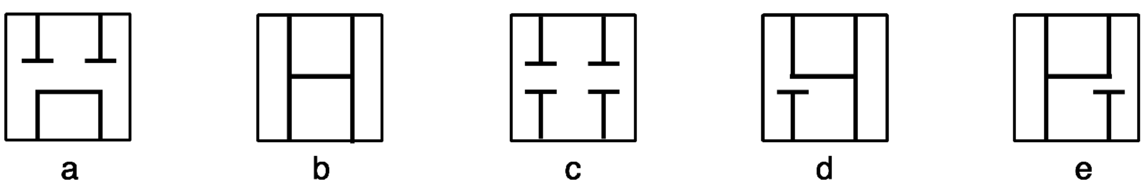


Figure 6-2: Five Common Directional Control Valve Center Configurations

Tandem center and open center spools are used with fixed displacement pumps in order to unload the pump when the actuator is not receiving flow. Closed center, float or motor center, and regen center spools are used with pressure compensated variable volume pumps. When the valve is de-energized, the blocked pressure port causes the pump to deadhead and go into pressure compensation. In the case of the regen spool, the pump will not compensate until the cylinder fully extends and deadheads.

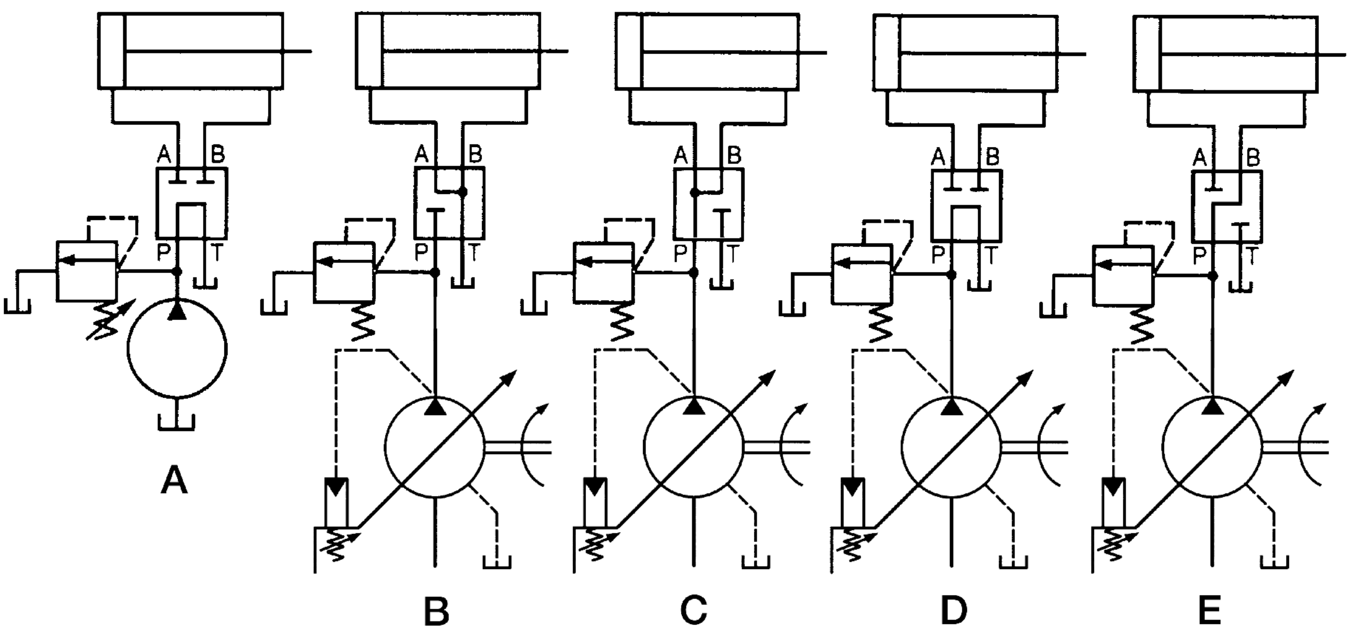
Many variable volume, pressure compensated pumps require backpressure, usually in the range of 400 psi, in order to provide internal lubricatoin in the pump. Deadheading into a blocked pressure center valve or into a circuit that deadheads is one means of providing the required backpressure.

Job Responsibility 6.0: Analyze and Troubleshoot Hydraulic Systems

A regen spool can also be used in a motor circuit. The motor will be able to freewheel just as it would with a motor center spool, but the motor ports will remain pressurized. In most cases, the use of a regen spool with a hydraulic motor will require that the motor be externally drained so that the case pressure will not rise, causing leakage at the shaft seal. An advantage of using a regen spool with a motor is that the working lines are pressurized. If meter-out flow controls are being used to control the speed of the motor, the fluid between the motor port and the flow control valve will already be pressurized, which may offer finer control, in comparison to using a motor spool, upon initial actuation.

Review 6.2.1.1: Which of the following valve center - pump combinations is generally considered to be incorrect?

a. A
b. B
c. C
d. D
e. E



Outcome 6.2.2:

Distinguish between correct and incorrect component connections.

Most hydraulic components are sensitive to the direction of flow and must be connected properly for the circuit to operate. Unidirectional pumps, for example, usually have the direction of rotation shown on the case and the ports are labeled “inlet” and “outlet.” The inlet port is usually larger than the pressure port in order to reduce pressure drop in the inlet line.

Hydraulic motors may be direction sensitive or they may operate with equal efficiency in either direction of rotation. Both motor ports would be the same size. If the motor operates with the outlet port at atmospheric pressure, generally the case and shaft seals are internally drained. However, if both ports are pressurized at the same time, for example if the motor has a brake valve at the outlet or meter-out flow control is being used, in order to prevent a blown shaft seal, the motor must have a case drain line connected to the reservoir to keep from pressurizing the case internally. Some motors have low backpressure capabilities, in the range of 15 to 25 psi, while other motors are rated for backpressures to 1500 psi and would not need to have an external case drain line connected unless that pressure was exceeded.

Pressure control valves are sensitive to the direction of flow. For example, flow cannot pass through a relief valve from T to P. Some pressure control valves, such as sequence valves, pressure reducing valves, and load holding/load control valves (counterbalance, holding, overcenter, and brake valves) are located in circuits between the directional control valve and the actuator. In order to allow reverse flow around the pressure control section of the valve in order to reverse the direction of the actuator, these valves include reverse free flow check valves. If the valve is installed incorrectly (backwards), the desired actuator control will not be achieved.

Check valves are used in many circuits. If a check valve is incorrectly installed, the circuit typically will not function. The direction of free flow is often shown by an arrow marked on the side of the valve. While it is uncommon that a check valve will become reversed in an operating circuit, there is always the possibility that it may become reversed during construction of the circuit or during maintenance. Occasionally, a check valve will stick open.

In figure 6-3, the high-low circuit consists of two hydraulic pumps that work together, with check valve A located in the line connecting their pressure ports. The pump on the right is unloaded when the system pressure reaches the setting of the unloading valve. High-low circuits are used to provide a high flow rate at a low to intermediate pressure during rapid extension of the cylinder, followed by low flow at high pressure during the work portion of the cycle.

Check valve B located directly below the directional control valve provides a 65 psi pressure differential source for pilot operation of the directional control valve.

Check valves C and D control the regenerative portion of the cylinder circuit. If the area ratio of the cap end of the cylinder to the rod end were 2 to 1, the full time regenerative circuit would cause the cylinder rod to have the same velocity and force extending as it does retracting.

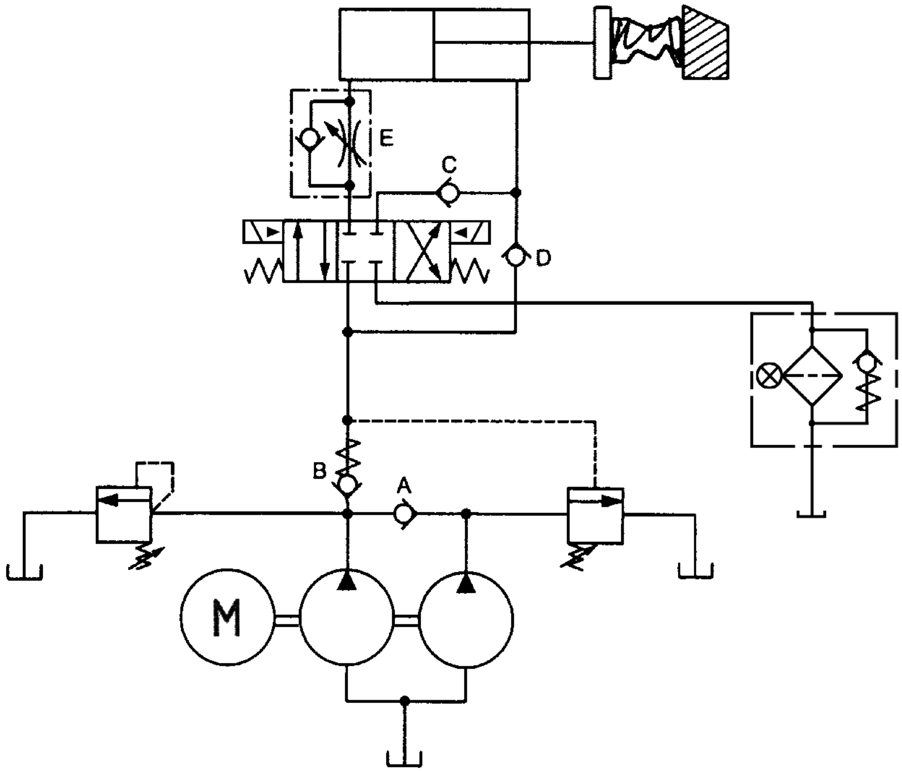


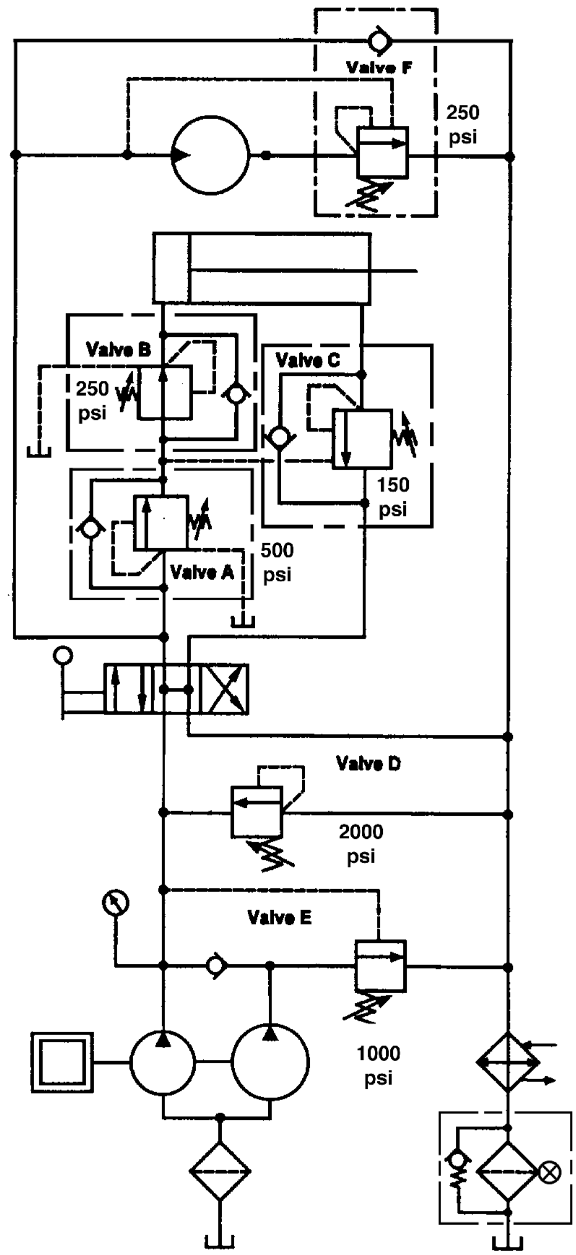
Figure 6-3: Circuit Schematic With Check Valves

When the cylinder rod extends, check valve C is closed and check valve D is open. Flow from the pump enters the cap end of the cylinder to extend the cylinder rod, while return flow from the rod end joins pump flow to increase the velocity of the cylinder extending. Thus, flow from rod end of the cylinder joins pump flow to extend the cylinder. When the directional control valve is shifted to retract the cylinder, check valve C is opened. Since the pressure at the cylinder side of check valve D is slightly lower then on its opposite side, due to pressure drop across the directional control valve, check valve D closes.

Check valve E provides a free flow path around the needle valve allowing the cylinder to extend at full speed, but checks flow as the cylinder rod is retracted, forcing fluid through the needle valve. Thus, check valve E controls the meter-out portion of the cylinder circuit when the cylinder rod retracts.

Review 6.2.2.1: In the schematic below, which of the pressure control valves is installed backwards?

a. A
b. B
c. C
d. D
e. E



Review 6.2.2.2: In Figure 6-3, what would happen if check valve A sticks open?

a. Both pumps would stall.
b. The accumulator would overfill.
c. Both pumps would unload at low pressure
d. The cylinder rod velocity would increase retracting.
e. The circuit would become inoperative.

Outcome 6.2.3:

Distinguish between correct and incorrect operation of components.

In order to determine whether or not a component is functioning correctly, the troubleshooter must understand what the behavior characteristics of each component. In addition, the troubleshooter should be familiar with the failure modes of each component and the operational characteristics of each of those modes. For example, the following items relate to pressure control valves:

1) The poppet and seat may be damaged or contamination can prevent the poppet from seating against the seat.

2) In spool type valves, sticktion may result from knurling caused by contamination or from varnish deposits caused by oxidized fluid, preventing the spool from partially or completely shifting, open or closed.

3) Pilot orifices may be blocked by contamination.

4) Pilot pressure and drain lines may be obstructed.

5) The spring may be broken or fatigued.

6) A valve design inappropriate for the circuit may have been installed

To be an effective troubleshooter, one must become familiar with the operating characteristics and failure modes of pumps, pressure control valves, flow control valves, directional control valves, actuators, and accessory equipment.

Task 6.3: Isolate hydraulic circuit and system malfunctions.

Outcome 6.3.1:

Trace the operation sequence of a circuit.

Hydraulic circuits are traced from the pump, through various control and pressure valves, to the actuator, and then back to reservoir. How the circuit operates is influenced by how components operate individually and in combination with other components. Tracing the flow of fluid through components, beginning with the pump, is useful in understanding how the circuit operates.

The circuit in Figure 6-4 is being used for illustrative purposes. The circuit performs three basic functions: a high-low pump circuit, a cylinder circuit, and a motor circuit.

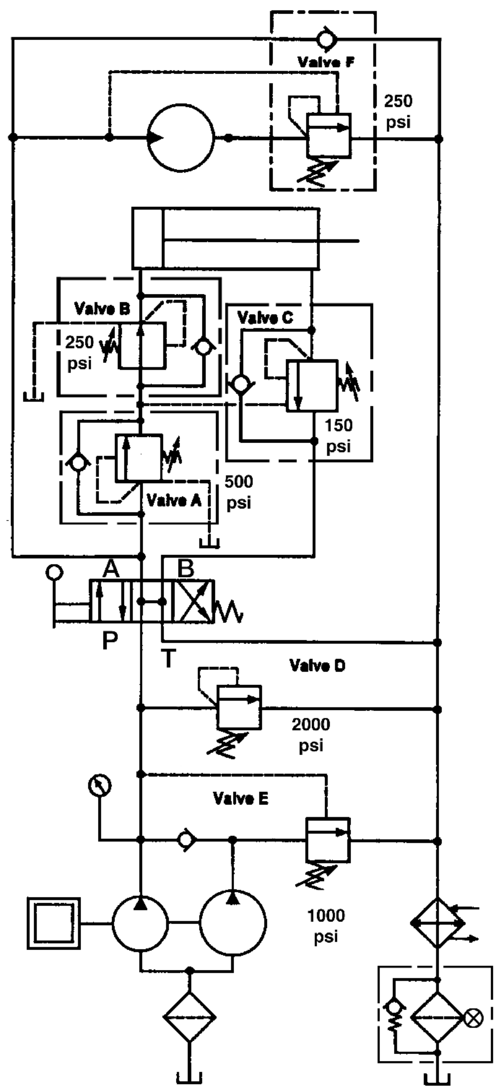


Figure 6-4: A Circuit Schematic with Three Functions

The high-low pump circuit includes two fixed displacement pumps, an open center directional control valve, and pressure relief valve D which will prevent damage to the system components when either the cylinder deadheads or the motor stalls. The cylinder is equipped with sequence valve A. Pressure reducing valve B controls the cylinder force during the extension stroke. Counterbalance valve C is located in the line connecting the rod end of the cylinder to the directional control valve. The bi-directional motor circuit is equipped with brake valve F that operates when the motor is turned in one direction by an over-running load. The open center spool in the directional control valve allows fluid to reach the opposite port of the motor, preventing the motor from cavitating during braking. A brake valve is not effective in controlling the deceleration of the load if a motor fails to receive fluid at the port opposite the brake valve.

For the circuit to operate properly pressure relief valve D must be set at a higher setting than valves A, B and E. Also, the pressure setting of sequence valve A is higher than the pressure setting of pressure reducing valve B, The pressure setting of counterbalance valve C and brake valve F are determined by the magnitude of the load, the strength of components, and safety factors.

Review 6.3.1.1: In Figure 6-4, assume the directional valve has been actuated connecting P to A and B to T and there is a low initial load on the hydraulic motor. As a result, brake valve F opens once its 250 psi setting is reached. After a short period of time the load pressure on the motor increases above 500 psi and the motor stalls. Which pressure control valve operates next?

a. A
b. B
c. C
d. D
e. E

Outcome 6.3.2:

Predict circuit malfunctions.

Circuits malfunction for a variety of reasons. Basic problems include components that fail, components that are mismatched, circuits that are improperly designed, and circuits that are not matched to the load.

The circuit shown in Figure 6-5 uses a two-stage, pilot operated three position directional control valve to operate a cylinder. The first stage is solenoid operated to pilot the second stage. Both first and second stage valves are spring centered.

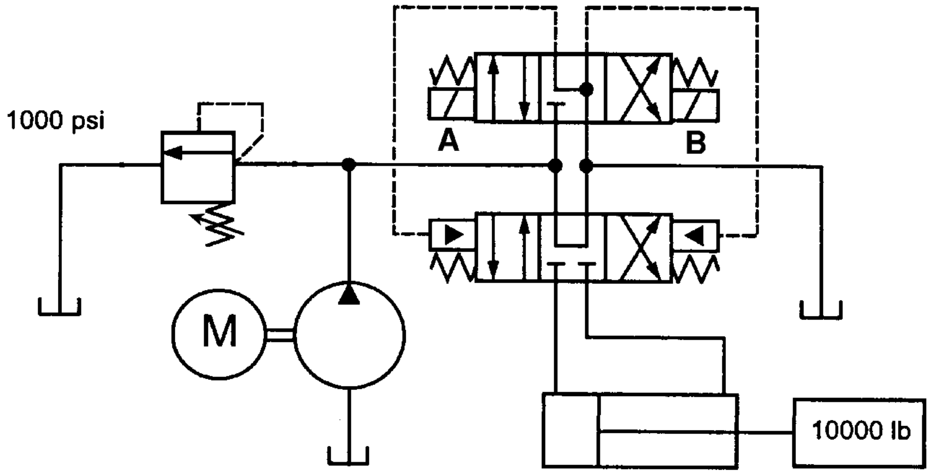


Figure 6-5: Pilot Operated Directional Control Valve Circuit

Review 6.3.2.1: The 4 inch bore x 1-1/2 inch rod cylinder shown in Figure 6-5 fails to move the load when the directional control valve solenoid control is energized. What is the likely cause of the circuit malfunction? Choose the best answer.

a. Solenoid A on the pilot valve malfunctions.
b. The pressure relief valve setting is too high.
c. The wrong spool center was used in the main valve stage.
d. The wrong spool center was used in the pilot valve.
e. The backpressure check valve in the P port of the main valve is missing.

The hypothetical circuit in Figure 6-6 is intended to explain possible interactions between two parts of a circuit, in this case a motor and cylinder that are connected to the same directional control valve.

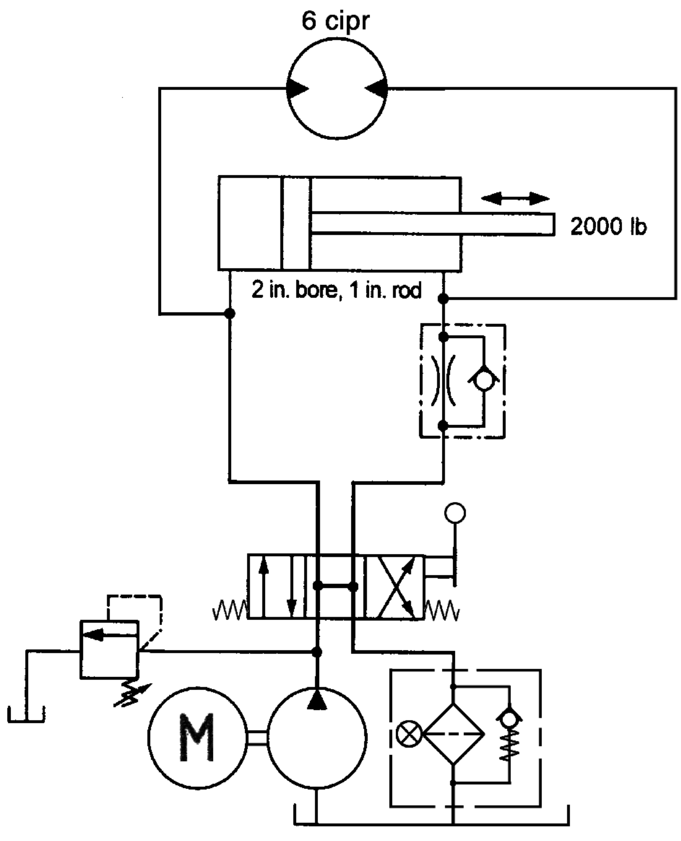


Figure 6-6: Cylinder - Motor Circuit 1

The motor is connected in parallel with the cylinder, meaning that both the motor and cylinder receive fluid from the same source when the directional control valve is shifted to extend or retract the cylinder. A flow control valve is connected in the line between the directional control valve and cylinder, metering fluid to the cylinder and motor when the cylinder rod is retracted.

