Online Instructor's Manual
*to accompany*

# Fluid Power with Applications

**Seventh Edition**

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# PREFACE

**The purpose of this manual for FLUID POWER WITH APPLICATIONS is threefold:**

1. **To provide the instructor with student-oriented learning objectives for each chapter. In this way the instructor can better organize teaching strategies and testing techniques.**

2. **To provide the instructor with answers to textbook questions, which are, designed to give the student the necessary practice for understanding the important concepts and applications.**

3. **To provide the instructor with solutions to textbook problems, which are, designed to give the student the necessary practice for mastering sound problem solving techniques.**

**Many of the textbook exercises (questions and problems) can be adapted directly for student testing purposes.**

**Considerable effort has been made to provide an instructor's manual that is helpful to both the instructor and the student. However there is always room for improvement. Therefore any suggestions for improving this manual are most welcome and are greatly appreciated.**

**I hope that this manual will help the instructor to more effectively use the Textbook so that he or she can provide the**

**student with a better education in the vast subject of Fluid Power.**

**Anthony Esposito**

# Part I  Overview of Text Objectives

## Chapter 1  Introduction to Fluid Power

This chapter introduces the student to the overall field of fluid power. It answers the question “What is fluid power?” and presents a corresponding historical background. Advantages and applications of fluid power systems are discussed in detail. Emphasis is placed on the fact that fluid power systems are designed to perform useful work. A complete hydraulic system and a complete pneumatic system are individually presented with identifications of the necessary components and their functions. The fluid power industry is examined in terms of its bright, expanding future and the need for fluid power mechanics, technicians and engineers.

## Chapter 2  Physical Properties of Hydraulic Fluids

This chapter deals with the single most important material in a hydraulic system: the working fluid. It introduces the student to the various types of hydraulic fluids and their most important physical properties. The differences between liquids and gases are outlined in terms of fundamental characteristics and applications. Methods for testing various fluid properties (such as bulk modulus, viscosity, and viscosity index) are presented. The student is introduced to the concepts of pressure, head and force. Units in the Metric System are described and compared to units in the English System. This will prepare the student for the inevitable United States adoption of the Metric System.

## Chapter 3 Energy and Power in Hydraulic Systems

This chapter introduces the student to the basic laws and principles of fluid mechanics, which are necessary for understanding the concepts presented in later chapters. Emphasis is placed on energy, power, efficiency, continuity of flow, Pascal's Law and Bernoulli's Theorem. Stressed is the fact that fluid power is not a source of energy but, in reality, is an energy transfer system. As such, fluid power should be used in applications where it can transfer energy better than other systems. Applications presented include the hydraulic jack and the air-to-hydraulic pressure booster. Problem solving techniques are presented using English and Metric units.

## Chapter 4 Frictional Losses in Hydraulic Pipelines

This chapter investigates the mechanism of energy losses due to friction associated with the flow of a fluid inside a pipeline. It introduces the student to laminar and turbulent flow, Reynold's Number and frictional losses in fittings as well as pipes. Hydraulic circuit analysis by the equivalent length method is presented. Stressed is the fact that it is very important to keep all energy losses in a fluid power system to a minimum acceptable level. This requires the proper selection of the sizes of all pipes and fittings used in the system. Problem solving techniques are presented using English and Metric units.

## Chapter 5 Hydraulic Pumps

This chapter introduces the student to the operation of pumps, which convert mechanical energy into hydraulic energy. The theory of pumping is presented for both positive displacement and non-positive displacement pumps. Emphasized is the fact that pumps do not pump pressure but instead produce the flow of a fluid. The resistance to this flow, produced by the hydraulic system, is what determines the pressure. The operation and applications of the three principal types of fluid power pumps (gear, vane and piston) are described in detail. Methods are presented for selecting pumps and evaluating their performance using Metric and English units. The causes of pump noise are discussed and ways to reduce noise levels are identified.

## Chapter 6  Hydraulic Cylinders and Cushioning Devices
## Chapter 7  Hydraulic Motors

These two chapters introduce the student to energy output devices (called actuators) which include cylinders and motors. Cylinders are linear actuators, whereas motors are rotary actuators. Emphasized is the fact that hydraulic actuators perform just the opposite function of that performed by pumps. Thus actuators extract energy from a fluid and convert it into a mechanical output to perform useful work. Included are discussions on the construction, operation and applications of various types of hydraulic cylinders and motors. Presented is the mechanics of determining hydraulic cylinder loadings when using various linkages such as first class, second class and third class lever systems. The design and operation of hydraulic cylinder cushions and hydraulic shock absorbers are discussed along with their industrial applications. Methods are presented for evaluating the performance of hydraulic motors and selecting motors for various applications. Hydrostatic transmissions are discussed in terms of their practical applications as adjustable speed drives.

## Chapter 8  Hydraulic Valves

This chapter introduces the student to the basic operations of the various types of hydraulic valves. It emphasizes the fact that valves must be properly selected or the entire hydraulic system will not function as required. The three basic types of hydraulic valves are directional control valves, pressure control valves and flow control valves. Each type of valve is discussed in terms of its construction, operation and application. Emphasis is placed on the importance of knowing the primary function and operation of the various types of valves. This knowledge is not only required for designing a good functioning system, but it also leads to the discovery of innovative ways to improve a fluid power system for a given application. This is one of the biggest challenges facing the hydraulic system designer. Also discussed are the functions and operational characteristics of servo valves, proportional control valves and cartridge valves.

## Chapter 9  Hydraulic Circuit Design and Analysis

The material presented in previous chapters dealt with basic fundamentals and system components. This chapter is designed to offer insight into the basic types of hydraulic circuits including their capabilities and performance. The student should be made aware that when analyzing or designing a hydraulic circuit, three important considerations must be taken into account: (1) Safety of operation, (2) Performance of desired function, and (3) Efficiency of operation. In order to properly understand the operation of hydraulic circuits, the student must have a working knowledge of components in terms of their operation and their ANSI graphical representations.

## Chapter 10  Hydraulic Conductors and Fittings

This chapter introduces the student to the various types of conductors and fittings used to conduct the fluid between the various components of a hydraulic system. Advantages and disadvantages of the four primary types of conductors (steel pipe, steel tubing, plastic tubing and flexible hose) are discussed along with practical applications. Sizing and pressure rating techniques are presented using English and Metric units. The very important distinction between burst pressure and working pressure is emphasized as related to the concept of factor of safety. The difference between tensile stress and tensile strength is also explained. Precautions are emphasized for proper installation of conductors to minimize maintenance problems after a fluid power system is placed into operation. The design, operation and application of quick disconnect couplings are also presented.

## Chapter 11  Ancillary Hydraulic Devices

Ancillary hydraulic devices are those important components that do not fall under the major categories of pumps, valves, actuators, conductors and fittings. This chapter deals with these ancillary devices which include reservoirs, accumulators, pressure intensifiers, sealing devices, heat exchangers, pressure gages and flow meters. Two exceptions are the components called

**filters and strainers which are covered in Chapter 12 Maintenance of Hydraulic Systems. Filters and strainers are included in Chapter 12 because these two components are specifically designed to enhance the successful maintenance of hydraulic systems.**

## Chapter 12 Maintenance of Hydraulic Systems

**This chapter stresses the need for planned preventative maintenance. The student is introduced to the common causes of hydraulic system breakdown. Stressed is the fact that over half of all hydraulic system problems have been traced directly to the fluid. Methods for properly maintaining and disposing of hydraulic fluids are discussed in terms of accomplishing pollution control and conservation of natural resources objectives. The mechanism of the wear of mating moving parts due to solid particle contamination of the fluid, is discussed in detail. Also explained are the problems caused by the existence of gases in the hydraulic fluid. Components that are presented include filters and strainers. Methods for trouble-shooting hydraulic circuits are described. Emphasized is the need for human safety when systems are designed, installed, operated and maintained.**

## Chapter 13 Pneumatics - Air Preparation and Components

**This chapter introduces the student to pneumatics where pressurized gases (normally air) are used to transmit and control power. Properties of air are discussed and the perfect gas laws are presented. Then the purpose, construction and operation of compressors are described. Methods are presented to determine the power required to drive compressors and the consumption rate of pneumatically driven equipment such as impact wrenches, hoists, drills, hammers, paint sprayers and grinders. Fluid conditioners such as filters, regulators, lubricators, mufflers and air dryers are discussed in detail. The student is then introduced to the design, operation and application of pneumatic pressure control valves, flow control valves, directional control valves and actuators (linear and rotary).**

## Chapter 14  Pneumatics - Circuits and Applications

**This chapter delves into the operation and analysis of basic pneumatic circuits and with corresponding applications. A comparison is made between hydraulic and pneumatic systems including advantages, disadvantages and types of applications. It is important for the student to appreciate the performance, operating characteristics, cost and application differences between pneumatic and hydraulic systems. The operation of pneumatic vacuum systems is discussed along with the analysis method for determining vacuum lift capacities. Techniques for evaluating the cost of air leakage into the atmosphere and frictional energy losses are presented. Methods are also provided for sizing gas-loaded accumulators. In addition, the trouble shooting of pneumatic circuits is discussed as a means of determining the causes of system malfunction.**

## Chapter 15  Basic Electrical Controls for Fluid Power Circuits

**This chapter introduces the student to fluid power systems where basic electrical devices are used for control purposes. There are seven basic electrical devices that are commonly used: manually actuated switches, limit switches, pressure switches, solenoids, relays, timers and temperature switches.**
**Each type of electrical device is discussed in terms of its construction, operation and function in various practical fluid power applications. Electrical circuits, containing these electrical components, are represented in ladder diagram format. This chapter delves into how the electrical ladder diagrams interact with corresponding hydraulic/pneumatic circuits. Shown for example is how the manual actuation of an electric push button switch can cause electrohydraulic/pneumatic equipment to perform a variety of industrial operations.**

## Chapter 16  Fluid Logic Control Systems

**This chapter introduces students to the theory and operation of MPL (Moving Part Logic) control systems. It is pointed out that successful miniaturization of MPL devices and also maintenance-free operation have resulted in increased utilization of MPL controls for a wide variety of industrial fluid power**

applications. Stressed is the fact that MPL is used for controlling fluid power systems. As such, the MPL portion of the system is the brain and the fluid power portion provides the brawn or muscle. Discussed in detail are the advantages and disadvantages of MPL control systems as compared to electronic control systems. Illustrations, graphical symbols and truth tables are provided to give the student a better understanding of how MPL control devices function. Examples of MPL logic circuits are presented to illustrate the numerous practical applications. Included are fluid logic circuits using general logic symbols and the application of logic systems design techniques using Boolean Algebra.

## Chapter 17 Advanced Electrical Controls for Fluid Power Systems

This chapter presents the theory, analysis and operation of electro-hydraulic servo systems. Such a system is closed-loop and, thus, provides very precise control of the movement of actuators. Also presented is the application of programmable logic controllers (PLCs) for the control of fluid power systems. Unlike general-purpose computers, PLCs are designed to operate in industrial environments where high ambient temperature and humidity levels may exist, as is typically the case for fluid power applications. Unlike electro-mechanical relays, PLCs are not hard-wired to perform specific functions. Thus when system operating requirements change, a PLC software program is readily changed instead of having to physically re-wire relays.

## Chapter 18 Automation Studio Computer Software

This chapter presents the salient features and capabilities of Automation Studio. Automation Studio is a computer software package that allows users to design, simulate and animate circuits consisting of various automation technologies including hydraulics, pneumatics, PLCs, electrical controls and digital electronics. Included with the Textbook is a CD that illustrates how Automation Studio is used to create, simulate and animate the following 16 fluid power circuits present throughout the book:

- **Four Hydraulic Circuits: Figures 9-3, 9-5, 9-9 and 9-16.**
- **Four Pneumatic Circuits: Figures 14-7, 14-11, 14-18 and 14-19.**
- **Four Electrohydraulic Circuits: Figures 15-11, 15-15, 15-18 and 15-24.**
- **Four Electropneumatic Circuits: Figures 15-14, 15-16, 15-20 and 15-21.**

**By playing this CD on a personal computer, the student obtains a dynamic and visual presentation of the creation, simulation, analysis and animation of many of the fluid power circuits studied in class or assigned as homework excecises.**

# Part II Answers and Solutions to Text Exercises

## Chapter 1

## Introduction to Fluid Power

1-1. Fluid power is the technology which deals with the generation, control and transmission of power using pressurized fluids.

1-2. Liquids provide a very rigid medium for transmitting power and thus can provide huge forces to move loads with utmost accuracy and precision.

1-3. The terms “fluid power” and “hydraulics and pneumatics” are synonymous.

1-4. Advantages of Fluid Power Systems

1. Not hindered by geometry of machine.
2. Provides remote control.
3. Complex mechanical linkages are eliminated.
4. Instantly reversible motion.
5. Automatic protection against overloads.
6. Infinitely variable speed control.

Advantages of Mechanical System:

1. No mess due to oil leakage problems.
2. No danger of bursting of hydraulic lines.
3. No fire hazard due to oil leaks.

1-5. Fluid transport systems have as their sole objective the delivery of a fluid from one location to another to accomplish some useful purpose such as pumping water to homes. Fluid power systems are designed specifically to perform work such as power steering of automobiles.

1-6. Hydraulic fluid power uses liquids which provide a very rigid medium for transmitting power. Thus huge forces can be provided to move loads with utmost accuracy and precision. Pneumatic systems exhibit spongy characteristics due to the compressibility of air. However pneumatic systems are less expensive to build and operate.

1-7. Hydraulic cylinder.

1-8. Hydraulic motor.

1-9.
1. Liquids provide a very rigid medium.
2. Power capacity of fluid systems is limited only by the strength capacity of the component material.

1-10. Pneumatic systems exhibit spongy characteristics due to the compressibility of air.

1-11. An electric motor or other power source to drive the pump or compressor.

1-12.
1. Reservoir.
2. Pump.
3. Prime mover.
4. Valves.
5. Actuators.
6. Piping.

1-13.
1. Compressed air tank.
2. Compressor.
3. Prime mover.
4. Valves.
5. Actuators.
6. Piping.

**1-14.** Plant tour.

**1-15.**
1. Power brakes.
2. Power steering.
3. Shock absorbers.
4. Air conditioning.
5. Automotive transmissions.

**1-16.** Air has entered the hydraulic oil line and has greatly reduced the Bulk Modulus (measure of stiffness or incompressibility) of the oil-air combination fluid.

**1-17.** Hydraulic applications are: automobile power steering and brakes, aircraft landing gear, lift trucks and front-end loaders.

Pneumatic applications are: packaging machinery, environmental test equipment, artificial heart, logic control systems and robotic materials handling devices.

**1-18.** Hydraulic sales – 75%
Pneumatic sales – 25%

**1-19.**
1. Fluid power mechanics.
2. Fluid power technicians.
3. Fluid power engineers.

**1-20.** The fluid power industry is huge as evidenced by its present annual sales figure of $13.6 billion registered by U.S. Companies and $35.5 billion worldwide. It is also a fast-growing industry with a 67% increase in terms of U.S. equipment sales during the period 1991-2000.

**1-21.** Research project.

**1-22.** Research project.

# Chapter 2

## Physical Properties of Hydraulic Fluids

2-1.
1. Transmit power.
2. Lubricate moving parts.
3. Seal clearances between mating parts.

2-2.
1. Good lubricity.
2. Ideal viscosity
3. Chemical and environmental stability.
4. Compatibility with system materials.
5. Large bulk modulus.
6. Fire resistance.
7. Good heat transfer capability.
8. Low density.
9. Foam resistance.
10. Non-toxic.

2-3. Generally speaking, a fluid should be changed when its viscosity and acidity increase due to fluid breakdown or contamination.

2-4. A liquid is a fluid which, for a given mass, will have a definite volume independent of the shape of its container. On the other hand, the volume of a gas will vary to fill the vessel which contains the gas. Liquids are considered to be essentially incompressible. Gases, on the other hand, are fluids which are readily compressible.

2-5. Advantages of air:
1. Fire resistant.
2. Not messy.

Disadvantages of air:
1. Due to its compressibility, it cannot be used in an application where accurate positioning or rigid holding is required.

2. Because it is compressible, it tends to be sluggish.

2-6. 1. Specific weight: weight per unit volume.
2. Density: mass per unit volume.
3. Specific gravity: the specific weight of given fluid divided by the specific weight of water.

2-7. Pressure is force per unit area.

2-8. Gage pressures are measured relative to the atmosphere, whereas absolute pressures are measured relative to a perfect vacuum such as that which exists in outer space. To distinguish between them, gage pressures are labeled psig or simply psi (Pa gage or kPa gage in Metric units). Absolute pressures are labeled psia (Pa abs or kPa abs in Metric units).

2-9. Bulk modulus is a measure of the incompressibility of a hydraulic fluid.

2-10. Viscosity is a measure of the sluggishness with which a fluid flows. Viscosity index is a relative measure of an oil's viscosity change with respect to temperature change.

2-11. 1. High resistance to flow which causes sluggish operation.
2. Increases power consumption due to frictional losses.

2-12. 1. Increased leakage losses past seals.
2. Excessive wear due to breakdown of the oil film between moving parts.

2-13. A Saybolt Universal Second is the viscosity an oil possesses which will allow it to fill a 60-cubic-centimeter container in one second through a standard metering orifice.

**2-14.** **Pour point is the lowest temperature at which a fluid will flow.**

**2-15.** **The height of a column of liquid that represents the pressure it develops at its base. For example a 10 foot head of oil, having a density of 56 $lb/ft^3$, produces a pressure of 0.40 psi at its base.**

**2-16.** **By atmospheric pressure at the base of the mercury column.**

**2-17.** **As the temperature increases, the viscosity decreases and vice versa.**

**2-18.** **When the fluid power system operates in an environment undergoing large temperature variations such as in outdoor machines like automobiles.**

**2-19.** **Decreases.**

**2-20.** **A high VI should be specified indicating small changes in viscosity with respect to changes in temperature.**

**2-21.** $$SG = \frac{\gamma_{fluid}}{\gamma_{water}} = \frac{55\,lb/ft^3}{62.4\,lb/ft^3} = \underline{0.881}$$

$$\rho_{fluid} = \frac{\gamma}{g} = \frac{55\ lb/ft^3}{32.2\ ft/s^2} = \underline{1.71\ slugs/ft^3}$$

**2-22.** $$\gamma\left(\frac{lb}{ft^3}\right) = \frac{372\ lb}{50\ gal} \times \frac{1\ gal}{231\ in^3} \times \frac{1728\ in^3}{1\ ft^3} = \underline{55.7\ lb/ft^3}$$

2-23. $\rho\left(\mathrm{kg/m^3}\right) = 515\,\rho\left(\mathrm{slugs/ft^3}\right) = 515(1.74) = \underline{896\ \mathrm{kg/m^3}}$

2-24. $SG = \dfrac{63}{50} = \underline{1.26}$

2-25. $(a)\ SG_{air} = \dfrac{\rho_{air}}{\rho_{water}} = \dfrac{1.23\,kg/m^3}{999.1\,kg/m^3} = \underline{0.00123}$

$(b)\ \dfrac{SG_{water}}{SG_{air}} = \dfrac{1}{0.00123} = \underline{813}$

2-26. $\text{Volume} = A_{base} \times h = \dfrac{\pi}{4}(0.5\ \mathrm{m})^2 \times 1\ \mathrm{m} = 0.196\ \mathrm{m^3}$

$\text{Weight} = \gamma V = 2000\ \mathrm{N/m^3} \times 0.196\ \mathrm{m^3} = 392\ \mathrm{N}$

$M = \dfrac{W}{g} = \dfrac{392\ \mathrm{N}}{9.81\ \mathrm{m/s^2}} = \underline{40.0\ \mathrm{kg}}$

2-27. (a) $\gamma = \dfrac{W}{V} = \dfrac{8.70\ \mathrm{N}}{0.001\ \mathrm{m^3}} = \underline{8700\ \mathrm{N/m^3}}$

(b) $\rho = \dfrac{\gamma}{g} = \dfrac{8700\ \mathrm{N/m^3}}{9.81\ \mathrm{m/s^2}} = \underline{888\ \mathrm{kg/m^3}}$

$(c)\ SG_{oil} = \dfrac{\rho_{oil}}{\rho_{water}} = \dfrac{888}{999} = \underline{0.889}$

**2-28.** $\gamma = (SG)\gamma_{H_2O} = 0.9 \times 9800 \, N/m^3 = \underline{8820 \, N/m^3}$

$\rho = (SG)\rho_{H_2O} = 0.9 \times 999 \, kg/m^3 = \underline{899 \, kg/m^3}$

$W = \gamma V = 8820 \, N/m^3 \times 125 \, m^3 = \underline{1{,}100{,}000 \, N}$

**2-29.** $p = \gamma h = 55 \, lb/ft^3 \times 30 \; ft = 1650 \; psf \; = \underline{11.4 psi}$

**2-30.** $p_{abs} = (26.0 + 14.7) psi \times \dfrac{1}{0.000145 \, psi/Pa} = 280{,}700 \, Pa \;\; abs$

**= <u>280.7 kPa abs</u>**

**2-31.** **-2 kPa + 101 kPa = <u>99 kPa abs</u>**

**2-32.** $V = A_{base} \times h$

$V(ft^3) = 2 \, ft \times 2 \, ft \times h(ft) = 4 \, h(ft)$

$V(ft^3) = V(gal) \times \dfrac{231 \, in^3}{1 \, gal} \times \dfrac{1 \, ft^3}{1728 \, in^3} = 0.134 \, V(gal)$

**Hence we have: 0.134 V(gal) = 4 h(ft)**

**Or** $h(ft) = \dfrac{0.134 \, V(gal)}{4} = \dfrac{0.134 \times 100}{4} = \underline{3.35 \, ft}$

**2-33.** $p = \gamma \; h = (62.4 \times 0.9) \, lb/ft^3 \times (100 \sin 30^\circ) \; ft = 2810 \; psf \; = \underline{19.5 psi}$

**2-34.** $p = \frac{F}{A} = \frac{13{,}300N}{\frac{\pi}{4}(0.250m)^2} = \frac{13{,}300N}{0.0491m^2} = 271{,}000\,Pa \;\; gage$

= **<u>271 kPa gage</u>**

**2-35.** $(\Delta V) = \frac{-V(\Delta p)}{\beta} = \frac{-20(950)}{300{,}000} = \underline{-0.0633 in^3}$

**2-36.** $\frac{\Delta V}{V} = \frac{-\Delta p}{\beta} = \frac{-49\times 101}{1{,}750{,}000} = -0.00283 = \underline{-0.283\%}$

**2-37.** $\beta = \frac{V(\Delta p)}{\Delta V}$

**Where** $\Delta V = \frac{\pi d^2}{4} L = \frac{\pi (2)^2}{4} \times 0.01 = 0.0314 \text{ in}^3$

$And\,\Delta p = \frac{\Delta F}{A} = \frac{5000}{\frac{\pi}{4}(2)^2} = 1592\,psi$

**Thus** $\beta = \frac{10(1592)}{0.0314} = \underline{507{,}000 \text{ psi}}$

**2-38.** $\beta = \frac{V(\Delta p)}{\Delta V} \quad where$

$V = 10 \text{ in}^3 \times \left(\frac{1 \text{ m}}{39.4 \text{ in.}}\right)^3 = 163 \times 10^{-6} \text{ m}^3$

$$\Delta V = 0.0314\ in^3 \times \left(\frac{1\ m}{39.4\ in.}\right)^3 = 0.513 \times 10^{-6}\ \ m^3$$

$$\Delta p = 1592\,psi \times \frac{1Pa}{0.000145\,psi} = 10.98 \times 10^6\ \ Pa$$

Thus $\beta = 163 \times 10^{-6} \times \frac{10.98 \times 10^6}{0.513 \times 10^{-6}} = 3489 \times 10^6\ Pa = \underline{3489\ MPa}$

**2-39.** $\nu\,(cS) = 0.220\,t - \frac{135}{t} = 0.220 \times 200 - \frac{135}{200} = \underline{43.3\ cS}$

$\mu\,(cP) = 0.9 \times \nu\,(cS) = 0.9 \times 43.3 = \underline{39.0\ cP}$

**2-40.** $VI = \frac{L - U}{L - H} \times 100$

$70 = \frac{375 - U}{375 - 125} \times 100$ Thus $U = \underline{200\ SUS}$

**2-41.** $\mu\left(\frac{N \bullet s}{m^2}\right) = \mu\left(\frac{lb - s}{ft^2}\right) \times \frac{4.448\ N}{1\ \ lb} \times \left(\frac{1\ ft}{0.3048\ m}\right)^2$

Therefore $1\ \frac{lb - s}{ft^2} = \underline{47.88\ \frac{N \bullet s}{m^2}}$

**2-42.** $\nu\left(\frac{m^2}{s}\right) = \nu\left(\frac{ft^2}{s}\right) \times \left(\frac{0.3048\ m}{1\ \ ft}\right)^2$

Therefore $1\,\frac{ft^2}{s} = \underline{0.0929\,\frac{m^2}{s}}$

**2-43.** $$\nu(cS) = \frac{\mu(cP)}{SG} = \frac{1200}{0.89} = 1348\ cS = 13.48\ Stokes = 13.48\frac{cm^2}{s}$$

$$\nu\left(\frac{ft^2}{s}\right) = \nu\left(\frac{cm^2}{s}\right) \times \left(\frac{1\ in.}{2.54\ cm}\right) \times \left(\frac{1\ ft}{12\ in.}\right)^2$$

$$= 13.48 \times \left(\frac{1}{2.54}\right)^2 \times \left(\frac{1}{12}\right)^2 = \underline{0.0145\,\frac{ft^2}{s}}$$

**2-44.** Using Eq. (2-13) yields

$$\mu = \frac{F/A}{v/y} = \frac{Fy}{vA} = \frac{6N \times 0.004m}{1\,m/s \times 0.49\,m^2} = 0.0490\,N\cdot s/m^2$$

Next we convert from units of $N\cdot s/m^2$ to $dyne\cdot s/cm^2$

$$\mu = 0.0490\frac{N\cdot s}{m^2} \times \frac{10^5\ dynes}{1N} \times \left(\frac{1m}{100\,cm}\right)^2 = 0.49\,dyne\cdot s/cm^2 = 0.49\ poise = 49\ cP$$

Finally using Eq. (2-17) we have

$$\mu(cS) = \frac{\mu(cP)}{SG} = \frac{49}{0.9} = \underline{54.4\ cS}$$

**2-45.** Using Eq. (2-13) we have

$$\mu = \frac{F/A}{v/y} = \frac{Fy}{vA}$$

**where**

**y = oil film thickness =** $\dfrac{4.004-4.0}{2} = 0.002\ in. = 0.000167\ ft$

**A = surface area of piston in contact with oil film**

$$= \frac{\pi D^2}{4} \times L = \frac{\pi}{4}\left(\frac{4}{12}\ ft\right)^2 \times \frac{2}{12}\ ft = 0.0145\ ft^2$$

**Substituting known values yields**

$$\mu = \frac{1.0\ lb \times 0.000167\ ft}{10\ {}^{ft}\!/_{s} \times 0.0145\ ft^2} = \underline{0.00115 \frac{lb \cdot s}{ft^2}}$$

# Chapter 3

## Energy and Power in Hydraulic Systems

3-1. Pressure applied to a confined fluid is transmitted undimished in all directions throughout the fluid and acts perpendicular to the surfaces of the container.

3-2. The total energy at upstream station 1 in a pipeline plus the energy added by a pump minus the energy removed by a motor minus the energy loss due to friction, equals the total energy at downstream station 2. If a section of horizontal pipe contains no pump or motor, the pressure at a small diameter location will be less than the pressure at a large diameter location. Pressure energy is transformed into kinetic energy in the small diameter location.

3-3. The weight flow rate is the same for all cross-sections of a pipe. Thus the smaller the pipe diameter, the greater the velocity and vice versa.

3-4. Ideally the velocity of a free jet of fluid is equal to the square root of the product of two times the acceleration of gravity times the head producing the jet.

3-5. As shown in Figure 3-22, in order for a siphon to work, the following two conditions must be met:

1. The elevation of the free end must be lower than the elevation of the liquid surface inside the container.
2. The fluid must be initially forced to flow up from the container into the center portion of the U-tube. This is normally done by temporarily providing a suction pressure at the free end of the siphon.

**3-6.** Energy can neither be created nor destroyed.

**3-7.** Per Figure 3-19, the volume of air flow is determined by the opening position of the butterfly valve. As the air flows through the venturi, it speeds up and loses some of its pressure. This produces a differential pressure between the fuel bowl and the venturi throat. This causes gasoline to flow into the air stream.

**3-8.** Using Figure 3-17 as a reference we have:

Z is called "elevation head" or elevation energy per lb of fluid.

$\frac{P}{\gamma}$ is called "pressure head" or pressure energy per lb of fluid.

$\frac{V^2}{2g}$ is called "velocity head" or kinetic energy per lb of fluid.

**3-9.**
1. A force is required to change the motion of a body.
2. If a body is acted upon by a force, the body will have an acceleration proportional to the magnitude of the force and inversely to the mass of the body.
3. If one body exerts a force on a second body, the second body must exert an equal but opposite force on the first body.

**3-10.** Energy is the ability to perform work. Power is the rate of doing work.

**3-11.** Torque equals the product of a force and moment arm which is measured from the center of a shaft (center of rotation) perpendicularly to the line of action of the force.

**3-12.** **Efficiency, another significant parameter when dealing with work and power, is defined as output power divided by input power.**

**3-13.** **Mechanical power equals force times velocity.**
**Electrical power equals voltage times electrical current.**
**Hydraulic power(or fluid power) equals pressure times volume flow rate.**

**3-14.** **Elevation head is potential energy per unit weight of fluid.**
**Pressure head is pressure energy per unit weight of fluid.**
**Velocity head is kinetic energy per unit weight of fluid.**

**3-15.** **p = constant = F/A**

$$\frac{F}{500} = \frac{100}{50} \quad \text{Hence} \quad F = \underline{1000\ N}$$

**3-16.**

$$V_{small\ piston} = V_{l\arg e\ piston}$$

$$(A\ S)_{small\ piston} = (A\ S)_{l\arg e\ piston}$$

$$(50 \times 10) = (500\ S) \quad \text{Hence} \quad S = \underline{1\ cm}$$

**3-17.** **First calculate the force applied to the air piston.**

$$F_{air} = p_{air}\ A_{air\ \ piston} = 500{,}000\ \frac{N}{m^2} \times \frac{\pi}{4}(0.100\,m)^2 = 3930\,N$$

**Next find the resulting oil pressure produced by the small diameter oil piston.**

$$p_{oil} = \frac{F_{air}}{A_{oil\,(small)}} = \frac{3930\,N}{\frac{\pi}{4}(0.050\,m)^2} = 2000\ kPa$$

**Finally the maximum load is found.**

**Max. Load =** $p_{oil}\ A_{oil\,(l\arg e)} = 2{,}000{,}000\dfrac{N}{m^2}\times\dfrac{\pi}{4}(0.100\,m)^2 =$ **<u>15,700 N</u>**

**3-18.** $\Delta\,p = \gamma\ (\Delta H) \qquad Assu\min g\ \ SG = 0.9,\ \ we\,have:$

$\Delta\,p = (62.4\times 0.9)\ {}^{lb}\!/\!{}_{ft^3}\times 20\ \ ft = 1120\ \ psf\ = 7.78\ \ psi$

$Thus\ \ F_{\tan k top} = 15 - 7.78 = 7.22\ \ psi$

$F\ = pA = 7.22\dfrac{lb}{in^2}\times\dfrac{\pi}{4}(10\times 12\,in.)^2 = \underline{81{,}500\ \ lb}$

**Pascal's Law states that pressure in a static body of fluid is transmitted equally only at the same elevation level. Pressure increases with depth and vice versa in accordance with the following equation:**

$\Delta p = \gamma\ (\Delta H)$

**Changes in pressure due to elevation changes can be ignored in a fluid power system as long as they are small compared to the magnitude of the system pressure produced at the pump discharge port. For example a pump discharge pressure of 1000 psi becomes 996 psi at an elevation 10 ft above the pump. This is only a 0.4% drop in pressure.**

**3-19.** $p = cons\tan t = \dfrac{F}{A}$

$\dfrac{F}{\frac{\pi}{4}(6)^2} = \dfrac{100}{\frac{\pi}{4}(2)^2}$ **Hence F = <u>900 lb</u>**

**3-20.** $V_{small\ piston} = V_{l\arg e\ piston}$

$(A\ S)_{small\ piston} = (A\ S)_{l\arg e\ piston}$

$$\frac{\pi}{4}(2)^2 \times 1.5 = \frac{\pi}{4}(6)^2\, S \quad \text{Hence} \quad S = \underline{0.167\ \text{in.}}$$

**3-21.** $p_{piston} = p_{air} - \gamma_{oil} H_{oil}$

$$= 550{,}000\ \text{Pa} - \left(0.90 \times 9797\ \frac{\text{N}}{\text{m}^3}\right) \times 1\ \text{m}$$

$$= 550{,}000\ \text{Pa} - 8820\ \text{Pa} = 541{,}180\ \text{Pa}$$

$$W = pA = 541{,}180 \times \frac{\pi}{4}(0.250)^2 = \underline{26{,}565\ N}$$

**Ignoring the 1 m head of oil, the maximum weight automobile that can be lifted is:**

$$W = pA = 550{,}000 \times \frac{\pi}{4}(0.250)^2 = 26{,}998\ N$$

$$\%\ \text{Error} = \frac{26{,}998 - 26{,}565}{26{,}565} \times 100 = \underline{1.63\ \%}$$

**3-22.** **(a)** $p_1 = \frac{F_1}{A_1} = \frac{500\, lb}{\frac{\pi}{4}(2\, in.)^2} = \underline{159\ psi}$

**(b)** $p_2 = p_1 = \underline{159\ psi}$

**(c)** $p_{pipe} = p_1 = p_2 = \underline{159\ psi}$

**(d)** $F_2 = p_2 A_2 = 159 \frac{lb}{in.^2} \times \frac{\pi}{4}(4\, in.)^2 = \underline{2000\ lb}$

**3-23.** **(a) First find the force acting on the rod of the pump.**

$$F_{rod} = \frac{8}{2} \times F_{input} = \frac{8}{2} \times 20 = 80\ \text{lb}$$

**Next calculate the pump discharge pressure p.**

$$p = \frac{rod\ \ force}{piston\ \ area} = \frac{F_{red}}{A_{pump\ piston}} = \frac{80lb}{\frac{\pi}{4}(2)^2 in^2} = 25.5psi$$

**We can now calculate the load-carrying capacity.**

$$F_{load} = pA_{load\ piston} = \left(25.5\frac{lb}{in^2}\right) \times \frac{\pi}{4}(4)^2 in^2 = \underline{320.4\ \ lb}$$

**(b) Total volume of oil ejected from the pump equals the volume of oil displacing the load cylinder.**

$$(A \times S)_{pump\ piston} \times (no.\ of\ cycles) = (A \times S)_{load\ piston}$$

$$\frac{\pi}{4}(2)^2\ in^2 \times 3\ in \times 20 = \frac{\pi}{4}(4)^2\ in^2 \times S_{load\ piston}(in)$$

**Thus** $S_{load\ piston} = \underline{15\ \ in.}$

**(c)** $Power = \frac{F \times s}{t} = \frac{320.4\ \ lb \times \frac{15}{12}\ ft}{15\ s} = 26.7\ \frac{ft - lb}{s}$

$$HP_{100\%efficiency} = \frac{26.7}{550} = 0.0485\ \ hp$$

$$HP_{90\%efficiency} = 0.90 \times 0.0485 = \underline{0.0437\ \ hp}$$

**3-24. Metric data are:**

$$\text{Pump piston diameter} = 2\ in \times \frac{2.54\ cm}{1\ in} = 5.08\ cm$$

$$\text{Load cylinder diameter} = 4\ in \times \frac{2.54\ cm}{1\ in} = 10.16\ cm$$

$$\text{Average hand force} = 20\text{ lb} \times \frac{1\text{ N}}{0.225\text{ lb}} = 88.9\text{ N}$$

$$\text{Pump piston stroke} = 3\text{ in} \times \frac{2.54\text{ cm}}{1\text{ in}} = 7.62\text{ cm}$$

(a) First find the force acting on the rod of the pump.

$$F_{rod} = \frac{8}{2} \times F_{input} = \frac{8}{2} \times 88.9\text{ N} = 355.6\text{ N}$$

Next calculate the pump discharge pressure p.

$$p = \frac{rod\ force}{piston\ area} = \frac{F_{rod}}{A_{pump\,piston}} = \frac{355.6\ N}{\frac{\pi}{4}(0.0508\,m)^2} = 175{,}000\ Pa$$

We can now calculate the load carrying capacity.

$$F_{load} = p \times A_{load\ piston} = 175{,}000\frac{N}{m^2} \times \frac{\pi}{4}(0.1016\,m)^2 = \underline{1419\ N}$$

(b) Total volume of oil ejected from the pump equals the volume of oil displacing the load cylinder.

$$(A \times S)_{pump\ piston} \times (\text{no. of cycles}) = (A \times S)_{load\ piston}$$

$$\frac{\pi}{4}(0.0508)^2\ m^2 \times 0.0762\text{ m} \times 20 = \frac{\pi}{4}(0.1016)^2\ m^2 \times S_{load\ piston}$$

$$S_{load\ piston} = 0.381\text{ m} = \underline{38.1\text{ cm}}$$

(c) $\text{Power} = \frac{F \times S}{t} = \frac{1419\text{ N} \times 0.381\text{ m}}{15\text{ s}} = 36.0\ \frac{N \bullet m}{s} = 36.0\text{ Watts}$

Power(@ 90% efficiency) = 0.90 x 36.0 = <u>32.4 Watts</u>

3-25. First find the booster discharge pressure $p_2$.

$$p_2 = \frac{p_1 \times A_1}{A_2} = \frac{125 \times 20}{1} = 2500\ psi$$

**Per Pascal's Law,** $p_3 = p_2 = 2500\ psi$

$$A_3 = \frac{F}{p_3} = \frac{75{,}000\,lb}{2500\,lb/in^2} = \underline{30\ in^2}$$

**3-26. Metric data are:**

$$p_1 = 125 psi \times \frac{101{,}000\,Pa}{14.7\,psi} = 859{,}000\ Pa$$

$$\mathbf{A_1 = 20\,in^2 \times \left(\frac{2.54\ cm}{1\ in.}\right)^2 = 129\ cm^2}$$

$$\mathbf{A_2 = 1\,in^2 \times \left(\frac{2.54\ cm}{1\ in}\right)^2 = 6.45\ cm^2}$$

$$\mathbf{F = 75{,}000\ lb \times \frac{1\ N}{0.225\ lb} = 333{,}000\ N}$$

$$p_2 = \frac{p_1 A_1}{A_2} = \frac{859{,}000\,Pa \times 129\,cm^2}{6.45\,cm^2} = 17{,}180{,}000\,Pa = 17.18\,MPa$$

$$p_3 = p_2 = 17.18\ MPa$$

$$A_3 = \frac{F}{p_3} = \frac{333{,}000\,N}{17.18 \times 10^6\ N/m^2} = \underline{0.0194\ m^2}$$

**3-27.** $$p_2 = \frac{p_1 A_1}{A_2} = \frac{1\,MPa \times 0.02\,m^2}{0.001\,m^2} = 20\,MPa$$

$$p_3 = p_2 = 20\ MPa$$

$$A_3 = \frac{F}{p_3} = \frac{300{,}000\,N}{20 \times 10^6\ N/m^2} = \underline{0.015\ m^2}$$

**3-28.** $H = \dfrac{p}{\gamma} = \dfrac{5 \times 144 \, lb/ft^2}{62.4 \times 0.9 \, lb/ft^3} = \underline{12.8 \; ft}$

**3-29.** $p = -\gamma H = -(62.4 \times 0.9) \, lb/ft^3 \times 4 \; ft = -225 \; psfg = \underline{-1.56 \; psig}$

**Frictional losses and changes in kinetic energy would cause the pressure at the pump inlet to increase negatively (greater suction pressure) because pressure energy decreases per Bernoulli's Equation. This would increase the chances for having pump cavitation because the pump inlet pressure more closely approaches the vapor pressure of the fluid (usually about 5 psi suction or -5 psig) allowing for the formation and collapse of vapor bubbles.**

**3-30.** $p = \dfrac{F_1}{A_1} = \dfrac{2000}{\frac{\pi}{4}(3)^2} = 283 \, psi$

$F_2 = p \times A_2 = 283 \times \dfrac{\pi}{4}(1)^2 = 222 \; lb$

**F(16) = 222(1) so F = <u>13.9 lb</u>**

$A_1S_1 = A_2S_2 \quad \text{so} \quad S_1 = \dfrac{A_2}{A_1} \times S_2 = \left(\dfrac{1}{3}\right)^2 \times \dfrac{5}{16} = \underline{0.0347 \text{ in.}}$

**3-31.** $F_1 = 2000 \text{ lb} \times \dfrac{1 \text{ N}}{0.225 \text{ lb}} = 8890 \text{ N}$

$A_1 = \dfrac{\pi}{4}(3 \text{ in.})^2 \times \left(\dfrac{1 \text{ m}}{39.4 \text{ in.}}\right)^2 = 0.00455 \text{ m}^2$

$$p = \frac{F_1}{A_1} = \frac{8890\ N}{0.00455\ m^2} = 1.95\ MPa$$

$$F_2 = pA_2 \quad where \quad A_2 = \frac{\pi}{4}(1\,in)^2 \times \left(\frac{1\,m}{39.4\,in}\right)^2 = 0.000506\ m^2$$

$$\mathbf{F_2 = (1.95 \times 10^6) \times (506 \times 10^{-6}) = 987\ N}$$

$$\mathbf{F = 987 \times \frac{1}{16} = \underline{61.7\ N}}$$

$$\mathbf{A_1S_1 = A_2S_2 \quad where \quad S_2 = \frac{5}{16}\,in \times \frac{1\ m}{39.4\ in} = 0.00793\ m}$$

$$\mathbf{S_1 = \frac{A_2}{A_1} \times S_2 = \frac{1}{9} \times 0.793\ cm = \underline{0.0881\ cm}}$$

**3-32.** **First calculate the lever force applied to the small diameter piston.**

$$F_{small\ piston} = 100\,N \times \frac{500\,mm}{250\,mm} = 200\,N$$

**Next find the resulting oil pressure produced.**

$$p_{oil} = \frac{F_{small\ piston}}{A_{small\ piston}} = \frac{200\,N}{\frac{\pi}{4}(0.050\,m)^2} = 102{,}000\ Pa$$

**Finally the clamping force is found.**

$$F_{clamping} = p_{oil}\ A_{large\ piston} = 102{,}000\frac{N}{m^2} \times \frac{\pi}{4}(0.100\,m)^2 = \underline{\mathbf{800\ N}}$$

**3-33.** $Q = Av = \frac{\pi}{4}\left(\frac{1}{12}\ ft\right)^2 \times 10\ ft/s = 0.0545\ ft^3/s = 0.0545 \times 449 = \underline{24.5\ gpm}$

**3-34**

$$Q = Av = \frac{\pi}{4} D^2 v$$

$$D = \sqrt{\frac{4Q}{\pi\ v}} = \sqrt{\frac{4\left(20/449\right) ft^3/s}{\pi \left(15\ ft/s\right)}} = 0.0615\ ft = 0.738\ in.$$

**3-35. Metric data are:**

$$\text{Velocity} = 15\ \frac{ft}{s} \times \frac{1\ m}{3.28\ ft} = 4.57\ \frac{m}{s}$$

$$\text{Flow rate}\left(m^3/s\right) = 0.0000632\ Q(gpm)$$

$$= 0.0000632 \times 20 = 0.001264\ m^3/s$$

$$\frac{\pi}{4} D^2 = A = \frac{Q}{v} = \frac{0.001264\ m^3/s}{4.57\ m/s} = 0.0002766\ m^2$$

$$D = \sqrt{\frac{4 \times 0.0002766}{\pi}} = \underline{0.0188\ m}$$

**3-36.** $$v = \frac{Q}{A} = \frac{0.040\ m^3/\text{min}}{\frac{\pi}{4}(0.025 m)^2} = 81.5\ m/\text{min} = 1.36\ m/s$$

**3-37.** $$v\left(m/s\right) = \frac{Q\left(m^3/s\right)}{A\left(m^2\right)} = \frac{Q\left(m^3/s\right)}{\frac{\pi}{4}\left[D\left(m\right)\right]^2} = \frac{C\ Q\left(m^3/s\right)}{\left[D\left(m\right)\right]^2}$$

$$\text{Therefore } C = \frac{4}{\pi} = \underline{1.273}$$

$$v\left(\frac{m}{s}\right) = \frac{1.273 \times 0.001896}{(0.0254)^2} = \underline{3.74\ \frac{m}{s}}$$

Velocity value agrees with that of Example 3-17.

