LAMPIRAN

Lampiran 1. Rencana Pembelajaran Semester (*Output* Kegiatan 1)

MATA KULIAH		KODE	DOSEN PENGAMPU	BOBOT (SKS)	SEMESTER	TANGGAL PENYUSUNAN
Sistem Hidraulik dan Pneumatik		TM201505	Gad Gunawan	3	7	
OTORISASI		KOORDINATOR MK		KOORDINATOR PROGRAM STUDI		
		Gad Gunawan		Andi Idhil Ismail, S.T., M.Sc., Ph.D		
CAPAIAN PEMBELAJARAN (CP)	CAPAIAN PEMBELAJARAN LULUSAN (CPL) YANG DITITIPKAN PADA MATA KULIAH					
	S9. Menunjukkan sikap bertanggung jawab atas pekerjaan di bidang keahliannya secara mandiri KU2. Mampu menunjukkan kinerja mandiri, bermutu, dan terukur P3. Mampu menguasai prinsip rekayasa untuk menemukan sumber masalah rekayasa kompleks pada sistem mekanik melalui proses penyelidikan, analisis, interpretasi data, dan informasi berdasarkan prinsip – prinsip rekayasa KK3. Mampu mengembangkan prinsip rekayasa untuk menemukan sumber masalah rekayasa kompleks pada sistem mekanik melalui proses penyelidikan, analisis, interpretasi data, dan informasi berdasarkan prinsip –prinsip rekayasa					
	CAPAIAN PEMBELAJARAN MATA KULIAH (CPMK)					
	Mampu menerapkan prinsip dasar hidraulik dan pneumatik dalam pemanfaatannya di industri					
DESKRIPSI SINGKAT MK	Memberikan dasar-dasar tentang pemanfaatan tenaga hydraulics & pneumatics. Prinsip pemindahan tenaga yang berkaitan dangan karakteristik fluida yang digunakan. Karakteristik komponen, operasi dan fungsinya. Pemahaman sirkuit hydraulics/pneumatics dan kontrol diskrit. Pemilihan komponen peralatan dari sirkuit yang ada. Pemanfaatan sistem hydraulics & pneumatics dalam industri, baik kekurangan maupun kelebihan dibanding sistem lainnya.					
BAHAN KAJIAN	1. Komponen hidraulik dan pneumatik 2. Sistem hidraulik 3. Sistem pneumatik					

PUSTAKA	UTAMA
	Esposito, A., (2014). Fluid Power with Applications, New York : Prentice Hall
	PENDUKUNG
	1. Rabie, M. Galal, (2009). Fluid Power Engineering, New York : Mc Graw Hill 2. Drs. Wirawan, M.T . Bahan Ajar Pneumatik – Hidraulik. Universitas Negeri Semarang 3. Barber, A. (1997). Pneumatic Handbook 8th Edition. Elsevier Science & Technology Books
MEDIA PEMBELAJARAN	1. Text Book 2. Google meeting, Moodlle 3. Video
MATA KULIAH PRASYARAT	Mekanika Fluida II

RENCANA PEMBELAJARAN SEMESTER

Minggu ke-	Sub-CPMK (Kemampuan akhir yg direncanakan)	Bahan Kajian	Bentuk/ Metode Pembelajaran	Aktivitas Belajar	Penilaian			Durasi (menit)	Pustaka
					Kriteria	Indikator	Bobot		
(1)	(2)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
1	**Mahasiswa mengetahui komponen-komponen Sistem Hidraulik**	Komponen-komponen sistem hidraulik: • Pengenalan • Karakteristik Fluida Hidraulik • Pompa • Aktuator • Control Valve • Conduktor and fitting • Ancilary hidraulic devices	Kuliah / Ceramah, *Problem Based Learning*	Tanya/ Jawab; Tugas Kelompok	Ketepatan dalam menjawab	• Mengidentifikasi sistem hidraulik • Mengetahui karakteristik fluida • Mengetahui komponen sistem hidraulik	-	**150**	1. Esposito(2000); 2. Rabie (2009); 3. Wirawan ; Sistem Hidraulik
2								150	
3								150	
4								150	

No	Kemampuan	Bahan Kajian	Metode		Penilaian		Indikator		Waktu	Referensi
5	**Mahasiswa mampu mendesain dan menganalisis sistem hidraulik**	Sistem hidraulik: Desain dan analisis Sistem Hidraulik	Tutorial/ Pembelajaran berbasis Tugas Besar	Tugas Besar	Ketepatan dalam menjawab		Mendesain dan menganalisis sistem hidraulik		**150**	1. Esposito(2000); 2. **Wirawan ; Sistem Hidraulik**
6									**150**	
7	**Mahasiswa mampu mengetahui perawatan sistem hidraulik**	Sistem hidraulik: • Perawatan Sistem Hidraulik	Kuliah / Ceramah, Diskusi Online	**Tanya Jawab**	Ketepatan dalam menjawab		Mengetahui perawatan sistem hidraulik	-	**150**	1. Esposito(2000);
8	**UJIAN TENGAH SEMESTER (UTS)**									
9	**Mahasiswa mengetahui komponen-komponen Sistem Pneumatik**	Komponen-komponen sistem pneumatik: • Pengenalan • Karakteristik Fluida Pneumatik • Kompresor • Aktuator • Control Valve • Konduktor and fitting • Fluid Conditioners	Kuliah / Ceramah, Problem Based Learning	Tanya/ Jawab; Tugas Kelompok	Ketepatan dalam menjawab		• Mengidentifikasi sistem pneumatik • Mengetahui karakteristik fluida • Mengetahui komponen sistem Pneumatik	-	**150**	1. Esposito(2000);**Ba b 13** 2. Rabie (2009) Bab 11 3. Wirawan , Sistem Pneumat ik
10									150	
11									**150**	

12								150	
13	**Mahasiswa mampu mendesain dan menganalisis sistem Pneumatik**	Sistem pneumatik: Desain dan analisis Sistem Pneumatik	Tutorial/ Pembelajaran berbasis Tugas Besar	Tugas Besar	Ketepatan dalam menjawab	Mendesain dan menganalisis sistem Pneumatik		150	1. Esposito(2000);**Ba b 14**
14								150	2. Rabie (2009) Bab 11 3. Wirawan , Sistem Pneumat ik
15	**Mahasiswa mampu mengetahui perawatan sistem pneumatik**	Sistem Pneumatik: • Perawatan Sistem Pneumatik	Kuliah / Ceramah, Diskusi Online	**Tanya Jawab**	Ketepatan dalam menjawab	Mengetahui perawatan sistem pneumatik		150	1. Barber (1997) Bagian 4 2. Wirawan , Sistem Pneumat ik
16	**UJIAN AKHIR SEMESTER (UAS)**								

KOMPOSISI NILAI EVALUASI

1. Tugas Kelompok   30%
2. Tugas Besar   40%
3. UTS   15%
4. UAS   15%

Lampiran 2. Bahan ajar/Presentasi mata kuliah Sistem Hidraulik dan Pneumatik (*Output* Kegiatan 2)

SISTEM HIDROLIK DAN PNEUMATIK

- **KETERLAMBATAN** kehadiran dalam kelas **LEBIH DARI 15 MENIT** setelah jam masuk kelas akan diberikan sanksi **TIDAK DIIZINKAN MENGIKUTI PERKULIAHAN** kepada mahasiswa yang bersangkutan.
- **KETERLAMBATAN** kehadiran dosen lebih dari 10 menit setelah jam masuk kelas maka kelas pada hari itu ditiadakan namun mahasiswa dianggap hadir.
- **KECURANGAN** yang meliputi kegiatan plagiat, curang, dan/atau menyontek dalam setiap **EVALUASI** (**UJIAN TULIS**) akan diberikan sanksi **NILAI 0 ATAU E** kepada mahasiswa yang bersangkutan.
- **KETIDAKHADIRAN** pada waktu tugas kelompok (presentasi) akan diberikan sanksi nilai 0 kepada mahasiswa yang bersangkutan.
- **KETERLAMBATAN** pengumpulan tugas individu dan tugas kelompok akan diberikan sanksi **PENGURANGAN NILAI EVALUASI** sebesar **5 POIN PER HARI** (maks 20 poin) kepada mahasiswa atau kelompok tugas mahasiswa yang bersangkutan.