The cylinder acts against a load of 2000 Ibs extending and retracting, while the bi-directional motor is shown unloaded. Unknown is the pressure setting of the relief valve and the setting of the flow control valve. The circuit is powered by a fixed displacement pump.

Review 6.3.2.2: In Figure 6-6, the cylinder rod does not extend when the directional control valve is actuated. The pressure relief valve is set at 1000 psi. What should be done to make the cylinder rod extend and retract? The torque required to turn the load is 575 lb-in.

a. Increase the pressure setting of the relief valve.
b. Resize the cylinder or the motor.
c. Reverse the flow control valve at the rod end of the cylinder.
d. Jog the circuit with the directional control valve.
e. The circuit cannot be made to operate properly.

The hypothetical circuit shown in Figure 6-7 is similar to the one in Figure 6-6, but is more complex because it contains pressure sensitive valves. The circuit might operate in a number of ways, depending upon system pressure and the pressure settings of valves A and B. One possibility is that the cylinder could be made to extend and retract the rod, independently of the operation of the motor. Another possibility is that the motor would operate in forward and reverse independently of the operation of the cylinder. A third possibility is that the cylinder and motor would operate at the same time, causing them to interact with each other.

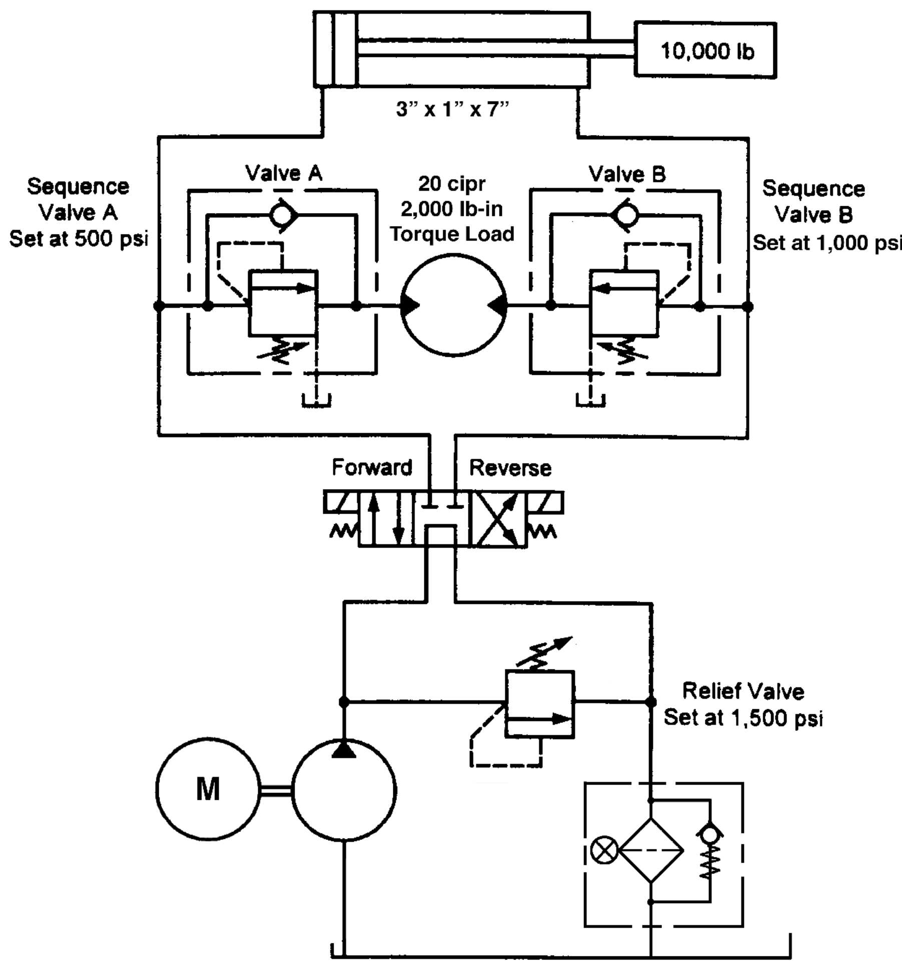


Figure 6-7: Cylinder - Motor Circuit 2

Review 6.3.2.3: Neglecting friction, why will the cylinder in Figure 6-7 fail to extend?

a. The pressure setting of valve A is too low.
b. The pressure setting of valve B is too high.
c. The load on the motor is too high.
d. The pressure relief valve is set too low.
e. The cylinder regenerates through the motor.

Job Responsibility 7.0: Interface Hydraulic Instrumentation and Control Systems

Task 7.1: Analyze the control sequence and machine operation.

Outcome 7.1.1:

Recognize the basics of electrical control systems.

Hydraulic system control functions regulate pressure, flow, position, and sequence of operations, including stopping, starting, and safety interlocks. Components include input, logic, and output devices. Input devices include electrical switches, transducers, and manual valve operators. Logic devices include programmable logic controllers (PLCs) and motion controllers (real time control, i.e., no scan time losses). Output devices include timers, relay coils, solenoids, lights motors, and hydraulic actuators such as cylinders and motors.

Automation makes use of an electrical control system to make decisions to turn output devices on or off based on signals received from various input devices. The control system combines the input and output devices with logic to accomplish a specific machine functions or operation. How these control components are connected determines the control logic which is the process through which the control system makes its decisions.

Schematics are used to show this control loop as clearly as possible so that someone can follow the thought process behind the logic. The most familiar schematic used is called a ladder diagram. The control console and associated control cabinets house the "brains" of the machine functions and operations. Part of being able to design and then provide maintenance and repair requires being able to read and understand ladder diagrams and relay logic.

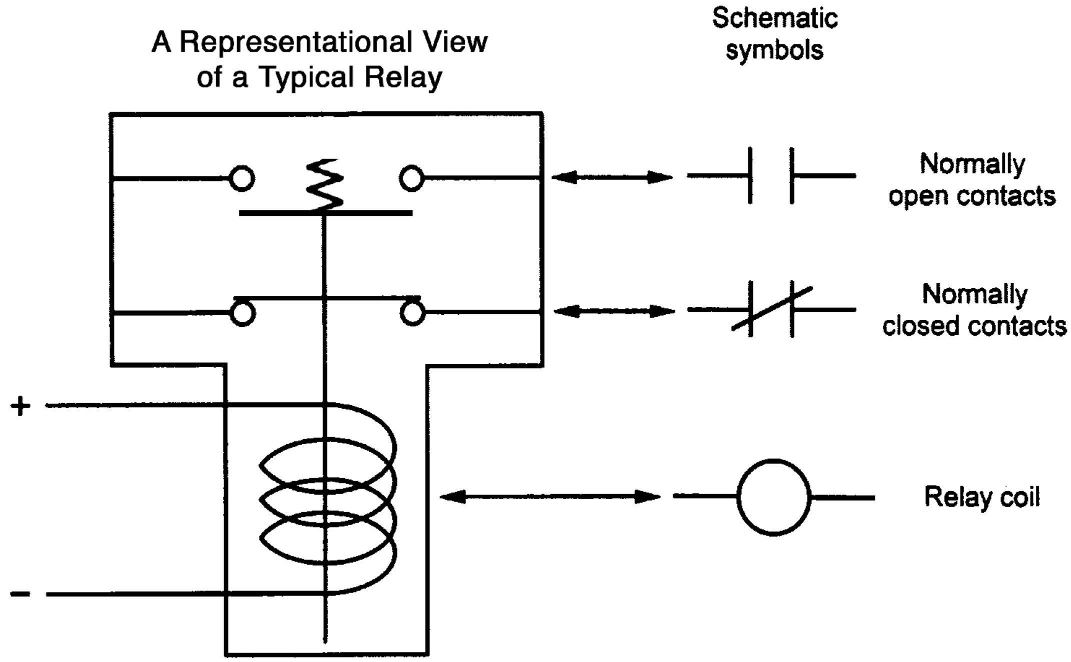
The three basic logic functions as they are used in digital systems (relay logic and PLCs) are:

1. The "AND" function, which is true (+) only if all inputs are true (+). This is read: Input A (+) and input B (+) will permit an output at C (result). For example, at the ATM you must have a card (+) and a pin number (+) and an account balance to receive cash (result).

2. The "OR" function, which is true (+) if at least one of the inputs is on. This is read: either A input or B input will permit an output at C (result). For example: your electronic car door opener (input) or key (input) will unlock the door (result).

3. The "NOT" function, which states that input must not be present for a result to occur. In other words the NOT function is true (+) if the input is not true and vice versa. For example, the switch on the telephone cradle must not be depressed for the telephone to operate (result).

Memory and Time are also logic functions, but they are treated differently in PLC logic. For example, there are "latching" relays as well as "latching" relay circuits for "memory functions."



Note that relays "pull in" rather than "push in." When the relay is actuated (actuated), the spring extends in tension. When the relay shown in this figure is de-actuated (de-energized), the spring pulls the contacts apart. Although this figure illustrates a relay with normally open (non-passing) contacts, relays are also available with normally closed (passing) contacts and also with a combination of normally open and normally closed contacts, offering application versatility.

Figure 7-1: Control Relay

The ladder diagram is the most common form of electrical schematic. It is a method which can be used to detach part of a device and place it where it logically falls within the control sequence. Figure 7-2 shows a ladder diagram with components. The two vertical lines are called “buses” and represent the voltage source; the horizontal lines are called “rungs.” The control logic circuit is drawn on the rungs from left to right using schematic symbols. The devices are arranged and numbered in the order in which they are actuated so that the machine sequence reads from beginning to end like a book or newspaper. If the actual devices were drawn with connections it would make for a complex drawing.

Rung number one in Figure 7-2 is an example of an AND circuit. Both input devices PB-1 (push button switch) and LS-1 (limit switch) are actuated in order to energize CR-1 (control relay) the output device. Output devices can be timers, counters, relay coils, solenoids, lights, motors, etc. In Figure 7-2, the coil of relay CR-1 energizes and pulls in the relay contacts which can be used as an input signal on other rungs in the ladder or sent directly to an output device. The output device is always the last item on the rung and is always drawn close to the right (low voltage) bus.

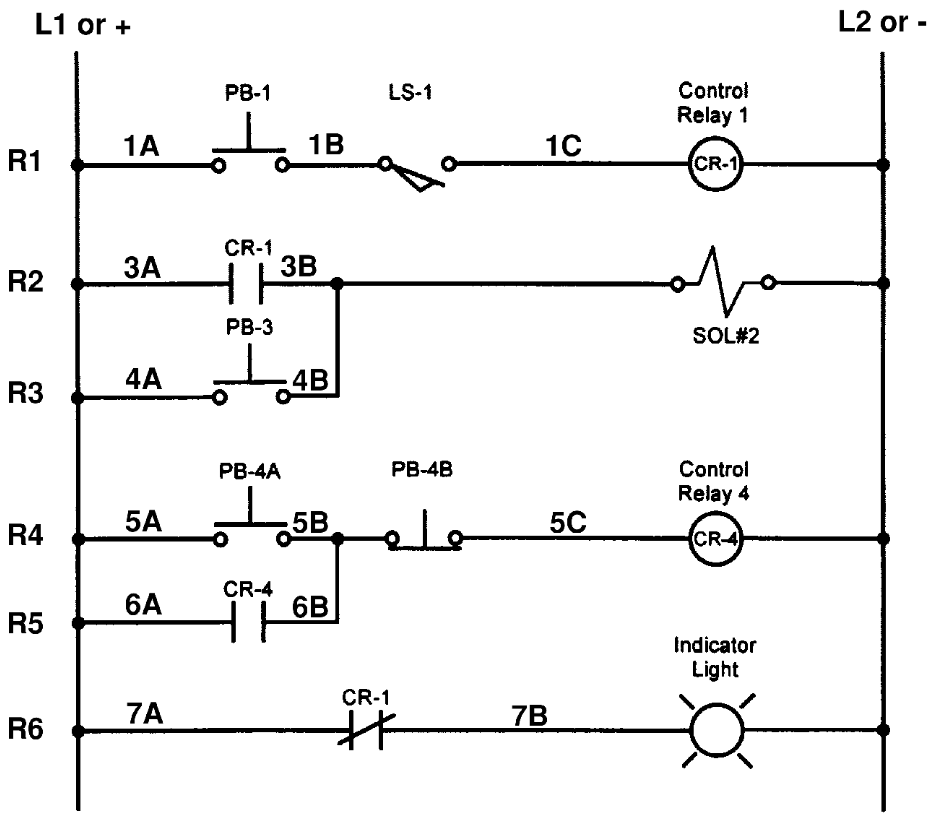


Figure 7-2: Typical Ladder Diagram

There are several methods of labeling the rungs in ladder diagrams. The method used in this review manual labels each wire. These wire labels would then correlate to the actual wire labels on the machine. Wire number 2, by convention, is generally reserved for the ground strip or terminal connection label. Also shown in Figure 7-2 are reference numbers to the left of each rung.

The vertical sides of a ladder diagram are generally labeled + and - for DC circuits while AC circuits use the labels L1 and L2 or H (high) and N (neutral).

The second rung, with wire labels 3A and 3B, is an example of an OR circuit. The solenoid, SOL #2, can be energized by control relay CR-1 or push button switch PB-3.

The example of a “specialized” circuit is shown on the forth rung (wires 5A and 5B) is a “latching” circuit. No power flows until push button PB-4A is pressed to energize CR-4. The normally open contact for control relay 4 closes and power flows through the contacts of PB-4A and CR-4. When push button PB-4A is released power continues to flow through CR-4 and the coil in CR-4 is “latched in.”

When a relay coil, such as that shown in Figure 7-1, is energized, all of the contacts in that relay connect which can cause a change in one or more of the logic rungs simultaneously. To make it easy to keep track of all of the rungs affected by the energizing of a relay, numbers are placed after the verbal description.

Motor starter relays contain a special normally closed contact called the overload contact. The overload contact is drawn on the rung between the motor starter coil and the bus. This is shown as OL-1 (meaning overload) in Figure 7-3. The overload contacts open if the motor current exceeds the maximum rated load current, interrupting power flow to the coil, thereby shutting down the motor. Overload contacts are physically connected after the coil for safety reasons.

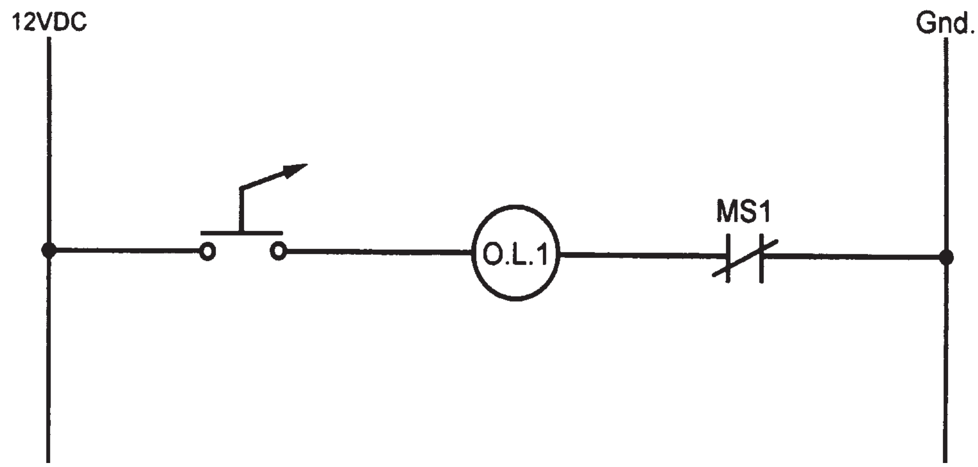


Figure 7-3: An Overload Switch in Series with a Motor

Once the relay logic diagram is constructed, the next step is to label the diagram by numbering the connections and the wires. For more than one different wire on the same line, use 1a, 1b, etc (the case of the letter is insensitive). Remember that wires do not change numbers unless they are separated by a device. The line segment to the right of the output device is not labeled.

Many systems are very complex and have hundreds of rungs. In these large drawings it simplifies finding wires on the control panel and in the drawing if reference numbers are used instead of rung numbers. For example, in Figure 7-2 reference line numbers are sequential, equally spaced, and not identified by rung.

There are some “Do’s and Don’ts” about ladder diagrams. Output devices carry a voltage rating. In the first rung in Figure 7-4 are two lightbulbs rated for 12 vdc. Lights should be wired in parallel. If you put two bulbs of equal wattage in series with a 12 vdc source, each bulb would see only 6 volts and would light up dimly. Likewise, a 12 vdc relay needs all 12 volts to latch the contacts properly.

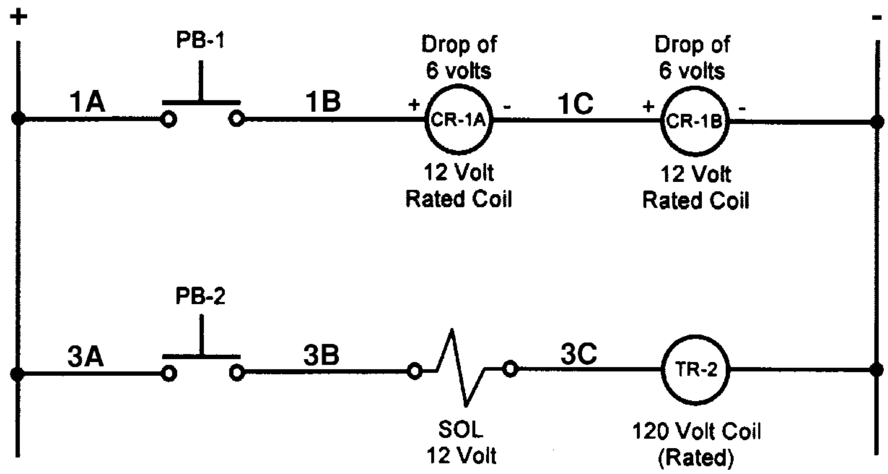


Figure 7-4: Output Components in Series

It is rare put two output devices in series on a rung. Do not mix output devices together that are not rated for the voltage between the left and right power bars. Do not put an input device across a rung without ending in an output device. This mistake is shown on rung 4 in Figure 7-5 as a "dead short." Depressing PB-4 will short the switch.

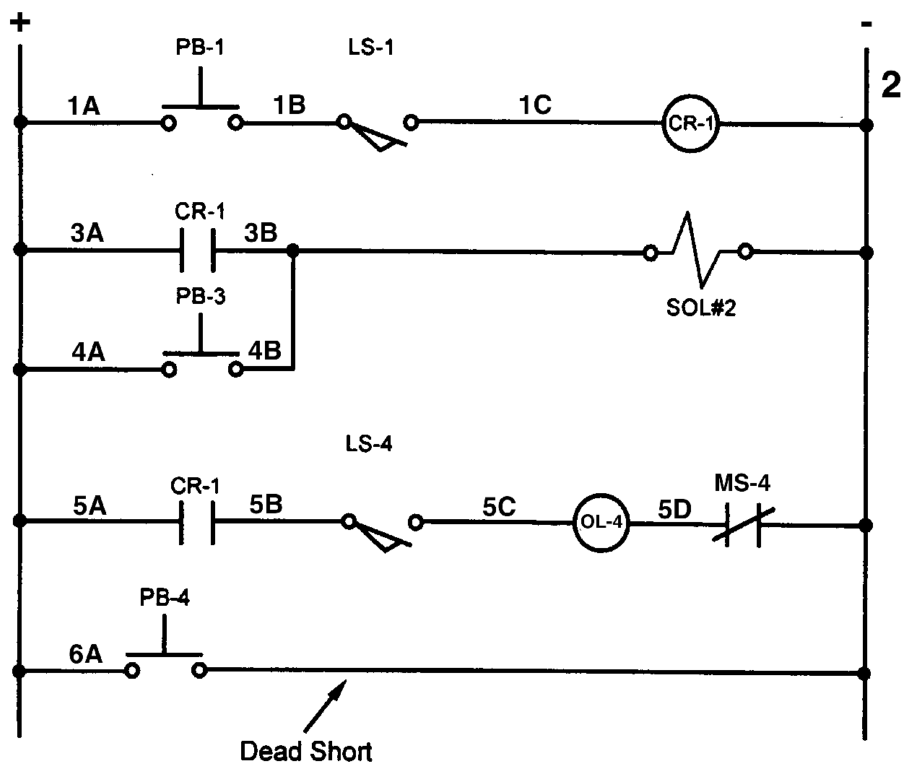
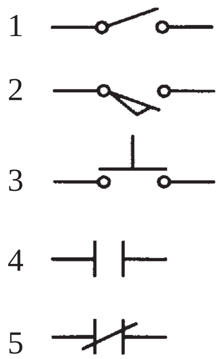


Figure 7-5: An Input Device Wired as a "Dead Short"

Review 7.1.1.1: Which one of the following symbols illustrates normally open relay contacts?

a. 1
b. 2
c. 3
d. 4
e. 5



Outcome 7.2.1:

Match programmable logic controller (PLC) devices with their application.