3-38. $$Q\left(\frac{m^3}{s}\right) = A\left(m^2\right) \times v\left(\frac{m}{s}\right) = \frac{\pi}{4}(0.10)^2 \times 3 = \underline{0.0236\ \frac{m^3}{s}}$$

3-39. $$v = \frac{Q}{A} = \frac{\left(\frac{20}{449}\right)\frac{ft^3}{s}}{\frac{\pi}{4}\left(\frac{4}{12}\ ft\right)^2} = 0.510\ \frac{ft}{s}$$

$$t = \frac{L}{v} = \frac{\left(\frac{20}{12}\right)ft}{0.510\ \frac{ft}{s}} = \underline{3.27\ s}$$

3-40. $$v = \frac{Q}{A} = \frac{20\ \frac{gal}{min} \times \frac{231\ in^3}{1\ gal} \times \frac{1\ min}{60\ s}}{\frac{\pi}{4}\left(4^2 - 2^2\right)in^2} = \frac{77\ \frac{in^3}{s}}{9.42\ in^2} = 8.17\ \frac{in}{s}$$

$$t = \frac{L}{v} = \frac{20\ in.}{8.17\ \frac{in}{s}} = \underline{2.45\ s}$$

3-41. $$\frac{V}{min} = A\ L\ N = \frac{\pi}{4}\left(3^2 - 2^2\right)in^2 \times (2 \times 20)in \times \frac{60}{min} = 4710\ \frac{in^3}{min}$$

$$Q = \frac{V}{min} = 4710\ \frac{in^3}{min} \times \frac{1\ gal}{231\ in^3} = \underline{20.4\ gpm}$$

**3-42.** Q = ALN Substituting values we have:

$$0.030\,\frac{m^3}{min} = \frac{\pi}{4}(0.08)^2\ m^2 \times 0.35\ m \times N\left(\frac{cycles}{min}\right) + \frac{\pi}{4}\left(0.08^2 - 0.03^2\right)\ m^2 \times 0.35\ m \times N\left(\frac{cycles}{min}\right)$$

$$0.030 = (0.00176 + 0.00151) \times N \quad so \quad N = \underline{9.2\ cycles/min}$$

**3-43.** $Q_1 = A_1v_1 = \frac{\pi}{4}\left(0.10\ m\right)^2 \times 5\ m/s = 0.0393\ m^3/s$

$$Q_2 = Q_3 = \frac{Q_1}{2} = 0.0197\,\frac{m^3}{s}$$

$$v_2 = \frac{Q_2}{A_2} = \frac{0.0197\ m^3/s}{\frac{\pi}{4}(0.07)^2\ m^2} = \underline{5.12\ m/s}$$

$$v_3 = \frac{Q_3}{A_3} = \frac{0.0197\ m^3/s}{\frac{\pi}{4}(0.06)^2\ m^2} = \underline{6.97\ m/s}$$

**3-44.** $Power(kW) = P(kPa) \times Q\left(m^3/s\right) = 10{,}000 \times \frac{0.050}{60} = \underline{8.33\ kW}$

**3-45.** $HP = \frac{P\left(psi\right) \times Q\left(gpm\right)}{1714}$ Substituting values we have:

$5 = \frac{1000 \times Q\left(gpm\right)}{1714}$ *Hence* $Q = \underline{8.57\ gpm}$

**3-46.** (a) $A = \frac{F_{load}}{p} = \frac{10{,}000\ lb}{1{,}000\ lb/in^2} = \underline{10\ in^2}$

(b) $Q\left(\frac{ft^3}{s}\right) = \frac{A\left(ft^2\right) \times S(ft)}{t(s)} = \frac{\frac{10}{144} \times 8}{8} = 0.0694\ \frac{ft^3}{s}$

$Q(gpm) = 449\ Q\left(\frac{ft^3}{s}\right) = 449 \times 0.0694 = \underline{31.1\ gpm}$

(c) $HHP = \frac{1000 \times 31.1}{1714} = \underline{18.2\ hp}$

(d) $OHP = HHP \times \frac{\eta}{100} = 18.2 \times \frac{100}{100} = \underline{18.2\ hp}$

(e) $A = \frac{F_{load} + F_{friction}}{p} = \frac{10{,}000 + 100}{1000} = \underline{10.1\ in.^2}$

$Q_{actual} = Q_{theoretical} + Q_{leakage} = \frac{10.1}{10} \times 31.1 + 0.3 = \underline{31.7\ gpm}$

$HHP = \frac{1000 \times 31.7}{1714} = \underline{18.5\ hp}$

$OHP = \frac{F(lb) \times v\left(ft/s\right)}{550} = \frac{10{,}000 \times 1}{550} = \underline{18.2\ hp}$

$\eta = \frac{OHP}{HHP} \times 100 = \frac{18.2}{18.5} \times 100 = \underline{98.4\%}$

**3-47.** $H_p(m) = \frac{Pump\ \ Power(W)}{\gamma\left(N/m^3\right) \times Q\left(m^3/s\right)}$ where $\gamma = \gamma_{water} \times SG = 9797(SG)$

Thus $H_p(m) = \frac{1000 \times Pump\ \ Power(kW)}{9797(SG) \times Q\left(m^3/s\right)} = \frac{0.1021 \times Pump\ \ Power(kW)}{Q\left(m^3/s\right) \times (SG)}$

**3-48.** $HP = \dfrac{F(lb) \times v(ft/s)}{550}$

Therefore $v = \dfrac{550 \times HP}{F(lb)} = \dfrac{550 \times 10}{5000} = \underline{1.1\ ft/s}$

**3-49.** $Power(kW) = F(kN) \times v(m/s)$

Thus $v(m/s) = \dfrac{Power(kW)}{F(kN)} = \dfrac{10}{20} = \underline{0.5\ m/s}$

**3-50.** $Flowrate(m^3/s) = \dfrac{Power(kW)}{Pressure(kPa)} = \dfrac{5}{10{,}000} = \underline{0.0005\ m^3/s}$

**3-51.** (a) $A = \dfrac{F}{p} = \dfrac{40{,}000\ N}{10 \times 10^6\ N/m^2} = \underline{0.004\ m^2}$

(b) $Q = A\,v = 0.004\ m^2 \times \dfrac{3\ m}{8\ s} = \underline{0.0015\ m^3/s}$

(c) $Power = p\,Q = (10 \times 10^3\ kPa) \times (0.0015\ m^3/s) = \underline{15\ kW}$

(d) $output\ power = hydraulic\ power \times \dfrac{\eta}{100} = 15\,kW \times \dfrac{100}{100} = \underline{15\,kw}$

(e) $A = \dfrac{F_{load} + F_{friction}}{p} = \dfrac{40{,}000 + 400}{10 \times 10^6} = \underline{0.00404\ m^2}$

$$Q_{actual} = Q_{theoretical} + Q_{leakage} = \frac{0.00404}{0.004} \times 0.0015 + \frac{0.001}{60}$$

$$= 0.001515 + 0.0000167 = \underline{0.00153\ m^3/s}$$

$$hydraulic\ \ power = p\ Q_{act} = 10 \times 10^3 \times 0.00153 = \underline{15.3\ \ kW}$$

$$output\ power = F(kN) \times v(m/s) = 40 \times \frac{3}{8} = \underline{15\ kW}$$

$$\eta = \frac{output\ power}{hydraulic\ power} \times 100 = \frac{15}{15.3} \times 100 = \underline{98.0\%}$$

**3-52.** (a) $W = F\ L = 3500\ lb \times 7\ ft = 24{,}500\ ft \bullet lb$

(b) $$p = \frac{F}{A} = \frac{3500\ lb}{\frac{\pi}{4}(8)^2\ in^2} = \underline{69.7\ psi}$$

(c) $$Power = \frac{24{,}500\ ft \bullet lb}{10\ s} = 2450\ \frac{ft \bullet lb}{s} = \frac{2450}{550}\ HP = \underline{4.45\ HP}$$

(d) $$v = \frac{Q}{A} = \frac{10\ \frac{gal}{min} \times \frac{231\ in^3}{1\ gal} \times \frac{1\ min}{60\ s}}{\frac{\pi}{4}\left(8^2 - 4^2\right)in^2} = \underline{1.02\ in/s}$$

(e) $$Q = A\ v = \frac{\pi}{4}\left(8^2 - 4^2\right)in^2 \times \frac{7 \times 12\ in}{10\ s} \times \frac{1\ gal}{231\ in^3} \times \frac{60\ s}{1\ min}$$

$$= \underline{82.2\ gpm}$$

**3-53.** (a) $$F = 3500\ lb \times \frac{1\ N}{0.225\ lb} = 15{,}600\ N$$

$$L = 7\ ft \times \frac{1\ m}{3.28\ ft} = 2.13\ m$$

$$W = F\ L = 15{,}600\ N \times 2.13\ m = \underline{33{,}200\ N \bullet m}$$

(b) $A = \frac{\pi}{4}(8)^2 \text{ in}^2 = 50.2 \text{ in}^2 = 50.2 \text{ in}^2 \times \left(\frac{1 \text{ m}}{39.4 \text{ in.}}\right)^2 = 0.0324 \text{ m}^2$

$$p = \frac{F}{A} = \frac{15{,}600\ N}{0.0324\ m^2} = 481{,}000\ N/m^2 = \underline{481\ kPa}$$

(c) $\text{Power} = \frac{33{,}200 \text{ N} \bullet \text{m}}{10 \text{ s}} = 3320 \frac{\text{N} \bullet \text{m}}{\text{s}} = 3320 \text{ W} = \underline{3.32 \text{ kW}}$

(d) $Q = 10 \frac{\text{gal}}{\text{min}} \times \frac{231 \text{ in}^3}{1 \text{ gal}} \times \left(\frac{1 \text{ m}}{39.4 \text{ in}}\right)^3 \times \frac{1 \text{ min}}{60 \text{ s}} = 0.000629 \text{ m}^3/\text{s}$

$$v = \frac{Q}{A} = \frac{0.000629 \text{ m}^3/\text{s}}{\frac{\pi}{4}(8^2 - 4^2)\text{in}^2 \times \left(\frac{1 \text{ m}}{39.4 \text{ in}}\right)^2} = \frac{0.000629 \text{ m}^3/\text{s}}{0.0243 \text{ m}^2}$$

$$= \underline{0.0259 \text{ m/s}}$$

(e) $Q = A\,v = 0.0243 \text{ m}^2 \times \left(\frac{2.13 \text{ m}}{10 \text{ s}}\right) = \underline{0.00518 \text{ m}^3/\text{s}}$

**3-54.** $Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$

$$v_1 = \frac{Q}{\frac{\pi}{4}D_1^2} = \frac{(100/449)\ ft^3/s}{\frac{\pi}{4}\left(\frac{2}{12}\ ft\right)^2} = 10.2\ \frac{ft}{s}$$

$$v_2 = \frac{(100/449)}{\frac{\pi}{4}\left(\frac{1}{12}\ ft\right)^2} = 40.8\ \frac{ft}{s}$$

$$\frac{p_1 - p_2}{\gamma} = \frac{v_2^2 - v_1^2}{2g} = \frac{40.8^2 - 10.2^2}{64.4} = \frac{1665 - 104}{64.4} = 24.2\ \ ft$$

$$p_1 - p_2 = 24.2 \times 62.4 \times 0.9 = 1359\ lb/ft^2 = 9.4\ \ psi$$

$p_2 = p_1 - 9.4 = 10 - 9.4 = \underline{0.6\ psig}$

**3-55.** $$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

$$p_1 = 10\ psi \times \frac{6895\,Pa}{1\ psi} = 68{,}950\ Pa$$

$$Q\left(m^3/s\right) = 0.00006309\,Q(gpm) = 0.00006309 \times 100 = 0.006309\ m^2/s$$

$$D_1 = 2\ in \times \frac{1\,ft}{12\ in} \times \frac{0.3048\,m}{1\ ft} = 0.0508\ m \qquad Thus\ \ D_2 = 0.0254\ m$$

$$v_1 = \frac{Q}{A_1} = \frac{0.006309\ m^3/s}{\frac{\pi}{4}(0.0508)^2\ m^2} = 3.113\ m/s$$

$$v_2 = \frac{Q}{A_2} = \frac{0.006309}{\frac{\pi}{4}(0.0254)^2} = 12.45\ m/s$$

$$\frac{p_1 - p_2}{\gamma} = \frac{v_2^2 - v_1^2}{2g} = \frac{12.45^2 - 3.113^2}{2 \times 9.81} = \frac{155.0 - 9.69}{2 \times 9.81} = 7.41\ m$$

$$p_1 - p_2 = 7.41 \times 9800 \times 0.9 = 65{,}360\ Pa$$

$$p_2 = p_1 - 65{,}360 = 68{,}950 - 65{,}360 = \underline{3590\ Pa\ \ gage}$$

**3-56.** $$W\,Z = 1000\,\text{gal} \times \frac{231\,\text{in}^3}{1\,\text{gal}} \times \frac{1\,\text{ft}^3}{1728\,\text{in}^3} \times 62.4\,\frac{\text{lb}}{\text{ft}^3} \times 100\,\text{ft}$$

$$= \underline{834{,}000\ \text{ft}\bullet\text{lb}}$$

**3-57.** $\frac{W v^2}{2g} = 1\,gal \times \frac{231\,in^3}{1\,gal} \times \frac{1\,ft^3}{1728\,in^3} \times 62.4\,\frac{lb}{ft^3} \times \left(20\,\frac{ft}{s}\right)^2 \times \frac{1}{64.4\,\frac{ft}{s^2}}$

$= \underline{51.8\ ft \bullet lb}$

**3-58.** $v_2 = \sqrt{2gh} = \sqrt{64.4 \times 25} = 40.1\ ft/s$

$$Q = Av = \frac{\pi}{4}\left(\frac{1}{12}\ ft\right)^2 \times 40.1\ ft/s = 0.219\ ft^3/s = 0.219 \times 449 = \underline{98.3\ gpm}$$

**3-59.** $v_2 = \sqrt{2gh}$ where $h = 25\,ft \times \frac{1\,m}{3.28\,ft} = 7.62\,m$

Thus $v_2 = \sqrt{2 \times 9.81 \times 7.62} = 12.2\ m/s$

Also $d = 1\,in \times \frac{1\,ft}{12\,in} \times \frac{1\,m}{3.28\,ft} = 0.0254\,m$

Therefore $Q = A\,v = \frac{\pi}{4} d^2\,v = \frac{\pi}{4} \times 0.0254^2 \times 12.2 = \underline{0.00618\ m^3/s}$

**3-60.** Writing Bernoulli's Equation between stations 1 and 2, we have:

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

*We have* $H_m = 0,\ v_1 = 0,\ Z_2 - Z_1 = 20,\ H_L = 40\ ft$ *and* $p_1 = 0.$

Substituting known values we have:

$$Z_1 + 0 + 0 + H_P - 0 - 40 = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

**Knowing that $Z_2 - Z_1 = 20$ ft, we have:**

$$\frac{p_2}{\gamma} = H_P - \frac{v_2^2}{2g} - 60$$ **Then using Equation 3-29 yields:**

$$H_P = \frac{3950 \times 4}{25 \times 0.9} = 702.2 \ ft$$ *Next solve for* $v_2$.

$$v_2 = \frac{Q}{A} = \frac{\left(25/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{0.75}{12} \ ft\right)^2} = 18.1 \ ft/s$$

**Substituting values we have:**

$$\frac{p_2}{\gamma} = 702.2 - \frac{18.1^2}{2 \times 32.2} - 60 = 702.2 - 5.09 - 60 = 637.1 \ ft$$

$$p_2 = 637.1 ft \times \gamma\left(\frac{lb}{ft^3}\right) = 637.1 \times 0.9 \times 62.4 = 35{,}780 \frac{lb}{ft^2}$$

**Changing to units of psi yields:** $p_2 = \frac{35{,}780}{144} = \underline{248 \ psi}$

**3-61. (a) Writing Bernoulli's Equation between stations 1 and 2 we have:**

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

*We have* $H_m = 0$, $v_1 = 0$, $Z_1 - Z_2 = 10 \ ft$, $H_L = 0$ *and* $p_1 = 10 \times 144$

$$= 1440 \frac{lb}{ft^2}$$

**Since there is no pump between stations 1 and 2, $H_P = 0$.**

**Solving for $v_2$ we have:**

$$v_2 = \frac{Q}{A_2} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{1.5}{12}\ ft\right)^2} = 5.44\ \frac{ft}{s}$$

**Substituting known values, we have:**

$$Z_1 + \frac{1440}{62.4 \times 0.9} + 0 + 0 - 0 - 0 = Z_2 + \frac{p_2}{\gamma} + \frac{5.44^2}{64.4}$$

**Knowing that $Z_1 - Z_2 = 10$ ft, we have:**

$$\frac{p_2}{\gamma} = 10 + \frac{1440}{62.4 \times 0.9} - \frac{5.44^2}{64.4} = 10 + 25.6 - 0.5 = 35.1\ ft$$

**Thus** $$p_2 = 35.1\ ft \times \frac{0.9 \times 62.4\ \frac{lb}{ft^3}}{\frac{144\ in^2}{1\ ft^2}} = \underline{13.7\ psig}$$

**(b) In this case, $H_L$ = 25 ft. Therefore we have from the previous equation:**

$$\frac{p_2}{\gamma} = 10 + 25.6 - 0.5 - 25 = 10.1\ ft$$

$$p_2 = 10.1 \times 0.9 \times \frac{62.4}{144} = \underline{3.94\ \ psig}$$

**3-62. Writing Bernoulli's Equation between stations 1 and 2 we have:**

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

*We have* $H_m = 0$, $v_1 = 0$, $Z_2 - Z_1 = 6.096\ m$, $H_L = 12.19\ m$ *and* $p_1 = 0$.

**Substituting known values we have:**

$$Z_1 + 0 + 0 + H_P - 0 - 12.19 = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

**Knowing that $Z_2 - Z_1 = 6.096$ m, we have:**

$$\frac{p_2}{\gamma} = H_P - \frac{v_2^2}{2g} - 18.29$$ **We next solve for the pump head.**

$$H_P(m) = \frac{\text{Pump Power (W)}}{\gamma\left(N/m^3\right) \times Q\left(m^3/s\right)} = \frac{2984}{(0.9 \times 9797) \times 0.00158} = 214.3\ m$$

**Next solve for $v_2$ and $\frac{v_2^2}{2g}$:**

$$v_2\left(m/s\right) = \frac{Q\left(m^3/s\right)}{A\left(m^2\right)} = \frac{0.00158}{\frac{\pi}{4}\left(0.01905\ m\right)^2} = 5.54\ m/s$$

$$\frac{v_2^2}{2g} = \frac{\left(5.54\ m/s\right)^2}{2 \times 9.81\ m/s^2} = 1.566\ m$$ **Substituting values we have.**

$$\frac{p_2}{\gamma} = 214.3 - 1.566 - 18.29 = 194.4\ m$$

$$p_2\left(\frac{N}{m^2}\right) = 194.4m \times \gamma\left(\frac{N}{m^3}\right) = 194.4m \times 8817\frac{N}{m^3} = 1{,}714{,}000\frac{N}{m^2}$$

$$= \underline{1714\ kPa}$$

**3-63. (a) Writing Bernoulli's Equation between stations 1 and 2 we have:**

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

We have $H_m = 0$, $v_1 = 0$, $Z_1 - Z_2 = 3.0489$ m, $H_L = 0$ and $p_1 = 68.97\ kPa$

Since there is no pump between stations 1 and 2, $H_P = 0$.

Solving for $v_2$ we have:

$$v_2\left(\text{m}/\text{s}\right) = \frac{Q\left(\text{m}^3/\text{s}\right)}{A\left(\text{m}^2\right)} = \frac{0.001896\ \text{m}^3/\text{s}}{\frac{\pi}{4}\left(0.0381\ \text{m}\right)^2} = 1.66\ \text{m}/\text{s}$$

Substituting known values we have:

$$Z_1 + \frac{68{,}970\ N/m^2}{8{,}817\ N/m^3} + 0 + 0 - 0 - 0 = Z_2 + \frac{p_2}{\gamma} + \frac{\left(1.66\ m/s\right)^2}{2\times 9.81\ m/s^2}$$

Knowing that $Z_1 - Z_2 = 3.048$ m, we have:

$$\frac{p_2}{\gamma} = 3.048 + \frac{68{,}970}{8{,}817} - \frac{1.66^2}{2\times 9.81} = 3.048 + 7.82 - 0.141 = 10.73\ m$$

$$p_2 = 10.73\,m \times 8817\,\frac{N}{m^3} = 94{,}610\,\frac{N}{m^2} = 94{,}610\ Pa = 94.6\ kPa$$

(b) In this case, $H_L = 7.622$ m. Therefore, we have from the previous equation:

$$\frac{p_2}{\gamma} = 10.73 - 7.622 = 3.11\,m$$

$$p_2 = 3.11\times 8817 = 27{,}400\ Pa = \underline{27.4\ kPa}$$

3-64. Per continuity equation: $Q_{in} = Q_{out} = \underline{30\ \text{gpm}}$

(a) Per Bernoulli's equation: $\underline{p_B - p_A = 0}$

**(b) Writing Bernoulli's equation we have:**

$$Z_A + \frac{p_A}{\gamma} + \frac{v_A^2}{2g} + H_P = Z_B + \frac{p_B}{\gamma} + \frac{v_B^2}{2g}$$

**where** $$H_P = \frac{3950 \times HP}{Q\,S_g} = \frac{3940 \times 2}{30 \times 0.9} = 293\text{ ft}$$

$$v_A = \frac{Q}{A} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{2}{12}\,ft\right)^2} = 3.06\ ft/s$$

$$v_B = \frac{Q}{A} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{1}{12}\,ft\right)^2} = 12.2\ ft/s$$

**Substituting values we have:**

$$\frac{p_B - p_A}{\gamma} = H_P + \frac{v_A^2 - v_B^2}{2g} = 293 + \frac{3.06^2 - 12.2^2}{2 \times 32.2} = 291\,ft$$

$$p_B - p_A = 291\gamma = 291 \times 56.2 = 16{,}350\,lb/ft^2 = \underline{114\,psi}$$

**3-65.** $Q_{in} = Q_{out} = 0.0000632\,Q(gpm) = 0.0000632 \times 30 = \underline{0.00190\ m^3/s}$

$d_A = 2\text{ in} = 0.0508\text{ m}$ **and** $d_B = 1\text{ in} = 0.0254\text{ m}$

**(a) Per Bernoulli's equation:** $\underline{p_B - p_A = 0}$

**(b) Writing Bernoulli's equation we have:**

$$Z_A + \frac{p_A}{\gamma} + \frac{v_A^2}{2g} + H_P = Z_B + \frac{p_B}{\gamma} + \frac{v_B^2}{2g}$$

$$\text{Power} = 2\,\text{HP} \times \frac{747\,\text{W}}{1\,\text{HP}} = 1494\,\text{W}$$

$$H_P(m) = \frac{\text{Pump Power(W)}}{\gamma\left(\frac{N}{m^3}\right) \times Q\left(\frac{m^3}{s}\right)} = \frac{1494}{0.9 \times 9800 \times 0.00190} = 89.2\,\text{m}$$

$$v_A = \frac{Q}{A} = \frac{0.00190}{\frac{\pi}{4}(0.0508)^2} = 0.937\ {}^{m}\!/_{s}$$

and $$v_B = \frac{0.00190}{\frac{\pi}{4}(0.0254)^2} = 3.75\ {}^{m}\!/_{s}$$

**Substituting values we have:**

$$\frac{p_B - p_A}{\gamma} = H_P - \frac{v_B^2 - v_A^2}{2g} = 89.2 - \frac{3.75^2 - 0.937^2}{2 \times 9.81} = 88.5\ m$$

$$p_B - p_A = 88.5\gamma = 88.5 \times 9800 \times 0.90 = 781{,}000\ Pa = \underline{781\ kPa}$$

# Chapter 4

## Frictional Losses in Hydraulic Pipelines

**4-1.** It is very important to keep all energy losses in a fluid power system to a minimum acceptable level.

**4-2.** Laminar flow is characterized by the fluid flowing in smooth layers. In turbulent flow, the movements of a particle becomes random and fluctuate up and down in a direction perpendicular as well as parallel to the mean flow direction. This causes a mixing motion as particles collide.

**4-3.**
1. If $N_R$ is less than 2000, the flow is laminar.
2. If $N_R$ is greater than 4000, the flow is turbulent.
3. Reynolds numbers between 2000 and 4000 cover a transition region between laminar and turbulent flow.

**4-4.** Relative roughness is defined as the pipe inside surface roughness divided by the pipe inside diameter.

**4-5.** The K factor equals the head loss divided by the velocity head.

**4-6.** The equivalent length of a valve or fitting is that length of pipe which, for the same flow rate, produces the same head loss as the valve or fitting.

**4-7.** $\Delta P \propto K$, true

**4-8.** High velocity and large pipe roughness.

4-9. $N_R = \dfrac{7740 \times v\left(ft/s\right) \times D(in)}{\nu(cS)} = \dfrac{7740 \times 20 \times 1.5}{75} = \underline{3096}$

4-10. $N_R = \dfrac{v\,D}{\nu} = \dfrac{6\ m/s \times 0.030\ m}{0.0001\ m^2/s} = \underline{1800}$

Since $N_R$ is less than 2000, <u>the flow is laminar.</u>

4-11. $N_R = \dfrac{v\,D}{\nu}$ , <u>increase</u>

4-12. Assuming laminar flow we have:

$$H_L = \frac{64}{N_R} \times \frac{L}{D} \times \frac{v^2}{2g} = \frac{64}{3096} \times \frac{100}{1.5/12} \times \frac{20^2}{64.4} = 102.7\ ft$$

$$\Delta P = \gamma H_L = (62.4 \times 0.9) \times 102.7 = 5767\ psf = \underline{40.0\ psi}$$

If turbulent flow with smooth pipe we have:

f = 0.044, $\Delta p = \underline{85.0\ psi}$

4-13. (a) $N_R = \dfrac{7740 \times 15 \times 0.75}{100} = 871$ laminar flow

Thus $f = 64/N_R = 64/871 = \underline{0.0735}$

(b) $N_R = \dfrac{7740 \times 45 \times 0.75}{100} = 2612$

We therefore assume the flow to be turbulent and must calculate the relative roughness of the pipe.

$\varepsilon$ (from Figure 4-8) = 0.0018 in. Thus the relative roughness can now be found.

$$\frac{\varepsilon}{D} = \frac{0.0018}{0.75} = 0.0024$$

From the Moody Diagram (Figure 4-9): $f_{turbulent} = \underline{0.046}$

If the flow is laminar, the friction factor is:

$$f_{laminar} = \frac{64}{N_R} = \frac{64}{2612} = \underline{0.0245}$$

4-14. $$H_L = \frac{64}{N_R} \times \frac{L}{D} \times \frac{v^2}{2g} = \frac{64}{1800} \times \frac{100\ m}{0.030\ m} \times \frac{\left(6\ m/s\right)^2}{2 \times 9.81\ m/s^2} = 217.5\ m$$

$$\Delta p_L = \gamma H_L = (1000\ \frac{kg}{m^2} \times 0.90 \times 9.81\ \frac{m}{s^2}) \times 217.5\ m$$

$$= 1.92\frac{MN}{m^2} = 1.92\ MPa = \underline{19.2\ bars}$$

4-15. (a) $$N_R = \frac{v D}{\nu} = \frac{2 \times 0.020}{0.0001} = 400 \text{ laminar}$$

$$f = \frac{64}{N_R} = \frac{64}{400} = \underline{0.16}$$

(b) $$N_R = \frac{10 \times 0.020}{0.0001} = 2000 \text{ laminar}$$

$$f = \frac{64}{2000} = \underline{0.032}$$

**4-16.** $$\frac{\Delta p}{\rho g} = H_L = \frac{64}{N_R} \times \frac{L}{D} \times \frac{v^2}{2g} = \frac{64}{\frac{vD\rho}{\mu}} \times \frac{L}{D} \times \frac{v^2}{2g}$$

$$\frac{\Delta p}{\rho g} = H_L = \frac{32\mu Lv}{\rho g D^2} \quad so \quad \Delta p = \frac{32\mu Lv}{D^2}$$

**Hence for laminar flow, $\Delta p$ is proportional to v.**

**4-17.** $$\frac{\Delta p}{\rho g} = H_L = f \times \frac{L}{D} \times \frac{v^2}{2g} \quad so \quad \Delta p = \frac{\rho f L v^2}{2D}$$

**Thus $\Delta p$ varies as the square of the velocity (provided the flow is fully turbulent where f is a constant).**

**4-18.** $H_L = K \times \frac{v^2}{2g}$ **First calculate the velocity.**

$$v = \frac{Q}{A} = \frac{\left(100/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{1.5}{12} ft\right)^2} = 18.1 \ ft/s$$

**From Figure 4-10, K for a wide open gate valve is 0.19.**

$$H_L = 0.19 \times \frac{18.1^2}{64.4} = \underline{0.97 \text{ ft of oil}}$$

**4-19.** $H_L = K \times \frac{v^2}{2g}$ **and K = 0.19**

$$v = \frac{Q}{A} = \frac{0.004 \ m^3/s}{\frac{\pi}{4}\left(0.030 \ m\right)^2} = 5.66 \ m/s$$

Thus $H_L = 0.19 \times \dfrac{5.66^2}{2 \times 9.81} = 0.31$ m and the pressure loss is:

$$\Delta p_L = \gamma\, H_L = (1000 \times 0.90 \times 9.81) \times 0.31 = 2740\ {}^{N}\!/_{m^2} = \underline{0.0274\ bars}$$

**4-20.** $\Delta p \propto Q^2, \quad \underline{factor\ \ is\ \ 4}$

**4-21.** For a given opening position, a valve behaves as an orifice. Thus use Eq. 8-1 (See Section 8.4) for flow through an orifice, to determine the flow coefficient.

$Q = 38.1 CA\sqrt{\dfrac{\Delta p}{SG}}$ Substituting values we have:

$60 = 38.06\, C \times 0.5\sqrt{\dfrac{40}{0.90}}$ Thus $C = \underline{0.473}$

To determine the K factor use Eq. 4-8.

$$\frac{\Delta p}{\gamma} = H_L = K \times \frac{v^2}{2g} \quad so \quad K = \frac{2g\Delta p}{\gamma v^2}$$

$where \quad g = 32.2\ {}^{ft}\!/_{s^2},\ \ \Delta p = 40 \times 144 = 5760\ {}^{lb}\!/_{ft^2} \quad and$

$\gamma = 0.9 \times 62.4 = 56.2\ {}^{lb}\!/_{ft^3}$

$$v = \frac{Q}{A} = \frac{\left(60/449\right) {}^{ft^3}\!/_{s}}{\left(0.5/144\right) ft^2} = 38.6\ {}^{ft}\!/_{s}$$

Thus $K = \dfrac{2 \times 32.2 \times 5760}{56.2 \times 38.6^2} = \underline{4.43}$

**4-22.** **The flow coefficient and K factor values would be the same because these two parameters are dimensionless.**

**4-23.** **First find the velocity.**

$$v = \frac{Q}{A} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{0.75}{12}\,ft\right)^2} = 21.8\ ft/s$$

**Next find the Reynolds Number.**

$$N_R = \frac{7740 \times 21.8 \times 0.75}{75} = 1687$$

**Then find the friction factor.**

$$f = \frac{64}{N_R} = \frac{64}{1687} = 0.0379$$

**Finally we calculate the equivalent length where the K factor equals 0.19 from Figure 4-10.**

$$L_e = \frac{K\,D}{f} = \frac{0.19 \times \frac{0.75}{12}}{0.0379} = \underline{0.313\ ft}$$

**4-24.** $N_R = \frac{v\,D}{\nu}$ **where** $v = \frac{Q}{A} = \frac{0.002}{\frac{\pi}{4} \times 0.020^2} = 6.37\ m/s$

$$N_R = \frac{6.37 \times 0.020}{0.0001} = 1274 \quad \text{and} \quad f = \frac{64}{1274} = 0.0502$$

**Thus** $L_e = \frac{K\,D}{f} = \frac{0.19 \times 0.020}{0.0502} = \underline{0.076\ m}$

**4-25. For the system of Figure 4-20, we have the following data:**

$H_m = 0$ *between stations* 1 *and* 2, $v_1 = 0$, $\frac{p_1}{\gamma} = 0$ *and*
$Z_2 - Z_1 = 20$ *ft.*

**Writing Bernoulli's equation between stations 1 and 2, we have:**

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

**Let's first solve for** $v_2$: $v_2 = \frac{Q}{A} = \frac{\left(25/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{0.75}{12} ft\right)^2} = 18.1\ ft/s$

**The velocity head at station 2 is:** $\frac{v_2^2}{2\,g} = \frac{18.1^2}{64.4} = 5.09\ ft$

**Reynolds Number can now be found.**

$$N_R = \frac{7740 \times v\left(ft/s\right) \times D(in)}{\nu(cS)} = \frac{7740 \times 18.1 \times 0.75}{75} = 1400$$

**The flow is laminar. Thus** $f = \frac{64}{N_R} = \frac{64}{1400} = 0.0457$

**We can now find the head loss due to friction between stations 1 and 2.**

$H_L = f \times \frac{L}{D} \times \frac{v^2}{2g}$ **where L is found as follows:**

$$L = 16 + 1 + 4 + \left(\frac{K\ D}{f}\right)_{std\ elbow} = 21 + \frac{0.9 \times \frac{0.75}{12}}{0.0457} = 22.2\ ft$$

$$H_L = 0.0457 \times \frac{22.2}{0.75/12} \times 5.09 = 82.6 \text{ ft}$$

**We can now substitute values into Bernoulli's equation to solve for $p_2/\gamma$.**

$$\frac{p_2}{\gamma} = (Z_1 - Z_2) + H_P + \frac{p_1}{\gamma} - H_L - \frac{v_2^2}{2g}$$

$$\frac{p_2}{\gamma} = -20 + H_P + 0 - 82.6 - 5.09 = H_P - 107.7$$

**Using Equation 3-29 allows us to solve for the pump head.**

$$H_P = \frac{3950 \times (HHP)}{Q(gpm) \times (SG)} = \frac{3950 \times 4}{25 \times 0.9} = 702.2 \ \ ft$$

**Thus we can solve for the pressure head at station 2.**

$$\frac{p_2}{\gamma} = 702.2 - 107.7 = 594.5 \ \ ft \ of \ \ oil = H \ \ ft \ of \ \ oil$$

**Finally we solve for the pressure at station 2.**

$$p_2 = \gamma H = (62.4 \times 0.9) lb/ft^3 \times 594.5 \ ft \times \frac{1 \ psi}{144 \ psf} = \underline{231.7 \, psi}$$

**4-26. For the system of Figure 4-21, we have the following data:**

**$H_m = 0$, $v_1 = 0$, $Z_1 - Z_2 = 10$ ft and $H_P = 0$ between stations 1 and 2.**

**Writing Bernoulli's equation between stations 1 and 2, we have:**

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

**Let's first solve for** $v_2$: $v_2 = \frac{Q}{A} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{1.5}{12}\ ft\right)^2} = 5.44\frac{ft}{s}$

**The velocity head at station 2 is:** $\frac{v_2^2}{2g} = \frac{5.44^2}{64.4} = 0.46\ ft$

**The pressure head at station 1 is:** $\frac{p_1}{\gamma} = \frac{10\times144}{62.4\times0.9} = 25.6\ ft$

**The Reynolds Number can now be found.**

$$N_R = \frac{7740\ v\left(ft/s\right) \times D(in)}{\nu(cS)} = \frac{7740 \times 5.44 \times 1.5}{100} = 632$$

**The flow is laminar so the friction factor is:**

$$f = \frac{64}{N_R} = \frac{64}{632} = 0.101$$

**We can now find the head loss due to friction between stations 1 and 2.**

$$H_L = f \times \frac{L}{D} \times \frac{v^2}{2g} + \text{head loss across the strainer}$$

**where** $L = 20 + 3 \times \left(\frac{K\ D}{f}\right)_{\text{std elbow}} = 20 + 3 \times \frac{0.9 \times 1.5/12}{0.101} = 23.3\ ft$

**Head loss across strainer** $= \frac{(\Delta p)_{strainer}}{0.433(SG)} = \frac{1}{0.433\times0.9} = 2.6\ ft$

$$H_L = 0.101 \times \frac{23.3}{1.5/12} \times 0.46 + 2.6 = 8.6 + 2.6 = 11.2\ ft$$

**We can now substitute into Bernoulli' equation to solve for** $p_2/\gamma$.

$$\frac{p_2}{\gamma} = (Z_1 - Z_2) + \frac{p_1}{\gamma} - H_L - \frac{v_2^2}{2g}$$
$$= 10 + 25.6 - 11.2 - 0.46 = 23.9 \; ft \; of \; oil \; = \; H \; ft \; of \; oil$$

**Finally we can solve for the pressure at station 2.**

$$p_2 = \gamma H = (62.4 \times 0.9)lb/ft^3 \times 23.9 \; ft \times \frac{1 \; psi}{144 \; psf} = \underline{9.31 psi}$$

**4-27. For the system of Figure 4-20, we have the following data:**

$$H_m = 0 \; between \; stations \; 1 \; and \; 2, \quad v_1 = 0, \quad \frac{p_1}{\gamma} = 0 \quad and$$
$$Z_2 - Z_1 = 6.096 \; m$$

**Writing Bernoulli's equation between stations 1 and 2 we have:**

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

**Let's first solve for** $v_2$: $v_2 = \dfrac{0.00158}{\frac{\pi}{4} \times 0.01905^2} = 5.54 \; m/s$

**The velocity head at station 2 is:** $\dfrac{v_2^2}{2g} = \dfrac{5.54^2}{2 \times 9.81} = 1.57 \; m$

**The Reynolds Number can now be found where the kinematic viscosity is:**

$$\nu \left(m^2/s\right) = \frac{\nu \; (cS)}{1{,}000{,}000} = \frac{75}{1{,}000{,}000} = 75 \times 10^{-6} \; m^2/s$$

$$N_R = \frac{v\left(\frac{m}{s}\right) \times D(m)}{\nu\left(\frac{m^2}{s}\right)} = \frac{5.54 \times 0.01905}{75 \times 10^{-6}} = 1400$$

**The flow is laminar so the friction depends only on $N_R$.**

**Therefore** $f = \frac{64}{N_R} = \frac{64}{1400} = 0.0457$

**We can now find the head loss due to friction between stations 1 and 2.**

$H_L = f \times \frac{L}{D} \times \frac{v^2}{2g}$ **where**

$$L = 4.88 + 0.305 + 1.22 + \left(\frac{K\,D}{f}\right)_{std\ elbow} = 6.41 + \frac{0.9 \times 0.01905}{0.4057}$$

$$= 6.79\ m$$

$$H_L = 0.0457 \times \frac{6.79}{0.01905} \times 1.57 = 25.6\ m$$

**Next use Bernoulli's equation to solve for** $\frac{P_2}{\gamma}$.

$$\frac{p_2}{\gamma} = (Z_1 - Z_2) + H_P + \frac{p_1}{\gamma} - \frac{v_2^2}{2g} - H_L = -6.096 + H_P + 0 - 25.6 - 1.57$$

$$= H_P - 33.2$$

**Solving for the pump head we have:**

$$H_P(m) = \frac{\text{Pump Power(Watts)}}{\gamma\left(\frac{N}{m^3}\right) \times Q\left(\frac{m^3}{s}\right)} = \frac{2984}{8817 \times 0.00158} = 214.3\ m$$

**Next we solve for the pressure head at station 2.**

$$\frac{p_2}{\gamma} = 214.3 - 33.2 = 181.1\ m\ of\ oil$$

**Finally we solve for the pressure at station 2.**

$$p_2 = 181.1\ m \times 8817\ {}^{N}\!/_{m^3} = 1{,}600{,}000\ Pa = \underline{1600\ \ kPa}$$

4-28. For the system of Figure 4-21 we have the following data:

$H_m = 0$, $v_1 = 0$, $Z_1 - Z_2 = 3.048$ m and $H_P = 0$ between stations 1 and 2.