Kontrak kuliah

- Jika ada laporan **KEKURANG-AKTIFAN / KETIDAK-AKTIFAN** satu atau lebih mahasiswa dalam satu kelompok oleh pimpinan kelompok (kepada dosen pengajar) maka akan diberikan sanksi pengurangan nilai tugas kelompok sebesar maksimal 50% kepada mahasiswa yang bersangkutan.
- Mahasiswa yang **TIDAK MEMENUHI SYARAT KEHADIRAN 80%** akan mendapat **NILAI E**.
- Mahasiswa yang melakukan **KECURANGAN DALAM PENGISIAN DAFTAR HADIR** akan diberikan sanksi **TIDAK LULUS**.
- Mahasiswa yang membantu mahasiswa lain untuk melakukan **KECURANGAN DALAM PENGISIAN DAFTAR HADIR** akan diberikan sanksi **PENGURANGAN 20% SELURUH NILAI EVALUASI**.
- Mahasiswa yang **TIDAK HADIR** pada waktu kuliah maupun presentasi tugas karena alasan yang jelas harus membawa surat keterangan dari instansi yang berwenang. Surat ijin harus diserahkan kepada Tata Usaha paling lambat 1 (satu) minggu sejak ketidakhadiran mahasiswa yang bersangkutan.

RENCANA PEMBELAJARAN

MINGGU KE	MATERI	METODE
1	Kontrak Perkuliahan dan Pengenalan Komponen-Komponen Sistem Hidrolik	Kuliah
2 – 3	Komponen-komponen Sistem Hidrolik	*Problem Based Learning*
4 – 6	Desain dan Analisis Sistem Hidrolik	*Tugas Besar*
7	Perawatan Sistem Hidrolik	Kuliah
8	Ujian Tengah Semester	Tes Tulis

RENCANA PEMBELAJARAN

MINGGU KE	MATERI	METODE
9 – 11	Komponen-komponen Sistem Pneumatik	Kuliah (9); *Problem Based Learning(10-11)*
12 – 14	Desain dan Analisis Sistem Pneumatik	Tugas Besar
15	Perawatan Sistem Pneumatik	Kuliah
16	Ujian Akhir Semester	Tes Tulis

1   **Tugas Kelompok : 30%**

2   **Tugas Besar   :   40%**

3   **Ujian Tengah Semester : 15%**

4   **Ujian Akhir Semester : 15%**

Mampu menerapkan prinsip dasar hidrolik dan pneumatik dalam pemanfaatannya di industri

Utama :

Esposito, A., (2000). Fluid Power with Applications, New York : Prentice Hall

Pendukung :

1. Rabie, M. Galal, (2009). Fluid Power Engineering, New York : Mc Graw Hill
2. Drs. Wirawan, M.T . Bahan Ajar Pneumatik – Hidrolik. Universitas Negeri Semarang
3. Watton, John, (1997). Modelling, Monitoring and Diagnostic Techniques for Fluid Power Systems London : Springer
4. Barber, A. (1997). Pneumatic Handbook 8th Edition. Elsevier Science & Technology Books

Minggu 1

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

- Fluid power is called *hydraulics* when the fluid is a liquid and is called *pneumatics* when the fluid is a gas.

Figure 1-1. Hydraulic chain saw. *(Courtesy of Greenlee Textron, Inc., Rockford, Illinois.)*

Figure 1-2. Pneumatically powered link chain hoist. *(Courtesy of Ingersoll-Rand Corp., Southern Pines, North Carolina.)*

1650 - Pascal's law - pressure

1750 - Bernoulli - law of conservation of energy for a fluid flowing in a pipeline
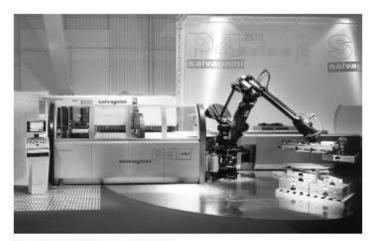
1870 - Industrial Revolution - cranes, presses, winches, extruding machines, hydraulic jacks, shearing machines, and riveting machines

1906 – hydraulic system was developed to replace electrical systems for elevating and controlling guns on the battleship *USS Virginia.*

1926 - United States developed the first unitized, packaged hydraulic system consisting of a pump, controls, and actuator.The military requirements leading up to World War II kept fluid power applications and developments going at a good pace.

Figure 1-3(a) is a photograph of a robotized panel bender system that bends metal sheets into parts called panels. The panels are produced by taking incoming flat sheets and bending them one or more times along one or more sides.The bending forces required for the operation of this machine (also called a press-brake) are provided by a hydraulic press cylinder with a 150-ton capacity. The piston of the hydraulic cylinder has a stroke of 14 in, a rapid traverse speed of 450 in per min, and a maximum bending speed of 47 in per min.The system illustrated is computer controlled and utilizes a robot whose movements are coordinated with the movements of the press-brake.This allows the robot to automatically feed the press-brake with the metal sheets to be formed into panels. The robot also unloads and stacks the processed panels so that the entire system can be operated unattended without interruption. The machine station where the bending operations occur is located just to the right of the computer console shown in Figure 1-3(a).

(a) Robotized panel bender system.

(a) B-2 in flight.

(b) B-2 hydraulic flight control servoactuator.

Figure 1-4. The B-2 Stealth Bomber. *(Courtesy of Moog, Inc., East Aurora, New York.)*

ADVANTAGES OF FLUID POWER

1. Ease and accuracy of control. By the use of simple levers and push buttons, the operator of a fluid power system can readily start, stop, speed up or slow down, and position forces that provide any desired horsepower with tolerances as precise as one ten-thousandth of an inch.

2. Multiplication of force. A fluid power system (without using cumbersome gears, pulleys, and levers) can multiply forces simply and efficiently from a fraction of an ounce to several hundred tons of output.

3. Constant force or torque. Only fluid power systems are capable of providing constant force or torque regardless of speed changes.This is accomplished whether the work output moves a few inches per hour, several hundred inches per minute, a few revolutions per hour, or thousands of revolutions per minute.

4. Simplicity, safety, economy. In general,
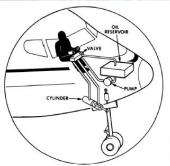
Figure 1-5. Hydraulic operation of aircraft landing gear. (*Courtesy of National Fluid Power Association, Milwaukee, Wisconsin.*)
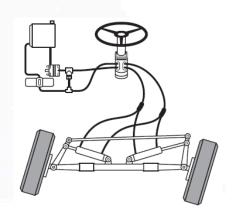
Ease and accuracy of control

Constant force or torque

Multiplication of force

Simplicity, safety, economy

Application

Fluid power drives high-wire overhead tram

Fluid power is the muscle in industrial lift trucks

Fluid power drives excavators

Fluid power is applied to harvesting soybeans
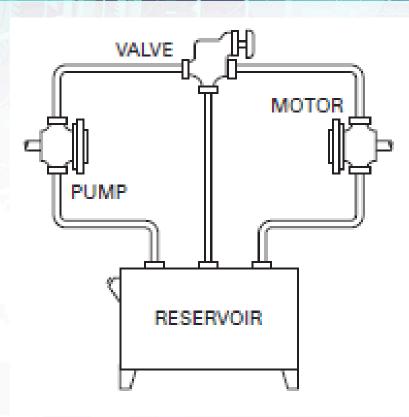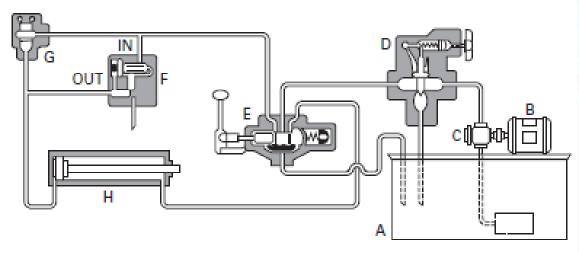
Figure 1-16. Basic hydraulic system with rotary hydraulic actuator (motor). *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

List of Components

A—Reservoir
B—Electric Motor
C—Pump
D—Maximun Pressure
(Relief) Valve

E—Directional Valve
F—Flow Control Valve
G—Right-Angle
Check Valve
H—Cylinder

Figure 1-15. Basic hydraulic system with linear hydraulic actuator (cylinder). (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Technical personnel who work in the fluid power field can generally be placed into three categories:

Fluid power mechanics. Workers in this category are responsible for repair and maintenance of fluid power equipment.They generally are high school graduates who have undertaken an apprenticeship training program. Such a program usually consists of three or four years of paid, on-the-job training plus corresponding classroom instruction.