Programmable logic controllers (PLCs) are software-based equivalents of hard wired relay logic control panels used to electrically control a machine or process. PLCs are composed of an input interface (or input module), output interface (or output module), logic processor unit with memory storage (CPU), power supply, and programming device. Programming devices may either be an integral part of the unit or separate so they can be used to program other units of the same type. The components of a PLC are shown in Figure 7-6.

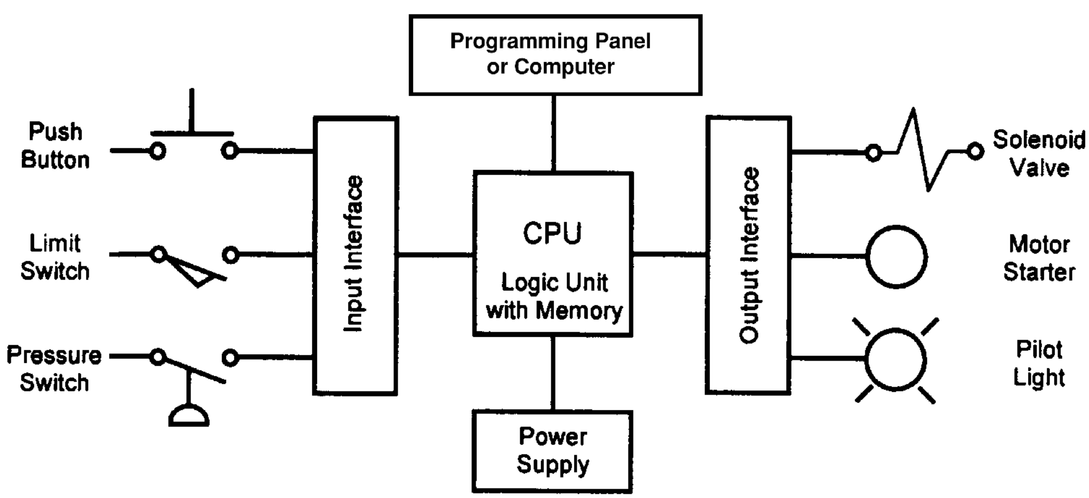


Figure 7-6: Programmable Logic Controller Components

Referring to Figure 7-6, the input interface, which is hard wired to the machine, receives signals or commands from the machine in either one of two forms:

1) Digital signals - on/off or closed/open. Examples include push buttons, limit switches, and position encoders, or:

2) Analog signals, which are a variable voltage or current signal. An example of an analog signal is the variable input to the amplifier card of a proportional valve. A digital-to-analog card converts the digital output signal from a PLC into the analog signal required by the valve's amplifier card.

The logic processor of the PLC is commonly referred to as the central processing unit (CPU). The CPU analyses and integrates the input data received from the input interface based upon the instructions that have been programmed into memory. When the various preprogrammed conditions have been met, the CPU will command the output interface to send a signal to the appropriate output component. An example of how the CPU executes a series of instructions is shown in Figure 7-7.

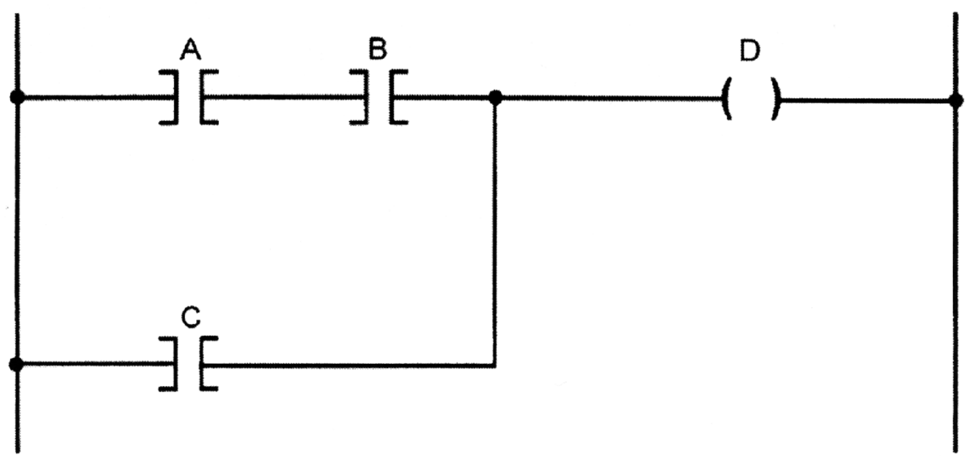


Figure 7-7: Processor Logic

Note that in the diagram in Figure 7-7, items A, B, and C are shown as brackets. These symbols may be shown instead as vertical lines. Also, item D is shown as parentheses. Item D might have been shown instead as a circle. The difference in symbols is based on whether or not the ladder diagram was generated by a typewriter or a text based computer program rather than a program dedicated to generating ladder diagrams.

1) The CPU will scan the input interfaces to determine the status of A, B, and C.

2) Based upon the program logic, if input A and input B, or if input C is true, the CPU will turn on output interface D.

3) The CPU will continue to scan the input interface and will act on the status of the inputs based upon the logic determined by the program.

The CPU also has memory storage capabilities. The program stored in memory tells the CPU what to do. The program is the logic of the PLC. The program is stored in memory by either the manufacturer (OEM) of the machine or by the end user of the machine. Most PLCs use random access memory (RAM), programmable read only memory (PROM), or erasable programmable read only memory (EPROM).

Random access memory (RAM) stores information in a solid state chip. The information is retained by supplying a constant electrical supply. Therefore, a backup battery is required in order to retain the memory storage in case of a power failure. This is currently the most commonly used type of memory.

Programmable read only memory (PROM) consists of solid state chips with information permanently stored in them. PROM chips do not require electrical power in order to retain memory. Normally, PROM chips are not re-programmable.

Erasable programmable read only memory (EPROM) is electrically erasable. As is the case with PROM chips, an electrical supply is not required in order to maintain memory storage.

All of these memory types file fixed format instructions. The fixed format consists of the instructions the PLC will process. The program's instructions are read sequentially, one instruction at a time, line by line. The time required to read the memory one time is called the memory scan time. One instruction takes up one unit of memory. The total memory capacity is the maximum number of instructions or data that the memory can store.

The power supply converts alternating current into the direct current that the processor needs for operation. A typical power supply will convert 120 vac current into +5 and +15 vdc current. Some PLCs do not have integral power supplies. If such is the case, an external power supply must be used.

A program may be uploaded into a PLC by a variety of means including, but not limited to a dedicated keypad, a special application programming device, a laptop computer, or a wired connection to a central computer system.

Programmable controllers have almost completely replaced the use of hard wired relay logic to control machines. There are several reasons for this. First, generally it takes less time to install and wire a PLC than it does to mount and wire a group of relays. Secondly, a program may be easily revised and uploaded to the PLC. To make changes in a relay logic system requires rewiring the relays. Also, small, low cost PLCs are available, making it cost effective to use a PLC rather than even a small number of relays. Finally, a PLC based system requires less panel space than does an equivalent relay logic system.

Electromechanical functions, such as timers and counters, are easily duplicated in PLC logic, negating the need for the additional cost and space for these components. Often, such components are configured for a specific adjustment range, such as 0 to 1 minute, 0 to 10 minutes, or 1 to 100 counts. If a different range is needed, the component needs to be changed out. In contrast, timer and counter functions programmed into the logic of a PLC do not have any limits on their ranges, and have no wearing parts, as do electromechanical components.

Another advantage of PLCs is that they can be used to perform arithmetical and logic operations. PLCs can store data collected from sensors, such as pressure, position, velocity, or temperature vs. time information. Some PLCs include self-diagnostic capabilities.

A PLC may be used to control just a segment of a machine, such as an electrohydraulic servo system, or it can control an entire machine. Furthermore, PLCs can be networked together.

The inputs to the PLC typically consist of all of the physical input devices. Each device is wired to a different input connection and is given its own unique number or label. The labels are used in the program to identify the input device. In the wiring diagram shown in Figure 7-8, input contacts MR-1 and MR-4 do not need to be wired to the input connections on the PLC circuit. Instead, these contacts, and their associated relay coils, are mimicked in the PLC's program using logic relays. The PLC inputs can be specified in rating for many voltage and amperage configurations: 12 vdc, 24 vdc, + or - 10 vdc, 120 vac, 240 vac, 0 to 20 mA, or 4 to 20 mA, for analog inputs. Digital inputs are acceptable as well. Each input is generally wired to a small light emitting diode (LED) so that one may visually determine whether the signal from the input device is "on" or "off."

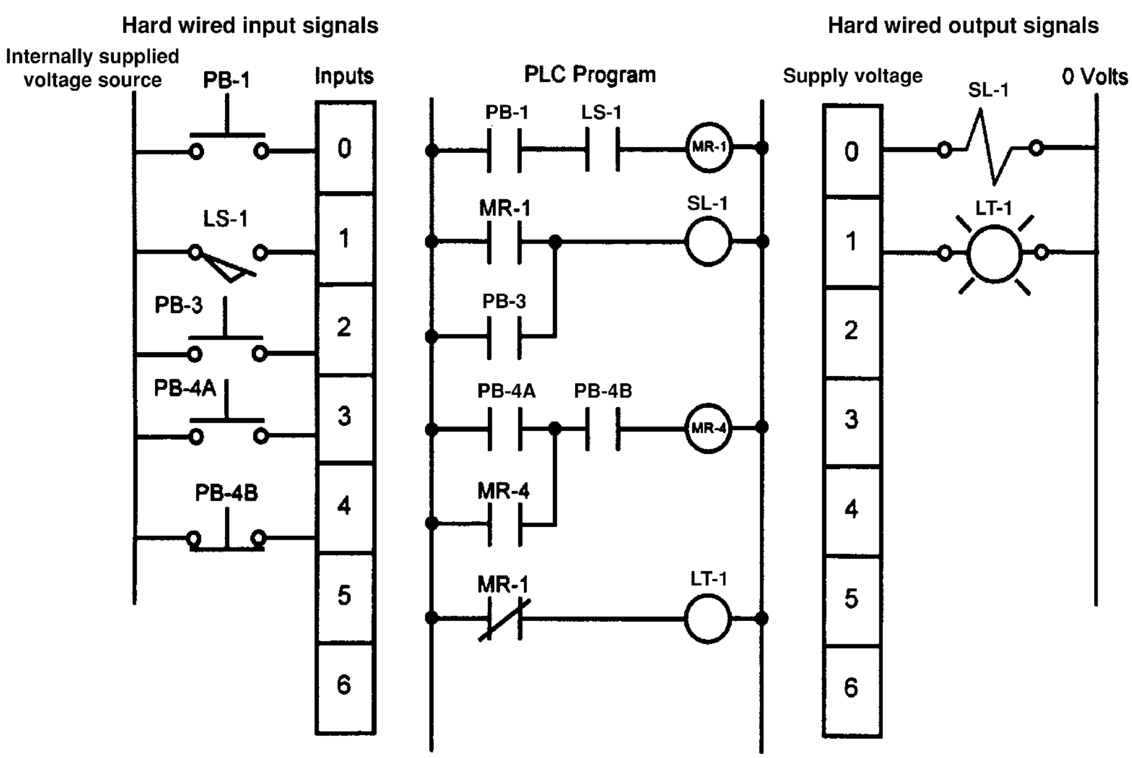


Figure 7-8: PLC Logic Wiring Diagram

The PLC's instruction manual details specifics related to the input devices such as the allowable voltage rating of the input devices, how the inputs are wired, and the numerical input address of the terminal connection numbers. The input address is the electronic memory address where the PLC stores the value (0 or 1) of the input device.

In many ways, PLC outputs are similar to PLC inputs. The outputs are physical devices that are wired to the PLC's output terminals. These terminals are numbered, as are the inputs. The output must match the rated voltage of the output device, for example, 120 vac. This is generally accomplished by an output relay device. Figure 7-9 illustrates a PLC output strip with variable outputs.

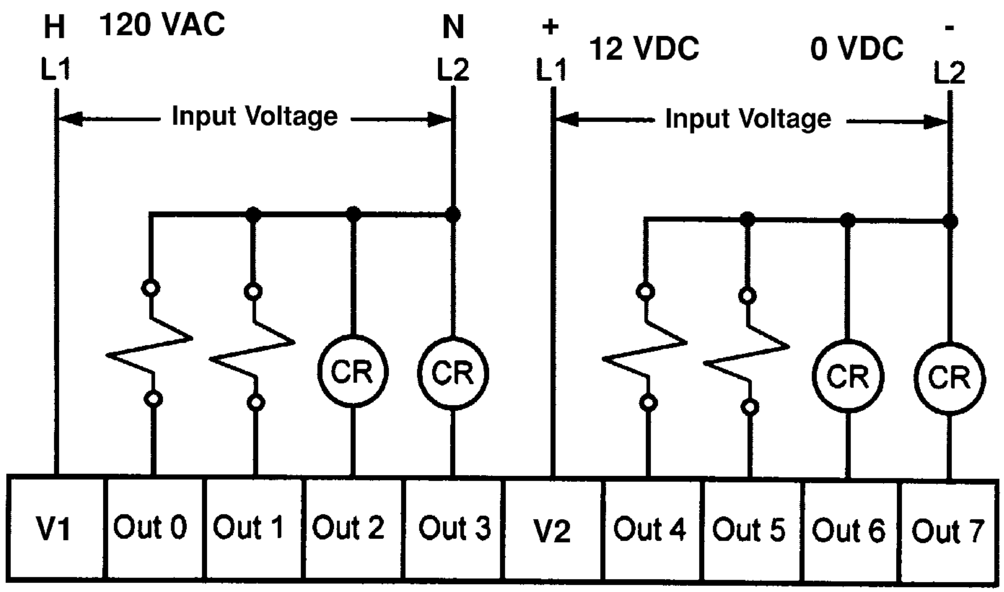


Figure 7-9: PLC Output Strip with Variable Outputs

PLC outputs are special in that they have a current rating. Consequently, each output device must not draw more current than the rating of the output strip. To protect the output circuit, some manufacturers provide fusing. This wiring diagram has a common voltage source connection provided for each group of four outputs. This configuration allows for the possibility of using one voltage configuration on the first group of four outputs, and a different voltage output for the next group of four outputs. This method is commonly uses groups of relay contacts rated for a specific output voltage and amperage range. Many times, the "common" connection is dedicated to only one output rather than to multiple outputs. The PLC does not supply the voltage or current to the loads.

Review 7.2.1.1: Analog output signals from a PLC require:

a. a relay
b. an indicator light
c. a parallel port
d. a special interface card
e. a serial port

Outcome 7.3.1:

Understand ladder logic diagrams: Identify probable causes of hydraulic circuit failure from the symptoms, the sequence of the circuit operations, the circuit schematic, and the electrical schematic.

Troubleshooting a hydraulic circuit failure begins by tracing the malfunction to a faulty component. This can be done by identifying most likely faults from symptoms and operational characteristics of the machine. The same is true if the fault is electrical, except that the problem may be masked by a combination of hydraulic as well as electrical faults.

The circuit in Fig. 7-10 consists of two cylinders operated by solenoid actuated two-position, four-way valves. The ladder diagram electrical schematic for the circuit is shown at the bottom. By convention, the left side of the ladder has a plus voltage, while the right side of the ladder has a minus voltage or ground. The rungs of the ladder diagram are numbered down the left side so that wiring and electrical components in complex circuits can be traced. Ladder diagrams use AND (A and B), OR (A or B) and NOT (output is on when input is off) logic functions.

In Fig. 7-10, rung 1 of the ladder logic diagram contains an emergency retract button labeled ER, a contact switch labeled CS, a limit switch labeled 1LS (shown closed) that is tripped open by Cylinder 1, and a control relay labeled 1CR coil. Rung 2 contains the contacts that are closed by 1CR that carry current to Solenoid 1. Rung 3 shows the same control relay 1CR-1 connected to Sol. 1. Rung 2A also contains relay coil 2CR which is energized when 1LS is opened at the end of Cylinder 1 extension stroke, closing 2CR-1contacts and energizing Solenoid 2.

The circuit operates as follows: when the cycle start button is pressed, control relay 1CR closes, actuating Solenoid 1 in Control Valve 1 to extend Cylinder Rod 1. At full extension (position P2), Cylinder Rod 1 contacts limit switch 1LS which opens the circuit to 1CR, retracting Cylinder Rod 1, and closes control relay 2CR, actuating Solenoid 2 in Control Valve 2 to extend Cylinder Rod 2. At full extension (position P4).

Cylinder Rod 2 contacts limit switch 2LS, opening the circuit to Control Solenoid 2 and Cylinder Rod 2 retracts. Both cylinders rods remain retracted until the start button restarts the cycle.

There are several methods that are currently used to label the rungs and wire segments in ladder logic diagrams. Thus, the method shown in Fig. 7-10 is one of the excepted methods.

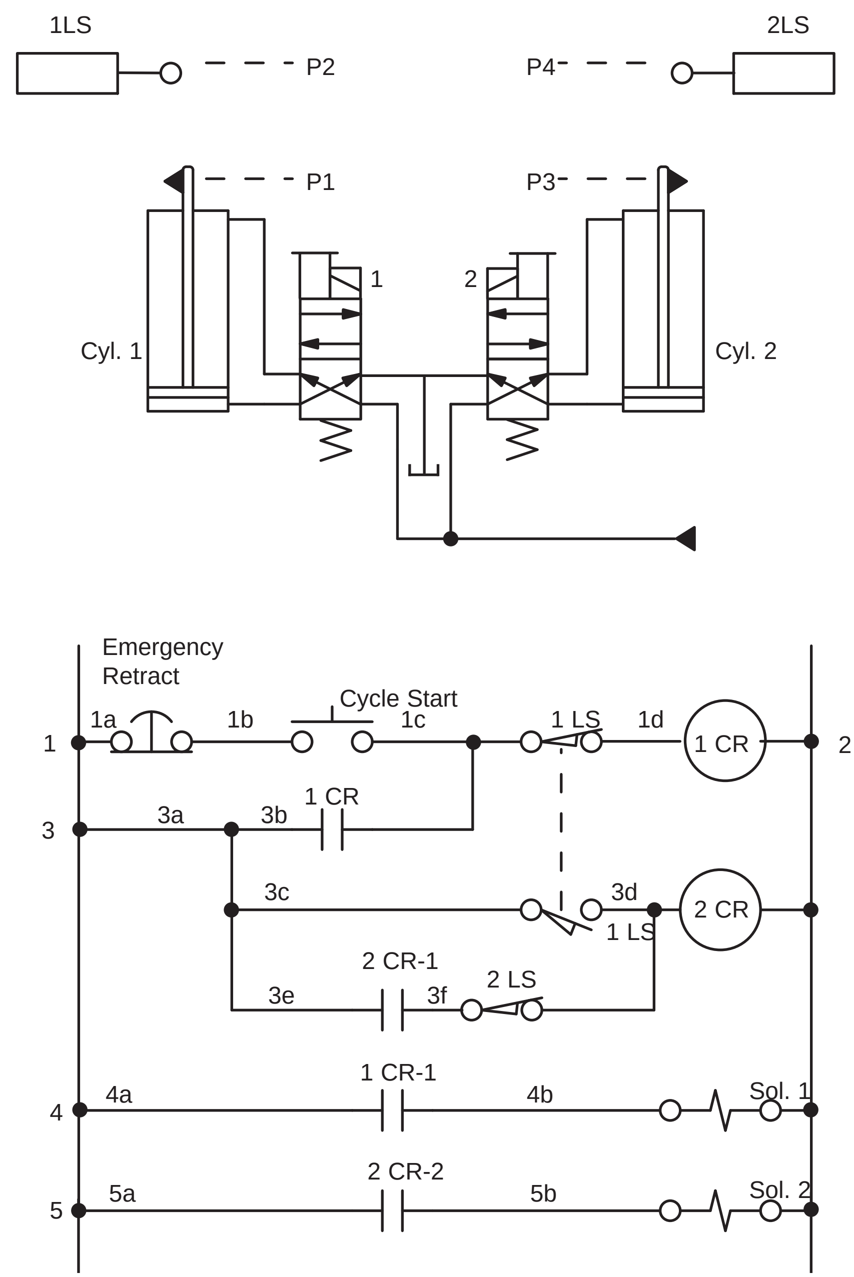


Figure 7-10: Hydraulic Circuit with Electronic Controls

Review 7.3.1.1:

What happens if the Emergency Retract button is pressed when Cylinder 1 rod is fully extended?

a. The circuit stops immediately.

b. Cyl. 1 retracts, then the circuit stops.

c. The circuit completes the normal cycle.

d. Cyl. 1 stops, Cyl. 2 extends, then the circuit stops.

e. Cyl. 1 retracts, Cyl. 2 extends, then the circuit stops.