Writing Bernoulli's equation stations 1 and 2, we have:

$$Z_1 + \frac{p_1}{\gamma} + \frac{v_1^2}{2g} + H_P - H_m - H_L = Z_2 + \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

Let's first solve for $v_2$: $v_2 = \dfrac{0.001896}{\frac{\pi}{4} \times 0.0381^2} = 1.66\ {}^{m}\!/_{s}$

The velocity head at station 2 is: $\dfrac{v_2^2}{2g} = \dfrac{1.66^2}{2 \times 9.81} = 0.141$ m

The pressure head at station 1 is: $\dfrac{P_1}{\gamma} = \dfrac{68{,}970}{8817} = 7.82$ m

The Reynolds Number can now be found:

$$N_R = \frac{v\left({}^{m}\!/_{s}\right) \times D(m)}{\nu\left({}^{m^2}\!/_{s}\right)} = \frac{1.66 \times 0.0381}{100/1{,}000{,}000} = 632$$

The flow is laminar so $f = \dfrac{64}{N_R} = \dfrac{64}{632} = 0.101$

We can now find the head loss due to friction between stations 1 and 2.

$$H_L = f \times \frac{L}{D} \times \frac{v^2}{2g} + \text{head loss across strainer}$$

where $L = 6.097 + 3\left(\frac{K\,D}{f}\right)_{std\ elbow} = 6.097 + 3 \times \frac{0.9 \times 0.0381}{0.101}$

$= 7.12\ m$

Head loss across strainer $= \frac{(\Delta p)_{strainer}}{\gamma} = \frac{6897}{8817} = 0.782\ m$

$H_L = 0.101 \times \frac{7.12}{0.0381} \times 0.141 + 0.782 = 3.44\ m$

Next use Bernoulli's equation to solve for $\frac{p_2}{\gamma}$.

$$\frac{p_2}{\gamma} = (Z_1 - Z_2) + \frac{p_1}{\gamma} - H_L - \frac{v_2^2}{2g} = 3.048 + 7.82 - 3.44 - 0.141$$
$$= 7.29\ m\ of\ oil$$

Finally we can solve for the pressure at station 2.

$p_2 = 7.29 \times 8817 = 64{,}300\ Pa = \underline{64.3\ kPa}$

**4-29.** $v = \frac{Q}{A} = \frac{\left(40/449\right)\ ft^3/s}{\frac{\pi}{4}\left(\frac{1}{12}\ ft\right)^2} = 16.3\ ft/s$

$N_R = \frac{7740\,v(ft/s) \times D(in)}{\nu(cS)} = \frac{7740 \times 16.3 \times 1}{100} = 1262$

$f = \frac{64}{N_R} = \frac{64}{1262} = 0.0507$

$L_{e\,T} = L_{pipe} + \left(\frac{K\,D}{f}\right)_{valve} = 50 + \frac{10 \times 1/12}{0.0507} = 50 + 16.4 = 66.4\ ft$

$H_{L\,T} = f \times \frac{L_{e\,T}}{D} \times \frac{v^2}{2g} = 0.0507 \times \frac{66.4}{1/12} \times \frac{16.3^2}{64.4} = 167\ ft\ of\ oil$

$$(\Delta p) = p_1 - p_2 = (62.4 \times 0.9)\,{}^{lb}\!/_{ft^3} \times 167\ ft \times \frac{1\ psi}{144\ psf} = 65.1\ psi$$

$$p_2 - p_1 = \underline{-65.1\ psi}$$

**4-30.** $$v = \frac{Q}{A} = \frac{0.0025}{\frac{\pi}{4} \times 0.025^2} = 5.09\ {}^{m}\!/_{s}$$

$$N_R = \frac{5.09 \times 0.025}{0.0001} = 1272$$

$$f = \frac{64}{1272} = 0.0503$$

$$L_{eT} = 16 + \frac{10 \times 0.025}{0.0503} = 16 + 5.0 = 21.0\ m$$

$$H_{LT} = 0.0503 \times \frac{21.0}{0.025} \times \frac{5.09^2}{2 \times 9.81} = 55.8\ m\ \text{of oil}$$

$$\Delta p = \gamma H_{LT} = (1000 \times 0.9 \times 9.81) \times 55.8 = 493{,}000\ {}^{N}\!/_{m^2}$$

$$p_2 - p_1 = -(\Delta p) = -493\ kPa = \underline{-4.93\ bars}$$

**4-31.** $$v = \frac{Q}{A} = \frac{\left({}^{30}\!/_{449}\right) {}^{ft^3}\!/_{s}}{\frac{\pi}{4}\left(\frac{1.5}{12}\ ft\right)^2} = 5.44\ {}^{ft}\!/_{s}$$

$$N_R = \frac{7740 \times 5.44 \times 1.5}{100} = 632$$

$$f = \frac{64}{632} = 0.101$$

$$L_e = L_{pipe} + \left(\frac{K\,D}{f}\right)_{valve} + 2\left(\frac{K\,D}{f}\right)_{elbow}$$

$$= 45 + \frac{10 \times 1.5}{0.001 \times 12} + 2 \times \frac{0.75 \times 1.5}{0.101 \times 12} = 45 + 12.4 + 1.9 = 59.3\ ft$$

$$H_L = 0.101 \times \frac{59.3}{1.5/12} \times \frac{5.44^2}{64.4} = 22.0\ ft\ \text{of oil}$$

$$\Delta p = (62.4 \times 0.9)\, lb/ft^3 \times 22.0\ ft \times \frac{1\ psi}{144\ psf} = 8.6\ psi \qquad Thus\ \ p_2 = \underline{91.4\ psi}$$

**4-32.** $$v = \frac{Q}{A} = \frac{0.002}{\frac{\pi}{4} \times 0.038^2} = 1.76\ m/s$$

$$N_R = \frac{1.76 \times 0.038}{0.0001} = 669$$

$$f = \frac{64}{669} = 0.096$$

$$L_e = 15 + \frac{10 \times 0.038}{0.096} + 2 \times \frac{0.75 \times 0.038}{0.096} = 15 + 4.0 + 0.6 = 19.6\ m$$

$$H_L = 0.096 \times \frac{19.6}{0.038} \times \frac{1.76^2}{2 \times 9.81} = 7.82\ ft\ \text{of oil}$$

$$\Delta p = \gamma\, H_L = (1000 \times 0.9 \times 9.81) \times 7.82 = 69{,}000\ N/m^2 = 0.69\ bars$$

*Therefore* $p_2 = \underline{6.31\ bars}$

**4-33.** $$H_L = \sum_{1}^{13}\left(f\,\frac{L}{D}\right)\frac{v^2}{2g} \quad and \quad v = \frac{Q}{A}$$

$$Q_{\text{return line}} = 40\left(\frac{8^2 - 4^2}{8^2}\right) = 30 \text{ gpm}$$

$$v_{1,2,3} = \frac{\left(40/449\right)\ ft^3/s}{\frac{\pi}{4}\left(\frac{1.5}{12}\ ft\right)^2} = 7.25\ ft/s$$

$$v_{4,5,6} = \frac{\left(40/449\right)}{\frac{\pi}{4}\left(\frac{1}{12}\right)^2} = 16.3\ ft/s$$

$$v_{7,8} = \frac{\left(30/449\right)}{\frac{\pi}{4}\left(\frac{1}{12}\right)^2} = 12.2\ ft/s$$

$$v_{9,10} = \frac{\left(40/449\right)}{\frac{\pi}{4}\left(\frac{0.75}{12}\right)^2} = 29.0\ ft/s$$

$$v_{11,12,13} = \frac{\left(30/449\right)}{\frac{\pi}{4}\left(\frac{0.75}{12}\right)^2} = 21.8\ ft/s$$

Now since $N_R = \frac{v\,D}{\nu}$, we have:

$$N_{R\ 1,2,3} = \frac{7.25 \times 1.5/12}{0.001} = 906\ , \quad N_{R\ 4,5,6} = \frac{16.3 \times 1.0/12}{0.001} = 1358$$

$$N_{R\ 7,8} = \frac{12.2 \times 1.0/12}{0.001} = 1017\ , \quad N_{R\ 9,10} = \frac{29.0 \times 0.75/12}{0.001} = 1813$$

$$N_{R\ 11,12,13} = \frac{21.8 \times 0.75/12}{0.001} = 1363$$

All flows are laminar. Hence $f = \frac{64}{N_R}$

$$H_{L\,1,2,3} = \left(\frac{64}{906} \times \frac{10}{1.5/12} + 1.5\right) \times \frac{7.25^2}{64.4} = 5.84\text{ ft of oil}$$

$$= \frac{5.84\text{ ft} \times 50\ \text{lb}/\text{ft}^3}{144\ \text{in}^2/\text{ft}^2} = 2.03\text{ psi} \quad \text{since } P = \gamma h$$

$$H_{L\,4,5,6} = \left(\frac{64}{1358} \times \frac{65}{1.0/12} + 10.5\right) \times \frac{16.3^2}{64.4} = 195\text{ ft} = 67.8\text{ psi}$$

$$H_{L\,7,8} = \left(\frac{64}{1017} \times \frac{10}{1.0/12} + 0.75\right) \times \frac{12.2^2}{64.4} = 19.1\text{ ft} = 6.66\text{ psi}$$

$$H_{L\,9,10} = \left(\frac{64}{1813} \times \frac{10}{0.75/12} + 0.75\right) \times \frac{29.0^2}{64.4} = 83.6\text{ ft} = 29.0\text{ psi}$$

$$H_{L\,11,12,13} = \left(\frac{64}{1363} \times \frac{90}{0.75/12} + 1.5\right) \times \frac{21.8^2}{64.4} = 510\text{ ft} = 177\text{ psi}$$

$$F = [1000 - (2.03 + 67.8 + 29.0)] \times \frac{\pi}{4} 8^2 - (6.66 + 177) \times \frac{\pi}{4}\left(8^2 - 4^2\right)$$

F = 45,300 - 6,900 = <u>38,400 lb</u>

**4-34.** $HP_{loss} = \frac{\Delta p(psi) \times Q(gpm)}{1714}$

Using values from Exercise 4-33, we have:

$$HP_{loss} = \frac{(2.03 + 67.8 + 29.0) \times 40}{1714} + \frac{(6.66 + 177) \times 30}{1714}$$

$$= 2.30 + 3.21 = 5.51 \text{ HP}$$

**Since 1 HP = 42.4 BTU/min, we have:**

**Heat generation rate = 42.4 x 5.51 = 234 BTU/min**

**= <u>14,000 BTU/hr</u>**

**4-35.** $Q = Q_{pump} = 40$ **gpm for both the extending and retracting speeds of the cylinder. Thus we have:**

$$v_{extending} = \frac{Q}{A_{piston}} = \frac{\left(40/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{8}{12}\ ft\right)^2} = \underline{0.255}\ ft/s$$

$$v_{retracting} = \frac{Q}{A_{piston} - A_{rod}} = \frac{\left(40/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{8^2 - 4^2}{144}\right) ft^2} = \underline{0.340}\ ft/s$$

**4-36.** $\Delta p_{pump} = 1000\ psi \times \frac{1\ Pa}{0.000145\ psi} = 6.90\ MPa$

$$Q_{pump} = 0.0000632 \times 40 = 0.00253\ m^3/s$$

$$\nu = 0.001\ \frac{ft^2}{s} \times \left(\frac{1\ m}{3.28\ ft}\right)^2 = 0.0000930\ m^2/s$$

$$\gamma = 50\,\frac{lb}{ft^3} \times \frac{1\ N}{0.225\ lb} \times \left(\frac{3.28\ ft}{1\ m}\right)^3 = 7840\,\frac{N}{m^3}$$

$$\text{Cylinder piston diameter} = 8\ in \times \frac{1\ ft}{12\ in} \times \frac{1\ m}{3.28\ ft} = 0.203\ m$$

Cylinder rod diameter = 0.102 m

All elbows are 90° with a K factor = 0.75

Pipe lengths and inside diameters are as follows:

| Pipe No. | Length(m) | Dia.(m) | Pipe No. | Length(m) | Dia.(m) |
|---|---|---|---|---|---|
| 1 | 0.610 | 0.0381 | 8 | 1.52 | 0.0254 |
| 2 | 1.83 | 0.0381 | 9 | 1.52 | 0.0190 |
| 3 | 0.610 | 0.0381 | 10 | 1.52 | 0.0190 |
| 4 | 15.2 | 0.0254 | 11 | 18.3 | 0.0190 |
| 5 | 3.05 | 0.0254 | 12 | 3.05 | 0.0190 |
| 6 | 1.52 | 0.0254 | 13 | 6.10 | 0.0190 |
| 7 | 1.52 | 0.0254 | | | |

The following equations are applicable:

$$H_l = \sum_{1}^{13}\left(f \times \frac{L}{D} + K\right)\frac{v^2}{2g}\ ,\ v = \frac{Q}{A}\ ,\ N_R = \frac{v\ D}{\nu}$$

$$Q_{\text{return line}} = 0.00253 \times \frac{\left(0.203^2 - 0.102^2\right)}{0.203^2} = 0.00189\ {}^{m^3}\!/_{s}$$

$$v_{1,2,3} = \frac{0.00253}{\frac{\pi}{4} \times 0.0381^2} = 2.22\ {}^{m}\!/_{s}$$

$$v_{4,5,6} = \frac{0.00253}{\frac{\pi}{4} \times 0.0254^2} = 4.99 \ {}^{m}\!/\!_{s}$$

$$v_{7,8} = \frac{0.00189}{\frac{\pi}{4} \times 0.0254^2} = 3.73 \ {}^{m}\!/\!_{s}$$

$$v_{9,10} = \frac{0.00253}{\frac{\pi}{4} \times 0.0190^2} = 8.92 \ {}^{m}\!/\!_{s}$$

$$v_{11,12,13} = \frac{0.00189}{\frac{\pi}{4} \times 0.0190^2} = 6.67 \ {}^{m}\!/\!_{s}$$

$$N_{R\ 1,2,3} = \frac{2.22 \times 0.0381}{0.000093} = 909, \quad N_{R\ 4,5,6} = \frac{4.99 \times 0.0254}{0.000093} = 1362$$

$$N_{R\ 7,8} = \frac{3.73 \times 0.0254}{0.000093} = 1018, \quad N_{R\ 9,10} = \frac{8.92 \times 0.019}{0.000093} = 1822$$

$$N_{R\ 11,12,13} = \frac{6.67 \times 0.019}{0.000093} = 1363$$

All flows are laminar. Hence $f = \frac{64}{N_R}$

$$H_{L\ 1,2,3} = \left(\frac{64}{909} \times \frac{3.05}{0.0381} + 1.5\right) \times \frac{2.22^2}{2 \times 9.81} = 1.79 \text{ m} = 14{,}000 \text{ Pa}$$

$$H_{L\ 4,5,6} = \left(\frac{64}{1362} \times \frac{19.8}{0.0254} + 10.5\right) \times \frac{4.99^2}{2 \times 9.81} = 59.8 \text{ m} = 469{,}000 \text{ Pa}$$

$$H_{L\ 7,8} = \left(\frac{64}{1018} \times \frac{3.05}{0.0254} + 0.75\right) \times \frac{3.73^2}{2 \times 9.81} = 5.89 \text{ m} = 46{,}200 \text{ Pa}$$

$$H_{9,10} = \left(\frac{64}{1822} \times \frac{3.05}{0.019} + 0.75\right) \times \frac{8.92^2}{2 \times 9.81} = 25.9\ m = 203{,}000\ Pa$$

$$H_{L\ 11,12,13} = \left(\frac{64}{1363} \times \frac{27.4}{0.019} + 1.5\right) \times \frac{6.67^2}{2 \times 9.81} = 157\ m = 1{,}230{,}000\ Pa$$

$$F = \left[6{,}900{,}000 - (14{,}000 + 469{,}000 + 203{,}000)\right] \times \frac{\pi}{4} \times 0.203^2$$

$$- (46{,}200 + 1{,}230{,}000) \times \frac{\pi}{4}\left(0.203^2 - 0.102^2\right)$$

= 201,000 - 30,900 = <u>170,000 N</u>

**4-37.** $Power\ \ Loss\,(Watts) = p\,(Pa) \times Q\left(m^3/s\right)$

Using the values from Exercise 4-36, we have:

Power Loss = (14,000 + 469,000 + 203,000)x(0.00253)

+ (46,200 + 1,230,000)x(0.00189)

= 1740 + 2410 = 4150 Watts = <u>4.15 kW</u>

**4-38.** $Q_{pump}\left(m^3/s\right) = 0.0000632\,Q(gpm) = 0.0000632 \times 40 = 0.00253\ m^3/s$

$$\text{Cylinder piston diameter} = 8\ in \times \frac{2.54\ cm}{1\ in} = 20.32\ cm$$

$$v_{extending} = \frac{Q}{A_{piston}} = \frac{0.00253\ m^3/s}{\frac{\pi}{4} \times (0.2032\ m)^2} = \underline{0.0780\ m/s}$$

$$\text{Cylinder rod diameter} = 4\ in \times \frac{2.54\ cm}{1\ in} = 10.16\ cm$$

$$v_{retracting} = \frac{Q}{A_{piston} - A_{rod}} = \frac{0.00253\ m^3/s}{\frac{\pi}{4} \times \left(0.2032^2 - 0.1016^2\right)} = \underline{0.104\ m/s}$$

# Chapter 5

# Hydraulic Pumps

5-1. 1. Gear
2. Vane
3. Piston

5-2. A positive displacement pump ejects a fixed amount of fluid into the hydraulic system per revolution of pump shaft rotation. Thus, for positive displacement pumps, pump flow rate is directly proportional to pump speed. However, for centrifugal pumps, flow output is reduced as circuit resistance is increased. Thus, the flow rate from a centrifugal pump not only depends on the pump speed, but also on the resistance of the external system.

5-3. All pumps operate on the principle whereby a partial vacuum is created at the pump inlet due to the internal operation of the pump. This allows atmospheric pressure to push the fluid out of the oil tank into the pump intake. The pump then mechanically pushes the fluid out the discharge line as shown by Figure 5-2.

5-4. Volumetric efficiency equals actual flow rate produced by a pump, divided by the theoretical flow rate based on volumetric displacement and pump speed. Actual flow rate is measured by a flow meter and theoretical flow rate is calculated from the equation: $Q_T = V_D N$

5-5. Mechanical efficiency is determined by using Equation 5-8 where pump discharge pressure P, pump input torque T and pump speed N are measured. The theoretical pump flow rate is calculated from the equation: $Q_T = V_D N$

**5-6.** **After the volumetric efficiency $\eta_v$ and mechanical efficiency $\eta_m$ have been found, the overall efficiency $\eta_o$ is determined from the equation:** $\eta_o = \eta_v \times \eta_m$

**5-7.** **A partial vacuum is created at the pump inlet due to the internal operation of the pump (See Figure 5-2). This allows atmospheric pressure to push the fluid out of the oil tank and into the pump intake because atmospheric pressure is greater than vacuum pressure.**

**5-8.** **A fixed displacement pump is one in which the amount of fluid ejected per revolution (displacement) cannot be varied. In a variable displacement pump, the displacement can be varied by changing the physical relationships of various pump elements. This change in pump displacement, produces a change in pump flow output even though pump speed remains constant.**

**5-9.**
1. **Spur gear**
2. **Helical gear**
3. **Herringbone gear**

**5-10.**
1. **Lobe**
2. **Gerotor**

**5-11.** **Three precision ground screws, meshing within a close-fitting housing, deliver non-pulsating flow (See Figure 5-13). The two symmetrically opposed idler rotors are in rolling contact with the central power rotor and are free to float in their respective housing bores on a hydrodynamic oil film.**

**5-12.**
1. **Flow rate requirements**
2. **Operating speed**
3. **Pressure rating**
4. **Performance**
5. **Reliability**
6. **Maintenance**
7. **Cost**

8. Noise

5-13. A pressure compensated vane pump is one in which system pressure acts directly on the cam ring via a hydraulic piston (See Figure 5-16). This forces the cam ring against the compensation spring-loaded piston. If the discharge pressure is large enough, it overcomes the compensator spring force and shifts the cam ring. As the discharge pressure continues to increase, zero eccentricity and thus, zero flow is achieved. Therefore, such a pump has its own protection against excessive pressure buildup.

5-14. Pump cavitation occurs when suction lift is excessive and the inlet pressure falls below the vapor pressure of the fluid (usually about 5 psi suction). As a result, vapor bubbles which form in the low pressure inlet region of the pump, are collapsed when they reach the high pressure discharge region. This produces high fluid velocities and impact forces which erode the surfaces of metallic components. The result is shortened pump life.

5-15. Pumps do not pump pressure. Instead they produce fluid flow. The resistance to this flow, produced by the hydraulic system, is what determines the pressure.

5-16. Cavitation can occur due to entrained vapor bubbles. This occurs when suction lift is excessive and the inlet pressure falls below the vapor pressure of the fluid (usually about 5 psi suction). Cavitation produces very large fluid impact forces which erodes the surfaces of metallic components and thus shortens pump life.

5-17. If there is no place for the fluid to go, the pressure will rise to an unsafe level unless a pressure relief valve opens to allow flow back to the oil tank. Thus, the relief valve determines the maximum pressure level which the system will experience.

5-18. The flow output of a centrifugal pump is reduced as circuit resistance is increased. Therefore, centrifugal pumps are rarely used in hydraulic systems.

5-19. A balanced vane pump is one that has two intake and two outlet ports diametrically opposite each other. Thus, pressure ports are opposite each other and a complete hydraulic balance is achieved. This eliminates the bearing side loads and, thus, permits higher operating pressures.

5-20. 1. Axial design
2. Radial design

5-21. 1. Pump speed
2. Pressure
3. Pump size
4. Entrained gas bubbles

5-22. The pressure rating is defined as the maximum pressure level at which the pump can operate safely and provide a good useful life.

5-23. 1. Keep the suction line velocities below 5 ft/s.
2. Keep the pump inlet lines as short as possible.
3. Minimize the number of fittings in the inlet line.
4. Mount the pump as close as possible to the reservoir.

5-24. Gear pumps are simple in design and compact in size. They are the least expensive. Vane pump efficiencies and costs fall in between gear and piston pumps. Piston pumps are the most expensive and provide the highest level of overall performance.

5-25. By specifying volumetric displacement and volumetric flow rate at a given pump speed.

5-26. Vane and piston pumps.

5-27. A balanced vane pump is one that has two intake and two outlet ports diametrically opposite each other. Thus, pressure ports are opposite each other, and a complete hydraulic balance is achieved eliminating bearing side loads and thus permitting higher operating pressures.

5-28. By varying the offset angle between the cylinder block centerline and the drive shaft centerline.

5-29. The eccentricity between the centerline of the rotor and the centerline of the cam ring can be changed by a hand wheel or by a pressure compensator.

5-30. The addition of pressure compensation prevents the manual setting of the rotor eccentricity to vary flow rate. Rather, the eccentricity is controlled by pump discharge pressure resulting in zero flow rate (zero eccentricity) at maximum pump discharge pressure. Thus the pump is protected against excessive pressure because it produces no flow at the maximum pressure level.

5-31. Noise is sound that people find undesirable.

5-32. Intensity and loudness are not the same because loudness depends on each person’s sense of hearing. The loudness of a sound may not be the same for two people sitting next to each other and listening to the same sound. However the intensity of sound, which represents the amount of energy possessed by the sound, can be measured and thus does not depend on who hears it.

5-33. One decibel equals the smallest change in intensity that can be detected by most people. The weakest sound intensity that the human ear can hear is designated as zero decibels. Since one bel represents a very large change in sound intensity, it has become standard practice to express sound intensity in units of decibels (a bel = 10 decibels).

5-34. 1. Prolonged exposure to loud noise can result in loss of hearing.

2. **Noise can mask sounds that people want to hear. These include voice communication between people and warning signals emanating from safety equipment.**

**5-35.** 1. **Make design changes to the source of the noise such as a pump.**
2. **Modify the path along which the noise travels such as by clamping hydraulic piping at specifically located supports.**
3. **Use sound absorption materials in nearby screens or partitions.**

**5-36.** $Q_T = \dfrac{V_D\,N}{231}$ **where** $V_D = 9 \times \dfrac{\pi}{4} \times 0.5^2 \times 0.75 = 1.33\ in^3$

$$Q_T = \frac{1.33 \times 2000}{231} = \underline{11.5\ gpm}$$

**5-37.** $$e = \frac{2\,V_D}{\pi(D_C + D_R)\,L} = \frac{2 \times 7}{\pi(4.5 + 2.5) \times 2} = \underline{0.371\ in}$$

**5-38.** $$Q_T = \frac{Q_A}{\eta_v} = \frac{30}{0.96} = 31.25\ gpm$$

$$\tan\theta = \frac{231Q}{DANY} = \frac{231 \times 31.25}{5 \times \frac{\pi}{4}\left(\frac{5}{8}\right)^2 \times 3000 \times 9} = 0.174$$

**Thus** $\theta = \underline{9.9°}$

**5-39.** $$Q = DANY = 0.020m \times \frac{\pi}{4}(0.015m)^2\left(\frac{2000}{60}\frac{rev}{s}\right) \times 9 = 0.00106\frac{m^3}{s}$$

**5-40.** $e = \dfrac{2\,V_D}{\pi\left(D_C + D_R\right)L} = \dfrac{2 \times \left(115 \times 10^3\right)}{\pi\,(88.9 + 63.5) \times 50.8} = 9.46\ mm$

**5-41.** $Q_T = \dfrac{Q_A}{\eta_v} = \dfrac{0.0019}{0.95} = 0.0020\ m^3/s$

$$\tan\theta = \frac{Q}{DANY} = \frac{0.0020}{0.127 \times \frac{\pi}{4}(0.0159)^2 \times \frac{3000}{60} \times 9} = 0.176$$

**Thus** $\theta = \underline{9.9°}$

**5-42.** $\eta_V = \dfrac{actual\ \ flowrate}{theoretical\ \ flowrate} = \dfrac{Q_A\left(m^3/\text{min}\right)}{V_D\left(m^3\right) \times N\left(rev/\text{min}\right)}$

**Substituting known values, we have:**

$$0.96 = \frac{0.029}{V_D \times 1000} \qquad Hence \qquad V_D = 0.0000302\ m^3 = \underline{0.0302\ L}$$

**5-43.** $Q\left(\dfrac{m^3}{min}\right) = V_D\left(m^3\right) \times N\left(\dfrac{rev}{min}\right)$

**5-44.** $\eta_o = \eta_v\,\eta_m \qquad Thus \qquad \eta_m = \dfrac{\eta_o}{\eta_V} = \dfrac{0.88}{0.92} = 0.957 = \underline{95.7\%}$

**5-45. First find the displacement volume.**

$$V_D = \frac{\pi}{4} \times \left(3.25^2 - 2.25^2\right) \times 1 = 4.32 \text{ in}^3$$

**Next find the theoretical flow rate.**

$$Q_T = \frac{V_D\, N}{231} = \frac{4.32 \times 1800}{231} = 33.7 \text{ gpm}$$

**The volumetric efficiency can now be found.**

$$\eta_V = \frac{29}{33.7} = 0.861 = \underline{86.1\%}$$

**5-46.** $V_D = \frac{\pi}{4}\left(D_o^2 - D_i^2\right) L = \frac{\pi}{4}\left(0.0826^2 - 0.0572^2\right) \times 0.0254 = 70.8 \times 10^{-6} \text{m}^3$

$$Q_T = V_D\, N = \left(70.8 \times 10^{-6} \text{ m}^3\right) \times \frac{1800}{60}\frac{\text{rev}}{\text{s}} = 2120 \times 10^{-6} \text{ m}^3/\text{s}$$

$$= 0.00212 \text{ m}^3/\text{s}$$

$$\eta_V = \frac{0.00183}{0.00212} = 0.863 = \underline{86.3\%}$$

**5-47.** $\eta_m = \frac{\eta_o}{\eta_V} = \frac{0.88}{0.92} = 0.96 = \underline{96\%}$

**Frictional HP = 0.12 x 8 = <u>0.96 HP</u>**

**5-48.** $\eta_o = \frac{\text{pump output power}}{\text{pump input power}}$

$$Output\ power = p\,Q = 10\times10^6\,\frac{N}{m^2}\times 40\,\frac{L}{\min}\times\frac{1\,m^3/s}{1000\,L/s}\times\frac{1\min}{60\,s}$$

$$= 6670\ W = 6.67\ kW$$

$$\text{Input power} = 10\ \text{HP}\times\frac{746\ \text{W}}{1\ \text{HP}} = 7460\ \text{W} = 7.46\ \text{kW}$$

$$\eta_o = \frac{6670}{7460} = 0.894 = \underline{89.4\%}$$

**5-49.** $$\text{dB Increase} = 10\times\log\frac{I(\text{final})}{I(\text{initial})} = 10\times\log 10 = \underline{10\ \text{dB}}$$

**5-50.** $$HP_{hydr} = \frac{p\,Q}{1714} = \frac{2000\times10}{1714} = \underline{11.7\ HP}$$

$$HP_{\text{elec motor}} = \frac{HP_{hydr}}{\eta_o} = \frac{11.7}{0.85} = \underline{13.8\ \text{HP}}$$

**5-51.** (a) First find the theoretical flow rate.

$$Q_T = \frac{V_D\,N}{231} = \frac{6\times1000}{231} = 26.0\ \text{gpm}$$

Next solve for the volumetric efficiency.

$$\eta_V = \frac{Q_A}{Q_T} = \frac{24}{26.0} = 0.923 = 92.3\%$$

Then solve for the mechanical efficiency.

$$\eta_m = \frac{p\,Q_T / 1714}{T\,N / 63{,}000} = \frac{1000 \times 26.0 / 1714}{1100 \times 1000 / 63{,}000} = \frac{15.17}{17.46}$$

$$= 0.869 = 86.9\%$$

**Finally we solve for the overall efficiency.**

$$\eta_o = \eta_v\,\eta_m = 0.923 \times 0.869 = 0.802 = \underline{80.2\%}$$

**(b)** $T_T = T_A \times \eta_m = 1100 \times 0.869 = \underline{956 \text{ in} \bullet \text{lb}}$

**5-52.** $$\textit{Hydraulic Power} = p\,Q = \left(140 \times 10^5\, N/m^2\right) \times 0.001\, m^3/s$$

$$= 14.0 \times 10^3 \frac{N \bullet m}{s} = \underline{14.0\ kW}$$

**Electric Power** $= \dfrac{14.0}{0.85} = \underline{16.5\ \text{kW}}$

**5-53.** **(a)** $Q_T = V_D\,N = \left(98.4 \times 10^{-6}\,\text{m}^3\right) \times \dfrac{1000}{60}\ \dfrac{\text{rev}}{\text{s}} = 0.00164\ \text{m}^3/\text{s}$

$$\eta_V = \frac{0.00152}{0.00164} = 0.927 = 92.7\%$$

$$\eta_m = \frac{p\,Q_T}{T\,\omega} = \frac{\left(70 \times 10^5\ N/m^2\right) \times 0.00164\ m^3/s}{\left(124.3\ N \bullet m\right) \times \left(\dfrac{1000}{60} \times 2\pi\ \ rad/s\right)}$$

$$= 0.882 = 88.2\%$$

$$\eta_o = \eta_v\,\eta_m = 0.927 \times 0.882 = 0.818 = \underline{81.8\%}$$

**(b)** $T_T = T_A\,\eta_m = 124.3 \times 0.882 = \underline{109.6\ \text{N} \bullet \text{m}}$

**5-54.** $$\eta_o = \frac{pump\ output\ HP}{pump\ input\ HP} = \frac{pQ/1714}{10} = \frac{3000\times 5/1714}{10} = 0.875 = \underline{87.5\%}$$

$$\text{Pump input HP} = \frac{T\ N}{63{,}000} = \frac{T\times 1000}{63{,}000} = 10 \quad \text{Thus } T = \underline{630\ in \bullet lb}$$

**5-55.** $$Q_T = \frac{V_D\ N}{231} = \frac{6\times 1200}{231} = 31.2\ gpm$$

$$\eta_V = \frac{Q_A}{Q_T} = \frac{29}{31.2} = 0.929 = 92.9\%$$

$$\eta_m = \frac{\eta_o}{\eta_V} = \frac{88}{92.9} = 0.947 = 94.7\%$$

$$T_T = \frac{V_D\ p}{2\pi} = \frac{6\times 500}{2\pi} = 477.5\ in \bullet lb$$

$$T_A = \frac{T_T}{\eta_m} = \frac{477.5}{0.947} = \underline{504.2\ in \bullet lb}$$

**5-56.** $$\text{Prime mover } HP = \frac{pQ}{1714} = \frac{2000\times 10}{1714} = \underline{11.7\ HP}$$

$$\text{Prime mover speed} = \frac{231\ Q}{V_D} = \frac{231\times 10}{1.5} = \underline{1540\ rpm}$$

**5-57.** (a) $Q_{pump\ act} = A_{piston}\ v_{piston\ ext}$

$$= \frac{\pi}{4} \times 8^2 \text{ in}^2 \times 3 \frac{\text{in}}{\text{s}} \times \frac{1 \text{ gal}}{231 \text{ in}^3} \times \frac{60 \text{ s}}{1 \text{ min}} = 39.2 \text{ gpm}$$

$$Q_{pump\ theor} = \frac{Q_{pump\ act}}{\eta_V} = \frac{39.2}{0.92} = 42.6 \text{ gpm}$$

$$Q_{pump\ theor} = \frac{V_D N}{231} = \frac{V_D \times 1800}{231} = 42.6 \quad \text{Thus } V_D = \underline{5.47 \text{ in}^3}$$

**(b)** $HP_{pump\ output} = \dfrac{(\Delta p) Q_{act}}{1714}$

$$p_{blank\ end} A_{piston} - p_{rod\ end}\left(A_{piston} - A_{rod}\right) = F_{ext.load}$$

$$p_{blank\ end} \times \frac{\pi}{4}\left(8^2\right) - 50 \times \frac{\pi}{4}\left(8^2 - 4^2\right) = 40{,}000$$

Therefore $\quad p_{blank\ end} = 833\ psi$

$$HP_{pump\ output} = \frac{(833 + 75 + 4) \times 39.2}{1714} = 20.9 \text{ HP}$$

$$HP_{pump\ input} = \frac{HP_{pump\ output}}{\eta_V\ \eta_m} = \frac{20.9}{0.92 \times 0.90} = \underline{25.2 \text{ HP}}$$

**(c)** $HP_{pump\ input} = \dfrac{T\ N}{63{,}000}$

$$25.2 = \frac{T \times 1800}{63{,}000} \quad \text{Thus } T = \underline{882 \text{ in} \bullet \text{lb}}$$

**(d)** Power to load $= F_{load}\ V_{cyl.\ ext.}$

$$= 40{,}000\ \text{lb} \times 3\frac{\text{in}}{\text{s}} \times \frac{1\ \text{ft}}{12\ \text{in}} \times \frac{1\ \text{HP}}{550\frac{\text{ft}\bullet\text{lb}}{\text{s}}}$$

= 18.2 HP

Percent of pump input power delivered to load

$$= \frac{18.2}{25.2} \times 100 = \underline{72.2\%}$$

5-58. The following metric data are applicable:

Cylinder piston diameter = $8\ \text{in} \times \frac{1\ \text{ft}}{12\ \text{in}} \times \frac{1\ \text{m}}{3.28\ \text{ft}} = 0.203\ \text{m}$

Cylinder piston rod diameter = 4 in = 0.102 m

Extending speed of cyl. = $3\frac{\text{in}}{\text{s}} \times \frac{1\ \text{ft}}{12\ \text{in}} \times \frac{1\ \text{m}}{3.28\ \text{ft}} = 0.0762\frac{\text{m}}{\text{s}}$

External load on cyl. = $40{,}000\ \text{lb} \times \frac{1\ \text{N}}{0.225\ \text{lb}} = 178{,}000\ \text{N}$

Pump volumetric efficiency = 92%

Pump mechanical efficiency = 90%

Pump speed = 1800 rpm

Pump inlet pressure = $-4.0\ \text{psi} \times \frac{1\ \text{Pa}}{0.000145\ \text{psi}} = -27{,}600\ \text{Pa}$

Total pressure drop in the line from the pump discharge port to the blank end of the cylinder is:

$75\ \text{psi} \times \frac{1\ \text{Pa}}{0.000145\ \text{psi}} = 517{,}000\ \text{Pa}$

Total pressure drop in the return line from the rod end of the cylinder = $50\ \text{psi} \times \frac{1\ \text{Pa}}{0.000145\ \text{psi}} = 345{,}000\ \text{Pa}$

(a) $Q_{pump\ act} = A_{piston}\, v_{piston\ ext} = \frac{\pi}{4} \times 0.203^2 \times 0.0762 = 0.00247\ m^3/s$

$$Q_{pump\ theor} = \frac{Q_{pump\ act}}{\eta_v} = \frac{0.00247}{0.92} = 0.00268\ m^3/s$$

$$Q_{pump\ theor} = V_D N = 0.00268$$

$$where \quad N = 1800 \frac{rev}{\min} \times \frac{1 \min}{60\ s} = 30\ rev/s$$

Thus $V_D = \frac{0.00268}{30} = 0.0000893\ m^3 = \underline{0.0893\ L}$

(b) Pump Output Power = $(\Delta p) Q_{act}$

$$p_{blank\ end} A_{piston} - p_{rod\ end}\left(A_{piston} - A_{rod}\right) = F_{est.load\ on\ cyl.}$$

$$p_{blank\ end} \times \frac{\pi}{4}\left(0.203^2\right) - 345{,}000 \times \frac{\pi}{4}\left(0.203^2 - 0.102^2\right) = 178{,}000$$

Thus $p_{blank\ end} = 5{,}758{,}000\ Pa = 5758\ kPa$

Pump Output Power = (5758 + 517 + 27.6)x(0.00247)
= 15.6 kW

$$\text{Pump Input Power} = \frac{\text{Pump Output Power}}{\eta_v\ \eta_m} = \frac{15.6}{0.92 \times 0.90}$$

= **18.8 kW**

(c) Pump Input Power = T N

$$\text{where } N = 1800 \frac{rev}{min} \times \frac{1\ min}{60\ s} \times \frac{2\pi\ rad}{1\ rev} = 188\ rad/s$$

$18{,}800 = T \times 188$ Thus $T = \underline{100\ N \bullet m}$

(d) Power delivered to load = $F_{load}\ v_{cyl\ ext.}$

= 178,000 x 0.0762 = 13,600 W = 13.6 kW

Percent of pump input power delivered to load

$$= \frac{13.6}{18.8} \times 100 = \underline{72.3\ \%}$$

5-59. (a) Per the solution to Exercise 5-57 we have the following while the cylinder is extending (pump discharge pressure = 833 psi + 75 psi = 908 psi):

Pump input power = $25.2\ hp \times \frac{0.746\ kW}{1hp} = 18.8\ kW$

Electric motor input power = $\frac{18.8\ kW}{0.85} = 22.1\ kW$

Thus with the cylinder fully extended (pressure relief valve set at 1000 psi):

Electric motor input power = $\frac{1000}{908} \times 22.1\ kW = 24.3\ kW$

Thus the yearly cost of electricity is:

$$Yearly\ cost = power\ rate \times time\ per\ year \times unit\ cost\ of\ electricity$$

$$= 0.30 \times 22.1\ kW \times 20 \frac{hr}{day} \times 250 \frac{days}{year} \times \frac{\$0.10}{kW\ hr}$$

$$+ 0.70 \times 24.3 kW \times 20 \frac{hr}{day} \times 250 \frac{days}{year} \times \frac{\$0.10}{kW\ hr}$$

$$= \$3315/yr + \$8505/yr = \underline{\$11{,}820/yr}$$

(b) The fixed displacement pump produces 39.2 gpm (per solution to Exercise 5-57) at 1000 psi when the cylinder is fully extended (1.0 gpm through cylinder + 38.2 gpm through relief valve). Thus when the cylinder is fully extended we have:

**hydraulic HP lost**

**with fixed displ. pump =** $\frac{pQ}{1714} = \frac{1000 \times 39.2}{1714} = 22.9 \ hp = 17.1 \ kW$

**Hence the electric motor input power =** $\frac{17.1\, kW}{0.828 \times 0.85} = 24.3\, kW$

**where the pump overall efficiency is 82.8%**

**The pressure compensated pump would produce only 1.0 gpm at 1000 psi when the cylinder is fully extended. For this case we have:**

**Hydraulic HP lost**

**with press-comp. pump =** $\frac{pQ}{1714} = \frac{1000 \times 1.0}{1714} = 0.583 \ hp = 0.44 \ kW$

**Hence the electric motor input power =** $\frac{0.44\, kW}{0.828 \times 0.85} = 0.63\, kW$

$$Thus\ the\ kW\ power\ saved\ while\ cylinder\ is\ fully\ extended = 24.3 - 0.63 = 23.67\ kW$$

$$Savings\ per\ year = 23.67\ kW \times 0.70\left(20\frac{hr}{day} \times 250\frac{days}{year}\right) \times \frac{\$0.10}{kW\ hr} = \$8284 \Big/ yr$$

**Time to pay for pump =** $\frac{\$2500}{\$8284 / yr} = \underline{0.30\ years}$

**5-60. Per the solution to Exercise 5-59 we have the following while the cylinder is extending:**

**pump discharge pressure =** $908\ psi \times \frac{101\ kPa}{14.7\ psi} = 6240\ kPa)$

$$Pump\ input\ power = 25.2\ hp \times \frac{0.746\ kW}{1\ hp} = 18.8\ kW$$

$$Electric\ motor\ input\ \ power = \frac{18.8\ kW}{0.85} = 22.1\ kW$$

**Thus with the cylinder fully extended (pressure relief valve set at 1000 psi = 6871 kPa):**

**Electric motor input power =** $\frac{6871}{6240} \times 22.1\ kW = 24.3\ kW$

**Thus per solution to Exercise 5-59, yearly cost of electricity = <u>$11,820/yr.</u>**

**(b) The fixed displacement pump produces 39.2 gpm = 2.48 L/s (per solution to Exercise 5-59) at 6871 kPa when the cylinder is fully extended. This is 1 gpm (0.0633 L/s) through the cylinder plus 2.417 L/s through the relief valve. Thus when the cylinder is fully extended we have:**

**Power lost with fixed displacement pump**

$$= p\,Q = 6871\ kPa \times 0.00248\ {m^3}/{s} = 17.1\ kW$$

$Hence\ the\ electric\ motor\ input\ \ power = \frac{17.1\ kW}{0.828 \times 0.85} = 24.3\ kW$ **where the pump overall efficiency is 82.8%**

**The pressure compensated pump would produce only 0.0633 L/s at 6871 kPa when the cylinder is fully extended. For this case we have:**

$Power\ lost = \ pQ = 6871\ kPa \times 0.0000633\ {m^3}/{s} = 0.44\ kW$ **with pressure compensated pump.**

$$Hence\ the\ electric\ motor\ input\ \ power = \frac{0.44\ kW}{0.828 \times 0.85} = 0.63\ kW$$

**Thus per solution to Exercise 5-59 the time to pay for the pump = <u>0.30 years</u>**

# Chapter 6

## Hydraulic Cylinders and Cushioning Devices

6-1. A single acting cylinder can exert a force in only the extending direction. Single acting cylinders do not retract hydraulically. Retraction is accomplished by using gravity or by the inclusion of a compression spring in the rod end. Double acting cylinders can be extended and retracted hydraulically.