Fluid power technicians. These people usually assist engineers in areas such as design, troubleshooting, testing, maintenance, and installation of fluid power systems. They generally are graduates of two-year technical and community colleges, which award associate degrees in technology.The technician can advance into supervisory positions in sales, manufacturing, or service management.

Fluid power engineers. This category consists of people who perform design, development, and testing of new fluid power components or systems.The fluid power engineer typically is a graduate of a four-year college program. Most engineers who work on fluid power systems are manufacturing, sales, or mechanical design oriented. They can advance into management positions in design, manufacturing, or sales.

Weight Versus Mass

$$F = W = mg$$

Specific Weight

$$\gamma = \frac{W}{V}$$

Specific Gravity

$$(SG)_{oil} = \frac{\gamma_{oil}}{\gamma_{water}}$$

Density

$$\rho = \frac{m}{V} \qquad SG = \frac{\gamma}{\gamma_{water}} = \frac{g\rho}{g\rho_{water}} = \frac{\rho}{\rho_{water}}$$

Force and Pressure

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

Head

$$p = \gamma H$$

1. A FOOT-SQUARE SECTION OF WATER 10 FEET HIGH CONTAINS 10 CUBIC FEET. IF EACH CUBIC FOOT WEIGHT 62.4 POUNDS ...

1 FT

0.433 psi

3. IF 10 FEET OF WATER IS EQUIVALENT TO 4.33 psi, ONE FOOT EQUALS 0.433, 5 FEET EQUALS 2.165 AND SO ON.

10 FT

2.165 psi

2. THE TOTAL WEIGHT HERE IS 624 POUNDS. THE PRESSURE DUE TO THE WEIGHT IS 624 ÷ 144 SQUARE INCHES OR 4.33 psi

4.33 psi
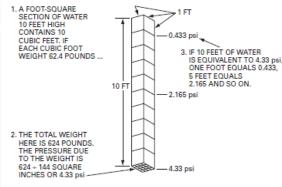
Figure 2-6. Pressure developed by a 10-ft column of water. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

1. A CUBIC FOOT OF OIL WEIGHS ABOUT 55–58 POUNDS.

2. IF THE WEIGHT IS DIVIDED EQUALLY OVER THE 144 SQUARE INCHES OF BOTTOM, THE FORCE ON EACH SQUARE INCH IS 0.4 POUNDS. PRESSURE AT THE BOTTOM THUS IS 0.4 psi.

1 FT

1 FT

1 FT

3. A TWO-FOOT COLUMN WEIGHS TWICE AS MUCH, THUS THE PRESSURE AT THE BOTTOM IS 0.8 psi.
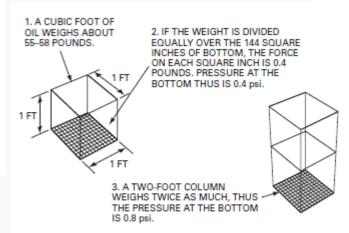
Figure 2-7. Pressures developed by 1- and 2-ft columns of oil. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Atmospheric Pressure
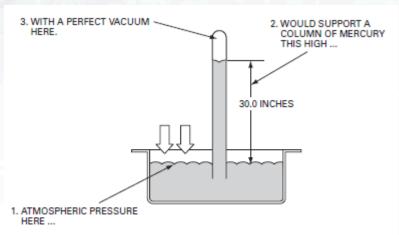
Figure 2-9. Operation of a mercury barometer. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

$$p_{abs} = p_{gage} + p_{atm}$$
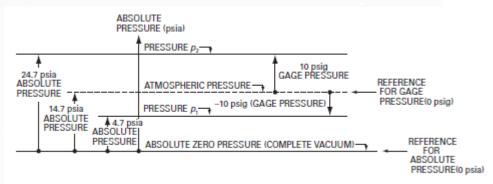
Figure 2-10. Difference between absolute and gage pressures.

Length, Mass, and Force Comparisons with English System

One meter equals 39.4 in = 3.28 ft.
One kilogram equals 0.0685 slugs.
One newton equals 0.225 lb.

$$1 \text{ N} = 1 \text{ kg} \times 1 \text{ m/s}^2$$

Since the acceleration of gravity at sea level equals 9.80 m/s$_2$, a mass of 1 kg weighs 9.80 N. Also, since 1 N 0.225 lb, a mass of 1 kg also weighs 2.20 lb.

Pressure Comparisons

$$1 \text{ Pa} = 1 \text{ N/m}^2$$

$$1 \text{ Pa} = 0.000145 \text{ psi}$$

$$1 \text{ bar} = 10^5 \text{ N/m}^2 = 10^5 \text{ Pa} = 14.5 \text{ psi}$$

- **Temperature Comparisons**

$$T(°F) = 1.8T(°C) + 32 \qquad (2\text{-}11)$$

Thus, to find the equivalent Celsius temperature corresponding to room temperature (68°F), we have:

$$T(°C) = \frac{T(°F) - 32}{1.8} = \frac{68 - 32}{1.8} = 20°C$$
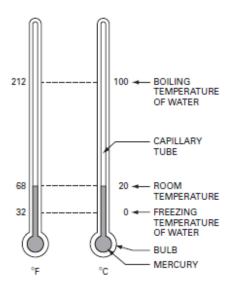
Figure 2-11. Comparison of the Fahrenheit and Celsius temperature scales.

Prefix Name	SI Symbol	Multiplication Factor
tera	T	10^{12}
giga	G	10^{9}
mega	M	10^{6}
kilo	k	10^{3}
centi	c	10^{-2}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}

Figure 2-12. Prefixes used in metric system to represent powers of 10.

BULK MODULUS

The highly favorable power-to-weight ratio and the stiffness of hydraulic systems make them the frequent choice for most high-power applications. The stiffness of a hydraulic system is directly related to the incompressibility of the oil. Bulk modulus is a measure of this incompressibility. The higher the bulk modulus, the less compressible or stiffer the fluid. Mathematically the bulk modulus is defined by Eq. below, where the minus sign indicates that as the pressure increases on a given amount of oil, the oil's volume decreases, and vice versa:

$$\beta = \frac{-\Delta p}{\Delta V/V}$$

VISCOSITY

Viscosity is probably the single most important property of a hydraulic fluid. It is a measure of a fluid's easily and is thin in appearance. A fluid that flows with difficulty has a high viscosity and is thick in appearance. In reality, the ideal viscosity for a given hydraulic system is a compromise.