Answers and Solutions

Answers and Solutions

Review 1.1.1.1: b

Review 1.1.1.2: e

Review 1.1.1.3: e

While the pressure compensated flow control shown by "d" does vary as a function of pressure, the thermometer symbol indicates that it also varies as a function of temperature.

Review 1.1.1.4: c

Review 1.1.1.5: d

Review 1.1.1.6: e

Review 1.1.1.7: a

Review 1.1.1.8: c

Review 1.1.1.9: e

Review 1.2.2.1: a

In a 2:1 area ratio cylinder, the bore diameter is 1.414 times the diameter of the rod.

Review 1.2.2.2: e

After the cylinder stalls, the pressure at the rod end of the cylinder will bleed down to tank across the bleed orifice. Therefore, backpressure will not act on the annular area of the cylinder.

First, solve equation 1-3 for the effective area:

$A = D^2 \times 0.7854 = (2.5 \text{ inches})^2 \times 0.7854 = 4.91$ sq-in

Next, solve equation 1-2 for force:

$F = P \times A = 1500$ psi x 4.91 = <u>7365 lbs</u>

Review 1.2.3.1: b

The relief valve must be set higher than the pressure reducing valve and the two sequence valves in order for the circuit to operate. While the counterbalance valve must be set approximately 150 psi higher than the induced load pressure of the work cylinder in order to hold the cylinder in place, the relief valve would have been set approximately 300 psi than the load pressure in order to provide smooth actuation of the work cylinder as it extends.

Answers and Solutions

Review 2.1.1.1: d

Based on Pascal's Law, F = P x A, only less pressure or less area would reduce force. The rod diameter does not affect extension force. Flow, cylinder stroke, and pump displacement do not affect system pressure.

Review 2.1.1.2: d

Using the conversion factor from Figure 2-1: 1750 psi x 6.895 = <u>12,066.25 psi.</u>

Review 2.1.1.3: e

Using the conversion factor from Figure 2-1: 150 bar x 14.5 = <u>2175 psi.</u>

Review 2.2.2.1: a

First, solving the pulley mechanism indicates that the effective load on the cylinder will be 4000 lbs.

Next, solve equation 1-3 for the effective area:

$A = D^2 \times 0.7854 = (3 \text{ inches})^2 \times 0.7854 = 7.07$ sq-in

Next, solve equation 1-2 for force:

$F = P \times A \leftrightarrow 4000 \text{ lbs} = P \times 7.07 \text{ sq-in} \leftrightarrow \text{Pressure} = \underline{565.88 \text{ psi}}$

Review 2.3.1.1: d

First, solve equation 2-5:

MA = TRL / SRL = 20 inches / 12 inches = 1.67

Next, solve equation 2-6:

RF = MA x SL = 1.67 x 90 lbs = <u>150 lbs</u>

Review 3.1.1.1: a

First, solve equation 1-2 for the effective area:

$F = P \times A \leftrightarrow 8000 \text{ lbs} = 1500 \text{ psi} \times A \leftrightarrow \text{Area} = 5.33$ sq-in

Next, solve equation 1-3 for the bore diameter: $D = (A / 0.7854)^{1/2}$

$D = \sqrt{5.33 / 0.7854} = \underline{2.61 \text{ inches}}$

Answers and Solutions

Review 3.1.2.1: d

First, solve equation 1-3 for the area of the cylinder:

$A = D^2 \times 0.7854 = (3 \text{ inches})^2 \times 0.7854 = 7.07$ sq-in

Next, solve equation 3-1 for force:

$F = C_f \times L = 0.25 \times 50{,}000$ lbs $= 12{,}500$ lbs

Finally, solve equation 1-2 for pressure:

$F = P \times A \leftrightarrow 12{,}500$ lbs $= P \times 7.07$ sq-in $\leftrightarrow$ Pressure = <u>1768 psi</u>

Review 3.1.2.2: c

First, solve equation 3-2:

TIF = $(Cf \times L \times \cos\theta) + (\sin\theta \times L) = (0.35 \times 75{,}000 \times \cos 30°) + (\sin 30° \times 75{,}000)$

TIF = $(0.35 \times 75{,}000 \times 0.866) + (0.50 \times 75{,}000) = 22733 + 37{,}500 = 60{,}233$ lbs

Next, solve equation 1-3 for the area of the cylinder:

$A = D^2 \times 0.7854 = (4 \text{ inches})^2 \times 0.7854 = 12.57$ sq-in

Finally, solve equation 1-2 for pressure:

$F = P \times A \leftrightarrow 60{,}233$ lbs $= P \times 12.57$ sq-in $\leftrightarrow$ Pressure = <u>4792 psi</u>

Review 3.1.3.1: c

First, solve for the opposite side using the Pythagorean Theorem:

$\text{hypotenuse}^2 = \text{adjacent}^2 + \text{opposite}^2$

$(12 \text{ inches})^2 = (1 \text{ inch})^2 + b^2 \leftrightarrow 144$ sq-in $= 1$ sq-in $+ 143$ sq-in

143 sq-in $= b^2 \leftrightarrow b = 11.958$ inches = opposite side (vertical axis)

Next, solve for the tangent:

$\tan\theta = \text{opposite} / \text{adjacent} \leftrightarrow 11.958 \text{ inches} / 1 \text{ inch} = 11.958$

Answers and Solutions

Next, solve equation 3-3 for force:

TF = CF x Tan θ

TF = 5,000 lbs x 11.958 = 59,790 lbs

Review 3.1.3.2: d

First, solve for the opposite side using the Pythagorean Theorem:

hypotenuse^2 = adjacent^2 + opposite^2

(15 inches)^2 = (0.5 inches)^2 + b^2 $\leftrightarrow$ 225 sq-in = 0.25 sq-in + 224.75 sq-in

224.75 sq-in = b^2 $\leftrightarrow$ b = 14.9917 inches = opposite side (vertical axis)

Next, solve for the tangent:

Tan θ = opposite / adjacent $\leftrightarrow$ 14.9917 inches / 0.5 inches = 29.9833

Next, solve equation 3-3 for force:

TF = CF x Tan θ

TF = 7,200 lbs x 29.9833 = 215,880 lbs

Review 3.1.4.1: c

First, solve equation 2-3 for the effort required to support the load:

L x LD = E x ED $\leftrightarrow$ 1500 lbs x 2 = E x 1 $\leftrightarrow$ Effort = 3000 lbs

Next, solve the right triangle problem:

Sin 30° = 0.5 = opposite / hypotenuse = 3000 lbs / hypotenuse $\leftrightarrow$ hypotenuse = 6000 lbs

The vertical force vector of 3000 lbs is proportional to the vector along the hypotenuse of 6000 lbs.

Next, solve equation 1-3 for the area of the cylinder:

A = D^2 x 0.7854 = (3 inches)^2 x 0.7854 = 7.07 sq-in

Finally, solve equation 1-2 for pressure:

F = P x A $\leftrightarrow$ 6000 lbs = P x 7.07 sq-in $\leftrightarrow$ Pressure = 849 psi

Answers and Solutions

Review 3.1.4.2: d

First, use the vertical angle between the load and the beam to compute the load that acts perpendicular to the end of the beam:

Sin 45° = 0.707 ↔ 0.707 x 10,000 lbs = 7072 lbs

Next, using equation 2-5, calculate the mechanical advantage of the load perpendicular to the end of the beam to the effort required perpendicular to the beam at the cylinder rod end pin:

MA = TRL / SRL = 24 feet / 6 feet = 4:1

Based on the mechanical advantage, calculate the load perpendicular to the beam at the rod pin of the cylinder:
7072 lbs x 4 = 28,288 lbs

Then, calculate the load to be supported by the cylinder:

Sin 30° = 0.50 = opposite / hypotenuse = 28,288 lbs / hypotenuse ↔ Hypotenuse = 56,576 lbs

Finally, solve equation 1-2 for the pressure needed in the 5 inch bore cylinder to support the load:

F = P x A ↔ 56,576 lbs = P x 19.64 sq-in ↔ Pressure = <u>2881.4 psi</u>

Review 3.2.1.1: e

First, solve equation 3-4 for each of the two intermediate speed reductions:

1) SR = OS / IS = 40 / 8 = 5:1

2) SR = OS / IS = 28 / 7 = 4:1

Then, multiply the two ratios to determine the final ratio:

5:1 x 4:1 = <u>20:1</u>

Review 3.2.1.2: b

Since the torque specification for the nut is given in lb-ft and the length of the wrench is given in inches, the two values must be reconciled. Either the length of the wrench must be changed to feet or the torque value must be changed to lb-in. Since the equation solves for values in inches and lb-in, change the torque specification to lb-in:

65 lb-in x 12 inches/foot = 780 lb-in

Then solve equation 3-5 for force:

Answers and Solutions

T = F x R ↔ 780 lb-ft = F x 20 inches ↔ Force = <u>39 lbs</u>

Review: 3.2.1.3: d

Solve equation 3-8 for torque in lb-ft:

HP = (T x N) / 5252 ↔ 2.5 hp = (T x 2250 rpm) / 5252 ↔ Torque = <u>5.84 lb-ft</u>

Review 3.2.1.4: d

Solve equation 3-9 for torque:

HMT = (D x P) / 6.28 = (0.66 cipr x 1250 psi) / 6.28 = 131.37 lb-in

The equation solves for torque in lb-in while the available answers are in units of lb-ft. Therefore, the units of torque must be converted:

131.37 lb-in / 12 inches/foot = <u>10.95 lb-ft</u>

Review 3.2.1.5: c

Using Table 3-1, enter the table at 9.0 cipr, read across to the column for 2500 psi, and note the torque value of 3583 lb-in. Next, repeat this process for displacements of 1.0 and 0.5 cipr, adding the torque values to 3583 lb-in for a result of <u>4180 lb-in.</u>

Review 3.2.1.6: c

Using Table 3-2, note that a speed of 1100 rpm is not listed. Therefore, enter the table at 1 hp and read across and note the torque outputs for 1000 and 1200 rpm. Then, average the two values:

(63.0 lb-in + 52.5 lb-in) / 2 = <u>57.75 lb-in</u>

Review 3.3.1.1: d

Solve equation 3-10 for flow:

Q = (D x N) / 231 = (1.69 cipr x 1200 rpm) / 231 = <u>8.78 gpm</u>

Review 3.3.1.2: e

First, solve equation 1-1 for flow:

V = (Q x 231) / A ↔ 540 in/min = (Q x 231) / 12.57 sq-in ↔ Q = 29.4 gpm

Then, solve equation 3-10 for displacement:

Answers and Solutions

$Q = (D \times N) / 231 = 29.4\ \text{gpm} = (D \times 1750\ \text{rpm}) / 231 \leftrightarrow \text{Displacement} = \underline{3.88\ \text{cipr}}$

Review 3.4.1.1: d

Solve equation 3-11 for C_v:

$Q = C_v \sqrt{PSID/Sg} = 20\ \text{gpm} = C_v \sqrt{150\ PSID}/0.90 = C_v \sqrt{166.67 = C}_v \times 12.91 \leftrightarrow C_v = \underline{1.549}$

Review 3.4.1.2: b

Solve equation 3-11 for pressure drop:

$Q = C_v \sqrt{PSID/Sg} = 20\ \text{gpm} = 2.85 \sqrt{PSID/0.92}\ \ 7.018 = \sqrt{PSID/0.92}$
$49.246 = PSID / 0.92 \leftrightarrow 49.246 \times 0.92 = PSID = \underline{45.31\ \text{psid}}$

Review 3.4.2.1: a

First, solve equation 1-3 for the area of the poppet bore, A_F:

$A = D^2 \times 0.7854 = (0.5\ \text{inches})^2 \times 0.7854 = 0.1964\ \text{sq-in}$

Since the pressure to open the valve is applied at port A_A, and the spring has to oppose the pressure applied on the area of A_A solve for the area of A_A:

$A_F = 1.6 \times A_A = A_F / 1.6 \leftrightarrow A_A = 0.1227\ \text{sq-in}$

Next, solve equation 3-12 for pressure:

$SF = P \times A \leftrightarrow 7\ \text{lbs} = P \times 0.1227\ \text{sq-in} \leftrightarrow \text{Pressure} = \underline{57\ \text{psi}}$

Review 3.4.2.2: b

Equation 3-13 states:

$(P_A \times A_A) + (P_B \times A_B) = (P_F \times A_F) + F_S$, where $A_A = 1$, $A_B = 0.6$ and, $A_F = 1.6$.

Reconfiguring equation 3-13:

$P_F = \{(P_A \times A_A) + (P_B \times A_B) - F_S\} / A_F$

Since $A_F = 1.6$:

$\underline{P_F = \{(P_A \times A_A) + (P_B \times A_B) - F_S\} / 1.6}$

Answers and Solutions

Review 3.4.2.3: d

Review 3.4.3.1: c

Since the force available from a force solenoid is proportional to the current over the range of operation, the force at 300 milliamps is:

(300 ma / 800 ma) x 14 lbs = <u>5.25 lbs</u>
Review 3.4.4.1: c

Review 3.4.5.1: a

The time to achieve 0 to 100% travel is 70 milliseconds (ms). Added to that figure is the dwell time of 3 seconds, plus the closing time of 50 ms. yielding a total cycle time of 3.12 seconds (70 ms + 3 sec + 50 ms = 3.12 sec).

Dividing this length of time into 60 seconds results in a maximum cycle rate of <u>19.23 cpm</u> (60 sec/min / 3.12 sec/cycle = 19.23 cycles/minute).

Review 3.4.6.1: b

Review 3.5.1.1: c

Solve equation 3-14 for volume:

V = (L x W x H x %) / 231 = (30 inches x 15 inches x 15 inches x 0.80) /231 = <u>23.4 gallons</u>

Review 3.5.2.1: a

First, solve for the vertical area:

A = (2 sides x 4 feet x 2 feet) + (2 ends x 2 feet x 2 feet) = 16 sq-ft + 8 sq-ft = 24 sq-ft

Next, since the hot fluid will not transfer much heat to the tank walls above the fluid level, adjust the surface are by the fluid level in the reservoir:

24 sq-ft x 0.75 = 18 sq-ft

Then, solve equation 3-15 for hp:

HP = 0.001 x ΔT x A = 0.001 x 100°F x 18 sq-ft = 1.8 hp

Finally, convert hp to Btu/hr:

1.8 hp x 2545 Btu/hr/hp = <u>4581 Btu/hr</u>

Answers and Solutions

Review 3.6.1.1: b

The total capacity of a one gallon accumulator is 231 cu-in, which is equal to V1. Before the system is started, this entire volume is filled with gas at the 1000 psi precharge pressure. After the system is started and the hydraulic pressure rises above 1000 psi, hydraulic fluid will begin to enter the accumulator. The difference in the gas volume between 1000 psi and 1750 psi is equal to the volume of hydraulic fluid that has entered the accumulator. Since the temperature has increased as the accumulator filled, the gas will take up a greater volume, proportional to the temperature rise. Gas law calculations require the use of absolute pressure and temperature values. The absolute scale for Fahrenheit is Rankin.

First, convert the pressures and temperatures to absolute values:

1) 1000 psig + 14.7 = 1014.7 psia

2) 1750 psig + 14.7 = 1764.7 psia

3) 80°F + 460 = 540°R

4) 150°F + 460 = 610°R

Next, substitute the appropriate values into equation 3-16 and solve for V2:

$P_1 \times V_1 \times T_2 = P_2 \times V_2 \times T_1$

1014.7 psia x 231 cu-in x 610°R = 1764.7 psia x V2 x 540°R ↔ 150.04 cu-in

Finally, since 150 cu-in is the volume of the precharge gas at 1750 psig, subtract this value from 231 to determine the volume of the hydraulic fluid:

231 cu-in - 150 cu-in = <u>81 cu-in</u>

Review 3.6.1.2: a

Since the process is isothermal, the effect of temperature changes can be ignored.

The gas capacity of the empty accumulator is 462 cu-in (2 gallons x 231 cu-in/gal = 462 cu-in). The gas volume will remain at 462 cu-in until a system pressure of 1000 psig is exceeded.

First, using equation 3-17, calculate the difference in gas volume between 1000 psig and 2000 psig:

$P_1 \times V_1 = P_3 \times V_3$ = 1014.7 psia x 462 cu-in = 2014.7 psia x V_3 ↔ 232.685 cu-in

Next, calculate the difference in gas volume between 1000 psig and 3000 psig:

$P_1 \times V_1 = P_2 \times V_2$ = 1014.7 psia x 462 cu-in = 3014.7 psia x V_2 ↔ 155.502 cu-in

Finally, subtract the gas volume at 3000 psig from the gas volume at 2000 psig:

232.685 cu-in - 155.502 cu-in = <u>77.183 cu-in</u>

Review 3.7.1.1: a

This is an area ratio problem. Pressure times area of the cylinder must equal pressure times area of the intensifier. But first, it is necessary to calculate the pressure that must be applied to the10 inch bore cylinder in order to achieve an output force of 100 tons:

2000 lbs/ton x 100 tons = 200,000 lbs of force

Next, calculate the pressure required to achieve this force:

$F = P \times A = 200{,}000 \text{ lbs} = P \times Area_{10\ inches} = P \times 78.54 \text{ sq-in} \leftrightarrow \text{Pressure} = 2546.5 \text{ psi}$

This is the pressure that must be developed by the 0.50 inch bore intensifier piston. Next, solve for the pressure that must be applied to 1.5 inch diameter intensifier input piston:

$2547 \text{ psi} \times Area_{0.50\ inches} = P \times Area_{1.5\ inches} = 2547 \text{ psi} \times 0.20 \text{ sq-in} = P \times 1.77 \text{ sq-in}$

P = <u>287.8 psi</u>

Review 3.8.1.1: c

First, solve equation 3-19 for fluid horsepower in order to determine the amount of horsepower being delivered to the motor:

HP = (Q x P) / 1714 = (10 gpm x 1500 psi) 1714 = 8.75 hp

Next, solve equation 3-18 for the energy loss:

Energy LossBtu/hr = 2545 x Thours x (HPinput - HPoutput)

EL = 2545 x 1 x (8.75 hp in - 7 hp out) = <u>4453.75 Btu/hr</u>

Review 3.9.1.1: b

First, calculate the ID of tubing and then solve equation 1-3 for the area of the ID:

1 inch - (2 x -.065 inch wall) = 1 inch - 0.130 inches = 0.870 inches

$A = D^2 \times 0.7854 = (0.870 \text{ inches})^2 \times 0.7854 = 0.594 \text{ sq-in}$

Next, solve equation 3-21 for velocity:

Answers and Solutions

V = (Q x 0.3208) / A = (20 gpm x 0.3208) / 0.594 sq-in = <u>10.8 ft/sec</u>

Review 3.9.1.2: d

Line E must handle the intensified return flow developed by the cylinder. To determine this flow rate, use equation 1-3 to calculate the area ratio of the cylinder:

A = D_2 x 0.7854

A_{piston} = (4 inches)^2 x 0.7854 = 12.57 sq-in

A_{rod} = (3 inches)^2 x 0.7854 = 7.07 sq-in

$A_{annulus}$ = A_{piston} - A_{rod} = 12.57 sq-in - 7.07 sq-in = 5.5 sq-in

12.57 sq-in / 5.5 sq-in = 2.285:1 ratio

Next, multiply the flow rate of the pump by the area ratio:

10 gpm x 2.285 = 22.85 gpm

Then, solve equation 3-21 for area:

V = (Q x 0.3208) / A ↔ 15 ft/sec = (22.85 gpm x 0.3208) / A ↔ Area = 0.489 sq-in

Then solve equation 1-3 for diameter:

A = D^2 x 0.7854 = 0.489 sq-in = D^2 x 0.7854 ↔ Diameter = 0.796 inches

Add two times the wall thickness (2 x 0.065 inches = 0.130 inches) to the diameter to determine the minimum tube outside diameter:

0.798 inches + 0.130 inches = <u>0.926 inches</u>

Review 3.9.2.1: c

First, solve equation 3-23 for burst pressure:

BP = (2 x WT x TS) / OD = (2 x 0.049 inches x 40,000 psi) / 0.75 inches = 5226.67 psi

Then, solve equation 3-23 for working pressure:

BP = WP x SF ↔ 5227 psi = WP x 4 ↔ Working Pressure = <u>1306.75 psi</u>

Answers and Solutions

Solution to 4.1.1.1: Ans. e

Solving equation 1-2 indicates that cylinder 1 can support a load of 707 lbs:

$F = P \times A \leftrightarrow 100$ psi x 7.07 sq-in = 707 lbs

Deducting the 100 lb load supported by the rod of cylinder 1, cylinder 1 can support another 607 lbs of load. The load acting on cylinder 2 is supported by the annular area of cylinder 1. The annular area of cylinder 1 is 3.93 sq-in:

($A_p - A_r = A_a \leftrightarrow A_{3\ inches} - A_{2\ inches} = A_a \leftrightarrow$ 7.07 sq-in - 3.14 sq-in = 3.93 sq-in)

Solve equation 1-2 for the pressure acting on the annular area of cylinder 1 as it supports 607 lbs:

$F = P \times A \leftrightarrow$ 607 lbs = P x 3.93 sq-in $\leftrightarrow$ Pressure = 154.5 psi

Finally, solve for the load cylinder 2 can support as 154.5 psi acts on its piston:

F = P x A = 154.5 psi x 7.07 sq-in = <u>1092 lbs</u>

Review 4.1.2.1: a

Solve equation 4-1 for the motor speed:

MN = PN x (PD / MD) = 1200 rmp x (2 cipr / 6 cipr) = <u>400 rpm</u>

Review 4.2.1.1: d

Review 5.1.1.1: a

Review 5.2.1.1: d

After solving for the individual variables, solve equation 5-1 for final squeeze:

IS = 0.300 inches x 0.15 = 0.045 inches

S = 0.300 inches x 0.20 = 0.060 inches

CS = 0.300 inches x 0.10 = 0.030 inches

FS = IS + S - CS = 0.045 inches + 0.060 inches - 0.030 inches = <u>0.075 inches</u>

Review 5.3.1.1: a

Solve equation 5-1 for the Beta Ratio:

Answers and Solutions

Beta Ratio = ß = (# of particles introduced / # of particles passed) = 100 / 50 = 2

Review 5.3.1.2: d

First, solve equation 5-1 for the number of particles passed:

Beta Ratio = (# of particles introduced / # of particles passed) = 100 / pp = 12 $\leftrightarrow$ 100 = 12 pp

$\leftrightarrow$ pp = 100 / 12 = 8.33 particles passed

Knowing that 8.33 particles passed though the filter, solve equation 5-2 for efficiency:

100 particles introduced minus 8.33 particles passed = 91.67 particles removed.