6-2.
1. Flange mount.
2. Trunnion mount.
3. Clevis mount.
4. Foot and centerline lug mounts.

6-3. Some cylinders contain cylinder cushions at the ends of the cylinder to slow the piston down near the ends of the stroke. This prevents excessive impact when the piston is stopped by the end caps as illustrated in Figure 6-16.

6-4. A double-rod cylinder is one in which the rod extends out of the cylinder at both ends. Since the force and speed are the same for either end, this type of cylinder is typically used when the same task is to be performed at either end.

6-5. Telescoping rod cylinders contain multiple cylinders which slide inside each other. They are used where long work strokes are required but the full retraction length must be minimized.

6-6. The effective cylinder area is not the same for the extension and retraction strokes. This is due to the effect of the piston rod.

6-7. Single acting cylinders are retracted by gravity or by the inclusion of a compression spring in the rod end of the cylinder.

6-8. A first class lever is characterized by the lever fixed hinge pin located between the cylinder and load rod pins. In a second class lever, the load rod pin is located between the fixed hinge pin and cylinder rod pin. For a third class lever, the cylinder rod pin lies in between the load rod pin and the fixed hinge pin.

6-9. A moment is the product a force and its moment arm relative to a given point.

6-10. A moment arm is the perpendicular distance from a given point to the line of action of a force.

6-11. The cylinder is clevis mounted to allow the rod pinned end to travel along the circular path of the lever as it rotates about its fixed hinge pin.

6-12. A torque is the product of a force and its torque arm relative to a given axis of rotation. The torque arm is the distance from the axis of rotation measured perpendicular to the line of action of the force.

Thus for example, for the first class lever of Figure 6-12, the axis of rotation is the fixed hinge pin centerline. The load torque that the cylinder must overcome thus equals the produce of the load force $F_{load}$ and its torque arm $L_2 \cos\theta$ relative to the hinge pin axis of rotation.

Hence a torque arm is a force’s distance to an axis of rotation and a moment arm is a force’s distance to a point. Hence a moment tends to bend a member about a point whereas a torque tends to rotate a member about an axis.

**6-13.** The purpose is to bring a moving load to a gentle rest through the use of metered hydraulic fluid. Two applications are moving cranes and suspension systems of automobiles.

**6-14.** $time = \frac{stroke}{velocity}$ or $t = \frac{s}{v}$ also Power = Fv = constant

and $A = \frac{\pi D^2}{4}$ where A = piston area

Q = constant because pump speed is constant

thus $v = \frac{Q}{A} = \frac{constant}{A}$ and hence vA = constant

a. $v = \frac{constant}{A}$ and A remains the same.

Therefore v does not change. Since the stroke is doubled the time increases by a factor of 2.

Also since $F = \frac{constant}{v}$, the force does not change.

b. Piston area A increases by a factor of 4. Therefore v decreases by a factor of 4. So the time imcreases by a factor of 4 and the force increases by a factor of 4.

c. Piston area A increases by a factor of 4. Therefore v decreases by a factor of 4. Since the stroke is doubled the time increases by a factor of 8 and the force increases by a factor of 4.

**6-15.** $A_{retraction} = \frac{\pi}{4}\left(D^2_{piston} - D^2_{rod}\right)$

If both piston rod diameters are doubled the new effective area during retraction is

$$A'_{retration} = \frac{\pi}{4}\left[\left(2D_{piston}\right)^2 - \left(2D_{rod}\right)^2\right] = \pi\left(D^2_{piston} - D^2_{rod}\right)$$

**Therefore just as in the case for extension, the effective cylinder area increases by a factor of 4. <u>Hence the answers are the same as those for the extension stroke</u>.**

**6-16.** $(a)\ v_{ext} = \dfrac{Q}{A_p} = \dfrac{\frac{100}{60}\times 10^{-3}\ m^3/s}{\frac{\pi}{4}(0.08\ m)^2} = \dfrac{1.667\times 10^{-3}\ m^3/s}{0.00503\ m^2} = \underline{0.331\ m/s}$

$$v_{ret} = \frac{Q}{A_p - A_R} = \frac{1.667\times 10^{-3}\ m^3/s}{\frac{\pi}{4}\left(0.08^2 - 0.04^2\right)m^2} = \frac{1.667\times 10^{-3}}{0.00377} = \underline{0.442\ m/s}$$

$$(b)\ F_{ext} = p\,A_p = 12\times 10^6\ N/m^2 \times 0.00503\,m^2 = 0.0604\times 10^6\ N = \underline{60{,}400\ N}$$

$$F_{ret} = p\left(A_p - A_R\right) = 12\times 10^6\ N/m^2 \times 0.00377\,m^2 = 0.0452\times 10^6\ N = \underline{45{,}200\ N}$$

**6-17.** $v = \dfrac{Q_{in}}{A_p} = \dfrac{Q_{out}}{A_p - A_R}$

$$Q_{out} = \frac{A_p - A_R}{A_p}\times Q_{in} = \frac{0.00377\,m^2}{0.00503\,m^2}\times 8\ gpm = \underline{6.00\ gpm}$$

**6-18.** $(a)\ \text{Pressure} = \dfrac{force\,(lb)}{piston\ area\,(in^2)} = \dfrac{1200}{\frac{\pi}{4}\times 1.5^2} = \dfrac{1200}{1.767} = \underline{679\ psi}$

$$\textbf{(b) Velocity} = \frac{\text{input flow}\left(ft^3/s\right)}{\text{piston area}\left(ft^2\right)} = \frac{25/448}{\frac{\pi}{4}\times 1.5^2/144} = \frac{0.0558}{0.0123}$$

**= <u>4.54 ft/s</u>**

(c) $HP = \dfrac{4.54\ ft/s \times 1200\ lb}{550} = \underline{9.91\ HP}$

(d) $Pressure = \dfrac{force\ (lb)}{piston\ area(in^2) - rod\ area(in^2)}$

$= \dfrac{1200}{1.767 - \frac{\pi}{4} \times 0.75^2} = \dfrac{1200}{1.33} = \underline{902\ psi}$

(e) $Velocity = \dfrac{input\ flow(ft^3/s)}{piston\ area(ft^2) - rod\ area(ft^2)}$

$= \dfrac{0.0558}{1.33/144} = \underline{6.04\ ft/s}$

(f) $HP = \dfrac{6.04\ ft/s \times 1200\ lb}{550} = \underline{13.18\ HP}$

**6-19.** (a) $p = \dfrac{F}{A} = \dfrac{5000}{\frac{\pi}{4} \times 0.040^2} = \underline{3.98\ MPa}$

(b) $v = \dfrac{Q}{A} = \dfrac{0.0016}{\frac{\pi}{4} \times 0.040^2} = \underline{1.27\ m/s}$

(c) $kW = pQ = 3980\ kPa \times 0.0016\ m^3/s = \underline{6.37\ kW}$

(d) $p = \dfrac{5000}{\frac{\pi}{4}\left(0.040^2 - 0.020^2\right)} = \underline{5.31\ MPa}$

(e) $v = \dfrac{0.0016}{\frac{\pi}{4}\left(0.040^2 - 0.020^2\right)} = \underline{1.70\ m/s}$

(f) $kW = 5310 \times 0.0016 = 8.50\ kW$

6-20. $$v\left(\frac{in}{min}\right) = \frac{Q\left(\frac{gal}{min}\right) \times \frac{231\ in^3}{1\ gal}}{A\left(in^2\right)}$$ Hence $C_1 = \underline{231}$

$$v\left(\frac{m}{s}\right) = \frac{Q\left(\frac{m^3}{s}\right)}{A\left(m^2\right)}$$ Hence $C_2 = \underline{1}$

6-21. $$F_{ext} - F_{ret} = pA_P - p\left(A_P - A_R\right) = pA_P - p\left(A_P - \frac{A_P}{4}\right)$$

$$= pA_P - \frac{3}{4}pA_P = \frac{1}{4}pA_P$$

Therefore the difference = $\underline{\frac{1}{4} \times \text{pressure} \times \text{piston area}}$

6-22. There would be a net force to extend the cylinder. This net force would have the following value which is the same as that obtained in Exercise 6-19.

$$F_{net\ extending} = \underline{\frac{1}{4} \times \text{pressure} \times \text{piston area}}$$

6-23.

$$F_{load} = 5000\ lb \times \sin 30° = 2500\ lb$$

$$F_{cyl} = F_{load} = 2500\ lb = p\ A_p = p\left(\frac{\pi}{4}D^2\right)$$

$$p = \frac{4 \times 2500\ lb}{\pi\ (3\ in.)^2} = \underline{354\ psi}$$

**6-24. Per Newton's Law of Motion we have**

$$\sum F = m\,a$$

**where** $a = \dfrac{100\dfrac{m}{\min}\times\dfrac{1\min}{60\,s}}{0.5\,s} = \dfrac{1.67\,m/s}{0.5\,s} = 3.34\,m/s^2$

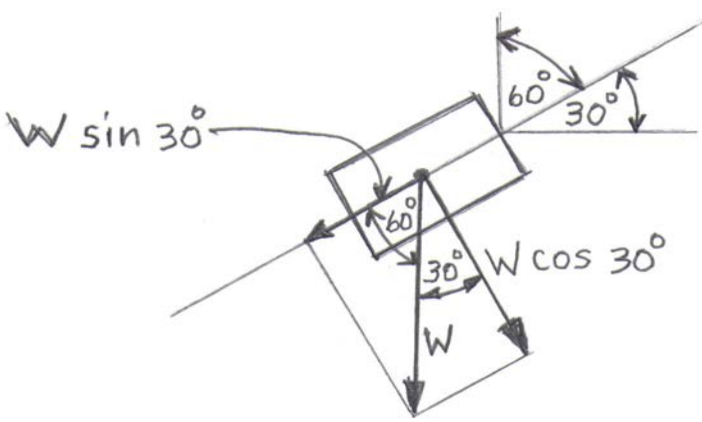
**Summing forces on the 10,000 N weight we have**

$$p\left(A_p - A_R\right) - 10{,}000\,N = \frac{10{,}000\,N}{9.80\,m/s^2}\times 3.34\,m/s^2$$

$$p\left(\frac{N}{m^2}\right)\times\frac{\pi}{4}\left(0.075^2 - 0.050^2\right)m^2 - 10{,}000\,N = 3408\,N$$

**p = 5,450,000 = <u>5450 kPa</u>**

**6-25.**




**The component of the weight W acting along the axis of the cylinder is W sin 30°. The component of the weight W acting normal to the incline surface is W cos 30°. The frictional force equals the coefficient of friction times the force normal to the sliding surfaces. Therefore the frictional force f acting along the axis of the cylinder is**

$$f = (CF)\times W\cos 30° = 0.15\times 6000\ lb\times\cos 30° = 779\ lb$$

The cylinder force equals $f + W\sin 30°$

Thus $F_{cyl} = 779\ lb + 6000\ lb \times \sin 30° = 3779\ lb$

And $p\,A_p = F_{cyl} = 3779\ lb$

$$1000\frac{lb}{in^2} \times A_p\left(in^2\right) = 3779\ lb$$

$$A_p = 3.779\ in^2 = \frac{\pi}{4}D^2\ , \quad D = \underline{2.19\ \ in.}$$

6-26. Per Newton's Law we have

$$\sum F = m\,a \qquad where \qquad a = \frac{5\ {ft}/{s}}{0.5\,s} = 10\ {ft}/{s^2}$$

Summing forces on the 6000 lb weight and using values determined in the solution to Exercise 6-23 we have

$$p\,A_p - 3779\ lb = \frac{6000\ lb}{32.2\ {ft}/{s^2}} \times 10\ {ft}/{s^2} = 1863\ lb$$

$$p\,A_p = F_{cyl} = 5642\ lb$$

$$A_p\left(in^2\right) = \frac{5642\ lb}{p\left({lb}/{in^2}\right)} = \frac{5642\ lb}{1000\ {lb}/{in^2}} = 5.642\ in^2 = \frac{\pi}{4}D^2\ , \quad D = \underline{2.68\ in.}$$

6-27. $L_1 = L_2 = 10\ in \times \dfrac{2.54\ cm}{1\ in} = 25.4\ cm$

$\phi = 0°$, $F_{load} = 1000\ lb \times \dfrac{1\ N}{0.225\ lb} = 4444\ N$

(a) First Class Lever:

$$F_{cyl} = \frac{L_2}{L_1 \cos\phi} \times F_{load} = \frac{25.4}{25.4 \times 1} \times 4444 = \underline{4444\ N}$$

**Second Class Lever:**

$$F_{cyl} = \frac{L_2}{(L_1 + L_2)\cos\phi} \times F_{load} = \frac{25.4 \times 4444}{(25.4 + 25.4) \times 1} = \underline{2222\ N}$$

**Third Class Lever:**

$$F_{cyl} = \frac{L_1 + L_2}{L_2 \cos\phi} \times F_{load} = \frac{25.4 + 25.4}{25.4 \times 1} \times 4444 = \underline{8888\ N}$$

**(b) Repeat part (a) with $\theta = 10°$**

**<u>Answers are the same as those given in part (a).</u>**

**(c) Re peat part (a) with $\phi = 5°$ and $20°$**

**First Class Lever:**

$$F_{cyl}(\phi = 5°) = \frac{4444}{\cos 5°} = \underline{4461\ N}$$

$$F_{cyl}(\phi = 20°) = \frac{4444}{\cos 20°} = \underline{4729\ N}$$

**Second Class Lever:**

$$F_{cyl}(\phi = 5°) = \frac{2222}{\cos 5°} = \underline{2231\ N}$$

$$F_{cyl}(\phi = 20°) = \frac{2222}{\cos 20°} = \underline{2365\ N}$$

**Third Class Lever:**

$$F_{cyl}(\phi = 5°) = \frac{8888}{\cos 5°} = \underline{8922\ N}$$

$$F_{cyl}(\phi = 20°) = \frac{8888}{\cos 20°} = \underline{9458\ N}$$

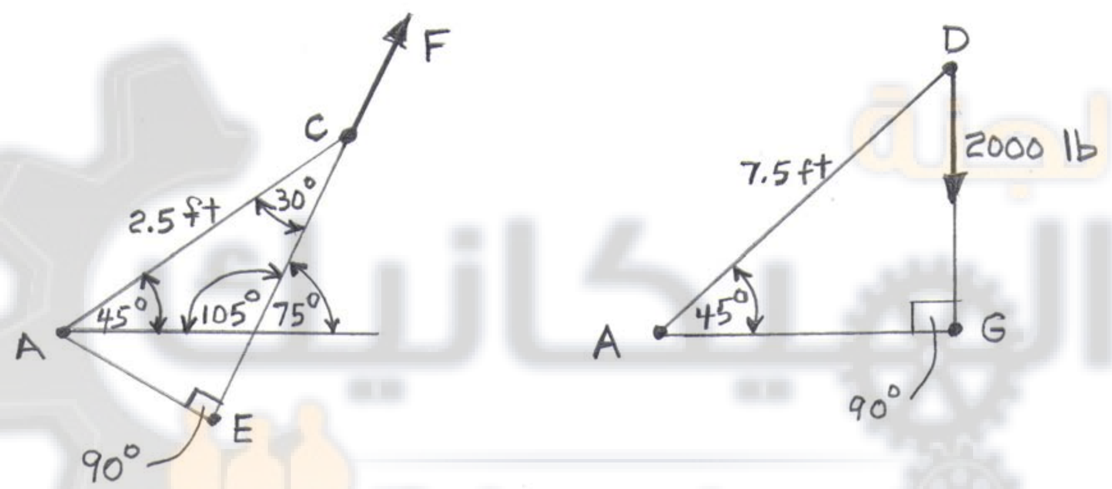
6-28. Equating moments about fixed pin C yields:

$$F_{cyl} \times 400\text{ mm} = F_{load} \times 500\text{ mm}$$

$$F_{cyl} = \frac{500}{400} \times F_{load} = 1.25 \times 1000\text{ N} = \underline{1250\text{ N}}$$

6-29. Equating moments about fixed pin A due to the cylinder force F and the 1000 lb weight yields:

2000 x perpend. dist. AG = F x perpend. dist. AE




From trigonometry of right triangles we have:

$$\cos 45° = \frac{AG}{7.5} \quad \text{Thus } AG = 7.5 \cos 45° = 5.30\text{ ft}$$

$$\sin 30° = \frac{AE}{2.5} \quad \text{Thus } AE = 2.5 \sin 30° = 1.25\text{ ft}$$

2000 x 5.30 = F x 1.25 Hence F = **8480 lb**

6-30. Setting the sum of the forces on pin C equal to zero (from Newton's Law of Motion, F = ma = 0 since a = 0 for constant velocity motion) yields the following for the X and Y axes:

Y axis: $F_{BC} \sin 60° - F_{BD} \sin 60° = 0$ Thus $F_{BC} = F_{BD}$

X axis: $F_{cyl} - F_{BC} \cos 60° - F_{BD} \cos 60° = 0$

Thus $F_{cyl} - 2\,F_{BC} \cos 60° = 0$ or $F_{BC} = \dfrac{F_{cyl}}{2 \cos 60°}$

Similarly setting the sum of forces on pin C equal to zero for the Y axis direction yields:

$F_{BC} \sin 60° - F_{load} = 0$ Therefore we have:

$$F_{load} = F_{BC} \sin 60° = \frac{\sin 60°}{2 \cos 60°} \times F_{cyl} = \frac{\tan 60°}{2} \times 1000 = \underline{866 \text{ lb}}$$

6-31. First, calculate the steady state piston velocity (V) prior to deceleration.

$$V = \frac{Q_{pump}}{A_{piston}} = \frac{\left(20/448\right) ft^3/s}{\left(\frac{\pi}{4} \times 2^2/144\right) ft^2} = \frac{0.0446}{0.0218} = 2.05\ ft/s$$

Next, calculate the deceleration (a) of the piston during the 1 inch displacement (S) using the constant acceleration (or deceleration) equation.

$$a = \frac{V^2}{2\,S} = \frac{\left(2.05\ ft/s\right)^2}{2 \times \frac{1}{12}\ ft} = 25.2\ ft/s^2$$

Substituting into Newton's Law of Motion Equation yields:

$$p_2\left(A_{piston} - A_{cushion\,plunger}\right) + \mu W - p_1 A_{piston} = \frac{W}{g} a$$

Solving for $p_2$ yields a usable equation.

$$p_2 = \frac{Wa/g + p_1 A_{piston} - \mu W}{A_{piston} - A_{cushion\ plunger}}$$

Substituting known values produces the desired result.

$$p_2 = \frac{1000 \times 25.2/32.2 + 500 \times \frac{\pi}{4} \times 2^2 - 0.15 \times 1000}{\frac{\pi}{4} \times 2^2 - \frac{\pi}{4} 0.75^2}$$

$$p_2 = \frac{783 + 1571 - 150}{3.14 - 0.442} = \frac{2204}{2.70} = \underline{816\ psi}$$

6-32. The following metric data are applicable:

Pump flow = 20 gpm = 0.0000632 × 20 = 0.00126 $m^3/s$

Hydr. cyl. dia. = 2 in = 2 in × $\frac{1\ ft}{12\ in}$ × $\frac{1\ m}{3.28\ ft}$ = 0.0508 m

Cushion plunger dia. = 0.75 in × $\frac{1\ ft}{12\ in}$ × $\frac{1\ m}{3.28\ ft}$ = 0.0191 m

Cushion plunger length = 1 in × $\frac{1\ ft}{12\ in}$ × $\frac{1\ m}{3.28\ ft}$ = 0.0254 m

Weight of cylinder load = 1000 lb × $\frac{1\ N}{0.225\ lb}$ = 4440 N

Coefficient of friction = 0.15

Pump pressure relief valve setting = 500 psi × $\frac{1\ Pa}{0.000145\ psi}$

$$= 3450 \text{ kPa}$$

**We now first calculate the steady state piston velocity (V) prior to deceleration.**

$$V = \frac{Q_{pump}}{A_{piston}} = \frac{0.00126 \ m^3/s}{\frac{\pi}{4} \times 0.0508^2 \ m^2} = 0.622 \ m/s$$

**Next we calculate the acceleration (a) of the piston during the 0.0254m displacement using the constant acceleration (or deceleration) equation.**

$$a = \frac{V^2}{2S} = \frac{\left(0.622 \ m/s\right)^2}{2 \times 0.0254 \ m} = 7.62 \ m/s^2$$

**Substituting into Newton's Law of Motion Equation and solving for $p_2$ yields:**

$$p_2 = \frac{Wa/g + p_1 A_{piston} - \mu W}{A_{piston} - A_{cushion\ plunger}}$$

**Substituting known values produces the desired result.**

$$p_2 = \frac{4440 \times 7.62/9.81 + 3{,}450{,}000 \times \frac{\pi}{4} \times 0.0508^2 - 0.15 \times 4440}{\frac{\pi}{4} \times 0.0508^2 - \frac{\pi}{4} \times 0.0191^2}$$

$$p_2 = \frac{3450 + 6990 - 666}{0.00203 - 0.000287} = \frac{9774}{0.001743} = \underline{5610 \ kPa}$$

# Chapter 7

## Hydraulic Motors

7-1. A limited rotation hydraulic actuator provides rotary output motion over a finite angle. A hydraulic motor is an actuator which can rotate continuously.

7-2. Simple design and subsequent low cost.

7-3. Since vane motors are hydraulically balanced, they are fixed displacement units.

7-4. The vanes must have some means other than centrifugal force to hold them against the cam ring. Some designs use springs while other types use pressure-loaded vanes.

7-5. Yes and either fixed or variable displacement units can be used.

7-6.
1. Volumetric efficiency equals the theoretical flow rate the motor should consume, divided by the actual flow rate consumed by the motor.
2. Mechanical efficiency equals the actual torque delivered by the motor divided by the torque the motor should theoretically deliver.
3. Overall efficiency equals the actual power delivered by the motor divided by the actual power delivered to the motor.

7-7. A motor uses more flow than it theoretically should because the motor inlet pressure is greater than the motor discharge pressure. Thus, leakage flow passes through a motor from the inlet port to the discharge port.

7-8. A hydrostatic transmission is a system consisting of a hydraulic pump, a hydraulic motor and appropriate valves and pipes, which can be used to provide adjustable speed drives for many practical applications. Four advantages of hydrostatic transmissions are:

1. Infinitely variable speed and torque in either direction and over the full speed and torque range.
2. Extremely high power per weight ratio.
3. Can be stalled without damage.
4. Low inertia of rotating members permits fast starting and stopping with smoothness and precision.

7-9. A hydraulic motor delivers less torque than it theoretically should because frictional losses exist in an actual hydraulic motor.

7-10. The theoretical torque output is proportional to inlet pressure and volumetric displacement which is independent of motor speed.

7-11. True since $T_A = T_T \eta_m = \dfrac{V_D P}{6.28} \times \eta_m$

7-12. Flow rate and volumetric displacement.

7-13. Displacement is the volume of oil required to produce one revolution of the motor. Torque rating is the torque delivered by the motor at rated pressure.

7-14. Some designs use springs, whereas other types use pressure-loaded vanes.

7-15. Pressure exerts a force on the pistons. The piston thrust is transmitted to the angled swash plate causing torque to be created in the drive shaft.

**7-16.** An increase in the working load results in an increase in volumetric displacement. This decreases motor speed for a constant pump flow rate.

**7-17.** Piston motor.

**7-18.** By using the following equation:

$$Q_{actual} = \frac{Q_{theoretical}}{\eta_V} = \frac{V_D\ N}{231\ \eta_V}$$

**7-19.** First, solve for the volumetric displacement.

$$V_D = \pi\left(1.25^2 - 0.4^2\right) \times 0.75 = 3.31\ in^3$$

Then solve for the pressure that must be developed to overcome the load.

$$p = \frac{6.28\,T}{V_D} = \frac{6.28 \times 750}{3.31} = \underline{1423\ psi}$$

**7-20.** $V_D = \pi\left(R_V^2 - R_R^2\right)L = \pi\left(0.032^2 - 0.010^2\right) \times 0.020 = 58.1 \times 10^{-6} m^3$

$$p = \frac{6.28\,T}{V_D} = \frac{6.28 \times 85}{58.1 \times 10^{-6}} = \underline{9.19\ MPa}$$

**7-21.** $HHP = \frac{pQ}{1714} = \frac{1800 \times 20}{1714} = 21.0\ hp$

$$Output\ HP = HHP = \frac{T\ N}{63{,}000} = 21.0$$

$$T = \frac{21.0 \times 63{,}000}{N} = \frac{21.0 \times 63{,}000}{800} = \underline{1654 \ \ in \cdot lb}$$

**7-22.** $$Hydraulic \ \ power = pQ = 12{,}000 \ kPa \times \frac{72}{60} \times 10^{-3} \ m^3/s = 14.4 \ kW$$

$$T\,(N \cdot m) \times \omega\,(rad/s) = 14{,}400 W$$

$$T = \frac{14{,}400}{\omega} = \frac{14{,}400}{800 \times 2\pi/60} = \underline{172 \ \ N \cdot m}$$

**7-23.** (a) $$N = \frac{231 Q_T}{V_D} = \frac{231 \times 15}{6} = \underline{577.5 \ rpm}$$

(b) $$T_T = \frac{V_D\, p}{6.28} = \frac{6 \times 2000}{6.28} = \underline{1911 \ \ in \bullet lb}$$

(c) $$HP_T = \frac{T_T N}{63{,}000} = \frac{1911 \times 577.5}{63{,}000} = \underline{17.5 \ \ HP}$$

**7-24.** (a) $$N = \frac{Q_T}{V_D} = \frac{0.001 \ m^3/s}{100 \times 10^{-6} \ m^3/rev} = 10 \ rev/s = \underline{600 \ \ rpm}$$

(b) $$T_T = \frac{V_D\, p}{6.28} = \frac{\left(100 \times 10^{-6}\right) \times \left(140 \times 10^5\right)}{6.28} = \underline{222.9 \ \ N \bullet m}$$

(c) $$Power = T_T N = (222.9) \times (10 \times 2\pi) = 14{,}000 \ W = \underline{14.0 \ \ kW}$$

**7-25.** *Equations are:* $Q = \frac{V_D N}{231}$, $T = \frac{p V_D}{6.28}$ *and* $HP = \frac{T N}{63{,}000}$

*Thus* $V_D = \frac{6.28\,T}{p} = \frac{6.28 \times (10 \times 1000)}{1000} = 62.8\ in^3$

**and** $Q = \frac{6.28 \times 30}{231} = \underline{8.16\ \text{gpm}}$

**and** $HP = \frac{(10 \times 1000) \times 30}{63{,}000} = \underline{4.76\ HP}$

**7-26.** $Q = V_D N$ *and* $T = \frac{p V_D}{6.28}$

*So* $V_D = \frac{6.28\,T}{p} = \frac{6.28(0.3 \times 4000)}{1 \times 10^8} = 0.0000754\ m^3 = 0.0754\ L$

$Q = V_D N = \frac{0.0000754 \times 30}{60} = \underline{0.0000377\ m^3/s}$

$Power = p\left(N/m^2\right) \times Q\left(m^3/s\right) = \left(1 \times 10^8\right) \times 0.0000377 = \underline{3.77\ kW}$

**7-27.** $HP = \frac{(\Delta p)Q}{1714} = \frac{1600 \times 100}{1714} = \underline{93.3\ HP}$

**7-28.** **The metric data are as follows:**

$$\text{Pump discharge pressure} = 2000\text{ psi} \times \frac{1\text{ kPa}}{0.145\text{ psi}} = 13{,}800\text{ kPa}$$

$$\text{Pump flow} = 100\text{ gpm} = 0.0000632 \times 100 = 0.00632\ m^3/s$$

$$\text{Pressure at motor inlet} = 1800\text{ psi} \times \frac{1\text{ kPa}}{0.145\text{ psi}} = 12{,}400\text{ kPa}$$

$$\text{Motor discharge pressure} = 200\text{ psi} \times \frac{1\text{ kPa}}{0.145\text{ psi}} = 1380\text{ kPa}$$

$$Power = (\Delta p)Q = (12{,}400 - 1{,}380)\ kPa \times 0.00632\ m^3/s = \underline{69.6\ kW}$$

**7-29.** **Friction**

**7-30.** **Friction**

**7-31.** $HP = \dfrac{T\,N}{63{,}000}$ **so** $T = \dfrac{63{,}000 \times HP}{N} = \dfrac{63{,}000 \times 4}{1750} = \underline{144\text{ in} \bullet \text{lb}}$

**7-32.** $(a)\quad T = \dfrac{pV_D}{6.28}$

**Since p and $V_D$ are both constant, torque T remains constant. This would, however, double the HP per the following equation:**

$$HP = \frac{T\,N}{63{,}000}$$

**(b) Torque T remains constant while the HP is cut in half.**

**7-33. (a)** $$HHP = \frac{p\,Q}{1714} = \frac{1800 \times 20}{1714} = 21.0\ hp$$

**Output HP = HHP – HP loss = 21.0 – 4 = 17.0 hp**

$$Output\ HP = \frac{T\,N}{63{,}000} = 17.0\ hp$$

$$T = \frac{17.0 \times 63{,}000}{800} = \underline{1339\ \ in \cdot lb}$$

**(b)** $$\eta_o = \frac{output\ HP}{HHP} = \frac{17.0}{21.0} = 0.809 = \underline{80.9\%}$$

**7-34. (a)** $$Hydraulic\ \ power = \ p\,Q = 14.4\ kW$$

$$T\omega = 14.4 - 3 = 11.4\ \ kW$$

$$T = \frac{11{,}400}{800 \times {}^{2\pi}\!/_{60}} = \underline{136\ N \cdot m}$$

**(b)** $$\eta_o = \frac{11.4}{14.4} = 0.792 = \underline{79.2\%}$$

**7-35. (a) First, calculate the theoretical flow rate.**

$$Q_T = \frac{V_D\ N}{231} = \frac{8 \times 2000}{231} = 69.3\ \text{gpm}$$

$$\eta_V = \frac{Q_T}{Q_A} = \frac{69.3}{75} = 0.924 = \underline{92.4\%}$$

**(b) To find $\eta_m$, we need to calculate the theoretical torque.**

$$T_T = \frac{V_D\ p}{6.28} = \frac{8 \times 1500}{6.28} = 1911\ in \bullet lb$$

$$\eta_m = \frac{T_A}{T_T} = \frac{1800}{1911} = 0.942 = \underline{94.2\%}$$

$$(c)\quad \eta_o = \eta_v \eta_o = 0.924 \times 0.942 = 0.870 = \underline{87.0\%}$$

$$(d)\ HP = \frac{T_A\ N}{63{,}000} = \frac{1800 \times 2000}{63{,}000} = \underline{57.1\ HP}$$

**7-36.** (a) $Q_T = V_D N = \left(130 \times 10^{-6}\ m^3/rev\right) \times \left(\frac{2000}{60}\ rev/s\right) = 0.00433\ m^3/s$

$$\eta_V = \frac{Q_T}{Q_A} = \frac{0.00433}{0.005} = 0.866 = \underline{86.6\%}$$

$$(b)\ T_T = \frac{V_D N}{6.28} = \frac{\left(130 \times 10^{-6}\right) \times \left(105 \times 10^5\right)}{6.28} = 217.4\ N \bullet m$$

$$\eta_m = \frac{T_A}{T_T} = \frac{200}{217.4} = 0.920 = \underline{92.0\%}$$

$$(c)\quad \eta_o = \eta_v \eta_m = 0.866 \times 0.920 = 0.797 = \underline{79.7\%}$$

(d) Power $= T_A N = 200 \times \left(\frac{2000}{60} \times 2\pi\right) = 41{,}900\ W = \underline{41.9\ kW}$

**7-37.** $$\eta_o = \frac{T_A N / 63{,}000}{p Q_A / 1714} = \frac{1300 \times 1750 / 63{,}000}{1000 \times 75 / 1714} = 0.824 = \underline{82.4\%}$$

**7-38.** $$T_{theor} = \frac{p V_D}{6.28} = T_{act} \quad if \quad \eta_m = 100\% \quad and$$

$$Q_{theor} = \frac{V_D N}{231} = Q_{act} \quad if \quad \eta_V = 100\%$$

Thus for $\eta_V$ = 100% we have:

$$\frac{Q_{act}}{N} = \frac{V_D}{231} = 0.075, \quad So \quad V_D = \underline{17.3\ in^3}$$

For $\eta_m = 100\%$ we have: $T_{act} = \frac{3000 \times 17.3}{6.28} = \underline{8260\ in \bullet lb}$

Note that the calculated values of $V_D$ and T are theoretical values. Actual values can be calculated as follows:

Since a motor consumes more flow than it theoretically should we have: $V_{D\,act} = \eta_V V_{D\,theor}$

Similarly since a motor produces less torque than it theoretically should we have: $T_{act} = \eta_m T_{theor}$

Hence we need values of $\eta_V$ and $\eta_m$ to obtain actual values of $V_D$ and T.

A relationship in terms of overall efficiency can be developed as follows:

$$V_{D\,act}T_{act} = \eta_m \eta_V V_{D\,theor}T_{theor} = \eta_o V_{D\,theor}T_{theor}$$

But this equation alone does not allow for the calculation of $V_{D\,act}$ and $T_{act}$ even if the value of $\eta_o$ is given.

7-39. (a) Pump theoretical flowrate $= \dfrac{\text{displ. of pump} \times \text{pump speed}}{231}$

$$= \frac{6 \times 100}{231} = 26.0 \text{ gpm}$$

Pump act. flowrate = pump theor. flowrate x pump vol. eff.
= 26.0 x 0.85 = 22.1 gpm

Motor theor flowrate = pump act flowrate x motor vol. eff.
= 22.1 x 0.94 = 20.8 gpm

$$\text{Motor displacement} = \frac{\text{motor theoretical flowrate} \times 231}{\text{motor speed}}$$

$$= \frac{20.8 \times 231}{600} = \underline{8.01 \text{ in}^3}$$

(b) $$HP_{del\ to\ motor} = \frac{\text{system pressure} \times \text{actual flowrate to motor}}{1714}$$

$$= \frac{1500 \times 22.1}{1714} = 19.3 \text{ HP}$$

$$HP_{del\ by\ motor} = 19.3 \times 0.94 \times 0.92 = 16.7 \text{ HP}$$

$$\text{Torque delivered by motor} = \frac{\text{HP delivered by motor} \times 63{,}000}{\text{motor speed}}$$

$$= \frac{16.7 \times 63{,}000}{600} = \underline{1756 \text{ in} \bullet \text{lb}}$$

7-40. (a) $Q_{TP} = V_{DP}N_P = \left(100 \times 10^{-6}\right) \times \frac{1000}{60} = 0.00167\ m^3/s$

$$Q_{AP} = Q_{TP}\eta_{VP} = 0.00167 \times 0.85 = 0.00142\ m^3/s$$

$$Q_{TM} = Q_{AP}\eta_{VM} = 0.00142 \times 0.94 = 0.00133\ m^3/s$$

$$V_{DM} = \frac{Q_{TM}}{N_M} = \frac{0.00133}{600/60} = 0.000133\ m^3 = \underline{133\ cm^3}$$

(b) $Power_{act\ to\ motor} = p\,Q_{AM} = \left(105 \times 10^5\right) \times 0.00142 = 14{,}900\ W$

$$Power_{act\ by\ motor} = 14{,}900 \times 0.94 \times 0.92 = 12{,}900\ W$$

$$T_{act\ by\ motor} = \frac{12{,}900}{600 \times 2\pi/60} = \underline{205\ N \bullet m}$$

# Chapter 8

# Hydraulic Valves

8-1. Directional control valves determine the path through which a fluid traverses within a given circuit.

8-2. A check valve is a directional control valve which permits free flow in one direction and prevents any flow in the opposite direction.

8-3. A pilot check valve always permits free flow in one direction, but permits flow in the normally blocked opposite direction only if pilot pressure is applied at the pilot pressure port of the valve.

8-4. A four-way directional control valve is one which has four different ports.