Too high a viscosity results in

1. High resistance to flow, which causes sluggish operation.

2. Increased power consumption due to frictional losses.

3. Increased pressure drop through valves and lines.

4. High temperatures caused by friction.

On the other hand, if the viscosity is too low, the result is

1. Increased oil leakage past seals.

2. Excessive wear due to breakdown of the oil film between mating moving parts. These moving parts may be internal components of a pump (such as pistons reciprocating in cylinder bores of a piston pump) or a sliding spool inside the body of a valve, as shown in Figure 2-13.resistance to flow.When the viscosity is low, the fluid flows

1. A TYPICAL SLIDING VALVE SPOOL MOVES BACK AND FORTH ...

2. INSIDE ITS BODY...

3. ON A THIN FILM OF HYDRAULIC FLUID (SHOWN GREATLY EXAGGERATED).

4. IF THIS PASSAGE IS UNDER PRESSURE, THE FLUID FILM SEALS IT FROM ADJACENT PASSAGE.
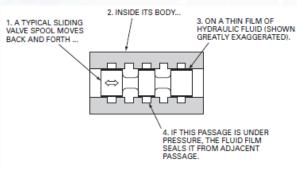
Figure 2-13. Fluid film lubricates and seals moving parts. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Absolute Viscosity

$$\mu = \frac{\tau}{v/y} = \frac{F/A}{v/y} = \frac{\text{shear stress in oil}}{\text{slope of velocity profile}}$$
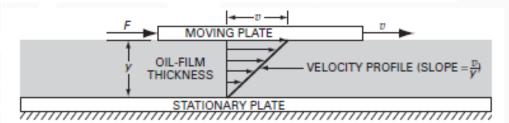
Figure 2-14. Fluid velocity profile between parallel plates due to viscosity.

Kinematic Viscosity

Calculations in hydraulic systems often involve the use of kinematic viscosity rather than absolute viscosity. Kinematic viscosity equals absolute viscosity divided by density:

$$\nu = \frac{\mu}{\rho}$$

Viscosity Index

For $t \leq 100$ SUS:
$$v = 0.226t - \frac{195}{t} \qquad \text{(2-15)}$$

For $t > 100$ SUS:
$$v = 0.220t - \frac{135}{t} \qquad \text{(2-16)}$$

Viscosity index:
$$VI = \frac{L - U}{L - H} \times 100 \qquad \text{(2-18)}$$

EXAMPLE 2-14

A sample of oil with a VI of 80 is tested with a 0-VI oil and a 100-VI oil whose viscosity values at 100°F are 400 and 150 SUS, respectively. What is the viscosity of the sample oil at 100°F in units of SUS?

Solution Substitute directly into Eq. (2-18):

$$VI = \frac{L - U}{L - H} \times 100$$

$$80 = \frac{400 - U}{400 - 150} \times 100$$

$$U = 200 \text{ SUS}$$

Fluid Characteristic
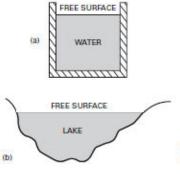
Figure 2-2. Free surface of a liquid.
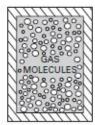
Figure 2-3. A gas always fills its entire vessel.
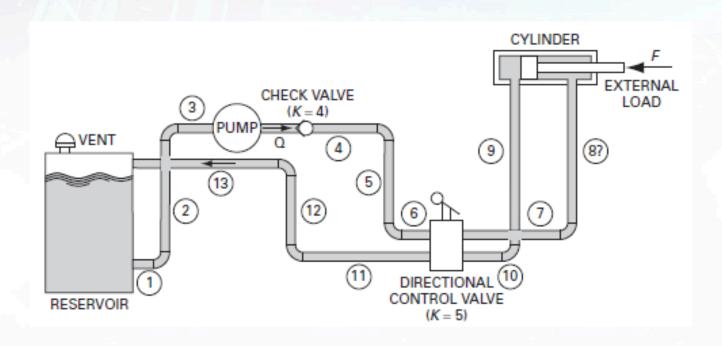
Parameter	Liquid	Gas
Volume	Has its own volume	Volume is determined by container
Shape	Takes shape of container but only to its volume	Expands to completely fill and take the shape of the container
Compressibility	Incompressible for most engineering applications	Readily compressible

Figure 2-4. Physical differences between liquids and gases.
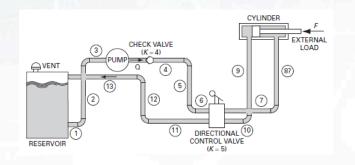
Minggu 2

Komponen Sistem Hidrolik
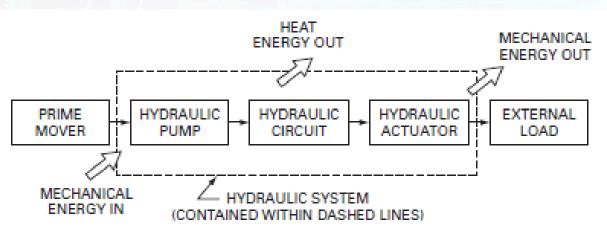
Figure 3-1. Block diagram of hydraulic system showing major components along with energy input and output terms.

Dynamic (nonpositive displacement) pumps. This type is generally used for low-pressure, high-volume flow applications.

Positive displacement pumps. This type is universally used for fluid power systems. As the name implies, a positive displacement pump ejects a fixed amount of fluid into the hydraulic system per revolution of pump shaft rotation. Such a pump is capable of overcoming the pressure resulting from the mechanical loads on the system as well as the resistance to flow due to friction. These are two features that are desired of fluid power pumps.

Advantages positive displacement pumps:

a. High-pressure capability (up to 12,000 psi)

b. Small, compact size

c. High volumetric efficiency

d. Small changes in efficiency throughout the design pressure range

e. Great flexibility of performance (can operate over a wide range of pressure requirements and speed ranges)
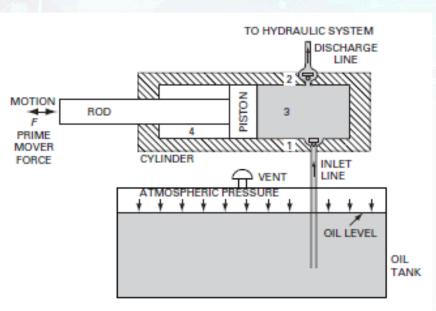
Figure 5-4. Pumping action of a simple piston pump.

External Gear Pump

4. OUTLET PRESSURE AGAINST TEETH CAUSES HEAVY SIDE-LOADING ON SHAFTS AS INDICATED BY ARROWS.

OUTLET

3. AND FORCED OUT OF PRESSURE PORT AS TEETH GO BACK INTO MESH.

DRIVE GEAR

2. OIL IS CARRIED AROUND HOUSING IN CHAMBERS FORMED BETWEEN TEETH, HOUSING AND SIDE PLATES …

INLET

1. VACUUM IS CREATED HERE AS TEETH UNMESH. OIL ENTERS FROM RESERVOIR.
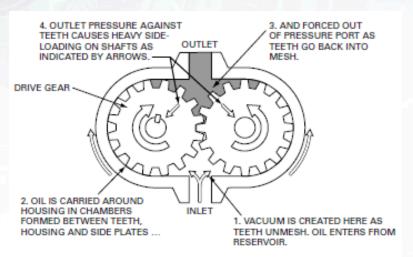
Figure 5-7. External gear pump operation. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)
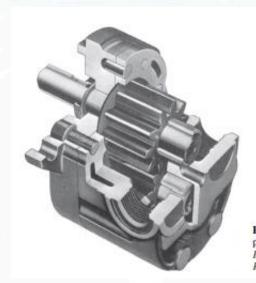
Figure 5-9. Detailed features of an external gear pump. (*Courtesy of Webster Electric Company, Inc., subsidiary of STA-RITE Industries, Inc., Racine, Wisconsin.*)

Internal Gear Pump
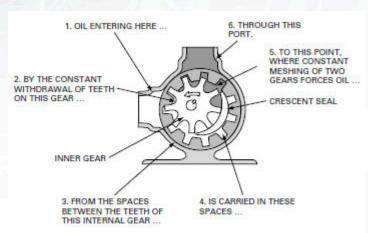
Figure 5-10. Operation of an internal gear pump. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)
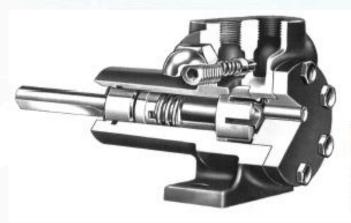
Figure 5-11. Cutaway view of an internal gear pump with built-in safety relief valve. (Courtesy of Viking Pump Division of Houdaille Industries Inc., Cedar Falls, Iowa.)