Efficiency = (# of particles removed / # of particles introduced) x 100 = (91.67 / 100) x 100 = 91.67% efficiency

Review 5.4.1.1: c

Review 6.1.1.1: b

Review 6.1.1.2: c

Review 6.1.1.3: d

In order for the cylinder to cycle at a rate of 10 cpm, a cycle could not last longer than six seconds (60 sec/min / 10 cpm = 6 sec).

First, calculate the volume required to extend the cylinder:

$\text{Volume}_{\text{cu-in}} = \text{Area}_{\text{sq-in}} \times \text{Stroke}_{\text{in}}$ = 12.57 sq-in x 20 inches = 251.40 cu-in

Next, calculate the volume required to retract the cylinder:

V = A x S = (12.57 sq-in - 3.14 sq-in) x 20 inches = 9.43 sq-in x 20 inches = 188.60 cu-in

Third, sum those two volumes: 251.40 cu-in + 188.6 cu-in = 440 cu-in

Then, multiply the volume per cycle times 10 cpm:

440 cu-in/cycle x 10 cpm = 4400 cu-in/min

Finally, divide the flow per minute by 231 cu-in/gal for the flow rate in gpm:

4400 cu-in/min / 231 cu-in/gal = 19.05 gpm

Answers and Solutions

Review 6.2.1.1: d

Review 6.2.2.1: d

Review 6.2.2.2: c

Review 6.3.1.1: a

Review 6.3.2.1: e

Review 6.3.2.2: b

Review 6.3.2.3: a

The 3 inch bore cylinder must extend against a load of 10,000 lbs. To do so requires 1414 psi (F = P x A = 10,000 lbs = 7.07 sq-in x 1414 psi).

Solve equation 3-9 for pressure:

HMT = (D x P) / 6.28 ↔ 2000 lb-in = (20 cipr x P) / 6.28 ↔ Pressure = 628 psi

The motor will begin to operate when 628 psi is reached, which is less than the 1414 psi required for the cylinder to move. If sequence valve A was set above 1414 psi, the cylinder would extend. After the cylinder stalls and pressure rises, sequence valve A would open providing flow to the hydraulic motor. Pressure will be greater than the 628 psi required to rotate the load.

Review 7.1.1.1: d

Review 7.2.1.1: d

Review 7.3.1.1: c

Notice that leg 4 of the electrical circuit connects to the 2CR relay independent of the cycle start switch. When cylinder 1 is fully extended, the circuit to 1CR is opened and 2CR is closed. What happens is that cylinder 1 retracts while cylinder 2 extends. At the end of the stroke, cylinder 2 rod will open limit switch 2LS opening the circuit to 2CR and cylinder 2 will retract. Thus, the circuit will complete the normal cycle and then stop.

Hydraulic Specialist Certification Pretests

Introduction

Pretests are used to evaluate candidate preparedness for certification tests. Pretests may be either taken individually or in a group setting such as during a Review Training Session (RTS). As a part of an RTS, pretests are used to allow the instructor to tailor the subject matter coverage to the needs of the audience. When a candidate is studying individually or in a small group, pretests provide insight into which areas require further study and whether the candidate should consider other study options, such as an RTS.

Included in this manual are four separate pretests for the Hydraulic Specialist certification test. Each pretest has its own separate answer sheet which appears at the end of the pretests. Individual pretests are numbered HS-1, HS-2, HS-3, and HS-4. The answer key for all four pretests appears at the end of the manual.

Candidates are encouraged to take a pretest early in the study process. Pretests should be taken under timed conditions. A maximum of forty-five minutes should be allotted for each pretest. This should be sufficient time to answer all sixteen questions on the pre-test. The results of the pretest will guide the candidate to one of four possible courses of action regarding test preparation.

1. Take the test: Preparation is sufficient.
2. Study the material using the Study Manual.
3. Attend a Review Training Session (RTS): Preparation is good, but not sufficient to pass the test.
4. Participate in a formal (general) course: A Review Training Session would not provide adequate preparation to pass the test.

Additional pre-tests should be taken after individual study or attendance at an RTS to further evaluate test readiness. In some instances, it may be desirable to take all four pretests at different times during the study process to better access preparedness and effectiveness of study.

The answer sheets provided have been developed such that each question is referenced to a particular subject matter area of the study manual and of the test. The candidate is encouraged to fold the answer sheet vertically along the dotted line before taking the pre-test. This will eliminate any bias that may occur by having the appropriate outcome statement appear with the answers and more closely mimics actual test conditions. After checking the answers, the answer sheet may be opened to reveal the areas where further study is needed. This should enable directed study in the areas where a deficiency exists.

Candidates should be advised that each pre-test covers only a representative sample of the types of questions found on the test. Due to the need to keep the pretest brief, not all subject matter is covered on every pretest. Thorough preparation for the certification test is strongly encouraged.

The experience of taking pretests under timed conditions should reduce test anxiety associated with the actual certification test. If necessary, candidates may wish to retake the pretests after some period of time has elapsed to recheck their knowledge.

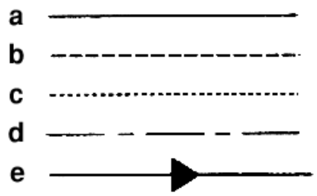
Suggestions or comments for improvements of these pre-tests and other certification materials should be sent to:

Fluid Power Society Education Institute
c/o FPS
3245 Freemansburg Avenue
Palmer, PA 18045
Phone: 610-923-0386, Fax: 610-923-0389
Web: www.ifps.org • E-Mail: askus@ifps.org

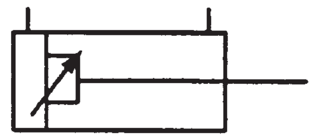
Hydraulic Specialist Certification Pretest HS-1

1: Which of the following symbols illustrates a pilot pressure line?

a. a
b. b
c. c
d. d
e. e

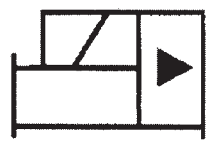


2: How is the cylinder shown by the graphic symbol equipped?



a. with a servo positioner
b. with a fixed cushioning device
c. with a cushioning device retracting
d. with an adjustable cushioning device extending
e. with an adjustable cushioning device extending and retracting

3: Based on the directional control valve symbol shown, in order for the valve to be actuated, the valve must receive:



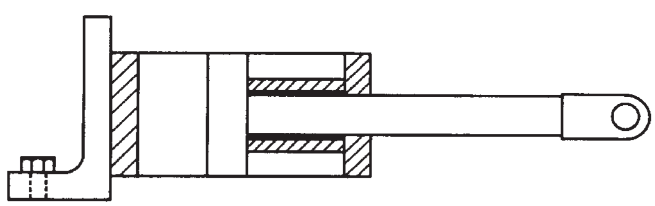
a. a pilot signal or manual actuation
b. solenoid actuation and manual actuation
c. solenoid actuation or a pilot signal, and manual actuation
d. solenoid actuation and a pilot signal, and manual actuation
e. solenoid actuation and a pilot signal, or manual actuation and a pilot signal

4: Which one of the following would be reduced to lower the extension force of a hydraulic cylinder?

a. Flow
b. Rod diameter
c. Cylinder stroke
d. System pressure
e. Displacement of the pump

5: If a hydraulic cylinder, similar to the one shown in the figure, has a 20 inch stroke, is extended 8 inches, and has a 90 lb side load acting against the end of the rod, what would be the reaction force acting against the rod bushing?

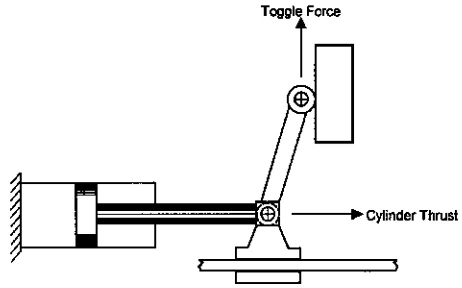
a. 36 lbs
b. 54 lbs
c. 90 lbs
d. 150 lbs
e. 225 lbs



Hydraulic Specialist Certification Pretest HS-1

6: If a cylinder rod exerts 5000 lb of force extending, how much force would a single lever toggle exert when the 12 inch lever arm (h) is 1 inch (a) from the vertical axis (o)?

a. 8000 lbs
b. 45312 lbs
c. 59790 lbs
d. 60210 lbs
e. 62312 lbs



7: A hydraulic motor drives a shaft through a double reduction chain drive. The hydraulic motor has a sprocket with 8 teeth driving an intermediate shaft with 40 teeth. The second sprocket on the intermediate shaft has 7 teeth connected to a final drive sprocket with 28 teeth. What is the final speed ratio between the hydraulic motor and final drive sprocket?

a. 4 :1
b. 5 :1
c. 9 :1
d. 15 :1
e. 20 :1

8: Using Table 3-2, what would be the expected torque from hydraulic motor with a displacement of 10.5 cipr at 2500 psi?

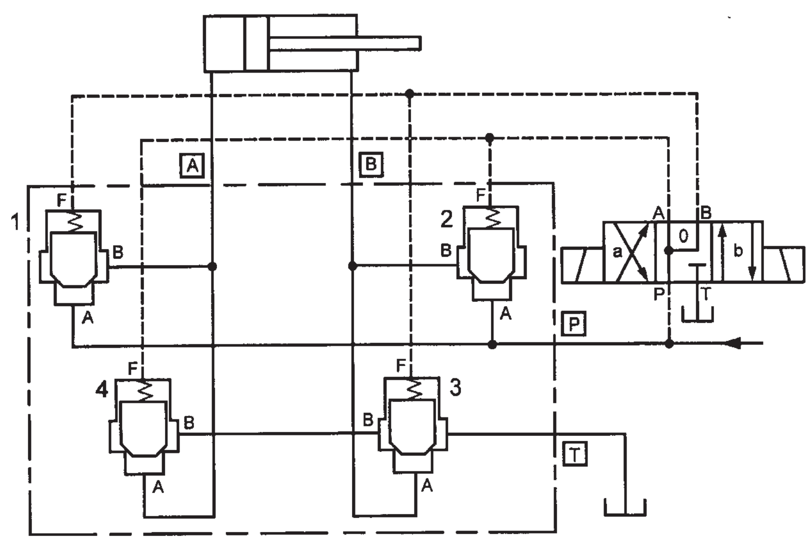
a. 3583 lb-in
b. 3981 lb-in
c. 4180 lb-in
d. 4379 lb-in
e. 4777 lb-in

9: The pressure drop across a directional control valve flowing 20 gpm is 150 psid. If the fluid has an Sg of 0.90 at 140° F, what is the flow coefficient for the valve?

a. 1.101
b. 1.286
c. 1.443
d. 1.549
e. 1.728

10: What would occur if an over-running load were applied to extend and retract the cylinder rod when the directional control pilot valve in the figure below is in the center position? Assume the induced pressure of the over-running load exceeds the pressure at the P port.

a. The cylinder rod will drift in when the pilot valve is in the center position.
b. The cylinder rod will drift out when the pilot valve is in the center position.
c. The cylinder rod is free to float because the pilot valve has a float center.
d. Pilot pressure to all spring chambers locks the cylinder in position.
e. Valve springs in the cartridge valves lock the cylinder rod in place.

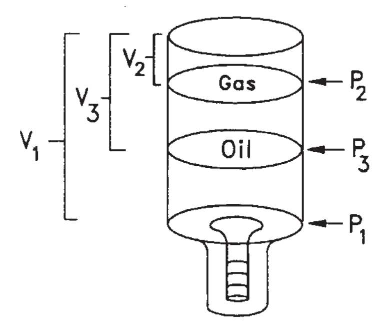


11: For the proportional valve amplifier card circuit shown in Figure 3-20, which setting would be adjusted to increase the power signal?

a. p1
b. p2
c. p3
d. p4
e. p5

12: A 2 gallon accumulator supplies fluid to a hydraulic system between 3000 psi and 2000 psi. If the precharge pressure is 1000 psi, how many cubic inches of hydraulic fluid is available from the accumulator if the process is isothermal as the accumulator fills?

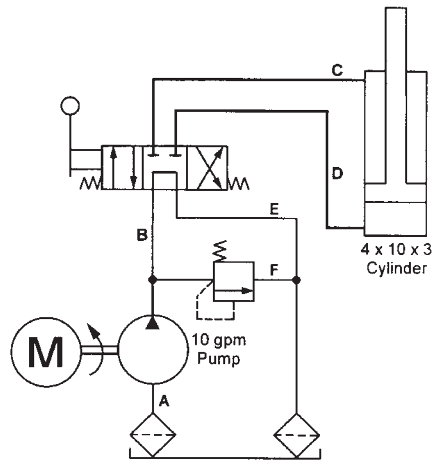
a. 77.2 cu-in
b. 81.4 cu-in
c. 155.5 cu-in
d. 180.6 cu-in
e. 232.7 cu-in



Hydraulic Specialist Certification Pretest HS-1

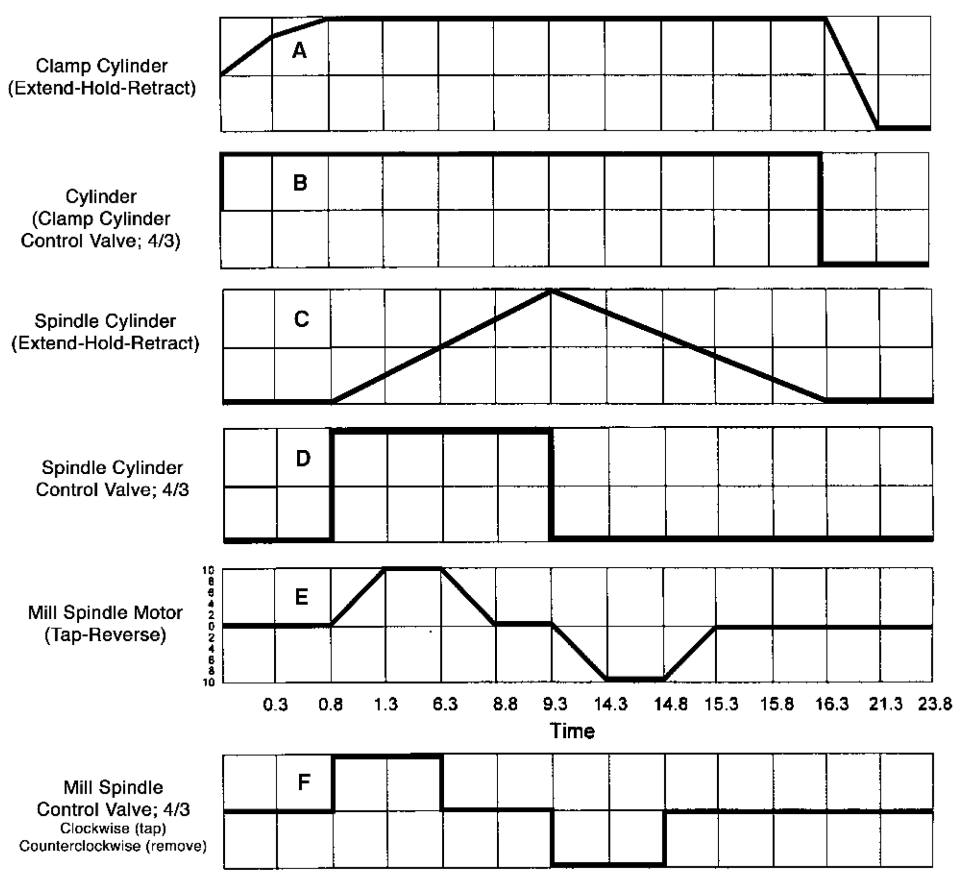
13: The circuit shown in the figure directs fluid from a 10 gpm pump to cycle a double acting cylinder. If the wall thickness is 0.065 inches, what minimum inch size tube would be required to prevent the average fluid velocity from exceeding 15 ft/sec in line E?

a. -8
b. -10
c. -12
d. -16
e. -20



14: In the figure below how long is the work piece clamped?

a. 13.5 seconds
b. 14.0 seconds
c. 14.5 seconds
d. 15.5 seconds
e. 16.3 seconds



Hydraulic Specialist Certification Pretest HS-1

15: Assume that a hydraulic fluid being analyzed for 5 micron sized particles contains 100 particles of contaminant upstream of the filter and 50 particles of contaminant downstream after passing through the filter. What is the Beta Ratio of the filter?

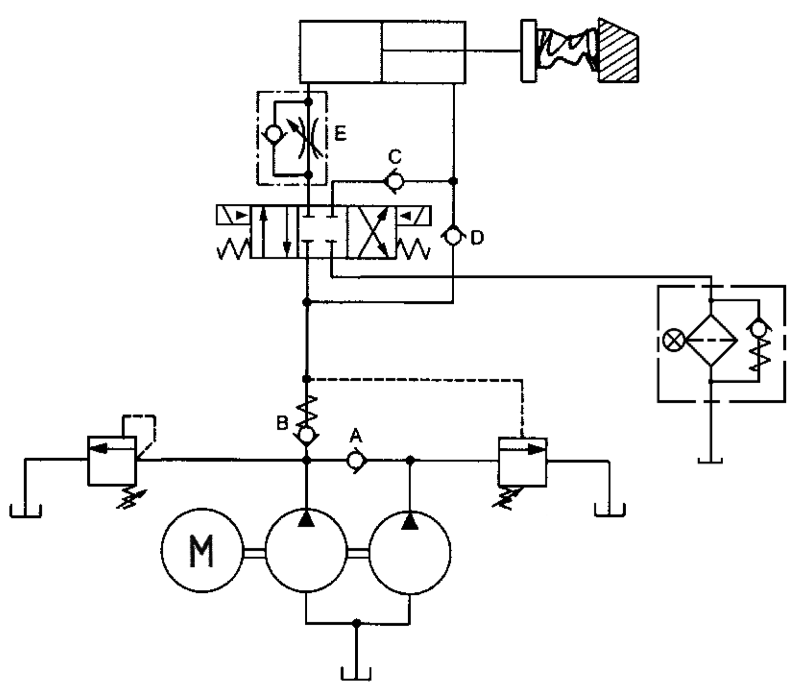
a. $ß_5 = 2$
b. $ß_5 = 4$
c. $ß_5 = 5$
d. $ß_5 = 10$
e. $ß_5 = 50$

16: Which one of the following calculations would determine if cylinder pressure is sufficient to lift a load resistance?

a. $N_{rpm} = (Q_{gpm} \text{ x } 231 \text{ cu-in/gal}) / \text{Displacement}_{cipr}$
b. $HP_{input} = (P \text{ x } Q)\ 1714$
c. $\text{Force}_{lbs} = \text{Pressure}_{psi} \text{ x } \text{Area}_{sq\text{-}in}$
d. $\text{Velocity}_{in/min} = (Q_{gpm} \text{ x } 231 \text{ cu-in/gal}) / \text{Area}_{sq\text{-}in}$
e. $\text{Hydraulic Motor Torque}_{lb\text{-}in} = \text{Pressure}_{psig} \text{ x } \text{Displacement}_{cipr} / 2\pi$

17: In the figure below what would happen if check valve A sticks open?