8-5. This valve contains a spool which can be actuated into three different functioning positions. The center position is obtained by the action of the springs alone.

8-6.
1. Manually
2. Air piloted
3. Solenoid actuated

8-7. A solenoid is an electric coil. When the coil is energized, it creates a magnetic force that pulls the armature into the coil. This causes the armature to push on the push rod to move the spool of the valve.

8-8. The open-center type connects all ports together when the valve is unactuated. The closed-center design has all ports blocked when the valve is unactuated.

8-9. A shuttle valve is another type of directional control valve. It permits a system to operate from either of two fluid power sources. One application is for safety in the event that the main pump can no longer provide hydraulic power to operate emergency devices.

8-10. To limit the maximum pressure experienced in a hydraulic system.

8-11. A pressure reducing valve is another type of pressure control valve. It is used to maintain reduced pressures in specified locations of hydraulic systems.

8-12. An unloading valve is used to permit a pump to build up to an adjustable pressure setting and then allow it to discharge to the tank at essentially zero pressure as long as pilot pressure is maintained on the valve from a remote source.

8-13. A sequence valve is a pressure control device. Its purpose is to cause a hydraulic system to operate in a pressure sequence.

8-14. To maintain control of a vertical cylinder so that it does not descend due to gravity.

8-15. Flow control valves are used to regulate the speed of hydraulic cylinders and motors by controlling the flow rate to these actuators.

8-16. In English units capacity coefficient is defined as the flow rate of water in gpm that will flow through the valve at a pressure drop of 1 psi.

**In Metric units capacity coefficient is defined as the flow rate of water in Lpm that will flow through the valve at a pressure drop of 1 kPa.**

**8-17. English Units:** $gpm/\sqrt{psi}$

**Metric Units:** $Lpm/\sqrt{kPa}$

**8-18. A pressure compensated flow control valve is one which provides the desired flow rate regardless of changes in system pressure.**

**8-19. A servo valve is a directional control valve which has infinitely variable positioning capability. Servo valves are coupled with feedback sensing devices which allow for the vary accurate control of position, velocity and acceleration of an actuator.**

**8-20. Mechanical-hydraulic servo valves use only mechanical components. Electrical-hydraulic servo valves typically use an electrical torque motor, a double-nozzle pilot stage and a sliding spool second stage.**

**8-21. A hydraulic fuse prevents hydraulic pressure from exceeding an allowable value in order to protect circuit components from damage. It is analogous to an electric fuse.**

**8-22. The upstream presssure is higher than the downstream pressure. A measurement of this pressure drop can be used to determine the flow rate.**

**8-23. In the design of Figure 8-5, the check valve poppet has the pilot piston attached to the threaded poppet stem by a nut. The light spring holds the poppet seated in a no-flow condition by pushing against the pilot piston. The purpose of the separate drain port is to prevent oil from creating a pressure buildup on the bottom of the piston.**

**8-24.** **Pilot check valves are frequently used for locking hydraulic cylinders in position.**

**8-25.** **Flow can go through the valve in four unique ways depending on the spool position.**

**(a) <u>Spool Position 1</u>: Flow can go from P to A and B to T.**
**(b) <u>Spool Position 2</u>: Flow can go from P to B and A to T.**

**8-26.** **A compound pressure relief valve (See Figure 8-23) is one which operates in two stages. Referring to Figure 8-24, the operation is as follows:**

**In normal operation the balanced piston is in hydraulic balance. For pressures less than the valve setting, the piston is held on its seat by a light spring. As soon as pressure reaches the setting of the adjustable spring, the poppet is forced off its seat. This limits the pressure in the upper chamber.**

**The restricted flow through the orifice and into the upper chamber results in an increase in pressure in the lower chamber. This causes an imbalance in hydraulic forces which tends to raise the piston off its seat. When the pressure difference between the upper and lower chambers reaches 20 psi, the large piston lifts off its seat to permit flow directly to tank.**

**8-27.** **Unloading valve: see Figure 8-27.**
**Sequence valve: see Figure 8-29.**

**8-28.** **This design incorporates a hydrostat which maintains a constant 20 psi differential across the throttle which is an orifice whose area can be adjusted by an external knob setting. The orifice area setting determines the flow rate to be controlled. The hydrostat is held normally open by a light spring.**

**However, it starts to close as inlet pressure increases and overcomes the light spring force. This closes the**

opening through the hydrostat and, thereby, blocks off all flow in excess of the throttle setting.

As a result, the only oil that will pass through the valve is the amount which 20 psi can force through the throttle. Flow exceeding this amount can be used by other parts of the circuit or return to the tank via the pressure relief valve.

8-29. Two

8-30. Three

8-31. To shift the spool in directional control valves.

8-32. 1. Using non-pressure-compensated flow control valves.
2. Using pressure-compensated flow control valves.

8-33. The pressure at which a pressure relief valve begins to open.

8-34. One port connects to the pressure line from the pump.
Second port connects to the drain line to the oil tank.

8-35. Control direction of flow.
Control flow rate.
Control pressure.

8-36. A hydraulic fuse, as in the case of a pressure relief valve, prevents hydraulic pressure from exceeding an allowable value in order to protect circuit components from physical damage. A hydraulic fuse is analogous to an electrical fuse because they both are one-shot devices. On the other hand, a pressure relief valve is analogous to an electrical circuit breaker because they both are resetable devices.

8-37. Position is the location of the spool inside the valve.
Way is the flow path through the valve.

Port is the opening in the valve body for the fluid to enter or exit.

8-38. A cartridge valve is a valve that is designed to be assembled into a cavity of a ported manifold block (alone or along with other cartridge valves and hydraulic components) in order to perform the valve's intended function.

8-39. The slip-in design cartridge valve uses a bolted cover while a screw-type design uses threads for assembling into the manifold block.

8-40.
1. Reduced number of fittings to connect hydraulic lines between various components in a system.
2. Reduced oil leakage and contamination due to fewer fittings.
3. Lower system installation time and costs.
4. Reduced service time since faulty cartridge valves can be easily changed without disconnecting fittings.
5. Smaller space requirements of overall system.

8-41. Directional control, pressure relief, pressure reducing, unloading and flow control functions.

8-42. Integrated hydraulic circuits are compact hydraulic systems formed by integrating various cartridge valves and other components into a single, machined, ported manifold block.

8-43. (a) $F_{valve\,closed} = k\,S_{initial} = 2000\,{}^{lb}\!/_{in.} \times 0.15\ in. = 300\ lb$

$p_{cracking}A_{poppet} = 300\ lb$

$p_{cracking} \times 0.65\ in^2 = 300\ lb$ **so** $p_{cracking} = \underline{462\ psi}$

(c) $F_{fully\,open} = k\,S_{fully\,open} = 2000\,{}^{lb}\!/_{in.} \times 0.25\ in. = 500\ lb$

$$p_{full\ pump\ flow} \times A = 500\ lb$$

$$p_{full\ pump\ flow} \times 0.65\ in^2 = 500\ lb \quad so \quad p_{full\ pump\ flow} = \underline{769\ psi}$$

**8-44.** $$F_{valve\ closed} = k\left({lb}/{in.}\right) \times l\,(in.) = 2000\,l = p_{cracking}\left({lb}/{in^2}\right) \times A_{poppet}\left(in^2\right)$$

**so** $$p_{cracking} = \frac{2000\,l}{0.65} = 3077\,l$$

$$F_{fully\ open} = k\left(l + 0.1\right) = 2000\left(l + 0.1\right) = 2000\,l + 200 = p_{full\ pump\ flow} \times A_{poppet}$$

**so** $$p_{full\ pump\ flow} = \frac{2000\,l + 200}{0.65} = 3077\,l + 307.7$$

$$\frac{p_{full\ pump\ flow}}{p_{cracking}} = 1.40 = \frac{3077\,l + 307.7}{3077\,l}$$

$$3077\,l + 307.7 = 4308\,l \quad so \quad l = \underline{0.25\ in.}$$

**8-45.** **(a)** $$F_{valve\ closed} = k\,S_{initial} = 3200\,{N}/{cm} \times 0.50\,cm = 1600\,N$$

$$p_{cracking} A_{poppet} = 1600\,N$$

$$p_{cracking}\left(4.20 \times 10^{-4}\ m^2\right) = 1600\,N$$

**So** $$p_{cracking} = 381 \times 10^4\ {N}/{m^2} = \underline{3.81\ MPa}$$

**(b)** $$F_{fully\ open} = k\,S_{fully\ open} = 3200\,{N}/{cm} \times 0.80\,cm = 2560\,N$$

$$p_{full\ pump\ flow} A_{poppet} = 2650\,N$$

$$p_{full\ pump\ flow}\left(4.20 \times 10^{-4}\ m^2\right) = 2560\,N$$

**So** $p_{full\ pump\ flow} = 610 \times 10^4\ {}^{N}\!/_{m^2} = \underline{6.10\ MPa}$

**8-46.** $F_{valve\ closed} = k\left({}^{N}\!/_{cm}\right) \times l\,(cm) = 3200\,l = p_{cracking}\left({}^{N}\!/_{m^2}\right) \times A_{poppet}\left(m^2\right)$

$$p_{cracking} = \frac{3200\,l}{4.20 \times 10^{-4}} = 762 \times 10^4\ l$$

$$F_{fully\ open} = k\,(l + 0.30) = 3200\,(l + 0.30) = 3200\,l + 960 = p_{full\ pump\ flow} \times A_{poppet}$$

$$p_{full\ pump\ flow} = \frac{3200\,l + 960}{4.20 \times 10^{-4}} = (762\,l + 229) \times 10^4$$

$$\frac{p_{full\ pump\ flow}}{p_{cracking}} = 1.40 = \frac{(762\,l + 229) \times 10^4}{762\,l \times 10^4}$$

$762\,l + 229 = 1067\,l \quad so \quad l = \underline{0.75\ cm}$

**8-47.** $HP = \dfrac{p\,Q}{1714} = \dfrac{2000 \times 25}{1714} = \underline{29.2\ HP}$

**8-48.** $HP = \dfrac{p\,Q}{1714} = \dfrac{30 \times 25}{1714} = \underline{0.44\ HP}$

**8-49.** $kW\,Power = pQ = \left(140 \times 10^5\right) \times \left(0.0016 \times 10^{-3}\right) = \underline{22.4kW}$

**8-50.** $kW\ Power = p\,Q = \left(2 \times 10^5\right) \times \left(0.0016 \times 10^{-3}\right) = \underline{0.32\ kW}$

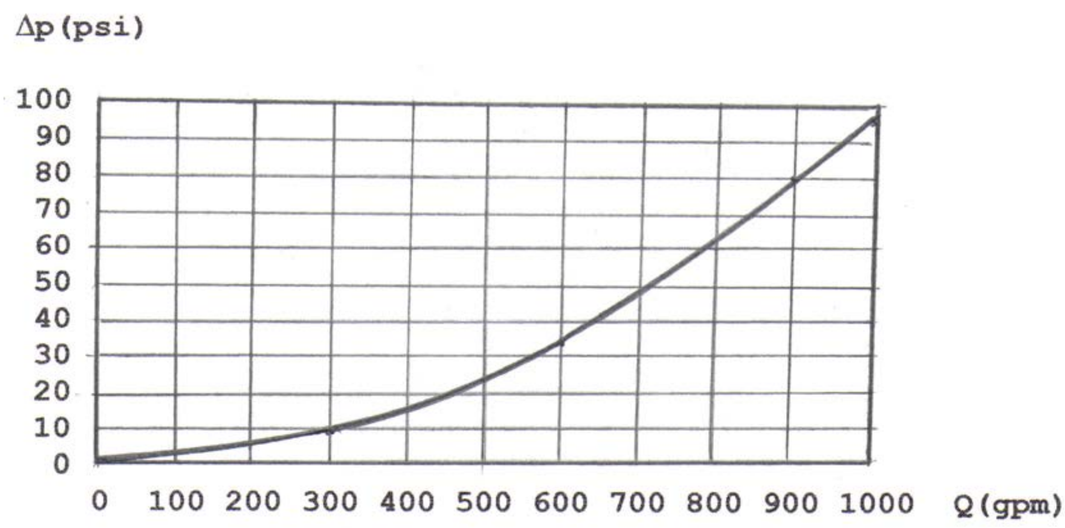
**8-51.** $Q = 38.1 C A\sqrt{\frac{\Delta p}{SG}} = 38.1 \times 0.80 \times \frac{\pi}{4} \times 2^2 \sqrt{\frac{50}{0.9}} = \underline{713\ gpm}$

**8-52.** $Q = 1.41 C A\sqrt{\frac{\Delta p}{SG}} = 1.41 \times 0.80 \times \frac{\pi}{4} \times 0.055^2 \sqrt{\frac{300}{0.9}} = \underline{0.0489\ m^3/s}$

**8-53.** $Q = 38.1 \times 0.80 \times \frac{\pi}{4} \times 2^2 \sqrt{\frac{\Delta p}{0.9}} = 100.8\sqrt{\Delta p}$

**Let's develop a table of values of Q versus Δp and plot the corresponding curve.**

| Q(gpm) | Δp | Q(gpm) | Δp |
|---|---|---|---|
| 0 | 0 | 781 | 60 |
| 319 | 10 | 843 | 70 |
| 451 | 20 | 902 | 80 |
| 552 | 30 | 956 | 90 |
| 638 | 40 | 1008 | 100 |
| 713 | 50 | | |



The graph is quicker to use but is not as accurate as the equation. A pressure gage can be calibrated (according to this relationship) to read Q directly rather $\Delta$p.

8-54. At a flow rate of 5 gpm the pressure drop is 47 psi. Using Equation 8-2 we have:

$$C_v = \frac{Q}{\sqrt{\frac{\Delta p}{SG}}} = \frac{5}{\sqrt{\frac{47}{0.9}}} = \frac{5}{7.23} = \underline{0.692\ gpm/\sqrt{psi}}$$

8-55. The valve identified by number 1 has the highest capacity coefficient because it has the lowest pressure drop for a given flow rate.

8-56. Substituting into Equation 8-2 using English units yields:

$$Q = C_v\sqrt{\frac{\Delta p}{(SG)}} = 1.5\sqrt{\frac{100}{0.90}} = \underline{15.8\ gpm}$$

8-57. Substituting into Equation 8-2 using Metric units yields:

$$Q = C_v\sqrt{\frac{\Delta p}{(SG)}} = 2.2\sqrt{\frac{687}{0.90}} = \underline{60.8\ Lpm}$$

8-58. For constant cylinder speed, the summation of forces on the hydrulic cylinder must equal zero. Thus we have

$$-W - p_1A_P + p_2\left(A_P - A_r\right) = 0$$

where $p_1$ = pressure relief valve setting = 750 psi

$$A_P = \frac{\pi}{4}(2\,in.)^2 = 3.14\ in.^2$$

$$A_P - A_r = 3.14\ in.^2 - \frac{\pi}{4}(1\ in.)^2 = 2.35\ in.^2$$

**(a) W = 2000 lb**

$$-2000\ lb - 750\frac{lb}{in.^2}\times 3.14\ in.^2 + p_2\left(2.35\ in.^2\right) = 0$$
$$-2000 - 2360 + 2.35\,p_2 = 0$$

$$p_2 = \underline{1860\ \ psi}$$

**(b)** $W = 0\ ,\quad p_2 = \underline{1000\ \ psi}$

**5-59. (a) Substituting into Equation 8-2 using English units yields:**

$$Q = C_v\sqrt{\frac{\Delta p}{(SG)}} = 0.5\sqrt{\frac{1860}{0.90}} = 22.7\ \ gpm$$

**where** $\Delta p = p_2$ **since the flow control valve discharges directly to the oil tank.**

**This is the flow rate through the flow control valve and thus the flow rate of the fluid leaving the hydraulic cylinder. Thus we have**

$$\left(A_p - A_r\right)v_p = Q$$

$$2.35\ in.^2 \times v_p\left(\frac{in.}{s}\right) = 22.7\frac{gal}{\min}\times\frac{231\ in.^3}{1\ gal}\times\frac{1\min}{60\ s}$$

$$v_p = \underline{37.2\ in./s}$$

**(b)** $Q = C_v\sqrt{\frac{\Delta p}{(SG)}} = 0.5\sqrt{\frac{1000}{0.90}} = 16.7\ gpm$

**so** $2.35\, v_p = 16.7 \times \frac{231}{60} \quad or \quad v_p = \underline{27.4\ in./s}$

**8-60.** **Cylinder piston dia. = 50.8 mm**
**Cylinder rod dia. = 25.4 mm**
**Pressure relief valve setting = 5150 kPa**

**For constant cylinder speed, the summation of forces on the hydraulic cylinder must equal zero. Thus we have**

$$-W - p_1 A_p + p_2\left(A_p - A_r\right) = 0$$

**where** $p_1$ **= pressure relief valve setting = 5150 kPa**

$$A_p = \frac{\pi}{4}(0.0508\,m)^2 = 0.00203\,m^2$$

$$A_p - A_r = 0.00203\,m^2 - \frac{\pi}{4}(0.0254\,m)^2 = 0.00152\ m^2$$

**(a) W = 8890 N**

$$-8890\,N - 5150\times10^3\,\frac{N}{m^2}\times2.03\times10^{-3}\,m^2 + p_2\left(0.00152\ m^2\right) = 0$$

$$-8890 - 10{,}450\,m^2 + 0.00152\,p_2 = 0, \qquad p_2 = \underline{12{,}700\ kPa}$$

**(b) W = 0,** $p_2$ **= <u>6880 kPa</u>**

**8-61.** **(a) Substituting into Equation 8-2 using Metric units yields:**

$$Q = C_v \sqrt{\frac{\Delta p}{(SG)}} = 0.72\sqrt{\frac{12{,}700}{0.90}} = 85.5\ Lpm$$

**where** $\Delta p = p_2$ **since the flow control valve discharges directly to the oil tank. This is the flow rate through the flow control valve and thus the flow rate of the fluid leaving the hydraulic cylinder. Thus we have**

$$\left(A_p - A_r\right)v_p = Q$$

$$0.00152\,m^2 \times v_p\left(\frac{m}{s}\right) = 85.5\frac{L}{\min} \times \frac{1m^3}{10^3\ L} \times \frac{1\min}{60\ s}$$

$$v_p = \underline{0.938\ m/s}$$

**(b)** $$Q = C_v \sqrt{\frac{\Delta p}{(SG)}} = 0.72\sqrt{\frac{6880}{0.90}} = 63.0\ Lpm$$

**So** $$0.00152 v_p = \frac{63.0}{60 \times 10^3} \quad or \quad v_p = \underline{0.691\ m/s}$$

# Chapter 9

## Hydraulic Circuit Design and Analysis

9-1. 1. Safety of operation.
2. Performance of desired function.
3. Efficiency of operation.

9-2. A regenerative circuit is used to speed up the extending speed of a double-acting hydraulic cylinder.

9-3. The load carrying capacity for a regenerative cylinder equals the pressure times the piston rod area rather than the pressure times the piston area.

9-4. Fail-safe circuits are those designed to prevent injury to the operator or damage to equipment. In general, they prevent the system from accidentally falling on an operator and they also prevent overloading of the system.

9-5. A hydraulic motor may be driving a machine having a large inertia. This would create a flywheel effect on the motor and stopping the flow of fluid to the motor would cause it to act as a pump. The circuit should be designed to provide fluid to the motor while it is pumping to prevent it from pulling in air.

9-6. Open circuit hydrostatic transmissions are drives in which the pump draws its fluid from a reservoir. Its output is then directed to a hydraulic motor and discharged from the motor back into the reservoir. In a closed circuit drive, exhaust oil from the motor is returned directly to the pump inlet.

**9-7.** **An air-over-oil system is one using both air and oil to obtain the advantages of each medium.**

**9-8.** **A mechanical hydraulic servo system is a closed-loop system using a mechanical feedback. One application is an automotive power steering system.**

**9-9.** **One relief valve (nearest pump) protects the system (pump to three-way valve) from over-pressure due to pump flow against a closed three-way valve. The other relief valve (nearest the accumulator) protects the system (rod end of cylinder to check valve and accumulator) from over-pressure while the cylinder is extending.**

**9-10.** **Yes. Use a regenerative circuit with a cylinder having a rod area equal to one-half the piston area. Also can use a double rod cylinder having equal area rods at each end.**

**9-11.** **Valve spool moves with the load.**
**Valve sleeve moves with the input.**

**9-12.** **Yes because** $v_1 = v_2$ **for both extension and retraction strokes when** $A_{P_1} - A_{R_1} = A_{P_2}$ **.**

**9-13.** **This is a regenerative system so the cylinder extends.**

$$v_{P\,ext} = \frac{Q_P}{A_r} = \frac{2\,\dfrac{gal}{\min} \times \dfrac{231\,in^3}{1\,gal} \times \dfrac{1\min}{60\,s}}{\dfrac{\pi}{4}\left(1\,in\right)^2} = \frac{7.7\,{in^3}/{s}}{0.785\,in^2} = \underline{9.81\,{in}/{s}}$$

$$F_{ext} = p\,A_r = 1000\,\frac{lb}{in^2} \times 0.785\,in^2 = \underline{785\,lb}$$

**9-14. This is a regenerative system so the cylinder extends.**

$$v_{P\,ext} = \frac{Q_P}{A_r} = \frac{0.008\,\frac{m^3}{\min} \times \frac{1\,\min}{60\,s}}{\frac{\pi}{4}(0.025\,m)^2} = \frac{0.000133\,m^3/s}{0.000491\,m^2} = \underline{0.271\,m/s}$$

$$F_{ext} = p\,A_r = 7\times10^6\,\frac{N}{m^2} \times 0.000491\,m^2 = \underline{3440\,N}$$

**9-15.** (a) $$v_{P\,ext} = \frac{Q_P}{A_r} = \frac{25\,\frac{gal}{min} \times \frac{231\,in^3}{1\,gal} \times \frac{1\,min}{60\,s}}{10\,in^2} = \underline{9.63\,in/s}$$

$$F_{load\,ext} = pA_r = 1500\,\frac{lb}{in^2} \times 10\,in^2 = \underline{15{,}000\,lb}$$

(b) $$v_{P\,ret} = \frac{Q_P}{A_p - A_r} = \frac{25 \times \frac{231}{60}}{20 - 10} = \underline{9.63\,in/s}$$

$$F_{load\,ret} = P\left(A_P - A_r\right) = 1500\,lb/in^2 \times 10\,in^2 = \underline{15{,}000\,lb}$$

**9-16.** (a) $$v_{P\,ext} = \frac{Q_P}{A_r} = \frac{0.0016\,m^3/s}{65 \times 10^{-4}\,m^2} = \underline{0.25\,m/s}$$

$$F_{load\,ext} = pA_r = \left(105\times10^5\right)\times\left(65\times10^{-4}\right) = \underline{68{,}300\,N}$$

(b) $$v_{P\,ret} = \frac{Q_P}{A_P - A_r} = \frac{0.0016}{130 \times 10^{-4} - 65 \times 10^{-4}} = \underline{0.25\,m/s}$$

$$F_{load\,ret} = p\left(A_P - A_r\right) = \left(105\times10^5\right)\times\left(65\times10^{-4}\right) = \underline{68{,}300\,N}$$

**9-17. Use the circuit of Figure 9-9 entitled “Hydraulic Cylinder Sequence Circuit”. The left cylinder of Figure 9-9 becomes**

the clamp cylinder of Figure 9-26 and the right cylinder of Figure 9-9 becomes the work cylinder of Figure 9-26.

9-18. A check valve is needed in the hydraulic line just upstream from where the pilot line to the unloading valve is connected to the hydraulic line. Otherwise, the unloading valve would behave like a pressure relief valve and thus, valuable energy would be wasted.

9-19. For cylinder 1 to extend, the directional control valve (DCV) must be in its left flow mode configuration. For this position of the DCV, the blank end of cylinder 2 is vented back to the reservoir. Therefore cylinder 2 does not move and <u>the answer is c</u>.

9-20. 1. Provides mid-stroke stop and hold of the hydraulic cylinder (during both the extension and retraction strokes) by de-activation of the four-way, three-position DCV.

2. Provides two speeds of the hydraulic cylinder during the extension stroke.
   (a) When the three-way, two-position DCV is unactuated in spring offset mode: extension speed is normal.
   (b) When this DCV is actuated: extension speed increases by the regenerative capability of the circuit.

9-21. Cylinder 1 extends, cylinder 2 extends.
Cylinder 1 retracts, cylinder 2 retracts.
Above cycle repeats.

9-22. Both manually actuated directional control valves must be actuated in order to extend or retract the hydraulic cylinder.

9-23. Both cylinder strokes would be synchronized.

9-24. Cylinder 2 will extend through its complete stroke receiving full pump flow while cylinder 1 does not move. As soon as cylinder 2 has extended through its complete stroke, cylinder 1 receives full pump flow and extends through its complete stroke. This is because system pressure builds up until load resistance is overcome to move cylinder 2 with the smaller load.

Then pressure continues to increase until the load on cylinder 1 is overcome. This causes cylinder 1 to then extend. In the retraction mode, the cylinders move in the same sequence.

**9-25.** $p_1 = \frac{F_1 + F_2}{A_{P1}} = \frac{5000 + 5000}{10} = \underline{1000\ psi}$

**9-26.** Cyl 2: $p_3 \left( A_{P_2} - A_{R_2} \right) - p_2\, A_{P_2} = F_2$

Cyl 1: $p_2 \left( A_{P_1} - A_{R_1} \right) = F_1$

But $A_{P_2} = A_{P_1} - A_{R_1}$

So $p_2\, A_{P_2} = F_1$

Thus we have $p_3 = \frac{F_1 + F_2}{A_{P_2} - A_{R_2}} = \frac{10{,}000\ lb}{6\ in^2} = \underline{1667\ psi}$

**9-27.** For cylinder 1 we have:

$$p_1 A_{P1} - p_2 \left( A_{P1} - A_{R1} \right) = F_1$$

Similarly for cylinder 2 we have:

$$p_2 A_{P2} - p_3 \left( A_{P2} - A_{R2} \right) = F_2$$

Adding both equations and noting that $A_{P2} = A_{P1} - A_{R1}$ yields:

$$p_1 A_{P1} - p_3 \left( A_{P2} - A_{R2} \right) = F_1 + F_2$$

**Solving for $p_1$ gives the desired result.**

$$p_1 = \frac{F_1 + F_2 + p_3(A_{P2} - A_{R2})}{A_{P1}} = \frac{5000 + 5000 + 50(8-2)}{10}$$

$$= \underline{1030\,psi}$$

**9-28.** $$p_1 = \frac{F_1 + F_2}{A_{P1}} = \frac{22{,}000 + 22{,}000}{65 \times 10^{-4}} = \underline{6.77\ MPa}$$

**9-29. Per solution to Exercise 9-26 we have**

$$p_3 = \frac{F_1 + F_2}{A_{P2} - A_{R2}} = \frac{44{,}000\,N}{35\,cm^2} = 1257\frac{N}{cm^2} \times \left(\frac{100\,cm}{1\,m}\right)^2 = 12{,}570{,}000\frac{N}{m^2}$$

$$= \underline{12.57\ MPa}$$

**9-30. Using the equation developed in Exercise 9-24 we have:**

$$p_1 = \frac{F_1 + F_2 + p_3(A_{P2} - A_{R2})}{A_{P1}}$$

$$p_1 = \frac{22{,}000\,N + 22{,}000\,N + 300{,}000\,\frac{N}{m^2} \times (50 - 15)\,cm^2 \times \left(\frac{1\ m}{100\ cm}\right)^2}{65\,cm^2 \times \left(\frac{1\ m}{100\ cm}\right)^2}$$

**= <u>6.93 MPa</u>**

**9-31.** **Unloading Valve**

**Back pressure force on cylinder equals pressure loss in return line times the effective area of the cylinder $(A_P - A_R)$.**

$$F_{back\ pressure} = 350{,}000\frac{N}{m^2}\times\frac{\pi}{4}\left(0.0375^2 - 0.0125^2\right)m^2 = 344\,N$$

**Pressure at the blank end of cylinder required to overcome back pressure force equals the back pressure force divided by the area of the cylinder piston.**

$$p_{cyl\ blank\ end} = \frac{344\,N}{\frac{\pi}{4}(0.0375m)^2} = 311\ kPa$$

**Thus pressure setting of unloading valve equals**

$$1.50\left(675+311\right)kPa = \underline{1480\ kPa}$$

**Pressure Relief Valve**

**Pressure required to overcome the punching operation equals the punching load divided by the area of the cylinder piston.**

$$p_{punching} = \frac{8000\,N}{\frac{\pi}{4}(0.0375m)^2} = 7240\ kPa$$

**Thus pressure setting of pressure relief valve equals**

$$1.50\times 7240\ kPa = \underline{10{,}860\ kPa}$$

**9-32. (a) At full pump flow pressure, spring force equals hydraulic force on poppet.**

$$k\,S = 1.5\times1000\frac{lb}{in^2}\times0.60\,in^2 = 900\ lb$$

**where S = total spring compression**

= initial spring compression ($l$) plus full poppet stroke
= $l$ + 0.15

Thus we have

$$k\left(l+0.15\right)=900\ lb \qquad or \qquad k\,l+0.15k=900\ lb$$

Also at cracking presssure, spring force equals hydraulic force on poppet. Thus we have

$$k\,l=1.10\times1000\frac{lb}{in^2}\times0.60\ in^2=660\ lb$$

substituting values we have

$$660+0.15k=900\ , \qquad k=\underline{1600\ lb/in.}$$

(b) From part (a) we have

$$1600\ lb/in\times l\left(in.\right)=660\ lb\ , \qquad l=\underline{0.41\ \ in}.$$

9-33. Cylinders 1 and 2 are identical and are connected by identical lines. Therefore they receive equal flows and can sustain equal loads ($F_1$ = $F_2$). Also

$$Q_4 = \frac{40}{2} = 20 \ gpm, \quad Q_6 = \frac{20\left(8^2 - 4^2\right)}{8^2} = 15 \ gpm$$

$$and \quad Q_8 = Q_9 = 2(15) = 30 \ gpm$$

**We have the following useable equations:**

$$v = \frac{Q}{A}, \quad N_R = \frac{vD}{\nu} \quad and \quad H_L = \sum\left(f \times \frac{L}{D} + K\right)\frac{v^2}{2g}$$

**Solving for velocities yields:**

$$v_1 = \frac{{}^{40}\!/_{449} \ {}^{ft^3}\!/_{s}}{\frac{\pi}{4}\left(\frac{2}{12} ft\right)^2} = 4.08 \ {}^{ft}\!/_{s}, \quad v_2 = v_3$$

$$v_3 = \frac{{}^{40}\!/_{449}}{\frac{\pi}{4}\left(\frac{1.25}{12}\right)^2} = 10.4 \ {}^{ft}\!/_{s}, \quad v_4 = \frac{{}^{20}\!/_{449}}{\frac{\pi}{4}\left(\frac{1}{12}\right)^2} = 8.16 \ {}^{ft}\!/_{s}$$

$$v_6 = \frac{{}^{15}\!/_{449}}{\frac{\pi}{4}\left(\frac{1}{12}\right)^2} = 6.12 \ {}^{ft}\!/_{s}, \quad v_8 = v_9 = \frac{{}^{30}\!/_{449}}{\frac{\pi}{4}\left(\frac{1.25}{12}\right)^2} = 7.83 \ {}^{ft}\!/_{s}$$

**We can now calculate the Reynolds numbers.**

$$N_{R1} = \frac{4.08 \times {}^{2.0}\!/_{12}}{0.001} = 680, \quad N_{R2} = N_{R3}$$

$$N_{R3} = \frac{10.4 \times {}^{1.25}\!/_{12}}{0.001} = 1083, \quad N_{R4} = \frac{8.16 \times {}^{1.0}\!/_{12}}{0.001} = 680$$

$$N_{R6} = \frac{6.12 \times {}^{1.0}\!/\!_{12}}{0.001} = 510, \quad N_{R8} = N_{R9} = \frac{7.83 \times {}^{1.25}\!/\!_{12}}{0.001} = 816$$

$$\text{Since all flows are laminar}, \quad f = {}^{64}\!/\!_{N_R}. \quad \text{Also} \quad \Delta p = \gamma H_L \quad \text{Thus}$$

$$H_{L1} = \left( \frac{64}{680} \times \frac{6}{2.0/12} + 0.75 \right) \times \frac{4.08^2}{64.4} = 1.07\,ft = \frac{50 \times 1.07}{144} psi = 0.37\ psi$$

$$H_{L2} = \left( \frac{64}{1083} \times \frac{30}{1.25/12} + 4 \right) \times \frac{10.4^2}{64.4} = 35.3\,ft = 12.3\,psi$$

$$\mathbf{H_{L3} = \left( \frac{64}{1083} \times \frac{20}{1.25/12} + 6.8 \right) \times \frac{10.4^2}{64.4} = 30.5\ ft = 10.6\ psi}$$

$$\mathbf{H_{L4} = \left( \frac{64}{680} \times \frac{10}{1.0/12} + 0 \right) \times \frac{8.16^2}{64.4} = 11.7\ ft = 4.05\ psi}$$

$$H_{L6} = \left( \frac{64}{510} \times \frac{10}{1.0/12} + 1.8 \right) \times \frac{6.12^2}{64.4} = 9.80\,ft = 3.40\,psi$$

$$H_{L8} + H_{L9} = \left( \frac{64}{816} \times \frac{80}{1.25/12} + 5.75 \right) \times \frac{7.83^3}{64.4} = 62.8\,ft = 21.8\,psi$$

$$F_1 = F_2 = (1000 - 0.37 - 12.3 - 10.6 - 4.05) \times \frac{\pi}{4}(8^2)$$

$$-(3.40 + 21.8) \times \frac{\pi}{4}(8^2 - 4^2) = 48{,}900 - 950 = \underline{47{,}900\ lb}$$

**9-34.** **Cylinders 1 and 2 are identical and are connected by identical lines. Therefore they receive equal flows and sustain equal loads $(F_1 = F_2)$.**

We have the following useable equations:

$$v = \frac{Q}{A}\ ,\quad N_R = \frac{v\,D}{\nu}\ ,\quad H_L = \sum\left(f \times \frac{L}{D} + K\right)\frac{v^2}{2g}$$

Values of system parameters are as follows:

$$\gamma = 50\,\frac{lb}{ft^3} \times \frac{1\ N}{0.225\ lb} \times \left(\frac{3.28\ ft}{1\ m}\right)^3 = 7840\ N/m^3$$

$$\nu = 0.001\,\frac{ft^2}{s} \times \left(\frac{1\ m}{3.28\ ft}\right)^2 = 0.0000930\ m^2/s$$

$$\text{Cylinder piston diameter} = 8\ in \times \frac{1\ m}{39.4\ in} = 0.203\ m$$

$$\text{Cylinder rod diameter} = \frac{4}{39.4} = 0.102\ m$$

$L_1 = 1.83\ m,\quad D_1 = 0.0508\ m \qquad L_6 = 3.05\ m,\quad D_6 = 0.0254\ m$

$L_2 = 9.15\ m,\quad D_2 = 0.0317\ m \qquad L_7 = 3.05\ m,\quad D_7 = 0.0254\ m$

$L_3 = 6.10\ m,\quad D_3 = 0.0317\ m \qquad L_8 = 12.2\ m,\quad D_8 = 0.0317\ m$

$L_4 = 3.05\ m,\quad D_4 = 0.0254\ m \qquad L_9 = 12.2\ m,\quad D_9 = 0.0317\ m$

$L_5 = 3.05\ m,\quad D_5 = 0.0254\ m$

$$\Delta p_{pump} = 1000\ psi \times \frac{1\,Pa}{0.000145\,psi} = 6.90\ MPa$$

$$Q_{pump} = 40\,\frac{gal}{min} \times \frac{231\ in^3}{1\ gal} \times \left(\frac{1\ m}{39.4\ in}\right)^3 \times \frac{1\ min}{60\ s} = 0.00252\ m^3/s$$

$$Q_4 = \frac{0.00252}{2} = 0.00126\frac{m^3}{s}\ ,\quad Q_6 = 0.00126 \times \frac{(8^2 - 4^2)}{8^2} = 0.000945\frac{m^3}{s}$$

$$Q_8 = Q_9 = 2 \times 0.000945 = 0.00189 \, m^3/s$$

**Solving for velocities yields:**

$$v_1 = \frac{0.00252}{\frac{\pi}{4} \times 0.0508^2} = 1.24 \, m/s \, , \quad v_2 = v_3$$

$$v_3 = \frac{0.00252}{\frac{\pi}{4} \times 0.0317^2} = 3.19 \, m/s \, , \quad v_6 = \frac{0.000945}{\frac{\pi}{4} \times 0.0254^2} = 1.86 \, m/s$$

$$v_4 = \frac{0.00126}{\frac{\pi}{4} \times 0.0254^2} = 2.49 \, m/s \, , \quad v_8 = v_9 = \frac{0.00189}{\frac{\pi}{4} \times 0.0317^2} = 2.39 \, m/s$$

**We can now solve for the Reynolds numbers.**

$$N_{R1} = \frac{1.24 \times 0.0508}{0.0000930} = 677 \, , \qquad N_{R2} = N_{R3}$$

$$N_{R3} = \frac{3.19 \times 0.0317}{0.0000930} = 1087 \, , \qquad N_{R6} = \frac{1.86 \times 0.0254}{0.0000930} = 508$$

$$N_{R4} = \frac{2.49 \times 0.0254}{0.0000930} = 680 \, , \qquad N_{R8} = N_{R9} = \frac{2.39 \times 0.0317}{0.0000930} = 815$$

**Since all flows are laminar, $f = 64/N_R$. Thus we have:**

$$H_{L1} = \left( \frac{64}{677} \times \frac{1.83}{0.0508} + 0.75 \right) \times \frac{1.24^2}{19.6} = 0.33 \, m = 7840 \times 0.33 \, Pa = 2560 \, Pa$$

$$H_{L2} = \left( \frac{64}{1087} \times \frac{9.15}{0.0317} + 4 \right) \times \frac{3.19^2}{19.6} = 10.9 \, m = 85{,}500 \, Pa$$

$$H_{L3} = \left( \frac{64}{1087} \times \frac{12.2}{0.0317} + 6.8 \right) \times \frac{3.19^2}{19.6} = 15.3 \, m = 120{,}000 \, Pa$$

$$H_{L4} = \left(\frac{64}{680} \times \frac{3.05}{0.0254} + 1.8\right) \times \frac{2.49^2}{19.6} = 4.14\ m = 32{,}500\ Pa$$

$$H_{L6} = \left(\frac{64}{508} \times \frac{3.05}{0.0254} + 0\right) \times \frac{1.86^2}{19.6} = 2.67\ m = 20{,}900\ Pa$$

$$H_{L8} + H_{L9} = \left(\frac{64}{815} \times \frac{6.10 + 12.2}{0.0317} + 5.75\right) \times \frac{2.39^2}{19.6} = 12.8\ m$$ **= 100,500 Pa**

$$F_1 = F_2 = \left(6.90 \times 10^6 - 2560 - 85{,}500 - 120{,}000 - 32{,}500\right) \times \frac{\pi}{4} \times$$
$$\left(0.203^2\right) - \left(20{,}900 + 100{,}500\right) \times \frac{\pi}{4} \times (0.203^2 - 0.102^2)$$

$$F_1 = F_2 = 216{,}000 - 2940 = \underline{213{,}000\ N}$$

**9-35.** $HP_{loss} = \frac{(\Delta p)Q}{1714}$ *and* $1\ HP = 42.4\ BTU/\text{min}$ **Thus**

$$HP_{loss} = \frac{(0.4 + 12.3 + 10.6) \times 40}{1714} + \frac{4.05 \times 40}{1714} + \frac{3.4 \times 30}{1714} + \frac{21.8 \times 30}{1714}$$

$$HP_{loss} = 0.54 + 0.10 + 0.06 + 0.38 = 1.08\ hp$$

$$Heat\ \ generation\ \ rate = 1.08 \times 42.4 = 45.8\ \ BTU/\text{min} = \underline{2750\ \ BTU/hr}$$

**9-36.** $v = \frac{Q_{cyl}}{A}$ **where each cylinder receives one half of pump flow.**

$$v_{ext} = \frac{Q_{blank\ end}}{A_{piston}} = \frac{20\ \frac{gal}{min} \times \frac{231\ in^3}{1\ gal} \times \frac{1\ min}{60\ s}}{\frac{\pi}{4} \times 8^2\ in^2} = \frac{77\ \frac{in^3}{s}}{50.3\ in^2} = \underline{1.53\ \frac{in}{s}}$$

$$v_{ret} = \frac{Q_{rod\ end}}{A_{piston} - A_{rod}} = \frac{77\ in^3/s}{\frac{\pi}{4}\left(8^2 - 4^2\right) in^2} = \underline{2.04\ in/s}$$