Lobe Pump
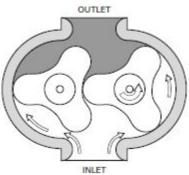
Figure 5-12. Operation of the lobe pump. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Gerotor Pump

CASE
GEROTOR ELEMENT
FEMALE GEAR ROTOR
DISCHARGE PORT
INLET PORT

Figure 5-13. Operation of the Gerotor pump. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)
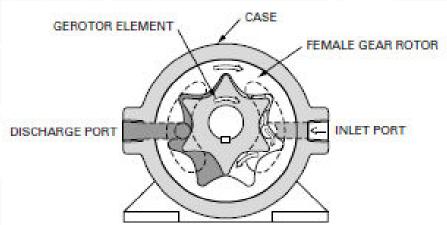
Figure 5-14. Gerotor pump. (*Courtesy of Brown & Sharpe Mfg. Co., Manchester, Michigan.*)

Screw Pump
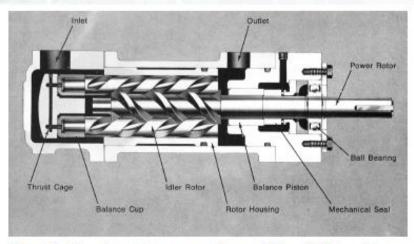
Figure 5-15. Nomenclature of a screw pump. (Courtesy of DeLaval, IMO Pump Division, Trenton, New Jersey.)
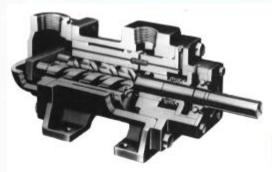
Figure 5-16. Screw pump. (Courtesy of DeLaval, IMO Pump Division, Trenton, New Jersey.)
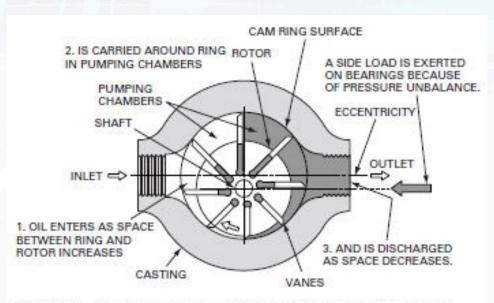
Figure 5-17. Vane pump operation. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Pressure-Compensated Vane Pump
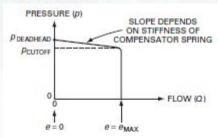
Figure 5-19. Pressure versus flow for pressure-compensated vane pump.

Figure 5-20. Cutaway photograph of pressure-compensated vane pump. (Courtesy of Continental Hydraulics, Division of Continental Machines, Inc., Savage, Minnesota.)
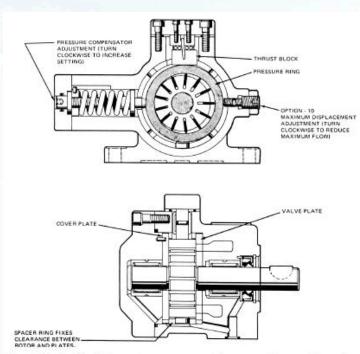
Figure 5-18. Variable displacement, pressure-compensated vane pump. (Courtesy of Brown & Sharpe Mfg. Co., Manchester, Michigan.)

Balanced Vane Pump
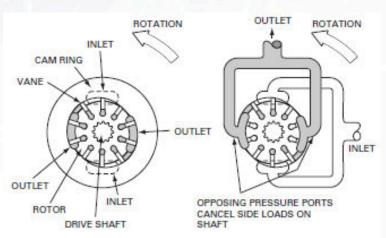
Figure 5-21. Balanced vane pump principles. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)
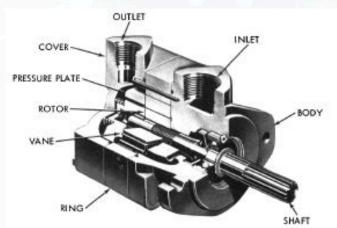
Figure 5-22. Cutaway view of balanced vane pump. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Axial Piston Pump (Bent-Axis Design)

Figure 5-24. Volumetric displacement changes with offset angle. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)
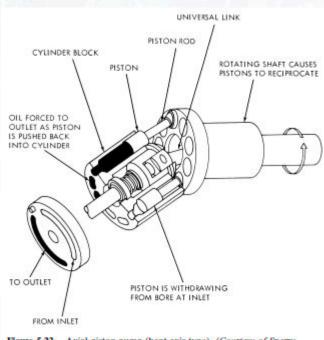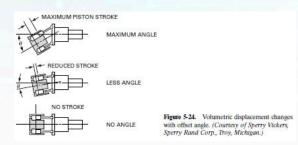
Figure 5-23. Axial piston pump (bent-axis type). (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)
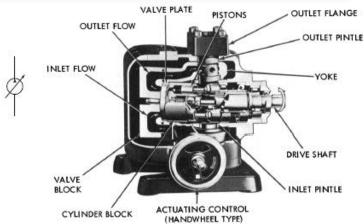
Figure 5-25. Variable displacement piston pump with handwheel. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

In-Line Piston Pump (Swash Plate Design)
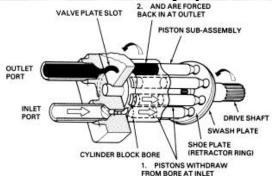
Figure 5-27. Swash plate causes pistons to reciprocate. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)
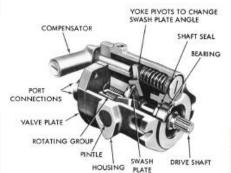
Figure 5-28. Variable displacement version of in-line piston pump. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)
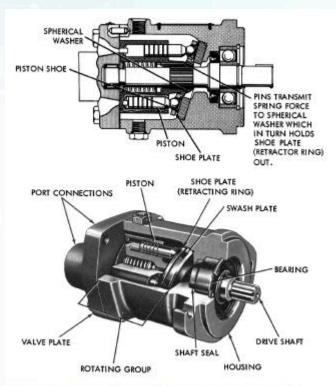
Figure 5-26. In-line design piston pump. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)

Radial Piston Pump
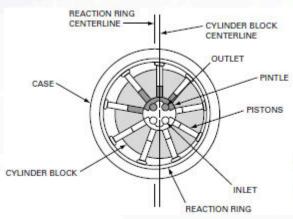
Figure 5-30. Operation of a radial piston pump. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Figure 5-31. Cutaway view of a radial piston pump. *(Courtesy of Deere & Co., Moline, Illinois.)*

Volumetric efficiency (η_v)

$$\eta_v = \frac{\text{actual flow-rate produced by pump}}{\text{theoretical flow-rate pump should produce}} = \frac{Q_A}{Q_T}$$

Mechanical efficiency (η_m)

$$\eta_m = \frac{\text{pump output power assuming no leakage}}{\text{actual power delivered to pump}}$$

$$\eta_m = \frac{\text{theoretical torque required to operate pump}}{\text{actual torque delivered to pump}} = \frac{T_T}{T_A}$$

Overall efficiency (η_o)

$$\text{overall efficiency} = \frac{\text{actual power deliveed by pump}}{\text{actual power delivered to pump}}$$

$$\eta_o = \eta_v \times \eta_m$$

- Hydraulic cylinders
- hydraulic motors

Hydraulic cylinders

(a) SCHEMATIC DRAWING   (b) GRAPHIC SYMBOL

Figure 6-2.   Single-acting hydraulic cylinder.
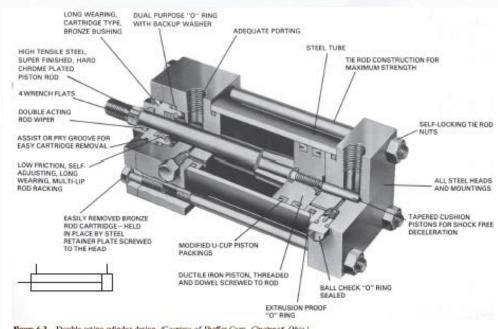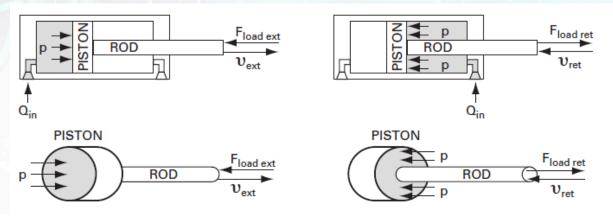
Figure 6-3.   Double-acting cylinder design. (*Courtesy of Sheffer Corp., Cincinnet, Ohio.*)

Figure 6-7. The effective area of double-acting cylinders is greater for the extension stroke than it is for the retraction stroke.