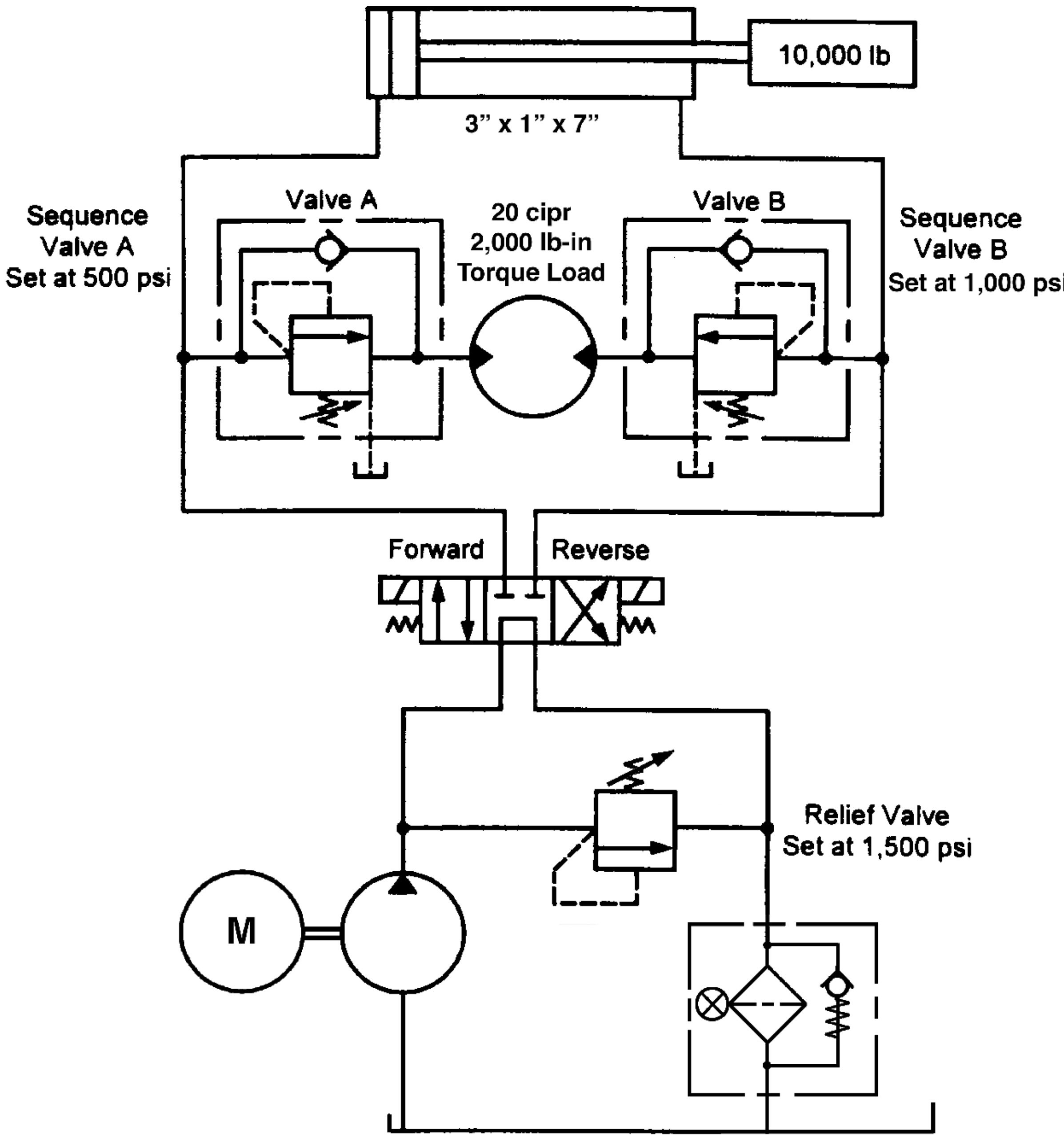
a. Both pumps would stall.
b. The accumulator would overfill.
c. Both pumps would unload at low pressure
d. The cylinder rod velocity would increase retracting.
e. The circuit would become inoperative.



Hydraulic Specialist Certification Pretest HS-1

18: Neglecting friction, why will the cylinder in the figure below fail to extend?

a. The pressure setting of valve A is too low.
b. The pressure setting of valve B is too high.
c. The load on the motor is too high.
d. The pressure relief valve is set too low.
e. The cylinder regenerates through the motor.



Hydraulic Specialist Certification Pretest HS-1

Pretest A Answer Sheet

Apply Hydraulic Circuits to Perform Desired Tasks.

Outcome:		Page	
1.1.1:	Recognize basic hydraulic symbols.	1 - 2	1. a b c d e
			2. a b c d e
			3. a b c d e

Analyze Loads and Motion.

2.1.1:	Recognize factors associated with hydraulic power systems.	2 - 2	4. a b c d e
2.3.1:	Understand the principles of levers.	2 - 12	5. a b c d e

Select Components for Hydraulic Applications.

3.1.3:	Compute the thrust for a toggle mechanism.	3 - 6	6. a b c d e
3.2.1:	Solve formulas for torque, speed and horsepower of hydraulic motors.	3 - 12	7. a b c d e
			8. a b c d e
3.4.1:	Calculate the flow coefficient for directional control valves.	3 - 18	9. a b c d e
3.4.2:	Recognize the characteristics of DIN valves.	3 - 19	10. a b c d e
3.4.6:	Identify compenents on the amplifier card.	3 - 30	11. a b c d e
3.6.1:	Size an accumulator using gas laws.	3 - 33	12. a b c d e
3.9.1:	Size fluid conductors.	3 - 38	13. a b c d e

Prepare Bills of Materials and Schematics.

4.2.1:	Recognize machine operations described by the Time Cycle Chart.	4 - 4	14. a b c d e

Recommend Fluid, Fluid Conductors and Fluid Filtration.

5.3.1:	Match filter specifications to a machine.	5 - 7	15. a b c d e

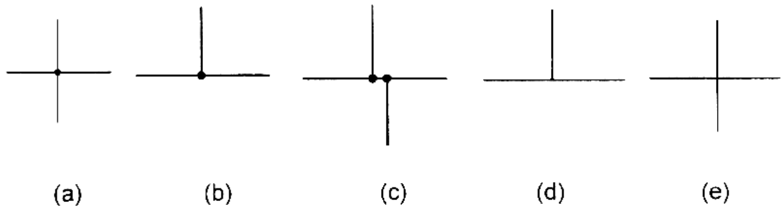
Analyze and Troubleshoot Hydraulic Systems.

6.1.1:	Identify troubleshooting parameters.	6 - 2	16. a b c d e
6.2.2:	Distinguish between correct and incorrect compenent connections.	6 - 8	17. a b c d e
6.3.2:	Predict circuit malfunctions.	6 - 13	18. a b c d e

Hydraulic Specialist Certification Pretest HS-2

1: Per ISO 1219-1, which one of the following symbols represents one hydraulic line crossing another hydraulic line?

a. a
b. b
c. c
d. d
e. e



2: Which circuit would require an unloading valve?

a. boom circuit
b. press circuit
c. motor circuit
d. sequence circuit
e. high-low pump circuit

3: In a full-time regenerative cylinder circuit, what approximate bore to rod diameter ratio would be required to extend and retract the cylinder with the same velocity?

a. 1.414 to 1.000
b. 1.375 to 2.125
c. 2.500 to 1.750
d. 3.250 to 1.750
e. 4.000 to 1.375

4: A pressure gauge indicates 1750 psi. What is the approximate pressure in kPa?

a. 12 kPa
b. 120 kPa
c. 1200 kPa
d. 12000 kPa
e. 120000 kPa

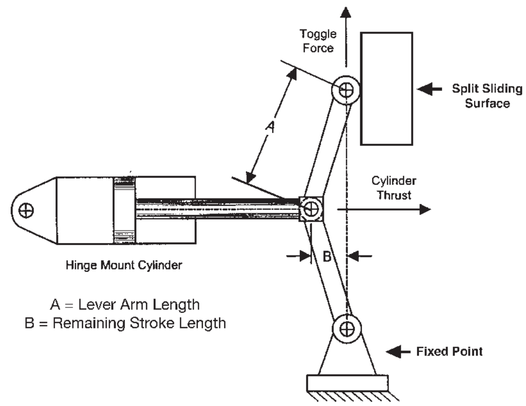
5: What minimum bore diameter cylinder is required to lift a load of 8000 lbs on the extension stroke at an operating pressure of 1500 psi?

a. 2.61 inches
b. 5.33 inches
c. 6.78 inches
d. 8.42 inches
e. 9.10 inches

Hydraulic Specialist Certification Pretest HS-2

6: A toggle mechanism with two 15 inch lever arms, such as the mechanism shown in the figure below is closed by a hydraulic cylinder exerting 7200 lbs of force. What would be the toggle force when the position of the cylinder rod eye is 1/2 inch from the vertical?

a. 107,880 lbs
b. 107,935 lbs
c. 215,750 lbs
d. 215,870 lbs
e. 216,000 lbs



7: The torque specification for a nut 65 is lb-ft. If the torque wrench handle is 20 inches long, how much force must the technician apply at the end of the handle to tighten the nut?

a. 26 lbs
b. 39 lbs
c. 45 lbs
d. 52 lbs
e. 65 lbs

8: Using Table 3-1 (on page 3-16), what would be the theoretical output torque from a 1 HP hydraulic motor rotating at 1100 rpm?

a. 52.50 lb-in
b. 55.30 lb-in
c. 57.75 lb-in
d. 59.75 lb-in
e. 63.00 lb-in

9: The literature from a manufacturer indicates a control valve has a C_v of 2.85 at a flow rate of 20 gpm. If the fluid has a Sg of 0.92 at 170° F, what would be the pressure drop across the valve?

a. 8.21 psid
b. 45.31 psid
c. 52.44 psid
d. 61.96 psid
e. 142.60 psid

10: If a force solenoid that actuates a proportional valve exerts a maximum force of 14 lbs at 800 milliamps, approximately how much force will the solenoid exert at 300 milliamps?

a. 3.25 lbs
b. 4.50 lbs
c. 5.25 lbs
d. 6.00 lbs
e. 7.50 lbs

11: The inside dimensions of a hydraulic reservoir are 30 inches long x 15 inches wide x 15 inches high. How much fluid would the reservoir hold if 20% of the volume above the fluid is left for expansion and contraction of the fluid?

a. 5.8 gallons
b. 20.8 gallons
c. 23.4 gallons
d. 29.2 gallons
e. 36.5 gallons

12: A 10 inch bore hydraulic cylinder operates a press and develops a force of 100 tons. The fixed displacement hydraulic pump advances the cylinder. At 200 psi a sequence valve opens to provide flow to a hydraulically powered intensifier which boosts the pressure to the cylinder. The bore size of the intensifier output piston is 0.50 inches and the bore size of the intensifier's input piston is 1.5 inches. At what pressure should the air pressure regulator be set in order to achieve the correct output pressure from the intensifier?

a. 288 psi
b. 636 psi
c. 719 psi
d. 1273 psi
e. 2547 psi

13: Using a safety factor of 4:1, determine the working pressure of a of a -12 x 0.049" wall hydraulic tube if the tensile strength is 40,000 psi.

a. 871 psi
b. 1287 psi
c. 1307 psi
d. 5227 psi
e. 6667 psi

Hydraulic Specialist Certification Pretest HS-2

14: What is the most important operational property of a hydraulic fluid?

a. viscosity
b. flash point
c. specific gravity
d. pour point
e. neutralization number

15: What is the efficiency of a filter with a Beta Ratio of $ß_{10} = 12$?

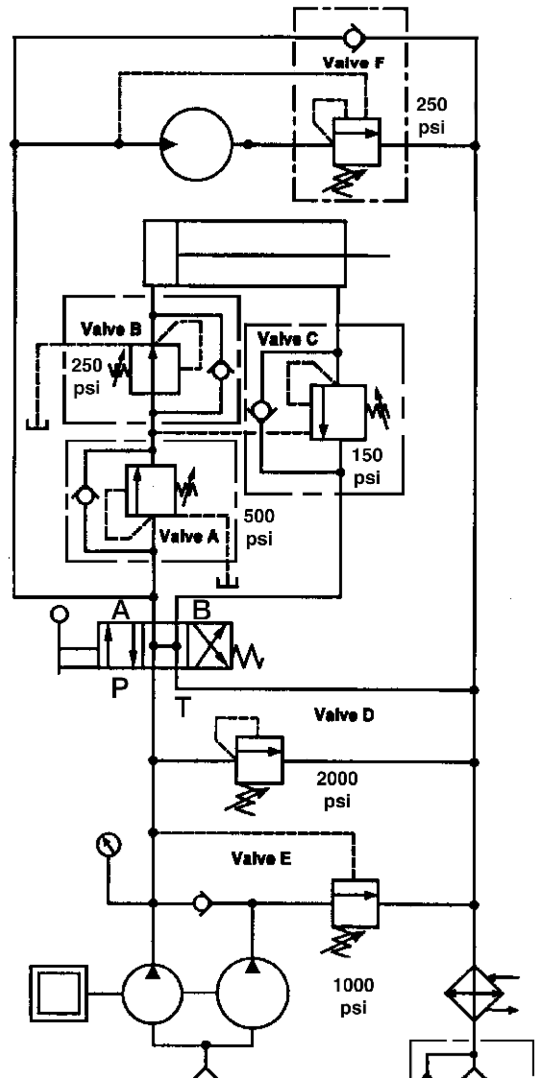
a. 76.8%
b. 88.2%
c. 89.5%
d. 91.7%
e. 92.7%

16: What gpm flow rate would be required to power a hydraulic cylinder with a 4 inch bore x 2 inch rod x 20 inch stroke operating at a rate of 10 cycles per minute (cpm)? There are no dwell times nor any acceleration or deceleration profiles at the ends of stroke.

a. 8.16 gpm
b. 10.88 gpm
c. 16.32 gpm
d. 19.04 gpm
e. 21.77 gpm

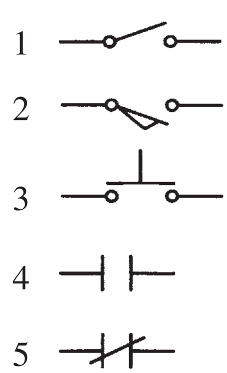
17: In the figure below when the motor is stalled and cylinder is extending, which of the following pressure valves would shift first?

a. A
b. B
c. C
d. D
e. E



18: Which one of the following symbols illustrates normally open relay contacts?

a. 1
b. 2
c. 3
d. 4
e. 5



Hydraulic Specialist Certification Pretest HS-2

Pretest B Answer Sheet

Apply Hydraulic Circuits to Perform Desired Tasks.

Outcome:		Page	
1.1.1:	Recognize basic hydraulic symbols.	1 - 2	1. a b c d e
			2. a b c d e
1.2.2:	Recognize the function of components in regenerative circuits.	1 - 31	3. a b c d e

Analyze Loads and Motion.

2.1.1:	Recognize factors associated with hydraulic power systems.	2 - 2	4. a b c d e

Select Components for Hydraulic Applications.

3.1.1:	Size cylinders.	3 - 2	5. a b c d e
3.1.3:	Compute the thrust for a toggle mechanism.	3 - 6	6. a b c d e
3.2.1:	Solve formulas for torque, speed and horsepower of hydraulic motors.	3 - 12	7. a b c d e
			8. a b c d e
3.4.1:	Calculate the flow coefficient for directional control valves.	3 - 18	9. a b c d e
3.4.3:	Recognize the characteristics of proportional control valves.	3 - 24	10. a b c d e
3.5.1:	Size hydraulic reservoirs from volume requirements.	3 - 31	11. a b c d e
3.7.1:	Calculate pressure intensification from piston sizes.	3 - 35	12. a b c d e
3.9.2:	Use Barlow's Formula to calculate the wall thickness and safety factor for tubing.	3 - 40	13. a b c d e

Recommend Fluid, Fluid Conductors and Fluid Filtration.

5.1.1:	Identify properties of hydraulic fluids.	5 - 2	14. a b c d e
5.3.1:	Match filter specifications to a machine.	5 - 7	15. a b c d e

Analyze and Troubleshoot Hydraulic Systems.

6.1.1:	Identify troubleshooting parameters.	6 - 2	16. a b c d e
6.3.1:	Trace the operation sequence of a circuit.	6 - 12	17. a b c d e

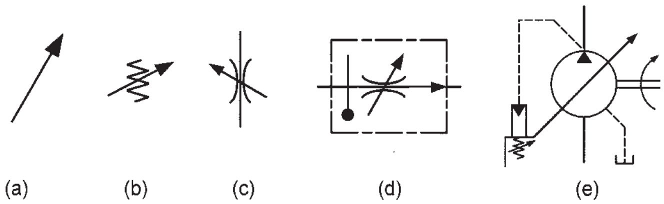
Interface Hydraulic Instrumentation and Control Systems.

7.1.1:	Recognize the basics of electrical control systems.	7 - 2	18. a b c d e

Hydraulic Specialist Certification Pretest HS-3

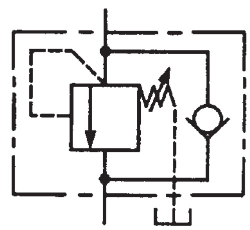
1: Which symbol indicates the component is varied only as a function of pressure?

a. a
b. b
c. c
d. d
e. e



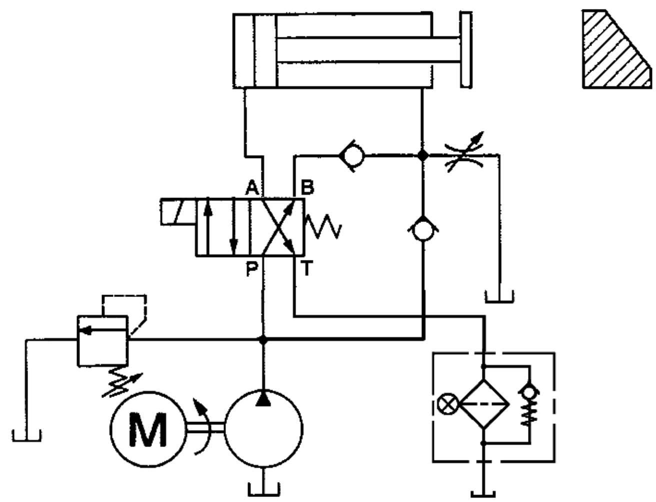
2: The valve symbol shown is a:

a. sequence valve
b. counterbalance valve
c. unloading valve
d. pressure reducing valve
e. brake valve



3: The regenerative circuit in the figure below has a 2.5 inch bore and a 1.750 inch diameter cylinder rod. If maximum cylinder pressure is set at 1500 psi, what force will the cylinder rod exert after the cylinder stalls?

a. 1178 Ibs
b. 3608 lbs
c. 3755 lbs
d. 6378 lbs
e. 7365 lbs



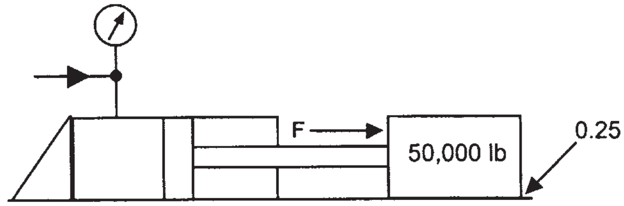
4: A pressure gauge indicates 150 bar. What is the approximate pressure in psi?

a. 10.34 psi
b. 21.75 psi
c. 217.5 psi
d. 1034 psi
e. 2175 psi

Hydraulic Specialist Certification Pretest HS-3

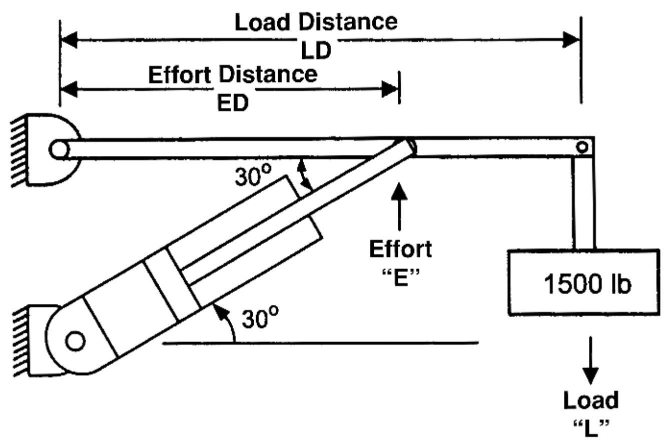
5: A 50000 lb load is pushed horizontally by a hydraulic cylinder. If the coefficient of friction between the surface and the load is 0.25 and the cylinder shown has a 3 inch bore diameter, what minimum pressure given would be required to move the load?

a. 360 psi
b. 520 psi
c. 640 psi
d. 1800 psi
e. 2000 psi



6: In the figure shown, the angle between the cylinder rod and boom is 30° and LD = 2 x ED, what minimum theoretical pressure would there be in the cap end of a 3 inch bore cylinder?

a. 350 psi
b. 500 psi
c. 850 psi
d. 1700 psi
e. 1850 psi



7: What torque could be expected from a hydraulic motor that is rated at 2.5 hp at 2250 rpm?

a. 0.93 lb-ft
b. 1.07 lb-ft
c. 1.97 lb-ft
d. 5.84 lb-ft
e. 8.45 lb-ft

8: What is the theoretical delivery in gpm of a gear pump with a displacement of 1.69 cipr turning at 1200 rpm?

a. 5.22 gpm
b. 6.54 gpm
c. 7.23 gpm
d. 8.78 gpm
e. 9.88 gpm

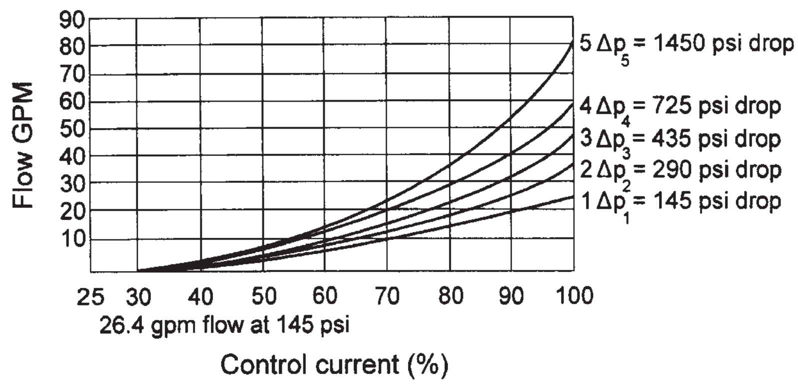
Hydraulic Specialist Certification Pretest HS-3

9: What pressure would be required to open a poppet valve with pressure at port A_A which has a bore diameter A_F of 0.5 inch, a 1.6:1 area differential, and a closing spring force of 7 Ibs? (Assume that the force holding the poppet closed comes from only the spring).

a. 57 psi
b. 122 psi
c. 245 psi
d. 1100 psi
e. 1220 psi

10: For the family of curves for control current vs. flow for the 26.4 gpm spool valve shown in the figure below what would be the approximate pressure drop across the valve for 100% control at a flow rate of 45 gpm? (Neglect load conditions)

a. 145 psi
b. 290 psi
c. 435 psi
d. 725 psi
e. 1450 psi



11: A raised hydraulic reservoir is 4 feet long x 2 feet high x 2 feet wide. If the reservoir is 3/4 full, and the operating temperature is expected to be 100°F above ambient temperature, how much cooling capacity will the reservoir provide in Btu/hr?

a. 4581 Btu/hr
b. 6108 Btu/hr
c. 7132 Btu/hr
d. 8102 Btu/hr
e. 9800 Btu/hr

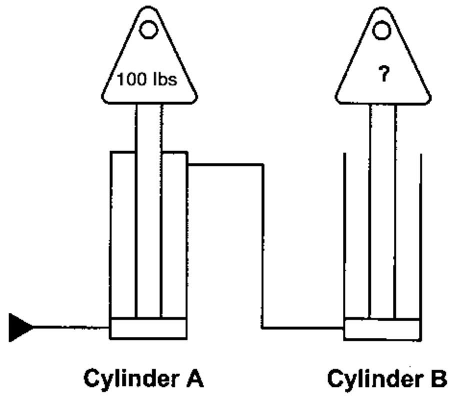
12: An in-plant hydrostatic transmission delivers 7 hp at the output shaft. A flow meter and a pressure gauge indicate the motor portion of the transmission is receiving 10 gpm at 1500 psi. If losses through the motor are converted to heat, how many Btu/hr is the unit generating?

a. 3385 Btu/hr
b. 4015 Btu/hr
c. 4454 Btu/hr
d. 5255 Btu/hr
e. 6011 Btu/hr

Hydraulic Specialist Certification Pretest HS-3

13: In Figure 4-1 fluid enters cylinder A at a pressure of 100 psi. Assuming 100% efficiency, what load on the rod of cylinder B would just stall cylinder B? The size of both cylinders are 3 inch bore x 2 inch rod x 8 inches in stroke.

a. 100 lb
b. 154 lb
c. 607 lb
d. 707 lb
e. 1092 lb



14: Referring to Table 5 -1, which seal material has the widest temperature range?