**9-37.** $$Power\ \ Loss(Watts) = (\Delta p)\frac{N}{m^2} \times Q\,\frac{m^3}{s}$$

= (2,560 + 85,500 + 120,000) x 0.00252

+ 2 x 20,900 x 0.000945

+ 2 x 32,500 x 0.00126

+ 100,500 x 0.00189

= 524 + 39.5 + 81.9 + 190

= 835 Watts = <u>0.835 kW</u>

**9-38.** $v = \frac{Q_{cyl}}{A}$ where each cylinder receives one half of pump flow.

$$v_{ext} = \frac{Q_{blank\ end}}{A_{piston}} = \frac{0.00126\ m^3/s}{\frac{\pi}{4} \times 0.203^2\ m^2} = \underline{0.0389\ m/s}$$

$$v_{ret} = \frac{Q_{rod\ end}}{A_{piston} - A_{rod}} = \frac{0.00126\ m^3/s}{\frac{\pi}{4} \times \left(0.203^2 - 0.102^2\right) m^2} = \underline{0.0521\ m/s}$$

**9-39.** We have the following useable equations:

$$v = \frac{Q}{A}, \quad N_R = \frac{vD}{\nu}, \quad H_L = \sum\left(f \times \frac{L}{D} + K\right)\frac{v^2}{2g} \qquad Also$$

$$Pump\ \ Power = \frac{\Delta p(psi) \times Q(gpm)}{1714} = 0.90 \times 25 = 22.5\ HP$$

$$\text{Thus} \quad Q_{pump} = \frac{22.5 \times 1714}{1000} = 38.6\ \text{gpm} \qquad \text{Also}$$

$$F_{regenerative} = p_{blank\ end} A_{piston} - p_{rod\ end}\left(A_{piston} - A_{rod}\right)$$

$$Q_1 = Q_2 = Q_{pump} = 38.6\ gpm \quad and \quad Q_3 = Q_{pump} + Q_4 \qquad Thus\ \ we\ \ have:$$

$$A_P v_{P\ ext} = Q_3 = Q_{pump} + \left(A_P - A_r\right) v_{P\ ext} \times \frac{A_P}{A_P}$$

$$\text{Therefore } Q_3 = Q_{pump} + \frac{A_P - A_r}{A_P} Q_3 \quad \text{so} \quad Q_3 = \frac{A_P}{A_r} Q_{pump}$$

$$\text{And} \quad Q_4 = \frac{A_P - A_r}{A_P} \times \frac{A_P}{A_r} Q_{pump} = \frac{A_P - A_r}{A_r} Q_{pump}$$

$$\text{Hence} \quad Q_3 = \frac{\pi/4 \times 8^2 \times 38.6}{\pi/4 \times 4^2} = 154\ \text{gpm}$$

$$\text{And} \quad Q_4 = \frac{\pi/4 \times \left(8^2 - 4^2\right) \times 38.6}{\pi/4 \times 4^2} = 116\ \text{gpm}$$

Solving for velocities yields:

$$v_1 == \frac{38.6/449 \ \frac{ft^3}{s}}{\frac{\pi}{4}\left(\frac{2}{12}\ ft\right)^2} = 3.94 \ ft/s \qquad v_2 = \frac{38.6/449}{\frac{\pi}{4}\left(\frac{1..75}{12}\right)^2} = 5.14 \ ft/s$$

$$v_3 = \frac{154/449}{\frac{\pi}{4}\left(\frac{1.75}{12}\right)^2} = 20.5 \ ft/s \qquad v_4 = \frac{116/449}{\frac{\pi}{4}\left(\frac{1.75}{12}\right)^2} = 15.5 \ ft/s$$

**We can now calculate the Reynolds numbers.**

$$N_{R1} = \frac{3.94 \times 2.0/12}{0.001} = 657 \qquad N_{R2} = \frac{5.14 \times 1.75/12}{0.001} = 750$$

$$N_{R3} = \frac{20.5 \times 1.75/12}{0.001} = 2990 \qquad N_{R4} = \frac{15.5 \times 1.75/12}{0.001} = 2260$$

**Also** $f = 64/N_R$ *and* $\Delta p = \gamma\, H_L$

$$H_{L1} = \left(\frac{64}{657} \times \frac{2}{2.0/12} + 10\right) \times \frac{3.94^2}{64.4} = 2.69 \ ft$$

$$= 50 \frac{lb}{ft^3} \times 2.69\, ft \times \frac{1\ psi}{144\ lb/ft^2} = 0.93 \ psi$$

$$H_{L2} = \left(\frac{64}{750} \times \frac{20}{1.75/12} + 5\right) \times \frac{5.14^2}{64.4} = 6.85 \ ft = \frac{50 \times 6.85}{144} = 2.38 \ psi$$

$$H_{L3} = \left(\frac{64}{2990} \times \frac{30}{1.75/12} + 0.75\right) \times \frac{20.5^2}{64.4} = 33.6 \ ft = \frac{50 \times 33.6}{144} = 11.7 \ psi$$

$$H_{L4} = \left( \frac{64}{2260} \times \frac{30}{1.75/12} + 0.75 \right) \times \frac{15.5^2}{64.4} = 24.5\, ft = \frac{50 \times 24.5}{144} = 8.52 \ \ psi$$

$$F = (1000 - 0.93 - 2.38 - 11.7) \times \frac{\pi}{4} \times 8^2$$
$$- (1000 - 0.93 - 2.38 + 8.52) \times \frac{\pi}{4} \times (8^2 - 4^2)$$

**Thus F = 49,500 - 37,900 = 11,600 lb**

**9-40. Metric data is as follows:**

**Electric Motor:** Power $= 25\ HP \times \frac{0.746\ kW}{1\ HP} = 18.65\ kW$

Overall efficiency = 90%

**Pump:** Discharge pressure $= 1000\ psi \times \frac{1\ kPa}{0.145\ psi} = 6897\ kPa$

**Oil:** Kin. Visc. $= \nu = 0.001\ \frac{ft^2}{s} \times \left( \frac{1\ m}{3.28\ ft} \right)^2 = 0.0000930\ m^2/s$

Weight Dens. $= \gamma\left(\frac{N}{m^3}\right) = 157 \times \gamma\left(\frac{lb}{ft^3}\right) = 157 \times 50 = 7850\ \frac{N}{m^3}$

**Cylinder:** Piston Diameter $= 8\ in \times \frac{1\ ft}{12\ in} \times \frac{1\ m}{3.28\ ft} = 0.203\ m$

Rod Diameter = 4 in = 0.102 m

**Elbows**: K factor = 0.75

**Pipes**:

| No. | Length(m) | Dia.(m) |
|---|---|---|
| 1 | 0.61 | 0.0508 |
| 2 | 6.10 | 0.0445 |
| 3 | 9.15 | 0.0445 |
| 4 | 9.15 | 0.0445 |
| 5 | 6.10 | 0.0445 |

We have the following useable equations:

$$v = \frac{Q}{A}, \quad N_R = \frac{v\,D}{\nu}, \quad \sum\left(f \times \frac{L}{D} + K\right) \times \frac{v^2}{2g}$$

$$Pump\ \ Power = \Delta p(kPa) \times Q\left(\frac{m^3}{s}\right) = 0.90 \times 18.65 = 16.79\ kW$$

$$Q_{pump} = \frac{16.79\ kW}{6897\ kPa} = 0.00243\ {}^{m^3}\!/_{s}$$

$$F_{regen} = p_{blank\ end} A_{piston} - p_{rod\ end}\left(A_{piston} - A_{rod}\right)$$

$$Q_1 = Q_2 = Q_{pump} = 0.00243\ {}^{m^3}\!/_{s}$$

Per the solution to Exercise 9-39, we have the following two equations to solve for the flow rates in lines 3 and 4:

$$Q_3 = \frac{A_P}{A_r} Q_{pump} = \frac{{}^{\pi}\!/_{4} \times 8^2 \times 0.00243}{{}^{\pi}\!/_{4} \times 4^2} = 0.00972\ {}^{m^3}\!/_{s}$$

$$Q_4 = \frac{A_P - A_r}{A_r} Q_{pump} = \frac{\pi/4 \times (8^2 - 4^2) \times 0.00243}{\pi/4 \times 4^2} = 0.00729\ m^3/s$$

**Solving for the velocities yields:**

$$v_1 = \frac{Q_1}{A_1} = \frac{0.00243}{\frac{\pi}{4} \times 0.00508^2} = 1.20\ m/s \qquad v_2 = \frac{0.00243}{\frac{\pi}{4} \times 0.0445^2} = 1.56\ m/s$$

$$v_3 = \frac{0.00972}{\frac{\pi}{4} \times 0.0445^2} = 6.24\ m/s \qquad v_4 = \frac{0.00729}{\frac{\pi}{4} \times 0.0445^2} = 4.69\ m/s$$

**We can now calculate the Reynolds numbers.**

$$N_{R1} = \frac{1.20 \times 0.0508}{0.0000930} = 655 \qquad N_{R2} = \frac{1.56 \times 0.0445}{0.0000930} = 746$$

$$N_{R3} = \frac{6.24 \times 0.0445}{0.0000930} = 2990 \qquad N_{R4} = \frac{4.69 \times 0.0445}{0.0000930} = 2240$$

**Also** $f = \frac{64}{N_R}$ *and* $\Delta p = \gamma\, H_L$

$$H_{L1} = \left(\frac{64}{655} \times \frac{0.610}{0.0508} + 10\right) \times \frac{1.20^2}{19.6} = 0.74\ m = 0.74 \times 7850 = 5{,}840\ Pa$$

$$H_{L2} = \left(\frac{64}{746} \times \frac{6.10}{0.0445} + 5\right) \times \frac{1.56^2}{19.6} = 2.08\ m = 16{,}300\ Pa$$

$$H_{L3} = \left( \frac{64}{2990} \times \frac{9.15}{0.0445} + 0.75 \right) \times \frac{6.24^2}{19.6} = 10.2\ m = 80{,}300\ Pa$$

$$H_{L4} = \left( \frac{64}{2240} \times \frac{9.15}{0.0445} + 0.75 \right) \times \frac{4.69^2}{19.6} = 7.43\ m = 58{,}400\ Pa$$

**Thus we have**

$$F(kN) = (6897\ kPa - 5.84 - 16.3 - 80.3) \times \frac{\pi}{4} \times 0.203^2$$
$$- (6897\ kPa - 5.84 - 16.3 + 58.4) \times \frac{\pi}{4} \times \left(0.203^2 - 0.102^2\right)$$

**F = 220 - 168 = <u>52 kN</u>**

**9-41. HP Loss =** $\sum Q(\Delta p)$ **= Pipe 1 Loss + Pump Loss + Pipe 2 Loss + Pipe 3 Loss + Pipe 4 Loss**

$$= \frac{38.6 \times 0.93}{1714} + (25 - 22.5) + \frac{38.6 \times 2.38}{1714}$$

$$+ \frac{154 \times 11.7}{1714} + \frac{116 \times 8.52}{1714}$$

**HP Loss = 0.02 + 2.50 + 0.05 + 1.05 + 0.58 = <u>4.20 HP</u>**

**Since 1 hp = 42.4 BTU/min we have:**

**Heat generation rate = 4.20 x 42.4**

**= 178 BTU/min = <u>10,700 BTU/hr</u>**

9-42. Power Loss = $\sum Q(\Delta p)$ kW

= Pipe 1 Loss + Pump Loss + Pipe 2 Loss

+ Pipe 3 Loss + Pipe 4 Loss

= 0.00243 x 5.84 + (18.7 - 16.8)

+ 0.00243 x 16.3 + 0.00972 x 80.3

+ 0.00729 x 58.4

= 0.014 + 1.90 + 0.040 + 0.78 + 0.43

Power Loss = Heat Generation Rate = 3.16 kW

9-43. Per the solution to Exercise 9-39, we have $Q_{pump} = 38.6\text{ gpm}$

Upper Position of DCV:

$$v_{ext} = \frac{Q_{pump}}{A_P} = \frac{38.6\,\frac{gal}{min} \times \frac{231\,in^3}{1\,gal} \times \frac{1\,min}{60\,s}}{\frac{\pi}{4} \times 8^2\,in^2} = \frac{149\,\frac{in^3}{s}}{50.3\,in^2} = \underline{2.96\,in/s}$$

Spring-Centered Position of DCV:

$$v_{ext} = \frac{Q_{pump}}{A_r} = \frac{149}{\frac{\pi}{4} \times 4^2} = \frac{149}{12.6} = \underline{11.8\,in/s}$$

Lower Position of DCV:

$$v_{ret} = \frac{Q_{pump}}{A_P - A_r} = \frac{149}{50.3 - 12.6} = \frac{149}{37.7} = \underline{3.95\,in/s}$$

**9-44.** Per solution of Exercise 9-40, we have $Q_{pump} = 0.00243\ m^3/s$.

<u>Upper Position of DCV</u>:

$$v_{ext} = \frac{Q_{pump}}{A_P} = \frac{0.00243}{\frac{\pi}{4} \times 0.203^2} = \underline{0.0751\ m/s}$$

<u>Spring-Centered Position of DCV</u>:

$$v_{ext} = \frac{Q_{pump}}{A_r} = \frac{0.00243}{\frac{\pi}{4} \times 0.102^2} = \underline{0.297\ m/s}$$

<u>Lower Position of DCV</u>:

$$v_{ret} = \frac{Q_{pump}}{A_P - A_r} = \frac{0.00243}{\frac{\pi}{4} \times \left(0.203^2 - 0.102^2\right)} = \underline{0.100\ m/s}$$

**9-45.** Per Eq.(9-9) we have

$$C_v = \frac{v_{cyl}\, A_{piston}}{\sqrt{\dfrac{p_{PRV} - F_{load}/A_{piston}}{(SG)}}}$$

Also from Eq.(9-7) the units for $C_v$ are $gpm/\sqrt{psi}$. Therefore we have the following units for the terms in Eq.(9-9):

$$Q = v_{cyl} A_{piston} = gpm, \quad P_{PRV} = psi, \quad F_{load}/A_{piston} = psi$$

Thus we have:

$$v_{cyl} A_{piston} = 10\frac{in.}{s} \times 3.14\ in.^2 = 31.4\frac{in.^3}{s} \times \frac{1gal}{231 in.^3} \times \frac{60s}{1\min} = 8.16\, gpm$$

$$p_{PRV} = 1000\ psi \quad and \quad \frac{F_{load}}{A_{piston}} = \frac{3000\ lb}{3.14\ in.^2} = 955\ psi$$

Substituting values yields

$$C_v = \frac{8.16}{\sqrt{\frac{1000-955}{0.9}}} = \frac{8.16}{7.07} = \underline{1.15\ gpm / \sqrt{psi}}$$

9-46. Converting to metric units we have:

1. Desired cylinder speed = 0.254 m/s
2. Cylinder piston diameter = 0.0508m (area = 0.00203 $m^2$)
3. Cylinder load = 13,340 N
4. Specific gravity of oil = 0.90
5. Pressure relief valve setting = 6895 kPa

Per Eq.(9-9) we have

$$C_v = \frac{v_{cyl}\ A_{piston}}{\sqrt{\frac{p_{PRV} - F_{load}/A_{piston}}{(SG)}}}$$

Also from Eq.(9-7) the units for $C_v$ are $Lpm / \sqrt{kPa}$.

Therefore we have the following units for the terms in Eq.(9-9):

$$Q = v_{cyl}\ A_{piston} = Lpm\ , \quad p_{PRV} = kPa\ , \quad F_{load}/A_{piston} = kPa$$

**Thus we have:**

$$v_{cyl}\,A_{piston} = 0.254\frac{m}{s}\times 0.00203m^2 \times \frac{1\,L}{0.001m^3}\times\frac{60\;s}{1\min} = 30.9\;Lpm$$

$$P_{PRV} = 6895\;kPa \quad and \quad F_{load}/A_{piston} = \frac{13{,}340\,N}{0.00203\,m^2}\times\frac{1\;kPa}{1000\,N/m^2} = 6570\;kPa$$

**Substituting values yields:**

$$C_v = \frac{30.9}{\sqrt{\dfrac{6895-6570}{0.9}}} = \frac{30.9}{19.0} = \underline{1.63\;Lpm/\sqrt{kPa}}$$

**9-47. (a)** $p_1$ **equals approximately the pressure relief valve setting of <u>1600 psi</u>.**

**The force acting on the cylinder piston is found next.**

$$F \;=\; 1600\,\frac{lb}{in^2}\;\times\;\frac{\pi}{4}\left(2.5\;in\right)^2 \;=\; 4910\;\;lb$$

**For a constant speed cylinder we have**

$$p_1\,A_P \;=\; p_2\,\left(A_P\,-\,A_R\right)$$

$$4910\;\;lb \;=\; p_2\;\times\;\frac{\pi}{4}\left(2.5^2\;-\;1.5^2\right)\quad so\;\; p_2 \;=\; \underline{1560\;\;psi}$$

$p_3$ **equals approximately <u>zero</u>.**

**(b)** $p_1 \;=\; \underline{1600\;psi} \quad and \quad p_3 \;=\; \underline{zero}$

$$4910\;\;lb\;+\;6000\;\;lb \;=\; p_2\;\times\;\frac{\pi}{4}\left(2.5^2\;-\;1.5^2\right)in^2 \quad so \quad p_2 \;=\; \underline{3470\;\;psi}$$

**9-48. (a)** $p_1$ **= 10 MPa – 0.3 MPa = <u>9.7 MPa</u>**

$$F = 9.7 \times 10^6 \; {}^{N}\!/_{m^2} \times \frac{\pi}{4}\left(0.050 \; m\right)^2 = 19{,}000 \; N$$

$$19{,}000 \; N = p_2 \times \frac{\pi}{4}\left(0.050^2 - 0.025^2\right) m^2$$

$$p_2 = \underline{12.9 \; MPa} \qquad and \qquad p_3 = \underline{zero}$$

**(b)** $p_1 = \underline{9.7 \; MPa}$

$$19{,}000 \; N + 20{,}000 \; N = p_2 \times \frac{\pi}{4}\left(0.050^2 - 0.025^2\right) m^2$$

$$p_2 = \underline{26.5 \; MPa} \qquad and \qquad p_3 = \underline{200 \; kPa}$$

# Chapter 10

## Hydraulic Conductors and Fittings

10-1. To carry the fluid from the reservoir through operating components and back to the reservoir.

10-2. <u>20 ft/s</u>

10-3. <u>4 ft/s</u>

10-4. Copper promotes the oxidation of petroleum oils.

10-5. Zinc, magnesium and cadmium.

10-6. It raises the pressure levels up to 4 times the steady state system design values.

10-7.
1. Tensile strength of conductor material.
2. Conductor outside diameter.
3. Operating pressure levels.

10-8. To handle pressure shocks and provide a factor of safety.

10-9.
1. When a joint is taken apart, the pipe must be tightened farther to reseal.
2. Pipes cannot be bent around obstacles.

10-10.
1. Steel pipe.
2. Steel tubing.
3. Plastic tubing.
4. Flexible hose.

**10-11. Average fluid velocity is defined as the volumetric flow rate divided by the pipe cross-sectional area.**

**10-12. Malleable iron can be used for hydraulic fittings for low-pressure lines such as inlet, return and drain lines.**

**10-13. Tubing can be bent into almost any shape, thereby reducing the number of required fittings. Tubing is also easier to handle and can be reused without any sealing problems.**

**10-14. Plastic tubing is relatively inexpensive. Also since it can readily be bent to fit around obstacles, it is easy to handle and can be stored on reels.**

**10-15. The quick-disconnect coupling is used mainly where a conductor must be frequently disconnected from a component.**

**10-16. When a joint is taken apart, the pipe must be tightened farther to reseal. This frequently requires replacing some of the pipe with slightly longer sections although this problem has been somewhat overcome by using Teflon tape to reseal the pipe joints.**

**10-17. Figure 10-10 shows the flared-type fitting which was developed before the compression-type for sealing against high pressures. Figure 10-9 shows a compression-type fitting which can be repeatedly taken apart and reassembled and remain perfectly sealed against leakage.**

**10-18. Flexible hoses are used when hydraulic components such as actuators are subject to movement.**

**10-19.**
1. **Install so there is no kinking during operation of system.**
2. **There should always be some slack to relieve any strain and allow for the absorption of pressure surges.**

3. If the hose is subject to rubbing, it should be encased in a protective sleeve.

10-20. Flexible hose is fabricated in layers of elastomer (synthetic rubber) and braided fabric or braided wire which permits operation at higher pressures. The outer layer is normally synthetic rubber and serves to protect the braided layer.

10-21. Increases.

10-22. By nominal size and schedule number.

10-23. Schedule number is a measure of how thick the wall of a pipe is. For a given nominal pipe size, the pipe outside diameter is fixed and changes in schedule number represent different wall thicknesses and thus different pipe inside diameter values. The larger the schedule number, the larger the wall thickness.

10-24. Thin-walled cylinders: $D_i/t$ *is greater than* $10$

Thick-walled cylinders: $D_i/t$ *equals or is less than* $10$

10-25. 1. The compression nut needs to be placed on the tubing before flaring the tube.
2. These fittings should not be over tightened. Too great a torque destroys the sealing surface and thus may cause leaks.

10-26. $A = \frac{\pi}{4} D^2 = \frac{Q}{v}$, $So \quad D = \sqrt{\frac{4Q}{\pi v}} = \sqrt{\frac{4 \times 20/449 \ ft^3/s}{\pi \times 4 \ ft/s}} = 0.119 \ ft$

= 1.428 inch inside diameter

**10-27.** $D = \sqrt{\dfrac{4Q}{\pi v}} = \sqrt{\dfrac{4 \times {}^{20}\!/_{449}\ {}^{ft^3}\!/_{s}}{\pi \times 20\ {}^{ft}\!/_{s}}} = 0.0533\ ft$ **= <u>0.639 inch inside dia.</u>**

**10-28. Fluid velocity limitation** $= 5\,\dfrac{ft}{s} \times \dfrac{1\ m}{3.28\ ft} = 1.52\,\dfrac{m}{s}$

$ID_{min} = \sqrt{\dfrac{4\,Q}{\pi\,v}} = \sqrt{\dfrac{4 \times 0.002}{\pi \times 1.52}} = 0.0409\ m = 40.9\ mm$

**<u>Select 42 mm ID</u>**

**10-29. Fluid velocity limitation** $= 20\,\dfrac{ft}{s} \times \dfrac{1\ m}{3.28\ ft} = 6.1\,\dfrac{m}{s}$

$ID_{min} = \sqrt{\dfrac{4\,Q}{\pi\,v}} = \sqrt{\dfrac{4 \times 0.002}{\pi \times 6.1}} = 0.0204\ m = 20.4\ mm$

**<u>Select 22 mm ID</u>**

**10-30. <u>Pipe Minimum Inside Diameter</u>** $\left(v_{max} = 7.5\,{}^{m}\!/_{s}\right)$

$$A_{min} = \frac{\pi}{4}(ID)^2_{min} = \frac{Q}{v_{max}} = \frac{{}^{0.075}\!/_{60}\ {}^{m^3}\!/_{s}}{7.5\ {}^{m}\!/_{s}} = 0.0001667\ m^2$$

$$(ID)_{min} = 0.0146\,m = \underline{14.6\ mm}$$

**<u>Pipe Maximum Inside Diameter</u>** $\left(v_{min} = 6\,{}^{m}\!/_{s}\right)$

$$A_{max} = \frac{\pi}{4}(ID)^2_{max} = \frac{Q}{v_{min}} = \frac{{}^{0.075}\!/_{60}\ {}^{m^3}\!/_{s}}{6\ {}^{m}\!/_{s}} = 0.0002083\ m^2$$

$$(ID)_{max} = 0.0163\,m = \underline{16.3\ mm}$$

10-31. $v = \frac{Q}{A} = \frac{Q}{\frac{\pi}{4} D^2}$ Answer is: <u>square</u>

10-32. $v = \frac{Q}{A} = \frac{Q}{\frac{\pi}{4} D^2}$ Thus $\frac{v_1}{v_2} = \frac{D_2^2}{D_1^2} = \left(\frac{D_2}{D_1}\right)^2$

$v_2 = v_1\left(\frac{D_1}{D_2}\right)^2 = v_1\left(\frac{1}{2}\right)^2 = \frac{v_1}{4}$ Thus answer is: <u>four</u>

10-33. $A = \frac{Q}{v}$ or $A(in^2) = \frac{Q\left(\frac{in^3}{s}\right)}{v\left(\frac{in}{s}\right)}$

$$Q\left(\frac{in^3}{s}\right) = Q\left(\frac{gal}{min}\right) \times \frac{231\,in^3}{1\,gal} \times \frac{1\,min}{60\,s} = 3.85\,Q(gpm)$$

$v\left(\frac{in}{s}\right) = v\left(\frac{ft}{s}\right) \times \frac{12\,in}{1\,ft} = 12\,v\left(\frac{ft}{s}\right)$ Thus we have:

$A(in^2) = \frac{3.85\,Q(gpm)}{12\,v\left(\frac{ft}{s}\right)}$ so $C_1 = \frac{3.85}{12} = \underline{0.321}$

Since units are satisfied by the following equation:

$A(m^2) = \frac{C_2\,Q\left(\frac{m^3}{s}\right)}{v\left(\frac{m}{s}\right)}$ Then $\underline{C_2 = 1}$

**10-34.** **First trial**: select 1-1/4 in. OD, 1.060 in. ID tube

$$v = \frac{Q}{A} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{1.060}{12}\ ft\right)^2} = 10.9\ ft/s$$

**Second trial**: select 1-1/2 in. OD, 1.310 in. ID tube

$$v = \frac{Q}{A} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{1.310}{12}\ ft\right)^2} = 7.13\ ft/s$$

**Third trial**: select 5/8 in. OD, 0.435 in. ID tube

$$v = \frac{Q}{A} = \frac{\left(30/449\right) ft^3/s}{\frac{\pi}{4}\left(\frac{0.435}{12}\ ft\right)^2} = 64.7\ ft/s$$

**Conclusion:**

The 1-1/2 in. OD, 1.310 in. ID size produces a velocity of 7.13 ft/s. Therefore need a larger size than given in Figure 10-7 for the pump inlet. The 1-1/4 in. OD, 1.060 in. ID size produces a velocity of 10.9 ft/s. Therefore this size is adequate for the pump outlet. The 5/8 in. OD, 0.435 in. ID size produces a velocity of 64.7 ft/s. Therefore this size is too small even for the pump outlet.

10 – 35. $$Q = 30\frac{gal}{\min}\times\frac{231in^3}{1gal}\times\frac{1\min}{60s}\times\left(\frac{1m}{39.4in}\right)^3 = 0.00189\,m^3/s$$

$$v = 5\,\frac{ft}{s}\times\frac{1\ m}{3.28\ ft} = 1.52\ m/s \quad \text{and} \quad v = 20\ ft/s = 6.08\ m/s$$

$$\text{Wall thickness} = 0.095\ in \times \frac{1\ mm}{0.0394\ in} = 2.41\ mm$$

**Can select from Figure 10-21, tube sizes having a wall thickness greater than 2.41 mm to withstand pressure.**

**First trial: select 30 mm OD, 3.0 mm wall, 24 mm ID**

$$v = \frac{Q}{A} = \frac{0.00189\ m^3/s}{\frac{\pi}{4}(0.024\ m)^2} = 4.17\ m/s$$

**Second trial: select 35 mm OD, 3.0 mm wall, 29 mm ID**

$$v = \frac{Q}{A} = \frac{0.00189\ m^3/s}{\frac{\pi}{4}(0.029\ m)^2} = 2.86\ m/s$$

**Third trial: select 42 mm OD, 3.0 mm wall, 36 mm ID**

$$v = \frac{Q}{A} = \frac{0.00189\ m^3/s}{\frac{\pi}{4}(0.036\ m)^2} = 1.86\ m/s$$

**Fourth trial: select 28 mm OD, 2.5 mm wall, 23 mm ID**

$$v = \frac{Q}{A} = \frac{0.00189\ m^3/s}{\frac{\pi}{4}(0.023\ m)^2} = 4.54\ m/s$$

**Fifth trial**: select 25 mm OD, 3.0 mm wall, 19 mm ID

$$v = \frac{Q}{A} = \frac{0.00189\ m^3/s}{\frac{\pi}{4}\left(0.019\ m\right)^2} = 6.67\ m/s$$

**Hence use 28 mm OD, 2.5 mm wall, 23 mm ID size tube for the pump outlet.**

**Since the 42 mm OD, 3.0 mm wall, 36 mm ID size tube produces a velocity of 1.86 m/s, need a larger size tube than given in Figure 10-21 for the pump inlet.**

**10-36.** **Pump Inlet**

$$A_{\min} = \frac{\pi}{4}(ID)^2_{\min} = \frac{Q}{v_{\max}} = \frac{0.075/60\ m^3/s}{1.2\ m/s} = 0.001042\ m^2$$

$$(ID)_{\min} = 0.0364\ m = 36.4\ mm$$

**Minimum commercial-size tubing: 42mm OD, 38mm ID**

**Pump Outlet**

$$A_{\min} = \frac{\pi}{4}(ID)^2_{\min} = \frac{Q}{v_{\max}} = \frac{0.075/60\ m^3/s}{6.1\ m/s} = 0.0002049\ m^2$$

$$(ID)_{\min} = 0.0162\,m = 16.2\,mm$$

**Minimum Commercial-size tubing: 22mm OD, 18mm ID**

10-37. First calculate the wall thickness of the tubing:

$$t = \frac{1.250 - 1.060}{2} = 0.095 \text{ in.}$$

Next find the burst pressure for the tubing.

$$BP = \frac{2\,t\,S}{D_i} = \frac{2 \times 0.095 \times 75{,}000}{1.060} = 13{,}440 \text{ psi}$$

Finally calculate the working pressure:

$$WP = \frac{13{,}440}{8} = \underline{1680 \text{ psi}}$$

10-38. Tensile Stress $= \sigma = \dfrac{P\,D_i}{2\,t} = \dfrac{1000 \text{ psi} \times 1.060 \text{ in}}{2\left(\dfrac{1.250 - 1.060}{2} \text{ in}\right)} = \underline{5580 \text{ psi}}$

10-39. First find the minimum inside diameter based on the fluid velocity limitation of 20 ft/s.

$$D = \sqrt{\frac{4Q}{\pi v}} = \sqrt{\frac{4 \times {}^{20}\!/_{449} \; {}^{ft^3}\!/_{s}}{\pi \times 20 \; {}^{ft}\!/_{s}}} = 0.0533 \; ft = 0.639 \; in.$$

From Figure 10-7, the smallest acceptable tube size based on flow rate requirements is:

3/4 in. OD, 0.049 in. wall thickness, 0.652 in. ID

Second find the burst pressures and working pressures for the above tubing for SAE 1010 and AISI 4130 materials.

(a) Material is SAE 1010

$$BP = \frac{2\,t\,S}{D_i} = \frac{2 \times 0.049 \times 55{,}000}{0.652} = 8267 \text{ psi}$$

$$WP = \frac{8267}{8} = 1030 \text{ psi}$$

This working pressure is adequate since it is greater than 1000 psi.

<u>**Use 3/4 in. OD, 0.049 in. wall thickness, 0.652 in. ID**</u>

(b) Material is AISI 4130

$$BP = \frac{2 \times 0.049 \times 75{,}000}{0.652} = 11{,}270 \text{ psi}$$

$$WP = \frac{11{,}270}{8} = 1410 \text{ psi} \quad \text{OK}$$

<u>**Use 3/4 in. OD, 0.049 in. wall thickness, 0.652 in. ID**</u>

**10-40.**

$$BP = \frac{2\,t\,S}{D_i} = \frac{2 \times 0.003\,m \times 517\,{}^{MN}\!/_{m^2}}{0.024\,m} = 129.3\ MPa$$

$$WP = \frac{BP}{FS} = \frac{129.3\,MPa}{8} = 16.2\,MPa = \underline{162\ Bars}$$

**10-41.** $Tensile\ \ Stress = \sigma = \dfrac{p\,D_i}{2t} = \dfrac{10\,MPa \times 0.024\,m}{2 \times 0.003\,m} = \underline{40\ \ MPa}$

**10-42.** **(a)** $ID = \sqrt{\dfrac{4 \times 0.001}{\pi \times 6.1}} = 0.0144\ m = 14.4\ mm$

**From Fig. 10-21 try 15 mm OD, 1.5 mm wall thickness, 12 mm ID tube size.**

$$BP = \frac{2 \times 0.0015 \times 379}{0.012} = 94.8\ MPa$$

$$WP = \frac{94.8}{8} = 11.9\ MPa = 119\ Bars$$

**OK since the WP is greater than 70 bars.**

**<u>Thus use 15 mm OD, 1.5 mm wall thickness, 12 mm ID tube.</u>**

**(b) Try 15 mm OD, 1.5 mm wall thickness, 12 mm ID tube.**

$$BP = \frac{2 \times 0.0015 \times 517}{0.012} = 129.3\ MPa$$

$$WP = \frac{129.3}{8} = 16.2\ MPa = 162\ Bars$$

**OK since the WP is greater than 70 bars.**

**<u>Thus use 15 mm OD, 1.5 mm wall thickness, 12 mm ID tube.</u>**

10-43. $$BP = \frac{2\,t\,S}{D_i} = 2\left(\frac{D_o - D_i}{2}\right) \times \frac{S}{D_i} = \frac{(D_o - D_i)\,S}{D_i}$$

Substituting values we have:

$$8000 = \frac{(D_o - 1)}{1} \times 55{,}000 = 55{,}000\,D_o - 55{,}000$$

$$D_o = \frac{55{,}000 + 8{,}000}{55{,}000} = \underline{1.145 \text{ in.}}$$

10-44. *From Exercise* 10–39, $BP = \frac{(D_o - D_i)S}{D_i}$

Substituting values we have:

$$50 = \frac{(D_o - 25) \times 379}{25} \qquad \text{Thus} \quad D_o = \underline{28.3 \text{ mm}}$$

# Chapter 11

## Ancillary Hydraulic Devices

**11-1.** 1. It must make allowance for dirt and chips to settle and for air to escape.
2. It must be able to hold all the oil that might drain into the reservoir from the system.
3. It must maintain the oil level high enough to prevent a "whirlpool" effect at the pump inlet line opening. Otherwise air will be drawn into the pump.
4. It should have a surface area large enough to dissipate most of the heat generated by the system.

**11-2.** This is a research project for students who should request literature from reservoir manufacturing firms.

1. Flat top design shown in Figure 11-1.

2. L-shaped design which consists of a vertical tank mounted on one side of a wide base. The other side of the base is used to mount the pump. Since the tank oil level is higher than the pump inlet, the possibility of cavitation is reduced due to the positive pump inlet pressure.

3. Overhead stack design which uses one or more modular frames which can be stacked in a vertical direction. Each frame contains its own pump and all the pumps (when more than one is used) receive oil from the single oil tank located on top. Since the tank oil level is higher than any of the pump inlets, the possibility of cavitation is reduced.

**11-3.** The purpose of a reservoir breather is to allow the reservoir to breathe as the oil level changes due to system demand requirements. In this way, the tank is always vented to the atmosphere.

**11-4.** The purpose of the baffle plate is to separate the pump inlet line from the return line to prevent the same fluid from re-circulating continuously within the tank. In this way all the fluid is uniformly used by the system.

**11-5.**
1. Weight loaded or gravity.
2. Spring loaded type.
3. Gas loaded type.

**11-6.**
1. Piston type: principal advantage is its ability to handle very high or low temperature system fluids through the utilization of compatible "O" ring seals.

2. Diaphragm type: primary advantage is its small weight-to-volume ratio which makes it suitable almost exclusively for air-born applications.

3. Bladder type: greatest advantage is the positive sealing between the gas and oil chambers.

**11-7.**
1. Auxiliary power source to store oil delivered by the pump during a portion of the work cycle.
2. Compensator for internal or external leakage during an extended period of time during which the system is pressurized but not in operation.
3. An emergency power source where a cylinder must be retracted even though the normal supply of oil pressure is lost due to a pump or electrical power failure.
4. Elimination or reduction of high-pressure pulsations or hydraulic shock.

**11-8.** A pressure intensifier is an auxiliary unit used to increase the pressure in a hydraulic system to a value above the pump discharge value. It accepts a high volume flow at relatively low pump pressure and converts a portion of this flow to high pressure. One application is for a punch press.

11-9. Positive seals do not allow any leakage whatsoever (external or internal). Non-positive seals (such as the clearance used to provide a lubricating film between a valve spool and its housing bore) permit a small amount of internal leakage.

11-10. Internal leak: leakage past piston rings in hydraulic cylinders.

External leak: leakage through pipe fittings which have become loose.

11-11. Static seals are used between mating parts which do not move relative to each other. Dynamic seals are assembled between mating parts which do move relative to each other.

11-12. At very high pressures, O-rings may extrude into the clearance space between mating parts as shown in Figure 11-24. This extrusion is prevented by installing a back-up ring as illustrated in Figure 11-24.

11-13.
1. V-ring packings.
2. Piston cup packings.
3. Piston rings.

11-14. Wiper seals are not designed to seal against pressure. Instead they are designed to prevent foreign abrasive or corrosive materials from entering a cylinder. As such, they provide insurance against rod scoring and add materially to packing life.

11-15.
1. Leather
2. Buna-N
3. Silicone
4. Neopreme

**11-16.** A durometer (See Figure 11-32) is an instrument used to measure the indentation hardness of rubber and rubber-like materials.

**11-17.** The purpose of a heat exchanger is to add heat or remove heat from the fluid of a hydraulic system so that the fluid temperature does not become too low or too high.

**11-18.** Heat generation rate, oil flow rate and allowable oil temperature.

**11-19.**
1. Rotameter.
2. Turbine flow meter.

**11-20.**
1. Bourdon gage.
4. Schrader gage.

**11-21.** Flowrate measurements are used to evaluate the performance of hydraulic components as well as troubleshooting a hydraulic system. They can be used to check the volumetric efficiency of pumps and also to determine leakage paths within a hydraulic circuit. Pressure measurements are used for testing and trouble-shooting purposes. They are used to adjust pressure settings of pressure control valves and to determine forces exerted by hydraulic cylinders and torques delivered by hydraulic motors.

**11-22.** Easier to read since values are given in digits rather than by a needle pointing along a scale.

**11-23.** There is a loss in pressure in the direction of flow and the greater the flow rate, the greater the pressure loss (pressure drop). Also the pressure drop is proportional to the square of the flow rate.