CYLINDER FORCE, VELOCITY, AND POWER

Extension Stroke

$$F_{ext}(\text{lb}) = p(\text{psi}) \times A_p(\text{in}^2)$$

$$F_{ext}(\text{N}) = p(\text{Pa}) \times A_p(\text{m}^2)$$

$$v_{ext}(\text{ft/s}) = \frac{Q_{in}(\text{ft}^3/\text{s})}{A_p(\text{ft}^2)}$$

$$v_{ext}(\text{m/s}) = \frac{Q_{in}(\text{m}^3/\text{s})}{A_p(\text{m}^2)}$$

Retraction Stroke

$$F_{ret}(\text{lb}) = p(\text{psi}) \times (A_p - A_r)\text{in}^2$$

$$F_{ret}(\text{N}) = p(\text{Pa}) \times (A_p - A_r)\text{m}^2$$

$$v_{ret}(\text{ft/s}) = \frac{Q_{in}(\text{ft}^3/\text{s})}{(A_p - A_r)\text{ft}^2}$$

$$v_{ret}(\text{m/s}) = \frac{Q_{in}(\text{m}^3/\text{s})}{(A_p - A_r)\text{m}^2}$$

$$\text{Power (HP)} = \frac{v_p(\text{ft/s}) \times F(\text{lb})}{550} = \frac{Q_{in}(\text{gpm}) \times p(\text{psi})}{1714}$$

$$\text{Power (kW)} = v_p(\text{m/s}) \times F(\text{kN}) = Q_{in}(\text{m}^3/\text{s}) \times p(\text{kPa})$$

Hydraulic motors
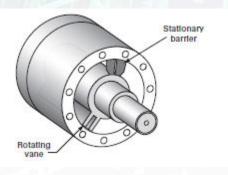
Figure 7-2. Rotary actuator. (Courtesy of Rexnord Inc., Hydraulic Components Division, Racine, Wisconsin.)
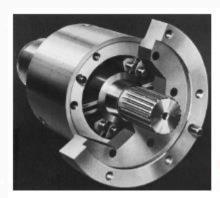
Figure 7-3. Rotary actuator. (Courtesy of Ex-Cell-O Corp., Troy, Michigan.)

4. THESE TWO TEETH HAVE ONLY TANK LINE PRESSURE OPPOSING THEM

3. PRESSURE BETWEEN TEETH IN THIS SEGMENT PUSHES BOTH WAYS AND DOES NOT AFFECT TORQUE AS OIL IS CARRIED AROUND TO OUTLET

3. PRESSURE BETWEEN TEETH IN THIS SEGMENT PUSHES BOTH WAYS AND DOES NOT AFFECT TORQUE AS OIL IS CARRIED AROUND TO OUTLET

1. THESE TWO TEETH ARE SUBJECT TO HIGH PRESSURE AND TEND TO ROTATE GEARS IN DIRECTION OF ARROWS

2. SEGMENTS OF TWO MESHING TEETH TEND TO OPPOSE ROTATION MAKING NET TORQUE AVAILABLE, A FUNCTION OF ONE TOOTH

Figure 7-5. Torque development by a gear motor. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Figure 7-6. External gear motor. *(Courtesy of Webster Electric Company, Inc., subsidiary of STA-RITE Industries, Inc., Racine, Wisconsin.)*
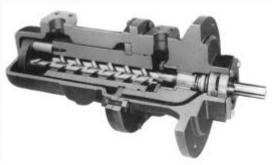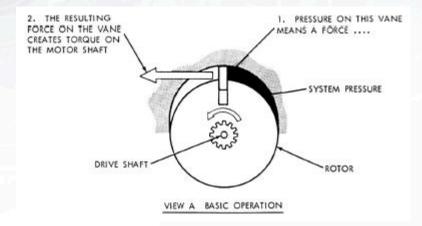
Figure 7-7. Screw motor. *(Courtesy of DeLaval, IMO Pump Division, Trenton, New Jersey.)*

2. THE RESULTING FORCE ON THE VANE CREATES TORQUE ON THE MOTOR SHAFT

1. PRESSURE ON THIS VANE MEANS A FORCE ....

SYSTEM PRESSURE

DRIVE SHAFT

ROTOR

VIEW A   BASIC OPERATION

1. THIS VANE IS SUBJECT TO HIGH PRESSURE AT THE INLET SIDE AND LOW PRESSURE OPPOSITE

OUTLET

ROTATION

INLET

3. THE INLET CONNECTS TO TWO OPPOSING PRESSURE PASSAGES TO BALANCE SIDE LOADS ON THE ROTOR.

2. THE RESULTING FORCE ON THE VANE CREATES TORQUE ON THE ROTOR SHAFT

VIEW B   BALANCED DESIGN

Figure 7-8. Operation of a vane motor. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

In-Line Piston Motor (Swash Plate Design)
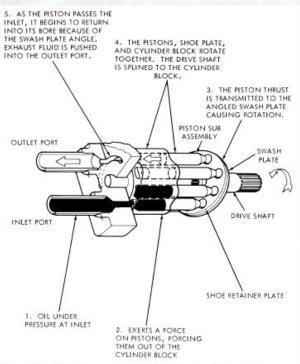
Figure 7-11. In-line piston motor operation. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*
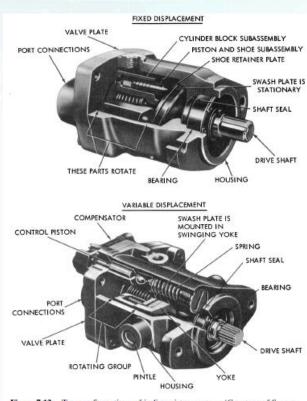
Figure 7-12. Two configurations of in-line piston motors. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Axial Piston Motor (Bent-Axis Design)

STROKE LENGTH IS MAXIMUM

STROKE LENGTH IS MINIMUM

(MAXIMUM DISPLACEMENT)

(MINIMUM DISPLACEMENT)

MAXIMUM SWASH PLATE ANGLE AND MAXIMUM TORQUE CAPABILITY

MINIMUM SWASH PLATE ANGLE AND MINIMUM TORQUE CAPABILITY
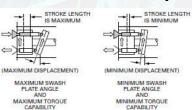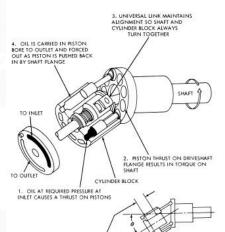
Figure 7-13. Motor displacement varies with swash plate angle. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

3. UNIVERSAL LINK MAINTAINS ALIGNMENT SO SHAFT AND CYLINDER BLOCK ALWAYS TURN TOGETHER

4. OIL IS CARRIED IN PISTON BORE TO OUTLET AND FORCED OUT AS PISTON IS PUSHED BACK IN BY SHAFT FLANGE

SHAFT

TO INLET

TO OUTLET

2. PISTON THRUST ON DRIVESHAFT FLANGE RESULTS IN TORQUE ON SHAFT

CYLINDER BLOCK

1. OIL AT REQUIRED PRESSURE AT INLET CAUSES A THRUST ON PISTONS

5. THEREFORE PISTON DISPLACEMENT AND TORQUE CAPABILITY DEPEND ON ANGLE

Figure 7-14. Bent-axis piston motor. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

DRIVESHAFT

PORTS

Figure 7-15. Fixed displacement bent-axis piston motor. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