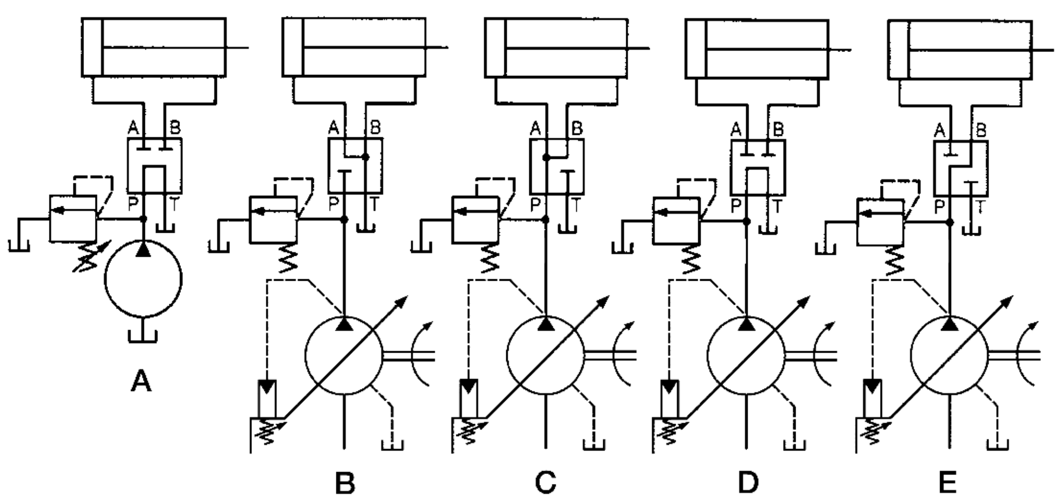
a. Polyurethane
b. Leather
c. Neoprene
d. Silicone rubber
e. Fluoro-elastomer

15: The TAN neutralization number of a hydraulic oil sample refers to the:

a. moisture content of the fluid
b. weight of the fluid sample
c. acidity of the fluid
d. contamination level
e. total analysis number

16: Which of the following valve center - pump combinations is generally considered to be incorrect?

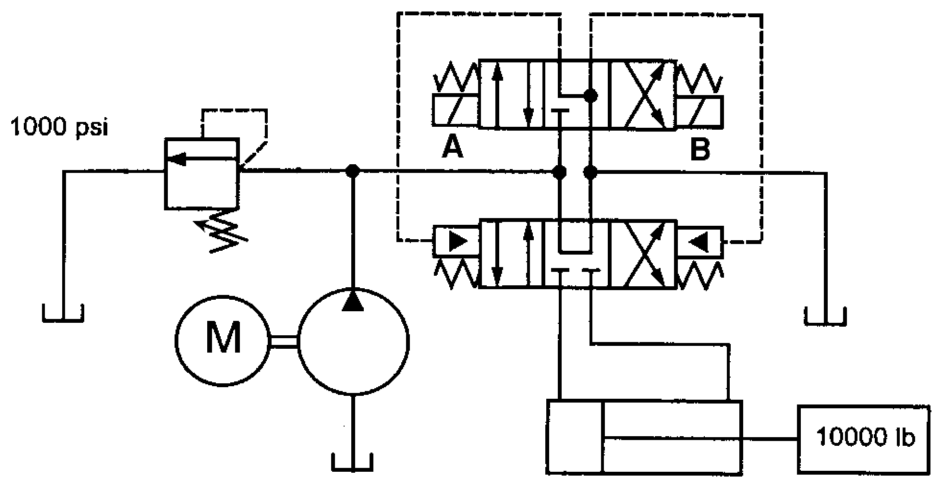
a. A
b. B
c. C
d. D
e. E



Hydraulic Specialist Certification Pretest HS-3

17: The 4 inch bore x 1-1/2 inch rod cylinder shown in the figure below fails to move the load when the directional control valve solenoid control is energized. What is the likely cause of the circuit malfunction? Choose the best answer.

a. Solenoid A on the pilot valve malfunctions.
b. The pressure relief valve setting is too high.
c. The wrong spool center was used in the main valve stage.
d. The wrong spool center was used in the pilot valve.
e. The backpressure check valve in the P port of the main valve is missing.



18: Analog output signals from a PLC require:

a. a relay
b. an indicator light
c. a parallel port
d. a special interface card
e. a serial port

Hydraulic Specialist Certification Pretest HS-3

Pretest C Answer Sheet

Apply Hydraulic Circuits to Perform Desired Tasks.

Outcome:		Page	
1.1.1:	Recognize basic hydraulic symbols.	1 - 2	1. a b c d e 2. a b c d e
1.2.2:	Recognize the function of components in regenerative circuits.	1 - 31	3. a b c d e

Analyze Loads and Motion.

2.1.1:	Recognize factors associated with hydraulic power systems.	2 - 2	4. a b c d e

Select Components for Hydraulic Applications.

3.1.2:	Compute the board diameter and pressure for a cylinder to move loads while allowing for a coefficient of friction (friction factor).	3 - 3	5. a b c d e
3.1.4:	Compute the hydraulic pressure to support jib-boom loads.	3 - 9	6. a b c d e
3.2.1:	Solve formulas for torque, speed and horsepower of hydraulic motors.	3 - 12	7. a b c d e
3.3.1:	Size pump circuits.	3 - 17	8. a b c d e
3.4.2:	Recognize the characteristics of DIN valves.	3 - 19	9. a b c d e
3.4.4:	Read operating curves for proportional directional control valves.	3 - 28	10. a b c d e
3.5.2:	Size hydraulic reservoirs from cooling requirements.	3 - 32	11. a b c d e
3.8.1:	Calculate the heat loss and temperature rise for a hydraulic pressure drop.	3 - 37	12. a b c d e

Prepare Bills of Material and Schematics.

4.1.1:	Calculate circuit force, distance, and sequence times.	4 - 2	13. a b c d e

Recommend Fluid, Fluid Conductors and Fluid Filtration.

5.2.1:	Match the seal materials with compatible fluids.	5 - 4	14. a b c d e
5.4.1:	Identify contaminants in a hydraulic fluid analysis.	5 - 11	15. a b c d e

Analyze and Troubleshoot Hydraulic Systems.

6.2.1:	Ensure the correct component has been specified for the application.	6 - 6	16. a b c d e
6.3.2:	Predict circuit malfunctions.	6 - 13	17. a b c d e

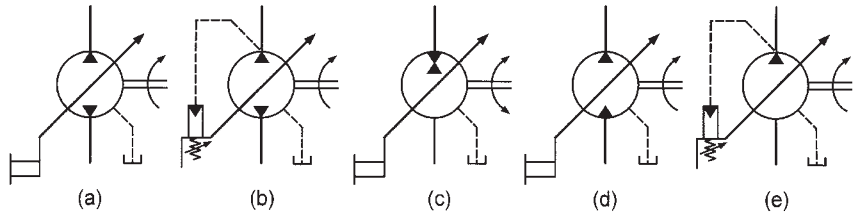
Interface Hydraulic Instrumentation and Control Systems.

7.2.1:	Match programmable logic controller (PLC) devices with their application.	7 - 7	18. a b c d e

Hydraulic Specialist Certification Pretest HS-4

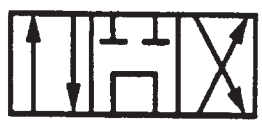
1: Which symbol indicates the hydraulic component operates in one direction as a pump, and in the other as a motor?

a. a
b. b
c. c
d. d
e. e



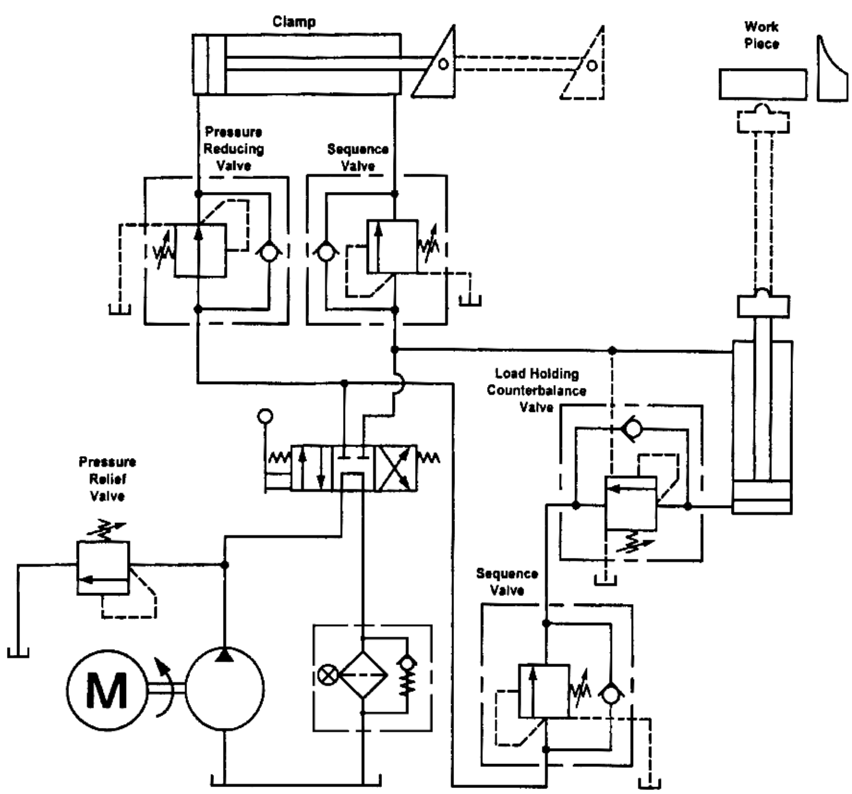
2: How many ports does the directional control valve shown have?

a. two
b. three
c. four
d. five
e. six



3: Which valve shown in the figure below would require the highest pressure setting?

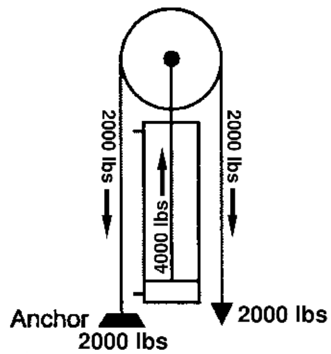
a. counterbalance
b. pressure relief
c. pressure reducing
d. work cylinder sequence valve
e. clamp cylinder sequence valve



Hydraulic Specialist Certification Pretest HS-4

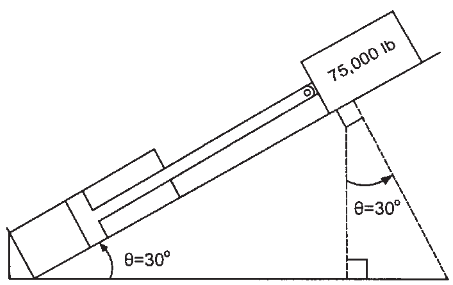
4: Using Figure 2-6, what minimum pressure would be required in a hydraulic cylinder with a 3 inch bore to raise a 2000 lb load? Assume 100% efficiency.

a. 566 psi
b. 650 psi
c. 2000 psi
d. 4000 psi
e. 8000 psi



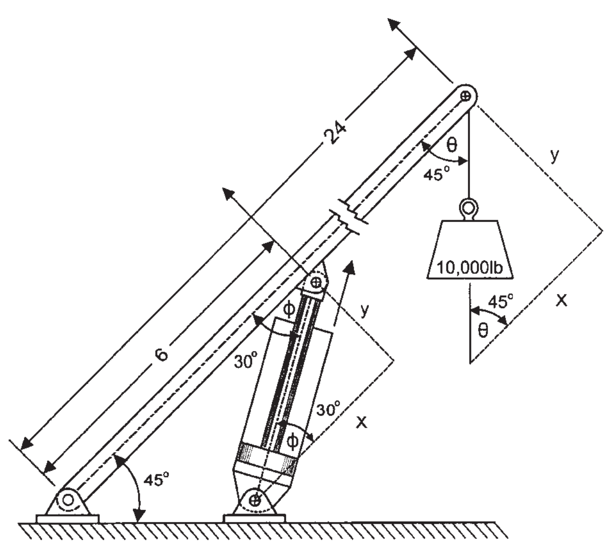
5: A 75000 lb load is to be pushed up a 30° incline by a hydraulic cylinder with a 4 in. bore. If the coefficient of friction between the surface and the load is 0.35, what minimum pressure will be required to extend the cylinder to move the load?

a. 2400 psi
b. 3500 psi
c. 4800 psi
d. 8500 psi
e. 10000 psi



6: The hydraulic cylinder in the figure below has a bore diameter of 5 inches. What minimum pressure in the hydraulic cylinder would be required to hold a load of 10,000 lb at the angles shown?

a. 1426 psi
b. 1440 psi
c. 2161 psi
d. 2882 psi
e. 4321 psi



Hydraulic Specialist Certification Pretest HS-4

7: A pressure gauge at the inlet of a hydraulic motor with a displacement of 0.66 cipr (cubic-inches per revolution) reads 1250 psi as the motor rotates the load at 100 RPM. What is the theoretical output torque of the hydraulic motor? Assume the pressure at the outlet of the motor is 0 psi.

a. 4.4 lb-ft
b. 6.6 lb-ft
c. 8.3 lb-ft
d. 10.9 lb-ft
e. 12.5 lb-ft

8: A pump operating at 1750 rpm is used to extend a 4 inch bore x 2 inch diameter rod x 24 inch stroke cylinder at an average velocity of 540 inches/minute. What is the theoretical displacement of the pump?

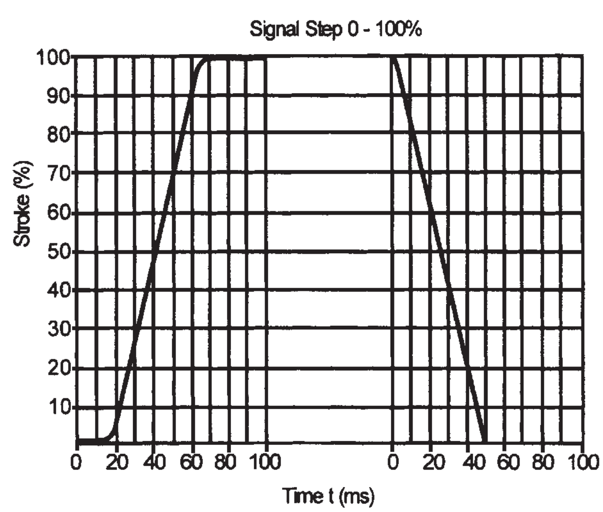
a. 0.62 cipr
b. 0.97 cipr
c. 1.23 cipr
d. 2.91 cipr
e. 3.88 cipr

9: In a cartridge valve with a differential area of 1.6 to 1 and valve closing spring force equal to F_S, what is the pressure at the F port to balance the poppet?

a. $P_F = (0.6\ P_A + P_B - F_S) / 1.6$
b. $P_F = (P_A + 0.6P_B - F_S) / 1.6$
c. $P_F = (P_A + P_B - F_S) / 0.6$
d. $P_F = (P_A + 1.6\ P_B - F_S) / 0.6$
e. $P_F = (P_A + 0.6\ P_B + F_S) / 1.6$

10: Assuming a cycle dwell time of 3 seconds, what is the maximum cycle rate in cycles per minute (cpm) for a proportional valve with response times given in the valve graph shown in the figure below?

a. 19.23 cpm
b. 28.35 cpm
c. 42.88 cpm
d. 53.57 cpm
e. 120.00 cpm



Hydraulic Specialist Certification Pretest HS-4

11: A one gallon capacity accumulator supplies fluid to a hydraulic system between 1750 psi and its precharge pressure of 1000 psi. Using the General Gas Law, how many cubic inches of hydraulic fluid are available from the accumulator if the temperature changes from 80°F to 150°F as the accumulator fills? Assume adiabatic compression and expansion of the gas.

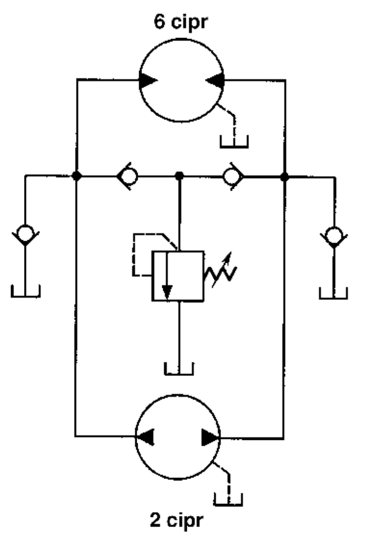
a. 17 cu-in
b. 81 cu-in
c. 150 cu-in
d. 180 cu-in
e. 231 cu-in

12: What is the average velocity of 20 gpm flowing through a 1 inch hydraulic tube if the wall thickness is 0.065 inch?

a. 6.4 ft/sec
b. 10.8 ft/sec
c. 12.7 ft/sec
d. 18.3 ft/sec
e. 21.6 ft/sec

13: In the hydrostatic transmission shown in the figure below the pump is driven at 1200 rpm. At what speed will the motor rotate?

a. 400 rpm
b. 800 rpm
c. 1200 rpm
d. 2400 rpm
e. 3600 rpm



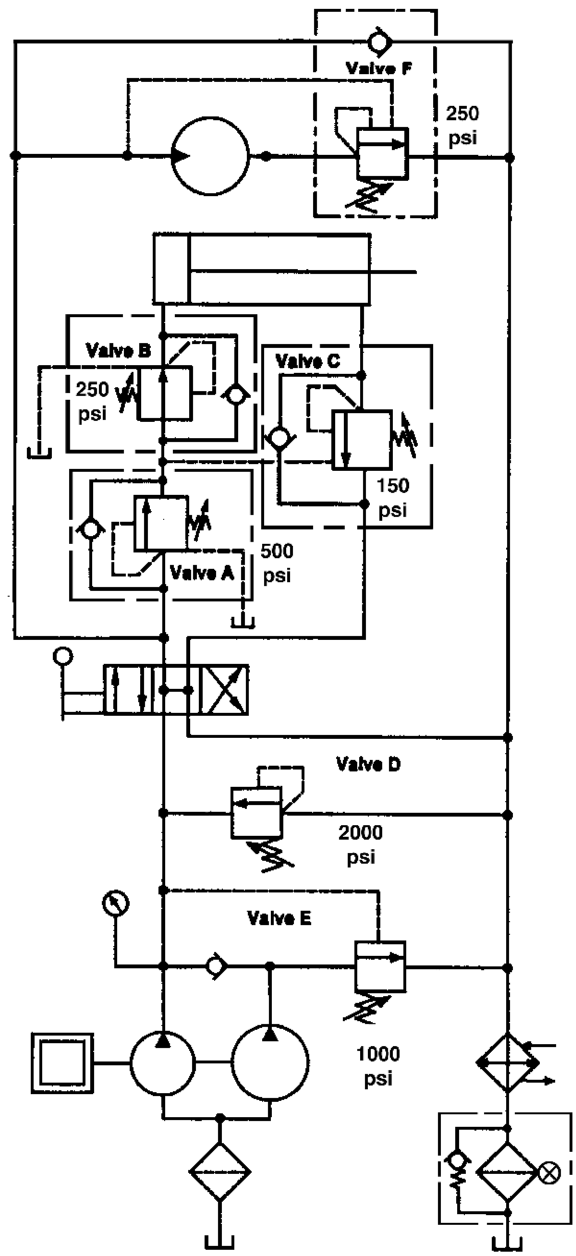
14: Note: This question has purposely been omitted.