**11-24.** **The accumulator should be place just upstream of the DCV rather than just downstream of the DCV. This change will allow the charged accumulator to assist the pump in driving the cylinder during its extension stroke.**

**11-25.** **The directional control valve (DCV) should have a center-closed configuration to allow the accumulator to charge when the DCV is in its unactuatad (spring-centered) mode. The existing center bypass configuration does not allow the accumulator to charge when the DCV is unactuated.**

**11-26.** $$\text{Reservoir Size (gal)} = 3 \times \text{Pump Flowrate (gpm)}$$

$$= 3 \text{ x } 15 = \underline{45 \text{ gal}}$$

**11-27.** $$\text{Reservoir Size } (m^3) = 3 \times \text{Pump Flowrate } \left(m^3/\text{min}\right)$$

$$= 3 \times 0.001 \times 60 = \underline{0.18\ m^3}$$

**11-28.** $$\frac{\text{High Discharge Pressure}}{1000 \text{ psi}} = \frac{3}{1} = \frac{21}{\text{Low Discharge Flowrate}}$$

**Solving for the unknown quantities, we have:**

**High Discharge Pressure = 3 x 1000 = <u>3000 psi</u>**

**Low Discharge Flowrate = 21/3 = <u>7 gpm</u>**

**11-29.** $$\frac{\text{High Discharge Pressure}}{70 \text{ bars}} = \frac{3}{1} = \frac{0.001 \, m^3/s}{\text{Low Discharge Flowrate}}$$

$$\text{Low Discharge Flowrate} = \frac{0.001}{3} = \underline{0.000333 \, m^3/s}$$

**High Discharge Pressure = 3 x 70 = <u>210 bars</u>**

**11-30.** **First, calculate the horsepower lost and convert to the heat generation rate in units of BTU/min.**

$$HP = \frac{p(psi) \times Q(gpm)}{1714} = \frac{2000 \times 15}{1714} = 17.5 \text{ HP}$$

**BTU/min = HP x 42.4 = 17.5 x 42.4 = 742 BTU/min**

**Next, calculate the oil flow rate in units of lb/min.**

**Oil flow rate(lb/min) = 7.42 x oil flow rate(gpm)**

**= 7.42 x 15 = 111.3 lb/min**

**The temperature increase is found using Equation 11-2.**

$$\text{Temperature Increase} = \frac{742}{0.42 \times 111.3} = 15.9^\circ F$$

**Downstream oil temperature = 130 + 15.9 = <u>145.9°F</u>**

**11-31.** $Power\,(kW) = \dfrac{p\,(Pa)\times Q\left(m^3/s\right)}{1000} = \dfrac{\left(14\times10^6\right)\times\left(1000\times10^{-6}\right)}{1000} = 10\ kW$

**Oil flow rate = 895 x 0.001 = 0.895 kg/s**

**Temperature increase =** $\dfrac{14}{1.8 \times 0.895} = 8.7°\ C$

**Downstream oil temperature = 60 + 8.7 = <u>68.7°C</u>**

**11-32. Per Example 11-4 we have**

$$Pump\ \ HP\ \ Loss = \left(\frac{1}{0.82} - 1\right)\times\frac{2000\times15}{1714} = 3.84$$

$$PRV\ \ Average\ \ HP\ \ Loss = 0.60 \times \frac{2000 \times 15}{1714} = 10.50$$

$$Line\ Average\ HP\ Loss = (1.00 - 0.60) \times 0.15 \times \frac{2000 \times 15}{1714} = 1.05$$

**Total Average HP Loss = 15.39 HP**

**Heat Exchanger Rating = 15.39 x 2544 = <u>39,150 BTU/hr</u>**

**11-33. Per Example 11-5 we have**

$$Pump\ \ kW\ \ Loss = \left(\frac{1}{0.82} - 1\right)\times\frac{\left(14\times10^6\right)\times\left(1000\times10^{-6}\right)}{1000} = 3.07$$

$$PRV\ Average\ kW\ Loss = 0.60 \times \frac{\left(14 \times 10^6\right) \times \left(1000 \times 10^{-6}\right)}{1000} = 8.40$$

$$Line\ Average\ kW\ Loss = 0.40 \times 0.15 \times \frac{\left(14 \times 10^6\right) \times \left(1000 \times 10^{-6}\right)}{1000}$$

$$= 0.84$$

**Total kW Loss = <u>12.31 kW</u>**

11-34. Specific heat of oil = $0.42\ ^{BTU}\!/_{lb}\ /°\ F = 1.8\ ^{kJ}\!/_{kg}\ /°\ C$

Thus $0.42\ \dfrac{BTU}{lb \bullet °\ F} = 1.8\ \dfrac{kJ}{kg \bullet °\ C}$

Hence $\dfrac{0.42}{0.42}\ BTU = \dfrac{1.8}{0.42}\ \dfrac{kJ}{kg \bullet °\ C} \times \dfrac{2.20\ lb}{2.20} \times \dfrac{1.8\ °\ F}{1.8}$

Therefore $1\ BTU = \dfrac{1.8}{0.42}\ \dfrac{kJ}{kg \bullet °\ C} \times \dfrac{1\ kg}{2.20} \times \dfrac{1°\ C}{1.8} = \underline{1.08\ kJ}$

11-35. Heat Loss $= 0.75 \times \left(2\ HP\right) \times \left(42.4 \times 60\ \dfrac{BTU}{hr \bullet HP}\right) \times \left(5\ hr\right)$

$= \underline{19{,}000\ BTU}$

11-36. $HP = \dfrac{p\left(psi\right) \times Q\left(gpm\right)}{1714}$ where 1 HP = 2544 BTU/hr

Operating rate of high pressure PRV HP loss = $\dfrac{3000 \times 12}{1714}$

= 21.0 HP

Avg rate of high press PRV HP loss = $21.0\ HP \times \dfrac{2}{6} = 7\ HP$

Oper rate of low press PRV HP loss = $\dfrac{600 \times 12}{1714} = 4.2\ HP$

Avg rate of low press PRV HP loss = $4.2\ HP \times \frac{4}{6} = 2.8\ HP$

Total average HP loss = 7 + 2.8 = 9.8 HP

$$= 9.8\ HP \times \frac{2544\ BTU/hr}{1\ HP} = \underline{24{,}900\ BTU/hr}$$

11-37. Pump Power $= \frac{0.020}{60}\frac{m^3}{s} \times 15{,}000\ kPa = 5\ kW$

Heat Loss = 0.80 x 1 kW = 0.80 kW

= 0.80 x 60 kJ/min = <u>48 kJ/min</u>

# Chapter 12

## Maintenance of Hydraulic Systems

**12-1.** 1. Clogged or dirty oil filters.
2. Inadequate supply of oil in the reservoir.
3. Leaking seals.
4. Loose inlet lines which cause the pump to take in air.

**12-2.** <u>Over half of all hydraulic system problems have been traced directly to the oil.</u>

**12-3.** Oxidation is caused by the chemical reaction of oxygen from the air with particles of oil. Corrosion is the chemical reaction between a metal and acid.

**12-4.** In applications where human safety is of concern.

**12-5.** 1. Flash point: the temperature at which the oil surface gives off sufficient vapors to ignite when a flame is passed over the surface.
2. Fire point: the temperature at which the oil will release sufficient vapor to support combustion continuously for 5 seconds when a flame is passed over the surface.
3. Autogenous ignition temperature: the temperature at which ignition occurs spontaneously.

**12-6.** 1. Water-glycol solutions.
2. Water-in-oil emulsions.
3. Straight systhetics.
4. High water content fluids.

**12-7.** 1. Special paints must be used.
2. Incompatibility with most natural or synthetic rubber seals.
3. High costs.

**12-8.** Air can become dissolved or entrained in hydraulic fluids. This can cause pump cavitation and also greatly reduce the bulk modulus of the hydraulic fluid. Foam resistant fluids contain chemical additives which break out entrained air to quickly separate the air from the oil while it is in the reservoir.

**12-9.** To prevent wear between the closely fitted working parts.

**12-10.** Coefficient of friction (CF) is the proportionality constant between a normal force (N) and the frictional force (F) it creates between two mating surfaces sliding relative to each other. (CF = F/N).

**12-11.** The neutralization number is a measure of the relative acidity or alkalinity of a hydraulic fluid and is specified by a Ph factor.

**12-12.** To prevent oxidation.

**12-13.** It may cause cavitation problems in the pump due to excessive vacuum pressure in the pump inlet line unless proper design steps are implemented.

**12-14.** Normally thorough draining, cleaning and flushing are required. It may even be necessary to change seals and gaskets on the various hydraulic components.

**12-15.** Controlling pollution and conserving natural resources are important goals to achieve for the benefit of society. Thus it is important to minimize the generation of waste hydraulic fluids and to dispose of them in an environmentally sound manner.

**12-16.** 1. Select the optimum fluid for the application involved.
2. Utilize a well designed filtration system to reduce contamination and increase the useful life of the fluid.
3. Follow proper storage procedures of the unused fluid supply.
4. Transporting of the fluids from the storage containers to the hydraulic systems, should be done carefully since the chances for contamination increase greatly with handling.
5. Operating fluids should be checked regularly for viscosity, acidity, bulk modulus, specific gravity, water content, color, additive levels, concentration of metals and particle contamination.
6. The entire hydraulic system including pumps, piping, fittings, valves, solenoids, filters, actuators and the reservoir should be maintained according to manufacturer's specifications.
7. Corrective action should be taken to reduce or eliminate leakage from operating hydraulic system.
8. Disposal of fluids must be done properly. An acceptable way to dispose of fluids is to utilize a disposal company that is under contract to pick up waste hydraulic fluids.

**12-17.** 1. The type of symptoms encountered, how they were detected, and the date.
2. A description of the maintenance repairs performed. This should include the replacement of parts, the amount of downtime and the date.

**12-18.** A filter is a device whose primary function is to retain, by some porous medium, insoluble contaminants from a fluid. Basically, a strainer is a coarse filter. Strainers are constructed of a wire screen which rarely contains openings less than 0.0059 inches. Thus, a strainer removes only the larger particles.

**12-19.** 1. Built into the system during component maintenance and assembly.
2. Generated within system during operation.
3. Introduced into system from external environment.

12-20. One micron is 1 millionth of a meter or 0.000039 inches. Therefore, ten microns is 0.00039 in. A ten-micron filter is one capable of removing contaminants as small as ten microns in size.

12-21. 1. Mechanical
2. Absorbent
3. Adsorbent

12-22. An indicating filter is one which contains an indicating element which signals the operator when cleaning is required.

12-23. 1. Proportional flow filter in separate drain line.
2. Full flow filter in suction line.
3. Full flow filter in pressure line.
4. Full flow filter in return line.

12-24. 1. Flow meters.
2. Pressure gages.
3. Temperature gages.

12-25. When troubleshooting hydraulic circuits, it should be kept in mind that a pump produces the flow of a fluid. However, there must be resistance to flow in order to have pressure.

12-26. 1. Air entering pump inlet.
2. Misalignment of pump and drive unit.
3. Excessive oil viscosity.
4. Dirty inlet strainer.
5. Chattering relief valve.

12-27. 1. Air in the fluid.
2. Pressure relief valve set too low.
3. Pressure relief valve not properly seated.
4. Leak in hydraulic line.

**12-28.** **1. Pump turning in wrong direction.**
**2. Ruptured hydraulic line.**
**3. Low oil level in reservoir.**
**4. Pressure relief valve stuck open.**

**12-29.** **1. Faulty pump.**
**2. Directional control valve fails to shift.**
**3. System pressure too low.**
**4. Defective actuator.**
**5. Actuator load is excessive.**

**12-30.** **1. Air in system.**
**2. Viscosity of fluid too high.**
**3. Worn or damaged pump.**
**4. Pump speed too low.**
**5. Excessive leakage through actuators or valves.**

**12-31.** **1. Heat exchanger turned off or faulty.**
**2. Undersized components or piping.**
**3. Incorrect fluid.**
**4. Continuous operation of pressure relief valve.**
**5. Overloaded system.**
**6. Reservoir too small.**

**12-32.** **OSHA stands for the Occupational Safety and Health Administration of the Department of Labor. OSHA is attempting to prevent safety hazards which can be harmful to the health and safety of personnel.**

**12-33.** **1. <u>Workplace Standards</u>: In this category are included the safety of floors, entrance and exit areas, sanitation and fire protection.**

**2. <u>Machines and Equipment Standards</u>: Important items are machine guards, inspection and maintenance techniques, safety devices and the mounting, anchoring and grounding of fluid power equipment. Of big concern are noise levels produced by operating equipment.**

**3. <u>Materials Standards</u>: These standards cover items such as toxic fumes, explosive dust particles and excessive atmospheric contamination.**

4. Employee Standards: Concerns here include employee training, personnel protective equipment and medical and first air services.

5. Power Source Standards: Standards are applied to power sources such as electric, hydraulic, pneumatic and steam supply systems.

6. Process Standards: Many industrial processes are included such as welding, spraying, abrasive blasting, part dipping and machining.

7. Administrative Regulations: Industry has many administrative responsibilities which it must meet. These include the displaying of OSHA posters stating the rights and responsibilities of both the employer and employees. Industry is also required to keep safety records on accidents, illnesses and other exposure-type occurrences. An annual summary must also be posted.

It is important that safety be incorporated into hydraulic systems to insure compliance with OSHA regulations. The basic rule to follow is that there should be no compromise when it comes to the health and safety of people at the place of their employment.

12-34. Pumps do not pump pressure. Instead they produce fluid flow. The resistance to this flow, produced by the hydraulic system, is what determines pressure. Low oil level in the reservoir could be a cause of no pressure even though there is nothing wrong with the pump.

12-35. An excessive pressure drop occurs across the filter resulting in reduced pressure downstream of the filter. This can adversely affect the operation of the pump (starved pump resulting in cavitation) and actuators (slow or no motion) depending on filter location.
A filter containing a bypass relief valve, assures non-excessive pressure drop and thus adequate flow no matter how dirt-clogged the filter might become. However filtration no longer is accomplished until the filter is replaced.

**12-36.** **Cylinder friction is influenced by the type of materials in sliding contact, the type of fluid lubricating the sliding surfaces, and the magnitude of the normal force between the mating surfaces.**

**12-37.** **The nominal rating is the micron value specified for which 95% of entering particles of size greater than the nominal rating will be trapped. The absolute rating represents the size of the largest opening or pore in the filter and thus indicates the largest size particle that could pass through the filter.**

**12-38.** **These required cleanliness levels can be used to select the proper filtration system for a given hydraulic application.**

**12-39.** **This means counting the particles per unit volume for specific particle sizes and comparing the resultss to a required cleanliness level. It uses an ISO code number that represents either the number of particles per milliliter of fluid of size greater than 5 micrometers or greater than 15 micrometers. In using the code, two numbers are used separated by a slash.**

**12-40.** **The prevention of the hydraulic fluid from providing lubrication of moving internal members of hydraulic components such as pumps, hydraulic motors, valves and actuators.**

**12-41.** **Contaminants can collect inside the clearance between moving mating parts and thus block lubricant flow. Also contaminants can rub against mating surfaces causing a breakdown in the fluid lubricating film.**

**12-42.**
1. **Free air.**
2. **Entrained gas.**
3. **Dissolved air.**

12-43. Vapor pressure is defined as the pressure at which a liquid starts to boil (vaporize) and thus begin changing into a vapor (gas).

12-44. As the vapor bubbles are exposed to the high pressure at the outlet port of a pump, the bubbles are collapsed thereby creating extremely high local fluid velocities. This high velocity fluid impacts on internal metal surfaces of the pump. The resulting high impact forces, cause flaking or pitting of the surfaces of the internal components such as gear teeth, vanes and pistons. This results in premature pump failure.

12-45. Pump manufacturers specify a minimum allowable vacuum pressure at the pump inlet port based on the type of fluid being pumped, the maximum operating temperature and the rated pump speed.

12-46.
1. Keep suction velocities below 5 ft/s (1.5 m/s).
2. Keep pump inlet lines as short as possible.
3. Mount the pump as close as possible to the reservoir.
4. Minimize the number of fittings in the pump inlet line.
5. Use low pressure drop pump inlet filters.
6. Use a properly designed reservoir.
7. Use the proper oil recommended by the pump manufacturer.
8. Keep the oil temperature from exceeding the recommended maximum level (usually 150°F/65°C).

12-47. Developing Biodegradable Fluids: This issue deals with preventing environmental damage caused by potentially harmful material leaking from fluid power systems. Oil companies are developing vegetable based fluids that are biodegradable and compatible with fluid power equipment.

Reducing Oil Leakage: Hydraulic fluid leakage can occur at pipe fittings in hydraulic systems and at mist-lubricators in pneumatic systems. This represents an environmental issue because the EPA has identified oil as a hazardous air pollutant. To resolve this issue, the fluid power industry is striving to produce zero-leakage systems.

Maintaining and Disposing of Hydraulic Fluids: It is important to minimize the generation of waste hydraulic fluids and to dispose of them in an environmentally sound manner. These results can be accomplished by implementing fluid-control and preventive maintenance programs along with proper fluid-disposal programs. Proper maintaining and disposing of hydraulic fluids represent a cost effective way to achieve a cleaner environment while conserging natural resources.

Reducing Noise Levels: Hydraulic power units such as pumps and motors can operate at noise levels exceeding the limits established by OSHA. New standards of reduced noise levels are being met by fluid power manufacturers in their efforts to produce safe, efficient, reliable, cost effective products.

12-48. The two key equations are:

$$v_{ret} = \frac{Q_{pump}}{A_P - A_R} \quad and \quad Q_{filter} = A_P\, v_{ret}$$ Thus we have

$$Q_{filter} = \frac{A_P}{A_P - A_R} \times Q_{pump} = \frac{\frac{\pi}{4}(0.125\ m)^2}{\frac{\pi}{4}(0.125^2 - 0.075^2)m^2} \times 75\ LPM = \underline{117\ LPM}$$

12-49. None because the cylinder operation includes both extension and retraction.

12-50. Beta ratio = $\frac{30{,}000}{1050} = \underline{28.6}$

12-51. Beta efficiency = $\frac{30{,}000 - 1050}{30{,}000} = \underline{96.5\ \%}$

**12-52.** Beta efficiency $= 1 - \dfrac{1}{\text{Beta ratio}}$

**12-53.** Identifies a particle size of 10 microns and a Beta ratio of 75 for a particular filter.

**12-54.** One

**12-55.** A code designation of ISO 26/9 indicates that per millimeter of fluid there are 640,000 particles of size greater than 5 micrometers and 5 particles of size greater than 15 micrometers.

**12-56.** The left most number corresponds to particle sizes greater than 5 micrometers and the right most number corresponds to particle sizes greater than 15 micrometers.

# Chapter 13

## Pneumatics - Air Preparation and Components

13-1. 1. Liquids exhibit greater inertia than do gases.
2. Liquids exhibit greater viscosity than do gases.
3. Hydraulic systems require special reservoirs and no-leak design components.

13-2. Standard air is sea level air having a temperature of 68°F, a pressure of 14.7 psia and a relative humidity of 36%.

13-3. 1. Boyle's Law states that, if the temperature of a given amount of gas is held constant, the volume of the gas will change inversely with the absolute pressure of the gas.
2. Charles' Law states that, if the pressure on a given amount of gas is held constant, the volume of the gas will change in direct proportion to the absolute temperature.
3. Gay-Lussac's Law states that, if the volume of a given gas is held constant, the pressure exerted by the gas is directly proportional to its absolute temperature.

13-4. 1. Piston type.
2. Screw type.
3. Sliding vane type.

13-5. Compressors having more than one cylinder are called multistage compressors. Staging means dividing the total pressure increase among two or more cylinders by feeding the exhaust from one cylinder into the inlet of the next cylinder. This improves pumping efficiency.

**13-6.** The function of an air filter is to remove contaminants from the air before it reaches pneumatic components such as valves and actuators.

**13-7.** An air pressure regulator is used so that a constant pressure is available for a given pneumatic system.

**13-8.** A lubricator insures proper lubricating of internal moving parts of pneumatic components.

**13-9.** A pneumatic indicator is a device which provides a two-color, two-position visual indication of air pressure.

**13-10.** A pneumatic exhaust silencer (muffler) is used to control the noise caused by a rapidly exhausting air stream flowing into the atmosphere.

**13-11.** An aftercooler is installed in the airline immediately downstream of the compressor. Compressors do not remove moisture. Thus, an aftercooler is essential to reduce the air temperature to convenient levels and to act as a first stage in removal of moisture prior to entering an air dryer.

Aftercoolers remove only about 80% of the moisture from the air leaving the compressor. An air dryer removes virtually all moisture by lowering the temperature of the pressurized air to a dew point of 50°F.

**13-12.** The dew point is the temperature at which air is saturated and thus the relative humidity is 100%.

**13-13.** 100%

**13-14.** Pneumatic actuators are of lighter construction making extensive use of aluminum and other non-ferrous alloys to reduce weight, improve heat transfer characteristics and minimize corrosive action of air.

**13-15.** 1. Determine pressure capacity requirement.
2. Establish number of stages required.
3. Determine scfm of air required.
4. Size the air receiver and compressor.
5. Determine type of compressor (piston, vane or screw).
6. Establish type of unloader control and pressure settings.

**13-16.** 1. Supply air at system steady flow rate requirements.
2. Supply air at essentially constant pressure.
3. Dampen pressure pulses either coming from the compressor or the pneumatic system during valve shifting and component operation.
4. Handle transient air demands exceeding compressor capability with a maximum and minimum pressure range.

**13-17.** Starting torque is the torque produced under load at zero speed.

**13-18.** Flow capacity constant is the proportionality constant between flow rate and valve pressure drop and downstream pressure. Thus for the same valve pressure drop and downstream pressure, the flow rate increases directly with the flow capacity constant. Hence a large flow capacity constant indicates a large size valve.

**13-19.** $V_1 = 20\ in^3, \quad p_1 = 30 + 14.7 = 44.7\ psia$

$$V_2 = 20 - \frac{\pi}{4} \times 2^2 \times 5 = 4.29\ in^3$$

From Boyle's Law, we have:

$$\frac{20}{4.29} = \frac{p_2}{44.7} \quad Thus \quad p_2 = 208.4 \ psia = \underline{193.7 \ psig}$$

**13-20.** **$T_1$ = 80 + 460 = 540° R, $T_2$ = 150 + 460 = 610° R, $V_1$ = 20 $in^3$**

**From Charles' Law, we have:**

$$\frac{20}{V_2} = \frac{540}{610} \qquad Thus \quad V_2 = \underline{22.6 \ in^3}$$

**13-21.** $p_1 = 30 + 14.7 = 44.7 \ psia$

**$T_1$ = 80 + 460 = 540° R, $T_2$ = 160 + 460 = 620° R**

**Using Gay-Lussac's Law, we obtain:**

$$\frac{44.7}{p_2} = \frac{540}{620} \quad Thus \quad p_2 = 51.3 \ psia = \underline{36.6 \ psig}$$

**13-22.** **Solve the general gas law for $P_2$ and substitute known values:**

$$p_2 = \frac{p_1 V_1 T_2}{V_2 T_1} = \frac{(1200 + 14.7) \times 2000 \times (250 + 460)}{1500 \times (120 + 460)} = 1983 \ psia$$

$$= \underline{1968 \ psig}$$

**13-23.** $\frac{V_1}{V_2} = \frac{p_2}{p_1} \quad where \quad V_1 = 300 \ cm^3$

$$V_2 = 300 - \frac{\pi}{4} \times 5^2 \times 13 = 300 - 255 = 45 \ cm^3$$

$$p_1 = 2 \times 10^5 + 1 \times 10^5 = 3 \times 10^5 \ Pa \ abs$$

$$\frac{300}{45} = \frac{p_2}{3\times10^5}$$

$$Thus \quad p_2 = 20\times10^5\,Pa\ abs = 20\ bars\ abs = \underline{19\ bars\ gage}$$

**13-24.** $T_1 = 30 + 273 = 303° K$ and $T_2 = 65 + 273 = 338° K$

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \quad so \quad \frac{130}{V_2} = \frac{303}{338} \quad and \quad V_2 = \underline{145\ cm^3}$$

**13-25.** $\frac{p_1}{p_2} = \frac{T_1}{T_2}$ where $p_1 = 2+1 = 3\ bars\ abs$

Also $T_1 = 25 + 273 = 298° K$ and $T_2 = 70 + 273 = 343° K$

$$Thus \quad \frac{3}{p_2} = \frac{298}{343} \quad and \quad p_2 = 3.45\ bars\ abs = \underline{2.45\ bars\ gage}$$

**13-26.** $p_2 = \frac{p_1V_1T_2}{V_2T_1} = \frac{81\times1290\times(120+273)}{1000\times(50+273)} = 127\ bars\ abs$

$= \underline{126\ bars\ gage}$

**13-27.** $°C = \frac{°F - 32}{1.8} = \frac{160 - 32}{1.8} = \underline{71.1°C}$

$°R = °F + 460 = 160 + 460 = \underline{620°R}$

$°K = °C + 273 = 71.1 + 273 = \underline{344.1°K}$

**13-28.** **Solve Equation 13-6 for $V_1$ and let subscript 1 represent atmospheric conditions.**

$$V_1 = \frac{V_2\, p_2\, T_1}{p_1\, T_2} = cfm\ of\ free\ air$$

$$V_1 = \frac{30 \times (150 + 14.7) \times (80 + 460)}{14.7 \times (100 + 460)} = \underline{324 \text{ cfm of free air}}$$

**13-29.** $(a)\quad V_r = \frac{14.7\, t\left(Q_r - Q_C\right)}{p_{\max} - p_{\min}} = \frac{14.7 \times 10 \times (30 - 0)}{120 - 100} = \underline{221\ ft^3}$

$$\text{(b)}\quad V_r = \frac{14.7 \times 10 \times (30 - 6)}{120 - 100} = \underline{176 \text{ ft}^3}$$

**13-30.** $V_1 = \frac{V_2\, p_2\, T_1}{p_1\, T_2} = \frac{1 \times (1000 + 101) \times (20 + 273)}{101 \times (40 + 273)} = \underline{10.2\ std\ m^3/\text{min}}$

**13-31.** (a) $\text{Consumption rate} = 30 \text{ scfm} \times \left(\frac{1 \text{ m}}{3.28 \text{ ft}}\right)^3 = 0.850 \text{ std } m^3/\text{min}$

$$p_{max} = 120\ psi \times \frac{1\ kPa}{0.145\ psi} = 828\ kPa$$

$$p_{min} = 100\ psi \times \frac{1\ kPa}{0.145\ psi} = 690\ kPa$$

$$V_r = \frac{101\,t\left(Q_r - Q_C\right)}{p_{max} - p_{min}} = \frac{101 \times 10(0.850 - 0)}{828 - 690} = \underline{6.22\ m^3}$$

**(b)** **Compressor delivery rate** $= 6\ \text{scfm} \times \left(\frac{1\ \text{m}}{3.28\ \text{ft}}\right)^3$

$= 0.170\ \text{std}\ \text{m}^3/\text{min}$

$$V_r = \frac{101 \times 10\,(0.850 - 0.170)}{828 - 690} = \underline{4.98\ \text{m}^3}$$

**13-32.** $$\text{Theoretical}\ HP = \frac{p_{in}\,Q}{65.4}\left[\left(\frac{p_{out}}{p_{in}}\right)^{0.286} - 1\right]$$

$$= \frac{14.7 \times 200}{65.4}\left[\left(\frac{134.7}{14.7}\right)^{0.286} - 1\right] = 39.8\ HP$$

$$\text{Actual HP} = \frac{HP_{Theor}}{\eta_o} = \frac{39.8}{0.72} = \underline{55.3\ \text{HP}}$$

**13-33.** **Theoretical Power (kW) = Actual Power (kW) × $\eta_o$**

$$= \frac{p_{in}\,Q}{17.1}\left[\left(\frac{p_{out}}{p_{in}}\right)^{0.286} - 1\right]$$

**Substituting known values yields:**

$$20 \times 0.75 = \frac{100 \times 4}{17.1}\left[\left(\frac{p_{out}}{100}\right)^{0.286} - 1\right]$$

$$Thus \left(\frac{p_{out}}{100}\right)^{0.286} = 0.641 + 1 = 1.641$$

$$And \quad \frac{p_{out}}{100} = 5.66 \quad so \quad \underline{p_{out} = 566\ kPa\ abs}$$

**13-34.** **(a) Per Fig. 13-29 the atmospheric air entering the compressosr contains 1.58 lb of moisture per 1000 $ft^3$. Thus the rate at which moisture enters the compressor can be found.**

$$Moisture\ rate\left(\frac{lb}{\min}\right) = Entering\ Moisture\ Content\left(\frac{lb}{ft^3}\right)$$

$$\times Entering\ scfm\ Flowrate\left(\frac{ft^3}{\min}\right)$$

$$= \frac{1.58\ lb}{1000\ ft^3} \times 50\ \frac{ft^3}{\min} = 0.079\ lb/\min$$

Since water weighs 8.34 lb/gal we have

$$gal/day = 0.079\frac{lb}{\min}\times\frac{60\min}{1\ hr}\times 16\frac{hr}{day}\times\frac{1\,gal}{8.34\ lb} = 9.09\ gal/day$$

(b) Per Fig. 13-29 if the compressed air is cooled back to 80°F, the maximum amount of moisture the air can hold per 1000 $ft^3$ of free air is 0.17 lb. Since (1.58 – 0.17)/1.58 = 0.892, then 89.2% of the moisture would condense out of the air. Thus the gallons of moisture per day received by the pneumatic system is

gal/day = (1-0.892)(9.09 gal/day) = <u>0.982 gal/day</u>

(c) Per Fig. 13-29 the max. amount of moisture the air leaving the air dryer can hold per 1000 $ft^3$ of free air is 0.05 lb. Since (1.58 – 0.05)/1.58 = 0.968, the moisture removal rate is 96.8%. Thus the moisture received per day by the pneumatic system is

gal/day = (1 – 0.968)(9.09 gal/day) = <u>0.29 gal/day</u>

13-35. Per Fig. 13-29 the entering moisture content

equals $2.13\ lb/1000\ ft^3$

Thus the leaving moisture content $= 0.15\times 2.13 = 0.320\ lb/1000\ ft^3$

At $0.320\ lb/1000\ ft^3$ $and\ 90°F,\ the\ pressure = \underline{80\ psig}$

13-36. T = 100 + 460 = 560° R

$p_1 = 125 + 14.7 = 139.7\ psia$

$p_2 = 0.53 \times 139.7 = 74.0\ psia$

**Substituting directly into Eqn. 13-10 yields the answer.**

$$Q = 22.7 \times 7\sqrt{\frac{(139.7 - 74.0) \times 74.0}{560}} = \underline{468\ scfm}$$

**13-37.** $\frac{Downstream\ pressure}{Upstream\ pressure} = \frac{p_2}{p_1} = \frac{180\ kPa\ abs}{400\ kPa\ abs} = 0.45$

$Since\ \frac{p_2}{p_1}\ is\ less\ than\ 0.53,\ \underline{the\ valve\ is\ choked}.$

**13-38.** $Q_1 = Q_2\left(\frac{p_2}{p_1}\right)\left(\frac{T_1}{T_2}\right)$

**where state 1 is the standard air state and constant temperature has been assumed.**

$$Q_2 = \frac{\frac{\pi}{4} \times 2.5^2 \times 12 \times 30}{1728} = 1.02\ ft^3/\min\ consumed\ by\ cyl\ at\ 100\ psig$$

$$Therefore\ Q_1 = \frac{p_2 Q_2}{p_1} = \frac{114.7 \times 1.02}{14.7} = \underline{7.96\ scfm}$$

**13-39.** $Q_1 = Q_2\left(\frac{p_2}{p_1}\right)\left(\frac{T_1}{T_2}\right)$

**where state 1 is the standard air state and constant temperature has been assumed.**

$Q_2 = \frac{\pi}{4}\times 0.06^2 \times 0.30\times 30 = 0.0254\,{m^3}/{\min}$ *consumed by the cyl* **at 700 kPa(gage).**

$Therefore \quad Q_1 = \frac{p_2 Q_2}{p_1} = \frac{800\times 0.0254}{100} = \underline{0.203\ std\ {m^3}/{\min}}$

**13-40. Ignoring the effect of the rod and assuming constant temperature we have:**

$Q_2 = \frac{\pi}{4}\times 0.050^2 \times (0.025\times 2)\times 80 = 0.00785\ {m^3}/{\min}$ **consumed by the cylinder at 600 kPa(gage).**

$Q_1 = \frac{V_1}{t} = \frac{p_2 Q_2}{p_1} \quad so \quad t = \frac{p_1 V_1}{p_2 Q_2} = \frac{100\times 100}{700\times 0.00785} = 1820\ \min = \underline{30.3\ hr}$

**13-41.** $Q_2 = \frac{\frac{\pi}{4}\times 2^2 \times (12\times 2)\times 200}{1728} = 8.72\,{ft^3}/{\min}$ **consumed by the cylinder at 100 psig.**

$Thus \quad Q_1 = \frac{p_2 Q_2}{p_1} = \frac{114.7\times 8.72}{14.7} = \underline{68.0\ \ scfm}$

**13-42.** $Q_1 = Q_2\left(\frac{p_2}{p_1}\right)\left(\frac{T_1}{T_2}\right)$ **where state 1 is the standard air state and constant temperature has been assumed.**

$$Q_2 = 4\frac{in^3}{rev}\times 1750\frac{rev}{\min}\times\frac{1\,ft^3}{1728 in^3} = 4.05\ ft^3/\min$$ **consumed by the air motor at 100 psig.**

$$Thus \qquad Q_1 = \frac{p_2 Q_2}{p_1} = \frac{114.7\times 4.05}{14.7} = \underline{31.6\ \ scfm}$$

$$Power(hp) = \frac{\Delta p(psi)\times Q(gpm)}{1714} = \frac{100(4.05\times 448)}{1714\times 60} = \underline{1.76\ hp}$$

**13-43.** $Q_1 = Q_2\left(\frac{p_2}{p_1}\right)\left(\frac{T_1}{T_2}\right)$

**where state 1 is the standard air state and constant temperature has been assumed.**

$$Q_2 = 0.000080\frac{m^3}{rev}\times 1750\frac{rev}{\min} = 0.14\ m^3/\min$$ **consumed by the air motor at 700 kPa gage.**

$$Q_1 = \frac{p_2 Q_2}{p_1} = \frac{801\times 0.14}{101} = \underline{1.11\ std\ \ m^3/\min}$$

$$Power(kW) = (\Delta p)\times Q = 700\ kPa\times\frac{0.14}{60}\ \frac{m^3}{s} = \underline{1.63\ kW}$$

**13-44.** $(a)\quad V = \frac{\pi}{4}\times 1^2\times 2 = \underline{1.57 in^3}$

(b) $V = \underline{1.57 \text{ in}^3}$

(c) $S = \dfrac{1.57}{\frac{\pi}{4} \times 1.5^2} = \underline{0.89 \text{ in}}$

(d) $V = \dfrac{\pi}{4} \times 8^2 \times 2 = \underline{101 \text{ in}^3}$

(e) $Q = 1.57 \times 1 = 1.57 \dfrac{\text{in}^3}{\text{s}} = 94.2 \dfrac{\text{in}^3}{\text{min}} = \underline{0.41 \text{ gpm}}$

(f) $p_{oil} = \dfrac{12{,}000}{\frac{\pi}{4} \times 1.5^2} = 6790 \ psi$ **and** $p_{air} = \dfrac{6790}{8^2} = 106 \ psi$ **g**

$$Q = 101 \frac{in^3}{s} \times \frac{106 + 14.7}{14.7} = 829 \ in^3/s \ \ of \ \ std \ \ air$$

$$Q = \frac{829 \times 60}{1728} = \underline{28.8 \ scfm}$$

**13-45.** The load $= 12{,}000 \text{ lb} \times \dfrac{1 \text{ N}}{0.225 \text{ lb}} = 53{,}300 \text{ N}$

Hydr. cyl. dia. $= 1.5 \text{ in} \times \dfrac{1 \text{ ft}}{12 \text{ in}} \times \dfrac{1 \text{ m}}{3.28 \text{ ft}} = 0.0381 \text{ m}$

Air piston diameter $= 8 \text{ in} \times \dfrac{1 \text{ ft}}{12 \text{ in}} \times \dfrac{1 \text{ m}}{3.28 \text{ ft}} = 0.203 \text{ m}$

Oil piston diameter $= 1 \text{ in} \times \dfrac{1 \text{ ft}}{12 \text{ in}} \times \dfrac{1 \text{ m}}{3.28 \text{ ft}} = 0.0254 \text{ m}$

Intensifier stroke $= 2 \text{ in} \times \dfrac{1 \text{ ft}}{12 \text{ in}} \times \dfrac{1 \text{ m}}{3.28 \text{ ft}} = 0.0508 \text{ m}$

Intensifier frequency = 1 stroke/s

(a) $V = \frac{\pi}{4} \times 0.0254^2 \times 0.0508 = 0.0000257\ m^3 = \underline{0.0257\ L}$

(b) $V = \underline{0.0257\ L}$

(c) $S = \frac{0.0000257}{\frac{\pi}{4} \times 0.0381^2} = 0.0225\ m = \underline{22.5\ mm}$

(d) $V = \frac{\pi}{4} \times 0.203^2 \times 0.0508 = 0.00164\ m^3 = \underline{1.64\ L}$

(e) $Q = 0.0257\ L \times \frac{1}{s} = \underline{0.0257\ L/s}$

(f) $p_{oil} = \frac{53{,}300\ N}{\frac{\pi}{4} \times 0.0381^2} = 46.8\ MPa$

$$p_{air} = \frac{46.8\ MPa}{\left(\frac{8}{1}\right)^2} = 0.731\ MPa = 731\ kPa$$

$$Q = 0.00164 \frac{m^3}{s} \times \frac{60\ s}{1\ \text{min}} \times \frac{731 + 101}{101} = \underline{0.811\ std\ m^3/\text{min}}$$

# CHAPTER 14

## Pneumatics - Circuits and Applications

14-1. 1. Safety of operation.
2. Performance of desired function.
3. Efficiency of operation.
4. Costs.

14-2. Because the air is clean and thus does not contaminate the environment.

14-3. This results in excessive pressure losses due to friction.

14-4. This results in excessive initial installation costs.

14-5. The compressor must operate at higher output pressure which requires greater input power.

14-6. The compressor must provide a greater flow rate to offset the air leaks into the atmosphere.

14-7. It costs money to provide the input power to drive a compressor for providing compressed air at greater than atmospheric pressure.

14-8. Systems where an air vacuum pressure is used to create a net force to perform a useful function.

14-9. 1. Materials handling.

2. Sealing.
3. Vacuum forming.

**14-10.** The exact amount of suction pressure developed can not be guaranteed.

**14-11.** Objects to be lifted can not generally weigh more than several hundred pounds because the maximum suction pressure equals one atmosphere of pressure.

**14-12.** It reduces the size and power requirements of a pump to handle system large transient flow rate conditions.

**14-13.**
1. Preload after charge gas has been added to ACC.
2. Charge after pump has been turned on and pressure reaches PRV setting.
3. Final position of ACC piston after load is fully driven.

**14-14.** First solve for the compression ratio.

$$\text{CR} = \frac{125 + 14.7}{14.7} = 9.50$$

$$\textit{From Figure } 14-3, \quad d^{5.31} = 1.2892$$

Finally, using the Harris Formula, the pressure loss is found.

$$p_f = \frac{0.1025\,L\,Q^2}{C\,R\,d^{5.31}} = \frac{0.1025 \times 150 \times \left(150/60\right)^2}{9.50 \times 1.2892} = \underline{7.85\ psi}$$

**14-15.** The total equivalent length of the pipe can be found using Figure 14-4.

$$L = 150 + 3 \times 0.56 + 2 \times 29.4 + 4 \times 1.50 + 5 \times 2.60$$

$$= 150 + 1.68 + 58.8 + 6.0 + 13.0 = 229.5\ \text{ft}$$

Substituting into the Harris Formula yields the answer.

$$p_f = \frac{0.1025 \times 229.5 \times \left(150/60\right)^2}{9.50 \times 1.2892} = \underline{12.0\ psi}$$

**14-16.** First solve for the compression ratio where 1000 kPa gage

$$= 1000\ \text{kPa gage} \times \frac{14.7\ \text{psi}}{101\ \text{kPa}} = 145.5\ \text{psig}$$

$$\text{Compression ratio} = CR = \frac{145.5 + 14.7}{14.7} = 10.9$$

$$\text{The pipe inside dia. is } d = 25\ \text{mm} \times \frac{1\ \text{in}}{25.4\ \text{mm}} = 0.984\ \text{in}$$

$$\text{Thus} \quad d^{5.31} = 0.984^{5.31} = 0.918$$

Finally using the Harris Formula, the pressure loss is found where

$$Q = 3\ \frac{\text{m}^3}{\text{min}} \times \left(\frac{3.28\ \text{ft}}{1\ \text{m}}\right)^3 = 105.9\ \text{scfm} \quad \text{and} \quad L = 100\ \text{m} = 328\ \text{ft}$$

$$p_f = \frac{0.1025 \times 328 \times \left(\frac{105.9}{60}\right)^2}{10.9 \times 0.918} = 10.5\ psi = 10.5\ psi \times \frac{101\ kPa}{14.7\ psi}$$

$$= \underline{72.1\ \text{kPa}}$$

**14-17.** **The total equivalent length of the pipe can be found using Figure 14-4 for a 1 inch nominal pipe size.**

**L = 328 + 2 × 0.56 + 3 × 29.4 + 5 × 1.50 + 4 × 2.60 + 6 × 1.23**

**= 328 + 1.12 + 88.2 + 7.5 + 10.4 + 7.38 = 442.6 ft**

**Substituting into the Harris Formula yields the answer.**

$$p_f = \frac{0.1025 \times 442.6 \times \left(\frac{105.9}{60}\right)^2}{10.9 \times 0.918} = 14.2 \ psi = \underline{97.6 \ kPa}$$

**14-18.** $CR = \frac{140 + 14.7}{14.7} = 10.5$

$d^{5.31} = \frac{0.0125\left(200\right)^2}{3600 \times 10.5 \times 0.10} = 1.085$ **so d = 1.015 in**

**Select:** **<u>1-in schedule 40 pipe</u>**

**14-19.** $CR = \frac{800 + 101}{101} = 8.92$

$$2000 \ Pa = 2000 \ Pa \times \frac{14.7 \ psi}{101{,}000 \ Pa} = 0.291 \ psi$$

$$\frac{p_f}{L} = \frac{2000 \ Pa}{1 \ m} = \frac{0.291 \ psi}{3.28 \ ft} = 0.0887 \ psi/ft$$

$$4 \ std \ m^3/\text{min} = 4 \times \left(3.28\right)^3 \ scfm = 141 \ scfm$$

$$d^{5.31} = \frac{0.1025\,(141)^2}{3600 \times 8.92 \times 0.0887} = 0.715$$

**d = 0.934 in = 23.7 mm**

**14-20.** $Actual\ HP\ to\ drive\ compressor = \frac{p_{in}Q}{65.4\eta_o}\left[\left(\frac{p_{out}}{p_{in}}\right)^{0.286} - 1\right]$

$$= \frac{14.7 \times 200}{65.4 \times 0.70}\left[\left(\frac{114.7}{14.7}\right)^{0.286} - 1\right] = 51.4\ \text{HP} = 38.3\ \text{kW}$$

**Electric power required to drive electric motor**

$$= \frac{38.3\ \text{kW}}{0.90} = 42.6\ \text{kW}$$

$$\text{Yearly cost} = 42.6\ \text{kW} \times 4000\ \frac{\text{hr}}{\text{yr}} \times \frac{\$0.10}{\text{kW hr}} = \$17{,}040/\text{yr}$$

**14-21. Actual HP to drive compressor**

$$= \frac{14.7 \times 250}{65.4 \times 0.70}\left[\left(\frac{126.7}{14.7}\right)^{0.286} - 1\right] = 80.3\ \text{HP} = 59.9\ \text{kW}$$

**Electric power required to drive electric motor**

$$= \frac{59.9\ \text{kW}}{0.90} = 66.6\ \text{kW}$$

$$\text{Yearly cost} = 66.6\ \text{kW} \times 4000\ \frac{\text{hr}}{\text{yr}} \times \frac{\$0.10}{\text{kW hr}} = \$26{,}640/\text{yr}$$

$$\text{Additional yearly cost} = \$26{,}640 - \$17{,}040 = \$9{,}600/\text{yr}$$

14-22. Actual kW to drive compressor $= \frac{p_{in}Q}{17.1\eta_o}\left[\left(\frac{p_{out}}{p_{in}}\right)^{0.286} - 1\right]$

$$= \frac{101 \times 6}{17.1 \times 0.70}\left[\left(\frac{791}{101}\right)^{0.286} - 1\right] = 40.6 \text{ kW}$$

Electric power required to drive electric motor

= 40.6 kW/0.90 = 45.1 kW

$$\text{Yearly cost} = 45.1 \text{ kW} \times 4000 \frac{\text{hr}}{\text{yr}} \times \frac{\$0.10}{\text{kW hr}} = \underline{\$18{,}040/\text{yr}}$$

14-23. Actual kW to drive compressor $= \frac{101 \times 7.5}{17.1 \times 0.70}\left[\left(\frac{891}{101}\right)^{0.286} - 1\right]$

= 54.7 kW

Electric power required to drive electric motor

= 54.7 kW/0.9 = 60.8 kW

$$\text{Yearly cost} = 60.8 \text{ kW} \times 4000 \frac{\text{hr}}{\text{yr}} \times \frac{\$0.10}{\text{kW hr}} = \$24{,}320/\text{yr}$$

Additional yearly cost = $24,320 - $18,040 = <u>$6,280/yr</u>

14-24. Cylinder extends and retracts continuously.