VALVE PLATE

OUTLET FLOW

PISTONS

OUTLET FLANGE

OUTLET PINTLE

INLET FLOW

YOKE

DRIVE SHAFT

VALVE BLOCK

INLET PINTLE

CYLINDER BLOCK

ACTUATING CONTROL (HANDWHEEL TYPE)
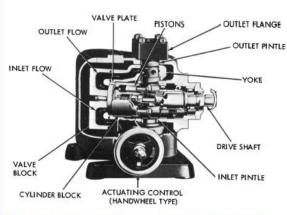
Figure 7-16. Variable displacement bent-axis piston motor with handwheel control. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

HYDRAULIC MOTOR PERFORMANCE

Volumetric and Mechanical efficiency

Volumetric efficiency of a hydraulic motor:

$$\eta_v = \frac{\text{theoretical flow rate motor should consume}}{\text{actual flow rate consumed by motor}} = \frac{Q_T}{Q_A}$$

Mechanical efficiency of a hydraulic motor:

$$\eta_m = \frac{\text{actual torque delivered by motor}}{\text{torque motor should theoretically deliver}} = \frac{T_A}{T_T}$$

Overall efficiency

$$\eta_o = \eta_v \eta_m$$

Minggu 3

1. DIRECTIONAL CONTROL VALVES
2. PRESSURE CONTROL VALVES
3. FLOW CONTROL VALVES
4. SERVO VALVES
5. PROPORTIONAL CONTROL VALVES
6. CARTRIDGE VALVES
7. HYDRAULIC FUSES

Figure 8-1. Welding machine application using hydraulic control valves. *(Courtesy of Owatonna Tool Co., Owatonna, Minnesota.)*
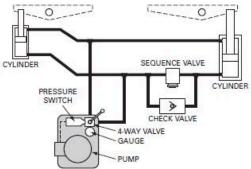
Figure 8-2. Hydraulic circuit showing control valves used for welding application. *(Courtesy of Owatonna Tool Co., Owatonna, Minnesota.)*

Check Valve

POPPET   SPRING

BODY

NO FLOW

IN   OUT

FREE FLOW

FREE-FLOW DIRECTION   NO-FLOW DIRECTION

GRAPHIC SYMBOL

SCHEMATIC DRAWINGS
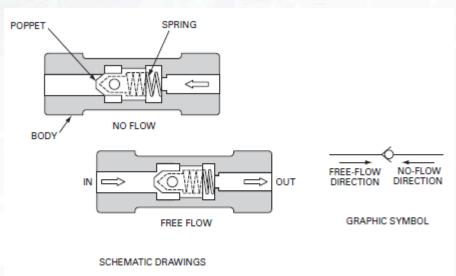
Figure 8-4. Operation of check valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Figure 8-3. Check valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Pilot-Operated Check Valve
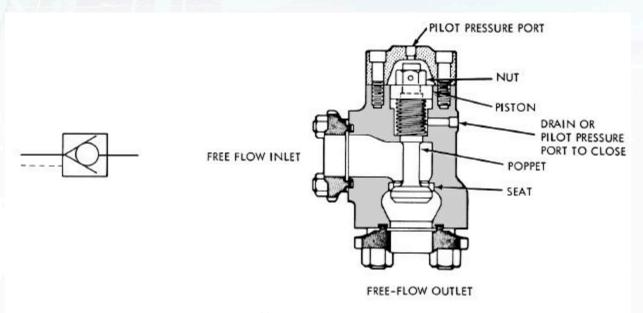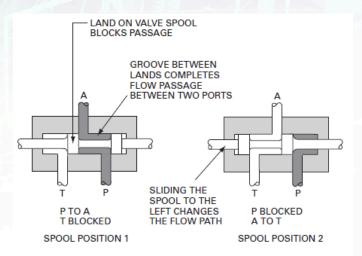
Figure 8-5. Pilot-operated check valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Three-Way Valves

Figure 8-6. Two spool positions inside a three-way valve.
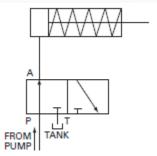
Figure 8-7. Three-way DCV controlling flow directions to and from a single-acting cylinder.

Four-Way Valves
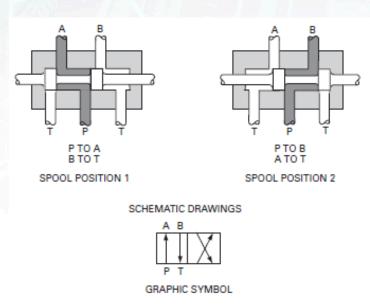
Figure 8-8. Two spool positions inside a four-way valve.
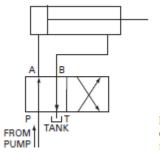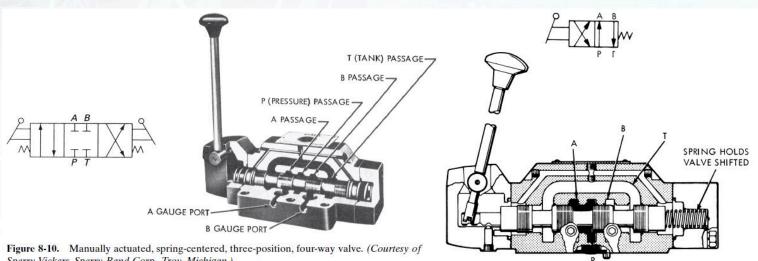
Figure 8-9. Four-way DCV controlling flow directions to and from a double-acting cylinder.

Manually Actuated Valves

Figure 8-10. Manually actuated, spring-centered, three-position, four-way valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Figure 8-11. Manually actuated, two-position, spring-offset, four-way valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Mechanically Actuated Valves
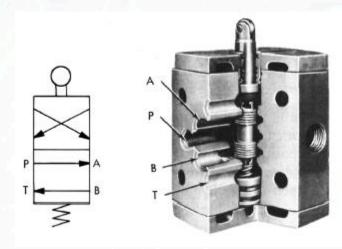
Figure 8-12. Mechanically actuated, spring-offset, two-position, four-way valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Pilot-Actuated Valves
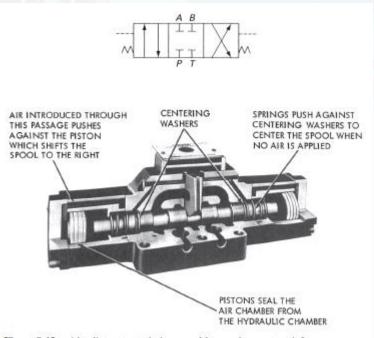
Figure 8-13. Air pilot–actuated, three-position, spring-centered, four-way valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Solenoid-Actuated Valves
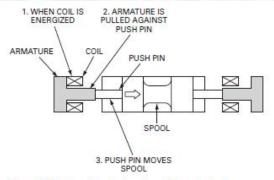
Figure 8-14. Operation of solenoid to shift spool of valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*
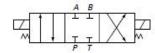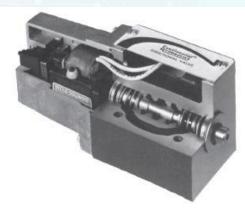
Figure 8-15. Solenoid-actuated, three-position, spring-centered, four-way directional control valve. *(Courtesy of Continental Hydraulics, Division of Continental Machines Inc., Savage, Minnesota.)*
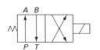
Figure 8-16. Single solenoid-actuated, four-way, two-position, spring-offset directional control valve. *(Courtesy of Continental Hydraulics, Division of Continental Machines Inc., Savage, Minnesota.)*
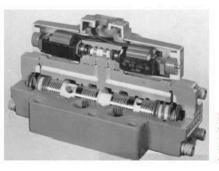
Figure 8-17. Solenoid-controlled, pilot-operated directional control valve. *(Courtesy of Continental Hydraulics, Division of Continental Machines Inc., Savage, Minnesota.)*

Center Flow Path Configurations for Three-Position, Four-Way Valves
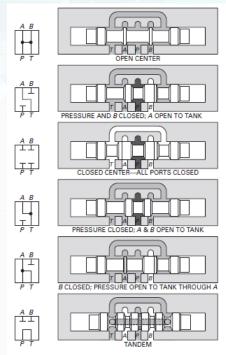
Figure 8-18. Various center flow paths for three-position, four-way valves. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Shuttle Valves
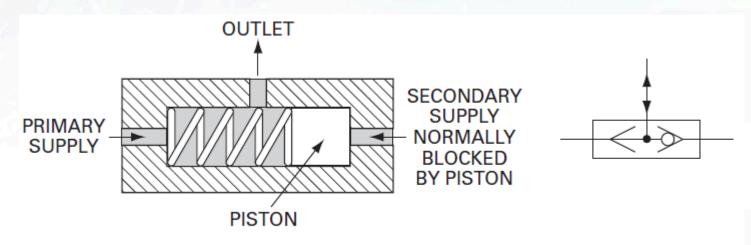
Figure 8-19. Shuttle valve (schematic and graphic symbol).