15: What type of problem would be indicated if a cylinder connected to a closed center directional control valve retracts under load in the center position?

a. pressure
b. leakage
c. flow
d. heat
e. noise and vibration

16: In the schematic below, which of the pressure control valves is installed backwards?

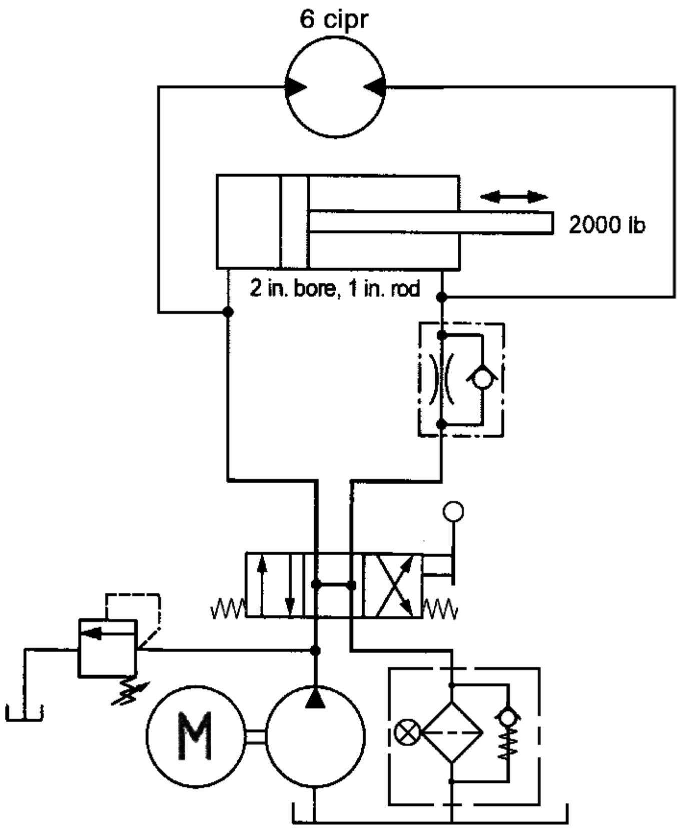
a. A
b. B
c. C
d. D
e. E



Hydraulic Specialist Certification Pretest HS-4

17: In the figure below, the cylinder rod does not extend when the directional control valve is actuated. The pressure relief valve is set at 1000 psi. What should be done to make the cylinder rod extend and retract? The torque required to turn the load is 575 lb-in."

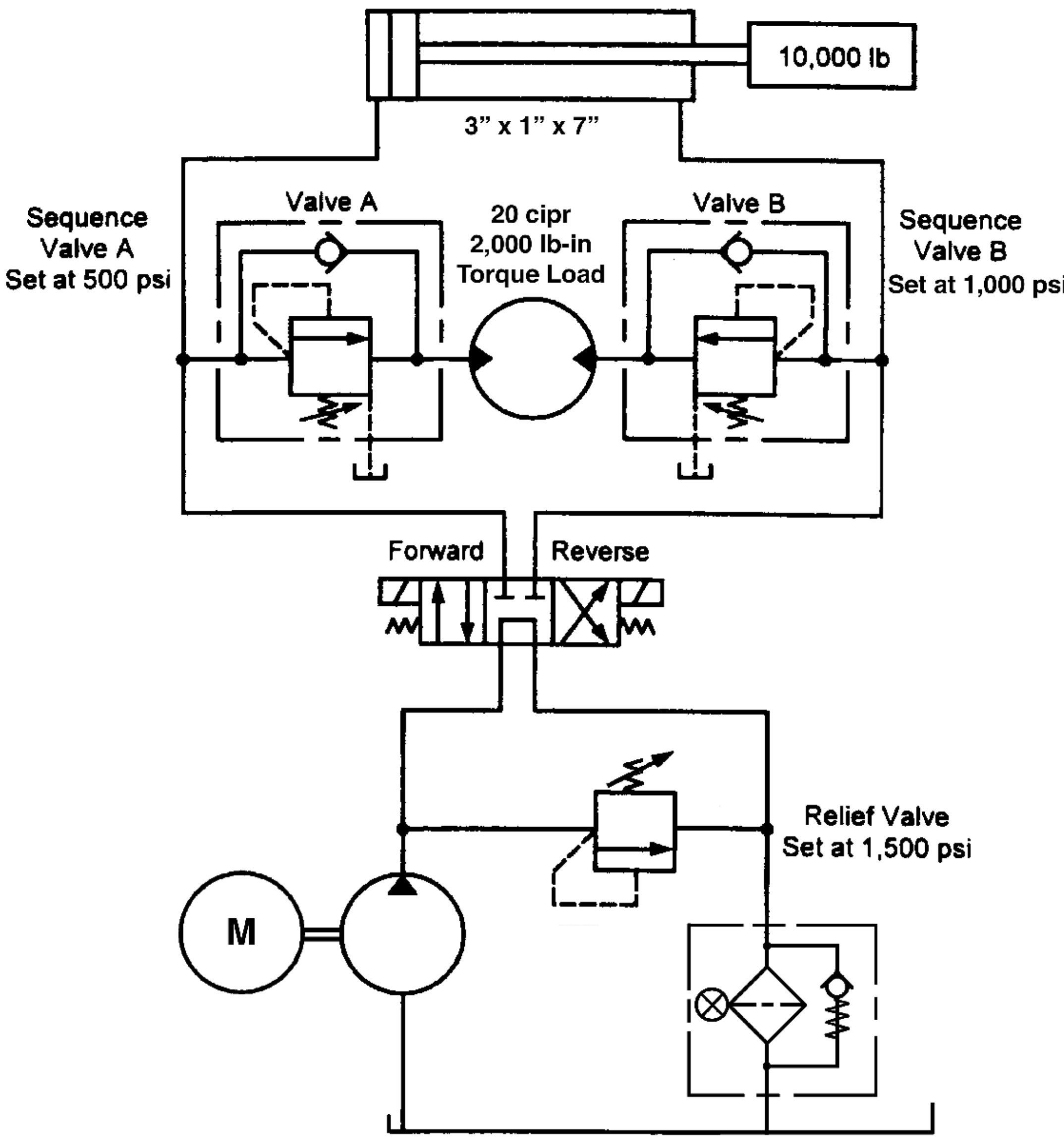
a. Increase the pressure setting of the relief valve.
b. Resize the cylinder or the motor.
c. Reverse the flow control valve at the rod end of the cylinder.
d. Jog the circuit with the directional control valve.
e. The circuit cannot be made to operate properly.



Hydraulic Specialist Certification Pretest HS-4

18: Neglecting friction, why will the cylinder in the figure below fail to extend?

a. The pressure setting of valve A is too low.
b. The pressure setting of valve B is too high.
c. The load on the motor is too high.
d. The pressure relief valve is set too low.
e. The cylinder regenerates through the motor.



Hydraulic Specialist Certification Pretest HS-4

Pretest D Answer Sheet

Apply Hydraulic Circuits to Perform Desired Tasks.

Outcome:		Page	
1.1.1:	Recognize basic hydraulic symbols.	1 - 2	1. a b c d e
			2. a b c d e
1.2.3:	Recognize the placement of pressure relief, pressure reducing, sequence, and counterbalance valves in a circuit.	1 - 36	3. a b c d e

Analyze Loads and Motion.

2.2.2:	Solve pulley systems for force at the point of applications.	2 - 9	4. a b c d e

Select Components for Hydraulic Applications.

3.1.2:	Compute the board diameter and pressure for a cylinder to move loads while allowing for a coefficient of friction (friction factor).	3 - 3	5. a b c d e
3.1.4:	Compute the hydraulic pressure to support jib-boom loads.	3 - 9	6. a b c d e
3.2.1:	Solve formulas for torque, speed and horsepower of hydraulic motors.	3 - 12	7. a b c d e
3.3.1:	Size pump circuits.	3 - 17	8. a b c d e
3.4.2:	Recognize the characteristics of DIN valves.	3 - 19	9. a b c d e
3.4.5:	Compute the cycle time for a proportional control valve.	3 - 29	10. a b c d e
3.6.1:	Size an accomulator using gas laws.	3 - 33	11. a b c d e
3.9.1:	Size fluid conductors.	3 - 38	12. a b c d e

Prepare Bills of Material and Schematics.

4.1.2:	Calculate motor rpm from the pump and motor displacement.	4 - 3	13. a b c d e
			14. omitted

Analyze and Troubleshoot Hydraulic Systems.

6.1.1:	Identify troubleshooting parameters.	6 - 2	15. a b c d e
6.2.2:	Distinguish between correct and incorrect component connections.	6 - 8	16. a b c d e
6.3.2:	Predict circuit malfunctions.	6 - 13	17. a b c d e
			18. a b c d e

Hydraulic Specialist Pretest Answer Key

	HS-1		HS-2		HS-3		HS-4	
No.	Ref.	Answer	Ref.	Answer	Ref.	Answer	Ref.	Answer
1	1.1.1.1.	b	1.1.1.2.	e	1.1.1.3.	e	1.1.1.4.	c
2	1.1.1.5.	d	1.1.1.6.	e	1.1.1.7.	a	1.1.1.8.	c
3	1.1.1.9.	e	1.2.2.1.	a	1.2.2.2.	e	1.2.3.1.	b
4	2.1.1.1.	d	2.1.1.2.	d	2.1.1.3.	e	2.2.2.1.	a
5	2.3.1.1.	d	3.1.1.1.	a	3.1.2.1.	d	3.1.2.2.	c
6	3.1.3.1.	c	3.1.3.2.	d	3.1.4.1.	c	3.1.4.2.	d
7	3.2.1.1.	e	3.2.1.2.	b	3.2.1.3.	d	3.2.1.4.	d
8	3.2.1.5.	c	3.2.1.6.	c	3.3.1.1.	d	3.3.1.2.	e
9	3.4.1.1.	d	3.4.1.2.	b	3.4.2.1.	a	3.4.2.2.	b
10	3.4.2.3.	d	3.4.3.1.	c	3.4.4.1.	c	3.4.5.1.	a
11	3.4.6.1.	e	3.5.1.1.	c	3.5.2.1.	a	3.6.1.1.	b
12	3.6.1.2.	a	3.7.1.1.	a	3.8.1.1.	c	3.9.1.1.	b
13	3.9.1.2.	d	3.9.2.1.	c	4.1.1.1.	e	4.1.2.1.	a
14	4.2.1.1.	d	5.1.1.1.	a	5.2.1.1.	d	omitted	
15	5.3.1.1.	a	5.3.1.2.	d	5.4.1.1.	c	6.1.1.1.	b
16	6.1.1.2.	c	6.1.1.3.	d	6.2.1.1.	d	6.2.2.1.	d
17	6.2.2.2.	c	6.3.1.1.	a	6.3.2.1.	e	6.3.2.2.	b
18	6.3.2.3.	a	7.1.1.1.	d	7.2.1.1.	d	6.3.2.3.	a

Bibliography

Closed Loop Electrohydraulic Systems Manual; First Edition, 1993; Vickers, Inc.
(now Eaton Corp., Eden Prairie, MN); ISBN: 0-9634162-1-9

Fluid Power Design Handbook; First Edition, 1996; B & T Hydraulics & Pneumatics, Inc.;
ISBN: 0-9653599-0-5

Industrial Hydraulics Manual; Fourth Edition, Second Printing, 2001; Eaton Corp., Eden Prairie, MN;
ISBN: 0-9634162-0-0

Lightening Reference Handbook; Eighth Edition, Rev. 2, 2001; Berendsen Fluid Power

Mobile Hydraulics Manual; First Edition 1998; Vickers Inc. (now Eaton Corp., Eden Prairie, MN);
ISBN: 0-9634162-5-1

Mobile Hydraulic Technology, Bulletin 0274-B1; 1999; Parker Hannifin Corporation, Cleveland, OH;
ISBN: 1-55769-039-1

Programmable Logic Controllers; 1989, Frank D. Petruzella; McGraw-Hill Book Company, New York;
ISBN: 0-07-049687-0

Using Industrial Hydraulics; Third Edition, 1990; T. C. Frankenfield; The Rexroth Corp.;
Hydraulics & Pneumatics Magazine, Cleveland, OH; ISBN: 0-932905-01-3

Machinery's Handbook, 26th Edition, 2000, Industrial Press, Inc., New York, NY

Help Improve This Guide

Updates, corrections and revisions to this Manual are requested and encouraged. This Manual is anon-going attempt at developing support materials for Certified Fluid Power candidates. It will undoubtedly require improvement. It is up to Certified Fluid Power candidates and Accredited Instructors to provide input and suggestions for improvement. The Fluid Power Certification Board, composed of industry volunteers, is responsible for determining what revisions and improvements are made to this Manual. The Manuals are updated on aregular basis and date stamped on each page.

Please send your suggestions for improvement to the executive director who is coordinating input on behalf of the Fluid Power Certification Board.

Thank you very much for helping us improve these materials for future candidates.

ATTN: Executive Director
Fluid Power Society
Mailing: P.O. Box 1420, Cherry Hill, NJ 08034-0054
Shipping: 1930 East Marlton Pike, Suite A-2, Cherry Hill, NJ 08003-2142
Phone: 856-424-8998 • Fax: 856-424-9248

Comments

FLUID POWER CERTIFICATION

Setting competitive standards for Fluid Power Professionals

Fluid Power Certification ... How Can I Benefit?

Fluid Power Certification is a fast-growing educational movement in the industry today - and it's not surprising why.

Much of the traditional training from manufacturers, technical schools, and universities has been of high quality, but limited in its availability. Consequently, few of the 350,000 people working in the industry have been able to take full advantage of Fluid Power training. Many of today's fluid power professionals learned about the technology on the job and often did not receive the recognition they deserved for their educational accomplishments.

If the majority of your professional training was on-the-job or limited to short courses and workshops, then fluid power certification may be just what you need to stay competitive in today's marketplace. Fluid power certification gives you an opportunity to demonstrate your extraordinary effort to enhance your professionalism through education, training, and peer review. It may provide you with the credential you need to open the door for career advancement. For fluid power distributors, manufacturers and end-users, certification offers a multitude of benefits:

- Provides another measure with which to assess new employees.
- Establishes a minimal level of Fluid Power knowledge and skills.
- Educates your customers - so you don't have to.
- Helps satisfy requirements for employee qualifications.
- Demonstrates an individual's efforts to achieve and maintain the highest professional proficiency available in the industry.

What's Involved in Certification?

Fluid power certification consists of an optional review session, followed by a three-hour written test and recognition upon successful completion. For Mechanics' and Technicians' certification, an additional three-hour job performance test is also required.

How Many Kinds of Tests Are Offered?

The Fluid Power Certification Board currently offers nine Certification Tests at four levels:

- **Mechanic:** fabricates, assembles, tests, maintains and repairs systems and components, etc.
  - ***Master Mechanic***
    - Mobile Hydraulic Mechanic
    - Industrial Hydraulic Mechanic
    - Pneumatic Mechanic
- **Technician:** troubleshoots systems, tests and modifies systems, prepares reports, etc.
  - ***Master Technician***
    - Mobile Hydraulic Technician
    - Industrial Hydraulic Technician
    - Pneumatic Technician
- **Specialist:** analyzes and designs systems, selects components, instructs others in operations and maintenance, etc.
  - ***Fluid Power Specialist***
    - Hydraulic Specialist
    - Pneumatic Specialist
- **Engineer:** has a technology or engineering degree or is a current Professional Engineer, has eight years of work experience and has passed the Hydraulic & Pneumatic Specialist exams.

What Technologies are Covered by the Tests?

Fluid power and motion control technologies include questions on hydraulics, pneumatics, electronic control, and vacuum.

Who May Organize a Review Training Session?

Educational institutions, end-user companies, fluid power distributors, fluid power component manufacturers, for-profit educational organizations and the Fluid Power Society (local chapters or the national Headquarters), can organize review training sessions.

Who Administers the Tests?

Written testing is conducted under the supervision of local proctors retained by the Fluid Power Certification Board. Job performance testing may only be administered by an FPS Accredited Instructor. Tests are scheduled throughout the world in over 138 cities throughout the year.

How Will My Accomplishments be Recognized?

Certified fluid power professionals are encouraged to include their certification on their business cards and letterhead - even on work vehicle signage. Certification patches are also available for use on uniforms, as well as other promotional items. All Certified Professionals receive a certificate suitable for framing, wallet card, are recognized in the Fluid Power Journal, are listed in the annual Certification Directory, and on the Fluid Power Society's web site.

Will I Have to Renew My Certification?

Yes - Certifications are valid for five years. After that time, you must apply for re-certification based on a point system. On the re-certification form, you will be asked to list job responsibilities, additional educational courses you have taken or taught, and professional involvement in Fluid Power or allied organizations.

What Will This Cost Me?

The Fluid Power Certification Board has made every effort to keep costs low and make Certification available to as many fluid power professionals as possible. Many manufacturers and distributors subsidize or even reward this program because it provides a great return on investment. A contribution to the fluid power certification program helps upgrade the skills of those professionals committed to the industry and elevates the level of professionalism throughout the entire Fluid Power Industry.

How Can I Receive More Information?

For fee schedules, review sessions, manuals and other information, please visit our web site at www.IFPS.org, contact Headquarters at 1-800-303-8520, or write to:

Fluid Power Certification Board
Mailing: P.O. Box 1420
Cherry Hill, NJ 08034-0054
Shipping: 1930 East Marlton Pike,
Suite A-2, Cherry Hill, NJ 08003-2142
Phone: 800-303-8520, 856-424-8998;
Fax: 856-424-9248
E-mail: askus@ifps.org
Web: http://www.ifps.org

Fluid Power Certification Board

Certification Coordinator, c/o Fluid Power Society • P.O. Box 1420, Cherry Hill, NJ 08034-0054 • Phone: (856) 424-8998 • Fax: (856) 424-9248

CERTIFICATION TEST APPLICATION

There is a three year window from date of application to take the test. After this time, fees are forfeited.

Personal Information:

Name

Address

City State Zip/Postal Code

Phone Fax

E-mail Address (home)

Social Security Number (Serves as Test ID Number)

Company Information:

Job Title

Company Name

Address

City State Zip/Postal Code

Phone Fax

E-mail Address (work)

Membership Status:

❑ Professional Member - Membership #___________ ❑ Student Member (Proof of full time included/12 credit minimum) ❑ Non-Member

Preferred Mailing Address: ❑ Home ❑ Work

Education Information:

- ❑ Grade School . . . . .Graduated ❑ Yes ❑ No
- ❑ High School . . . . .Graduated ❑ Yes ❑ No
- ❑ Technical Institute . . . . .Graduated ❑ Yes ❑ No
- ❑ College . . . .❑ Associates ❑ Bachelors ❑ Masters ❑ Doctorate/PhD

Please Note:
Payment must be received by the test deadline date to:
- Receive study materials.
- Have test available at testing site.
- Avoid incurring additional fees.

Which test do you intend to take? (Please note, Job Performance Test must be taken once for Mechanic and Technician Certifications.)

- ❑ Mobile Hydraulic Mechanic*
- ❑ Mobile Hydraulic Technician*
- ❑ Hydraulic Specialist
- ❑ Industrial Hydraulic Mechanic*
- ❑ Industrial Hydraulic Technician*
- ❑ Job Performance Test
- ❑ Pneumatic Mechanic*
- ❑ Pneumatic Technician
- ❑ Pneumatic Specialist

**Certification requires the Job Performance Test if it has not already been passed.*

Test Date: ________________________________ Test Site: ________________________________

❑ Test Date and Site to Be Determined

Item (Refer to Fee Schedule)	Amount
Written Test Fee	
Job Performance Test Fee (if required)	
Retake Fee - Written Test	
Retake Fee - Job Performance Test	
Short Notice Reschedule Fee	
Total Due	

For Office Use Only
Member #
Prior Test:
Fee Received:
Study Manuals Sent:

Payment Type:

Payment is required by test deadline date to receive study material and to have test available at location, without incurring additional fees.

Credit Card ❑ MasterCard ❑ Visa ❑ Check or Money Order Enclosed (in U.S. Funds)

Credit Card Number Exp. Date Signature

All FPS fees must be pre-paid and are non-refundable. Visit www.IFPS.org or call 800-303-8520 for complete Fee Policies.