14-25. (a) Nothing if fully extended. Extends and stops if fully retracted.
(b) Cylinder extends and retracts continuously when V4 and V5 are both depressed.

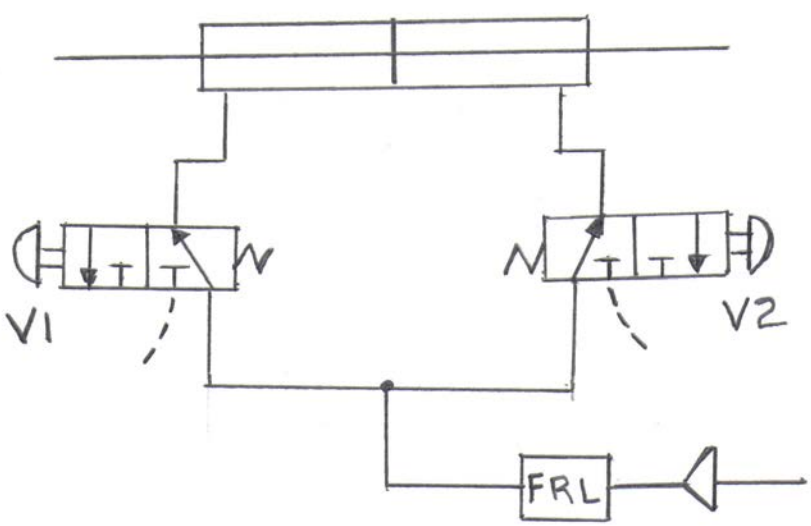
**14-26.** **(a) Cylinder 1 extends and then cylinder 2 extends.**
**(b) Cylinder 2 retracts and then cylinder 1 retracts.**

**14-27.** **Insert a pilot check valve in the line connected to the blank end of cylinder 1. The pilot line of the check valve should be connected to the line connected to the rod end of cylinder 1.**

**The direction of the pilot check valve should be such that free flow is always allowed into the blank end of the cylinder through the pilot check valve even though no pilot pressure exists. Reverse flow from the blank end of the cylinder requires pilot pressure to be exerted on the pilot check valve.**

**14-28.**
1. **Actuate V1 only: cylinder moves to the right.**
2. **Actuate V2 only: cylinder moves to the left.**
3. **Actuate both V1 and V2: cylinder is pneumatically locked since both ends are exposed to system air pressure.**
4. **Unactuate both V1 and V2: cylinder is free to move since both ends are vented to the atmosphere.**

**14-29.** **System redesign is shown below:**



**14-30.** **Cylinder 2 extends through full stroke while cylinder 1 does not move. Then cylinder 1 extends through full stroke.**

**By adding a properly adjusted flow control valve in each line leading to the blank end of each cylinder, the cylinders will extend and retract together at the same speeds.**

**14-31.** $(a) \quad p_{suction}(abs) = p_{suction}(gage) + p_{atm} = -8 + 14.7 = 6.7 \ psia$

**Using Equation 14-4 we have:**

$$F = p_{atm}A_o - p_{suction}A_i$$

$$= 14.7 \times \frac{\pi}{4} \times 7^2 - 6.7 \times \frac{\pi}{4} \times 6^2 = 566 - 189 = 377 \text{ lb}$$

**Using a factor of safety of 3, we have:**

**W = F/3 = 377/3 = <u>126 lb</u>**

$(b) \quad F = p_{atm}A_o = 566 \ lb \quad so \quad W = 566/3 = \underline{189 \ lb}$

**14-32.** $t = \frac{V}{Q} \ln\left(\frac{p_{atm}}{p_{vacuum}}\right) = \frac{5}{3} \ln\left(\frac{14.7}{5}\right) = \underline{1.80 \ \text{min}}$

**14-33.** $(a) \quad p_{suction}(abs) = p_{suction}(gage) + p_{atm}$

**= - 50 kPa + 101 kPa = 51 kPa abs**

**Using Equation 14-4, we have:**

$$F = p_{atm}A_o - p_{suction}A_i$$

$$= 101{,}000 \times \frac{\pi}{4} \times 0.100^2 - 51{,}000 \times \frac{\pi}{4} \times 0.080^2$$

= 793 - 256 = 537 N

Using a factor of safety of 3, we have:

W = F/3 = 537/3 = 179 N

$$(b) \quad F = p_{atm}A_o = 793 \ lb \quad so \quad W = 793/3 = \underline{264 \ N}$$

**14-34.** $F = W = p_{atm}A_o - p_{suction}A_i$

Since the factor of safety equals 2, we have:

$W_{total}$ = 2 x 1500 = 3000 N = 6W

Substituting values into the above equation yields :

$$\frac{3000}{6} = 101{,}000 \times \frac{\pi}{4} \times 0.1^2 - p_{suction} \times \frac{\pi}{4} \times 0.08^2$$

$$500 = 793 - 0.00503\, p_{suction} \quad or \quad P_{suction} = 58{,}300 \ Pa \ abs$$

$$Also \quad t = \frac{V}{Q} \ln\left(\frac{p_{atm}}{p_{suction}}\right)$$

$$\text{Thus} \quad 2 = \frac{0.20}{Q} \ln\left(\frac{101{,}000}{58{,}300}\right) \quad \text{so} \quad Q = \underline{0.0550 \text{ std } m^3/min}$$

**14-35.** Use Equation 13-3 where $V_1$ = required accumulator size.

$p_1V_1 = p_2V_2 = p_3V_3$ Also we have:

$V_{hydr\ cyl} = V_3 - V_2 = 450 \ in^3$ per statement of problem.

$$Thus \quad V_3 = \frac{p_2 V_2}{p_3} = \frac{3000 V_2}{1800} = 1.67 V_2$$

**Solving the preceding equations yields:**

**$1.67\ V_2 - V_2 = 450\ in^3$ or $V_2 = 672\ in^3$, Thus $V_3 = 1122\ in^3$**

**Therefore, we have a solution as follows:**

$$V_1 = \frac{p_2 V_2}{p_1} = \frac{3000 \times 672}{1200} = 1680\ in^3 = \underline{7.27\ gal\ \ accumulator}$$

**14-36.** $F_{load} = p_3 A_{hydr\ cyl} = 1800 \times \frac{\pi}{4} \times 6^2 = 50{,}894\ lb$

**$V_{hydr\ cyl} = \frac{\pi}{4} \times 6^2 \times stroke = 450\ in^3$, Thus stroke = $\underline{15.9\ \ in}$.**

**14-37.** $p_1 V_1 = p_2 V_2 = p_3 V_3 \quad where \quad V_{hydr\ cyl} = V_3 - V_2 = 7370\ cm^3$

$$V_3 = \frac{p_2 V_2}{p_3} = \frac{310 V_2}{126} = 1.67 V_2 \quad Thus\ we\ have:$$

**$1.67\ V_2 - V_2 = 7370$ or $V_2 = 11{,}000\ cm^3$ and $V_3 = 18{,}370\ cm^3$**

$$So \quad V_1 = \frac{p_2 V_2}{p_1} = \frac{210 \times 11{,}000}{84} = 27{,}500\ cm^3 = 0.0275\ m^3 = 27.5\ L$$

**14-38.** $F_{load} = p_3 A_{hydr\ cyl} = \left(126 \times 10^5\right) \times \frac{\pi}{4} \times 0.152^2 = 229{,}000\ N$

$$V_{hydr\ cyl} = \frac{\pi}{4} \times 0.152^2 \times \text{stroke} = 0.00737\ m^3$$

**Therefore the stroke = 0.406 m = 406 mm**

**14-39.** $\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$ **Thus we have:**

$$p_2 = \frac{T_2}{T_1} \times \frac{V_1}{V_2} \times p_1 = \frac{180+273}{40+273} \times \frac{0.04}{0.03} \times 10\ MPa = \underline{19.3\ MPa}$$

**14-40.** $\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$ **Thus we have:**

$$p_1 = \frac{T_1}{T_2} \times \frac{V_2}{V_1} \times p_2 = \frac{200+460}{100+460} \times \frac{275}{180} \times 1000\ psi = \underline{1800\ psi}$$

# Chapter 15

# Basic Electrical Controls for Fluid Power Circuits

15-1. One of the reasons for this trend is that more machines are being designed for automatic operation to be controlled with electrical signals from computers.

15-2. Pressure switches open or close their contacts based on system pressure. A temperature switch opens or closes an electrical switch when a predetermined temperature is reached.

15-3. Limit switches open and close circuits when they are actuated at the end of the retraction or extension strokes of hydraulic or pneumatic cylinders. Push button switches are actuated manually.

15-4. A relay is an electrically actuated switch. As shown in Figure 15-8(a), when switch 1-SW is closed, the coil is energized. This pulls on the spring-loaded relay arm to open the upper set of normally closed contacts and close the lower set of normally open contacts. Figure 15-8(b) shows the symbol for the relay coil and the symbols for the normally open and closed contacts.

15-5. Timers are used in electrical control circuits when a time delay is required from the instant of actuation to the closing of contacts.

15-6. Electrical switches possess virtually no resistance.

15-7. An indicator lamp is often used to indicate the state of a specific circuit component. For example, indicator lamps are used to determine which solenoid operator of a directional control valve is energized.

15-8. A normally open switch is one in which no electric current can flow through the switching element until the switching element is actuated. In a normally closed switch, electric current can flow through the switching element until the switch is actuated.

15-9. The cylinder extends, retracts and stops.

15-10. Cylinder 1 extends.
Cylinder 2 extends.
Both cylinders remain extended until 1-SW is opened.
Then both cylinders retract together and stop.

15-11. (a) Cylinder 1 extends.
Cylinder 2 extends.
(b) Cylinders 1 and 2 retract together.

15-12. When push button switch 1-PB is actuated, coil 1-CR is energized. This closes normally open contacts 1-CR which energizes SOL A and holds. Thus, the cylinder extends until limit switch 1-LS is actuated. This opens the contacts of 1-LS which de-energizes coil 1-CR. As a result, the contacts for 1-CR are returned back to their normally open mode and SOL A is de-energized. This shifts the DCV back into its spring offset mode to retract the cylinder.

If push button 2-PB is actuated while the cylinder is extending, the cylinder will immediately stop and then retract. This is because coil 1-CR is de-energized which returns the contacts for 1-CR back to their normally open mode. This de-energizes SOL A which shifts the DCV into its spring offset mode.

15-13. Initially cylinder 1 is fully retracted and cylinder 2 is fully extended. Cylinder 1 extends and stays fully extended. End of cycle.

15-14. Initially cylinder 1 is fully retracted and cylinder 2 is fully extended. Cylinder 1 fully extends while cylinder 2 fully retracts. End of cycle.

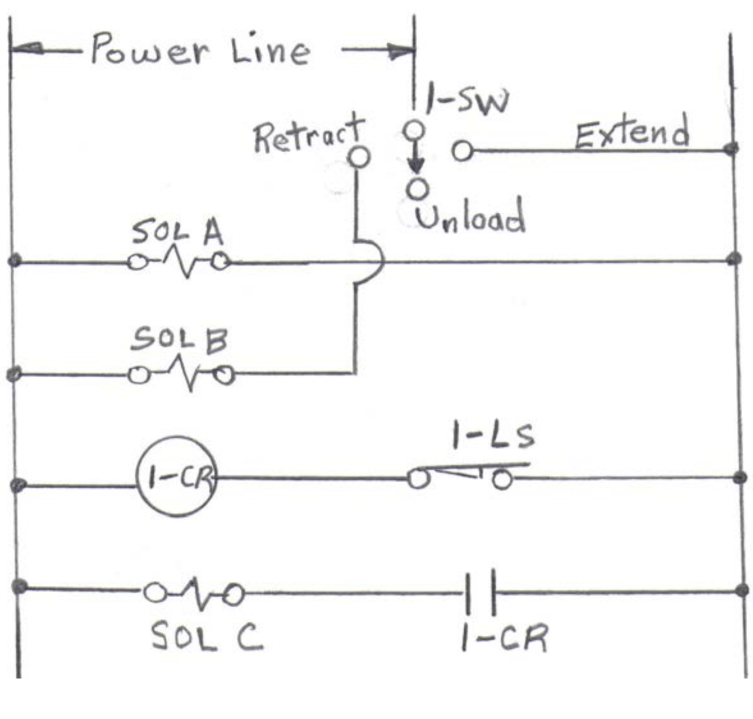
15-15. (a) Cylinder 1 extends.
Cylinder 2 extends.
Cylinders 1 and 2 start to retract together but as soon as limit switch 1-LS is actuated, SOL B and D are de-energized. Thus, both DCVs go into their spring-centered mode and both cylinders stop.

(b) If 2-PB is depressed momentarily while cylinder 1 is extending, cylinder 1 stops and nothing else happens. If 2-PB is depressed momentarily while cylinder 2 is extending, the system behaves the same as described in part (a) above. Thus, there is no effect on the operation of the system.

15-16. Design Change One: Add the series combination of 2-PS(NC) and 3-CR(NO) in parallel with the 1-LS(NO) that is currently in the ladder diagram.

Design Change Two: Add the series combination of 1-CR(NC) and 3-CR(NO) in parallel with the 1-LS(NO) that is currently in the ladder diagram.

15-17.



# Chapter 16

## Fluid Logic Control Systems

16-1. Moving part logic devices are miniature valve-type devices which by the action of internal moving parts, perform switching operations in fluid logic systems.

16-2. 1. Mechanical displacement.
2. Electric voltage.
3. Fluid pressure.

16-3. Fluidics is the technology that utilizes fluid flow phenomenon in components and circuits to perform a wide variety of control functions. These include sensing, logic, memory, timing and interfacing to other control media.

16-4. An AND function is one which requires that two or more control signals exist in order to obtain an output.

16-5. An OR function is one in which all control signals must be off in order for the output to not exist. Therefore, any one control signal will produce an output.

16-6. A flip-flop is a bistable digital control device. Thus, a flip-flop has two stable states when all control signals are OFF.

16-7. An OR gate is a device which will have an output if any one or any combination of control signals is ON. An exclusive OR gate is a device which will have an output only if one control signal (but not any combination of control signals) is ON.

16-8. A MEMORY function is one which has the ability to retain information as to where a signal to a control system originated.

16-9.
1. It provides a means by which a logic circuit can be reduced to its simplest form.
2. It allows for the quick synthesis of a circuit which is to perform desired logic operations.

16-10. Multiplication and addition are permitted.

16-11. Logic inversion is the process that makes the output signal not equal to the input signal in terms of ON versus OFF.

16-12. The commutative law states that the order in which variables appear in equations is irrelevant.

An example is: $A + B = B + A$

The associative law states that the order in which functions are performed is irrelevant, provided that the functions are unchanged. An example is as follows:

$$A + B + C = (A + B) + C = A + (B + C) = (A + C) + B$$

16-13. DeMorgan's Theorem allows for the inversion of functions as follows:

(I) $\overline{A + B + C} = \overline{A} \bullet \overline{B} \bullet \overline{C}$

The inversion of the function (A or B or C) equals the function (not A and not B and not C).

(II) $\overline{A \bullet B \bullet C} = \overline{A} + \overline{B} + \overline{C}$

The inversion of the function (A and B and C) equals the function (not A or not B or not C).

16-14. 2 variables produce $2^2 = 4$ possible combinations.

| A | B | A + B | A • (A + B) |
|---|---|---|---|
| 0 | 0 | 0 | 0 |
| 1 | 0 | 1 | 1 |
| 0 | 1 | 1 | 0 |
| 1 | 1 | 1 | 1 |

16-15. 3 variables produce $2^3 = 8$ possible combinations.

| A | B | C | A + B | B + C | (A + B) + C | A + (B + C) |
|---|---|---|---|---|---|---|
| 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 1 | 0 | 0 | 1 | 0 | 1 | 1 |
| 1 | 1 | 0 | 1 | 1 | 1 | 1 |
| 1 | 0 | 1 | 1 | 1 | 1 | 1 |
| 0 | 1 | 0 | 1 | 1 | 1 | 1 |
| 0 | 0 | 1 | 0 | 1 | 1 | 1 |
| 0 | 1 | 1 | 1 | 1 | 1 | 1 |
| 1 | 1 | 1 | 1 | 1 | 1 | 1 |

16-16.

| A | B | A • B | $\overline{A \bullet B}$ | $\overline{A}$ | $\overline{B}$ | $\overline{A} + \overline{B}$ |
|---|---|---|---|---|---|---|
| 0 | 0 | 0 | 1 | 1 | 1 | 1 |
| 1 | 0 | 0 | 1 | 0 | 1 | 1 |
| 0 | 1 | 0 | 1 | 1 | 0 | 1 |
| 1 | 1 | 1 | 0 | 0 | 0 | 0 |

16-17.

| A | B | C | $B + C$ | $A \bullet (B + C)$ | $A \bullet B$ | $A \bullet C$ | $(A \bullet B) + (A \bullet C)$ |
|---|---|---|---|---|---|---|---|
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 1 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 1 | 1 | 0 | 1 | 1 | 1 | 0 | 1 |
| 1 | 0 | 1 | 1 | 1 | 0 | 1 | 1 |
| 0 | 1 | 0 | 1 | 0 | 0 | 0 | 0 |
| 0 | 0 | 1 | 1 | 0 | 0 | 0 | 0 |
| 0 | 1 | 1 | 1 | 0 | 0 | 0 | 0 |
| 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |

16-18.

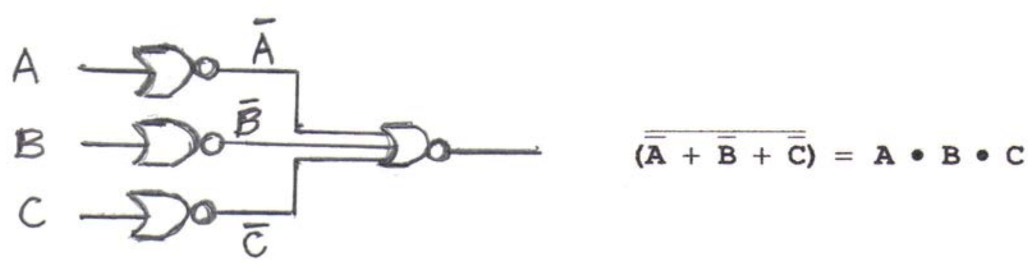
| A | B | $\overline{A}$ | $\overline{A} + B$ | $A \bullet (\overline{A} + B)$ | $A \bullet B$ |
|---|---|---|---|---|---|
| 0 | 0 | 1 | 1 | 0 | 0 |
| 0 | 1 | 1 | 1 | 0 | 0 |
| 1 | 0 | 0 | 0 | 0 | 0 |
| 1 | 1 | 0 | 1 | 1 | 1 |

16-19. From DeMorgan's Theorem we have:

$$\overline{A \bullet B \bullet C} = \overline{A} + \overline{B} + \overline{C}$$

Hence $A \bullet B \bullet C = (\overline{\overline{A \bullet B \bullet C}}) = (\overline{\overline{A} + \overline{B} + \overline{C}}) = \text{NOT}(\overline{A} + \overline{B} + \overline{C})$

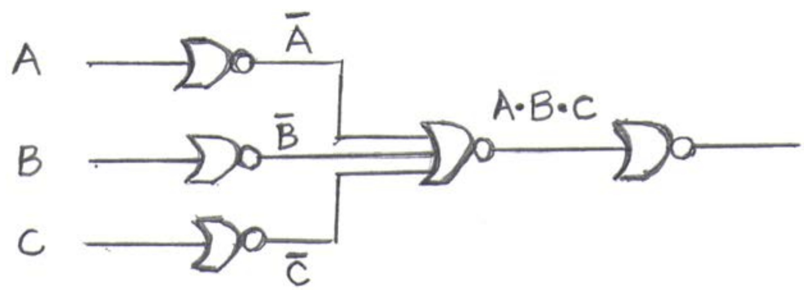
Therefore to generate the AND function, we invert individual inputs and connect the inverted inputs to a NOR gate. The NOR gates are used to invert the input signals as shown below.



16-20. From DeMorgan's Theorem we have:

$$Z = \overline{A} + \overline{B} + \overline{C} = (\overline{A \bullet B \bullet C})$$

This means we need to generate the NAND function of inputs A, B and C as shown below.




16-21. When the cylinder is fully retracted, the signals from A1 and A2 are both ON. The extension stroke begins when push button P is pressed since the output P • A1 of the AND gate produces an output Q from the Flip Flop. The push button can be released because the Flip Flop maintains its Q output even though P • A1 is OFF.

When the cylinder is fully extended, A2 is OFF, causing $\overline{A2}$ to go ON switching the Flip Flop to output $\overline{Q}$. This removes the signal to the DCV which retracts the cylinder. The push button must be again pressed to produce another cycle. If the push button is held depressed, the cycle repeats continuously.

16-22. When the guard is open (M is OFF and $\overline{M}$ is ON), the output of the Flip Flop is Q. Closing the guard turns M ON and produces an output to the DCV from the AND gate. This causes the cylinder to extend. When the cylinder is fully extended, signal A goes OFF, signal $\overline{A}$ goes ON, and the Flip Flop output shifts to $\overline{Q}$. This shifts the DCV which retracts the cylinder.

At the end of the cylinder retraction stroke, Flip Flop inputs S and R are both OFF, and Q remains OFF. Thus the cylinder remains fully retracted. When the guard is opened again (M is OFF), the Flip Flop switches to the Q output to prepare the system for the next cycle.

16-23. $P = A \bullet (A + B) = A \bullet A + A \bullet B$

$P = A + A \bullet B$ (using Theorem 6)

Thus output P is ON when A is ON, or A and B are ON. Therefore control Signal B (applied to valve 3) is not needed.

# Chapter 17

# Advanced Electrical Controls For Fluid Power systems

**17-1.** 1. **Higher pressures increase internal leakage inside pumps, actuators and valves.**

2. **Temperature changes affect fluid viscosity and thus, leakage.**

**17-2.** 1. **Velocity Transducer: senses the linear or angular velocity of the system output and generates a signal proportional to the measured velocity.**

2. **Positional Transducer: senses the linear or angular position of the system output and generates a signal proportional to the measured position.**

**17-3.** **A feedback transducer is a device which performs the function of converting one source of energy into another such as mechanical to electrical.**

**17-4.** **A servo valve replaces the flow control valve and directional control valve of an open-loop system.**

**17-5.** The transfer function of a component or a total system is defined as the output divided by the input.

**17-6.** Deadband is that region or band of no response where an input signal will not cause an output. Hysteresis is the difference between the response of a component to an increasing signal and the response to a decreasing signal.

**17-7.** Open loop gain is the gain (output divided by input) from the error signal to the feedback signal.

**17-8.** Closed loop transfer function is the system output divided by the system input.

**17-9.** Repeatable error is the discrepancy between the actual output position and the programmed output position.

**17-10.** Tracking error is the distance by which the output lags the input command signal while the load is moving.

**17-11.** The forward path contains the amplifier, servo valve and cylinder. The feedback path contains the transducer.

**17-12.** A programmable logic controller (PLC) is a user-friendly electronic computer designed to perform logic functions such as AND, OR and NOT for controlling the operation of industrial equipment and processes.

**17-13.** Unlike general purpose computers, a PLC is designed to operate in industrial environments where high ambient temperature and humidity levels may exist.

**17-14.** A PLC consists of solid–state digital logic elements (rather than electromechanical relays) for making logic decisions and providing corresponding outputs.

**17-15.**
1. Electromechanical relays have to be hard-wired to perform specific functions.
2. PLCs are more reliable and faster in operation.
3. PLCs are smaller in size and can be more readily expanded.

**17-16.**
(a) CPU: receives input data from various sensing devices such as switches, executes the stored program, and delivers corresponding output signals to various load control devices such as relay coils and solenoids.

(b) Programmer/Monitor: allows the user to enter the desired program into the RAM memory of the CPU as well as edit, monitor, and run the program.

(c) I/O Module: transforms the various signals received from or sent to the fluid power interface devices such as push button switches, pressure switches, limit switches, motor relay coils, solenoid coils and indicator lights.

**17-17.** ROM memory cannot be changed during operation or lost when electrical power to the CPU is turned off. RAM memory which is lost when electrical power is removed, can be programmed and altered by the user.

17-18. $\omega_H = A\sqrt{\frac{2\beta}{V\,M}} = 4\sqrt{\frac{2 \times 200{,}000}{40 \times 750/386}} = 287\ \text{rad}/\text{s}$

Open loop gain $= \omega_H/3 = 95.7/\text{s}$

$$G_{SV} = \frac{\text{open loop gain}}{G_A \times G_{CYL} \times H} = \frac{95.7}{G_A \times 0.15 \times 3.5} = \frac{182}{G_A}$$

$$RE = \frac{\text{system deadband}}{G_A \times H} = \frac{3.5}{G_A \times 3.5} = 0.002$$

Hence $G_A = 500\ \text{ma}/\text{V}$ and $G_{SV} = \underline{0.364\ \left(\text{in}^3/\text{s}\right)/\text{ma}}$

17-19. Closed loop transfer function $= \frac{G}{1 + G\,H}$

$G = G_A G_{SV} G_{CYL} = 55 \times 0.364 \times 0.15 = 27.3\ \text{in}/\text{V}$

Closed loop transfer function $= \frac{27.3}{1 + 27.3 \times 3.5} = \underline{0.283\ \text{in}/\text{V}}$

17-20. $\omega_H = A\sqrt{\dfrac{2\beta}{V\,M}} = 25 \times 10^{-4}\sqrt{\dfrac{2 \times (1400 \times 10^{6})}{(750 \times 10^{-6}) \times 300}} = 279\ \text{rad}/\text{s}$

Open loop gain $= \omega_H/3 = 93/\text{s}$

$$G_{SV} = \frac{\text{open loop gain}}{G_A \times G_{CYL} \times H} = \frac{93}{G_A \times 0.04 \times 1.75} = \frac{1330}{G_A}$$

$$RE = \frac{\text{system deadband}}{G_A \times H} = \frac{3.5}{G_A \times 1.75} = 0.004$$

Hence $G_A = 500\ \text{ma}/\text{V}$ and $G_{SV} = \underline{2.66\ (\text{cm}^3/\text{s})\ /\ \text{ma}}$

17-21. Closed loop transfer function $= \dfrac{G}{1 + G\,H}$

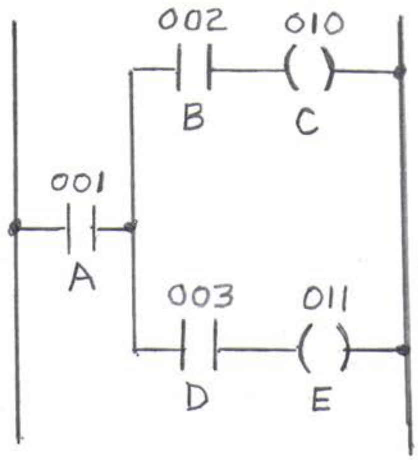
$$G = G_A G_{SV} G_{CYL} = 500 \times 2.66 \times 0.04 = 53.2\ \text{cm}/\text{V}$$

$$\text{Closed loop transfer function} = \frac{53.2}{1 + 53.2 \times 1.75} = \underline{0.565\ \text{cm}/\text{V}}$$

17-22. $TE = \dfrac{\text{S. V. max. current (ma)}}{G_A(\text{ma}/\text{V}) \times H(\text{V}/\text{in})} = \dfrac{250}{55 \times 3.5} = \underline{0.143\ \text{inches}}$

17-23. $TE = \dfrac{\text{S. V. max. current (ma)}}{G_A(\text{ma}/\text{V}) \times H(\text{V}/\text{cm})} = \dfrac{250}{500 \times 1.75} = 0.286\ \text{cm}$

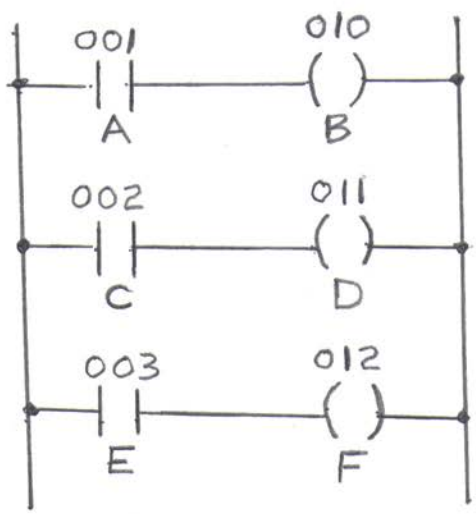
17-24.




C = A • B B is energized when A and B are actuated.

E = A • D E is energized when A and D are actuated.

17-25.

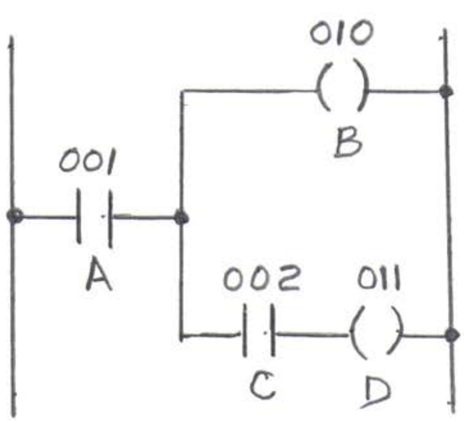



A = B B is energized when A is actuated.

C = D D is energized when C is actuated.

E = F F is energized when E is actuated.

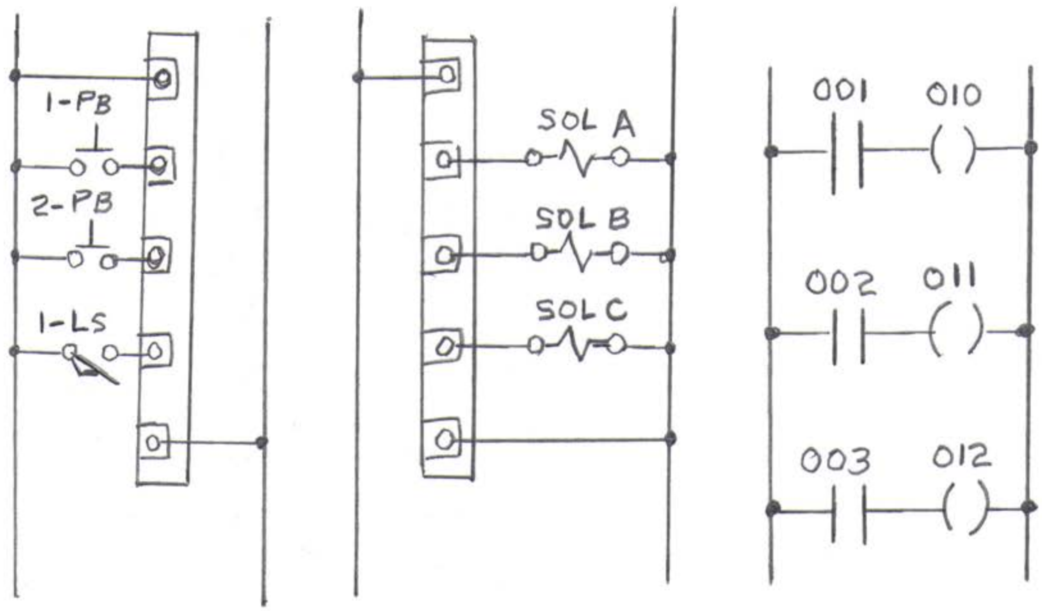
17-26.




B = A B is energized when A is actuated.

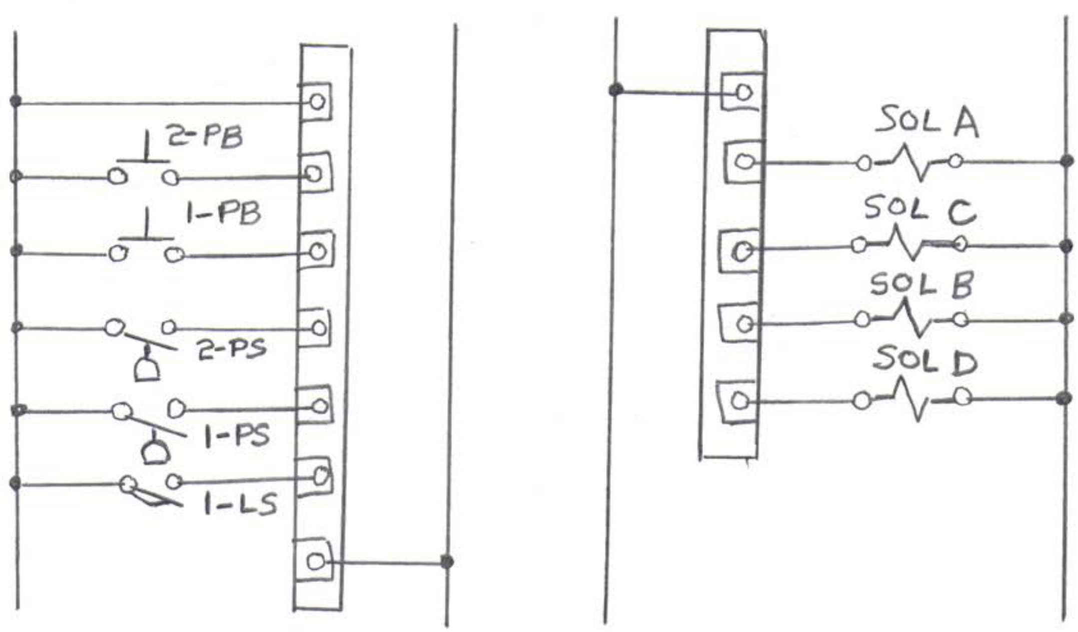
D = A • C D is energized when A and C are actuated.

17-27.

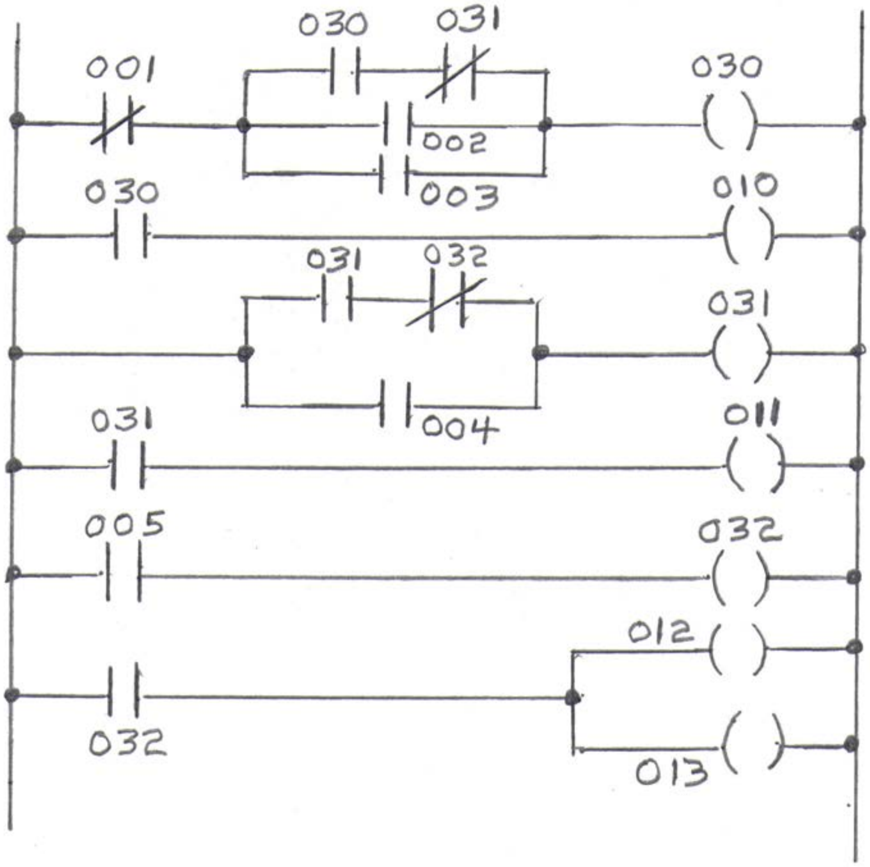



Input Connection Diagram | Output Connection Diagram | PLC Logic Ladder Diagram

17-28. Electrical relays are not included in the I/O connection diagram since their functions are replaced by internal control relays.

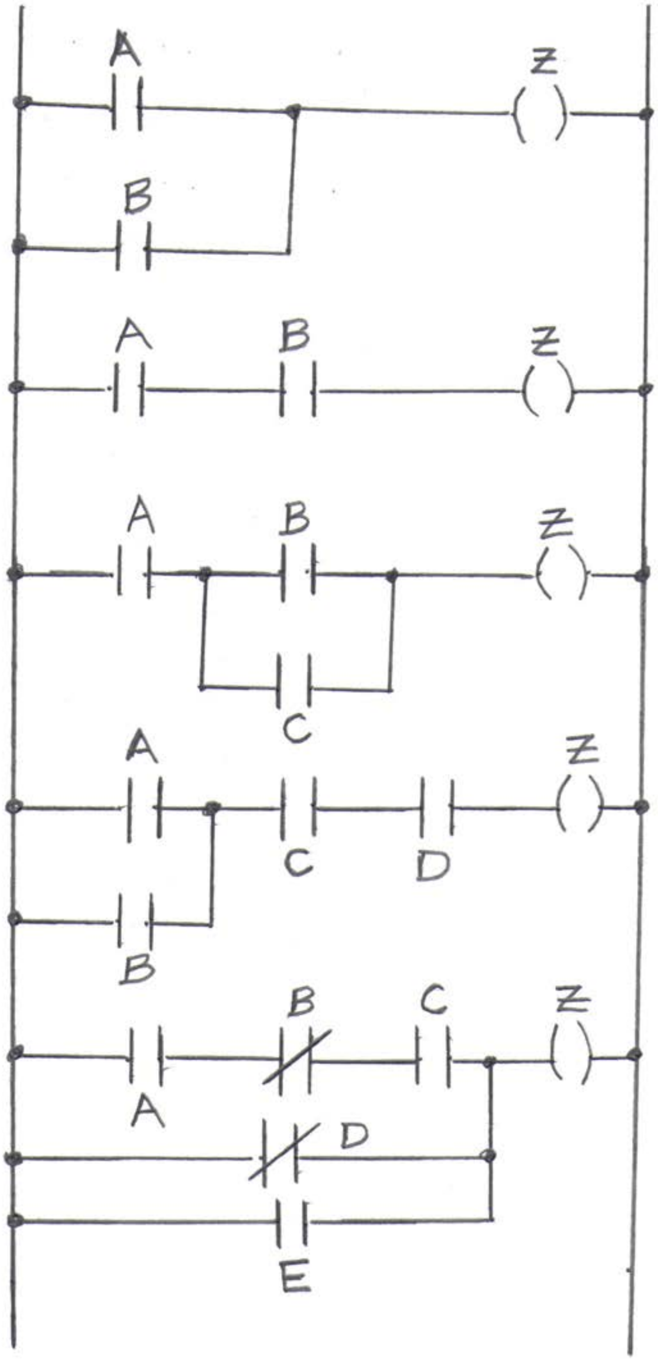



Input Connection Diagram | Output Connection Diagram




PLC Logic Ladder Diagram

17-29.




$Z = A + B$

$Z = A \bullet B$

$Z = A \bullet (B + C)$

$Z = (A + B) \bullet C \bullet D$

$Z = A \bullet \bar{B} \bullet C + \bar{D} + E$