Simple Pressure Relief Valves
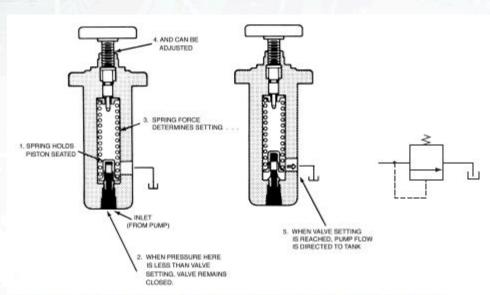
Figure 8-20. Simple pressure relief valve. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)
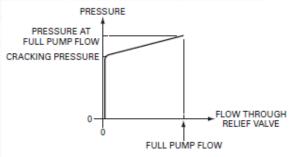
Figure 8-21. Pressure versus flow curve for simple relief valve.
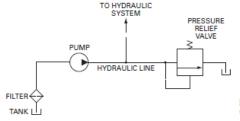
Figure 8-22. Symbolic representation of partial hydraulic circuit.

Compound Pressure Relief Valves
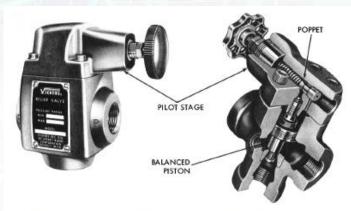
Figure 8-23. External and cutaway views of an actual compound relief valve. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Figure 8-25. Compound pressure relief valve with integral solenoid-actuated, two-way vent valve. (*Courtesy of Abex Corp., Denison Division, Columbus, Ohio.*)

4. WHEN THE VALVE SETTING IS REACHED, THE POPPET "OPENS" LIMITING PRESSURE IN UPPER CHAMBER.

7. VENT CONNECTION PERMITS UNLOADING PUMP THROUGH RELIEF VALVE.

3. SPRING HOLDS PISTON CLOSED.

1. INLET PRESSURE HERE ...

2. IS SENSED ABOVE PISTON AND AT PILOT VALVE THROUGH ORIFICE IN PISTON.

6. PISTON MOVES UP TO DIVERT PUMP OUTPUT DIRECTLY TO TANK.

5. WHEN THIS PRESSURE IS 20 psi HIGHER THAN IN UPPER CHAMBER ...

VIEW A CLOSED

VIEW B CRACKED

VIEW C RELIEVING

Figure 8-24. Operation of compound pressure relief valve. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Pressure-Reducing Valves
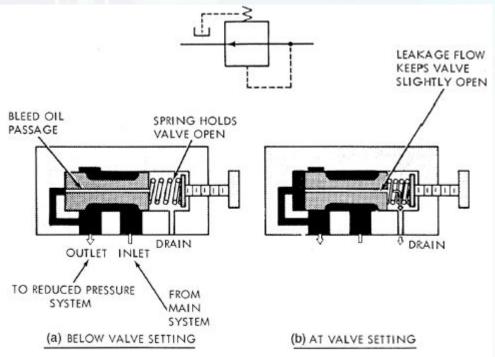
Figure 8-26. Operation of a pressure-reducing valve. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Unloading Valves
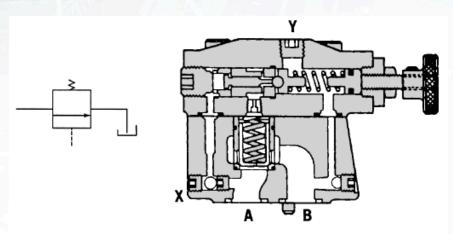
Figure 8-27. Schematic of unloading valve. (*Courtesy of Abex Corp., Denison Division, Columbus, Ohio.*)

Figure 8-28. Unloading valve. (*Courtesy of Abex Corp., Denison Division, Columbus, Ohio.*)

Sequence Valves
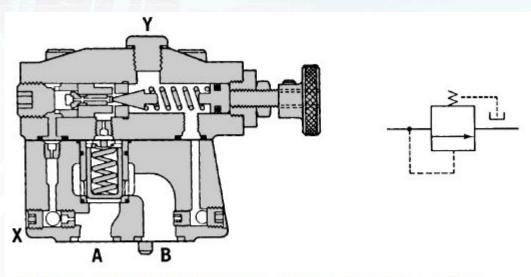
Figure 8-29. Schematic of sequence valve. *(Courtesy of Abex Corp., Denison Division, Columbus, Ohio.)*

Counterbalance Valves
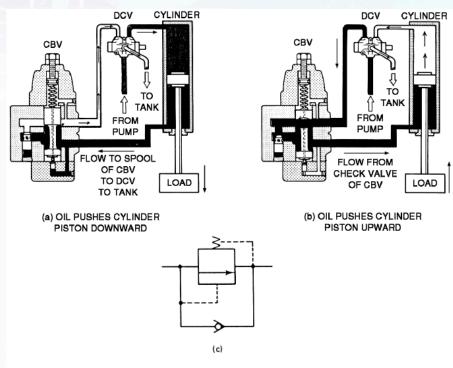
Figure 8-30. Application of counterbalance valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Orifice as a Flow Meter or Flow Control Device
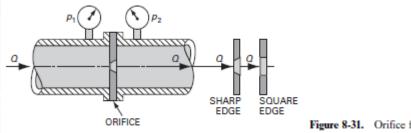
Figure 8-31. Orifice flowmeter.

Needle Valves
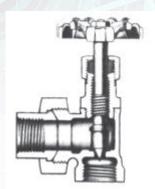
Figure 8-32. Needle valve. (*Courtesy of Crane Co., Chicago, Illinois.*)

Figure 8-33. Easy read and adjust flow control valve. (*Courtesy of Deltrol Corp., Bellwood, Illinois.*)
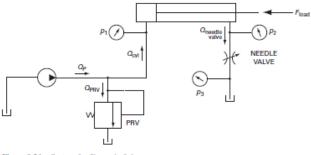
Figure 8-34. System for Example 8-6.

Non-Pressure-Compensated Valves
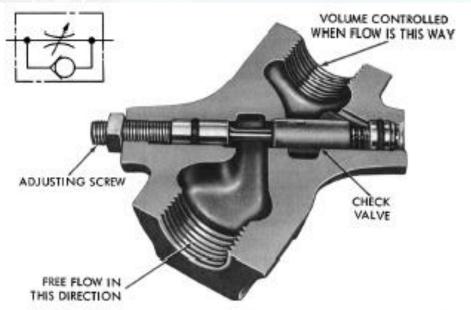
Figure 8-35. Non-pressure-compensated flow control valve. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Pressure-Compensated Valves

SPRING LOAD SETS
PRESSURE DIFFERENCE
ACROSS THROTTLE

TO LOAD

THIS AREA EQUALS
COMBINED AREAS OF
ANNULUS AND STEM

HYDROSTAT PISTON IS
BALANCED BETWEEN
INTERMEDIATE PRESSURE
BELOW AND LOAD PRESSURE
ABOVE

ANNULUS

FROM PUMP

THROTTLE
CONTROLS
FLOW

STEM

LAND BLOCKS
EXCESS FLOW AND
FORCES IT OVER
RELIEF VALVE
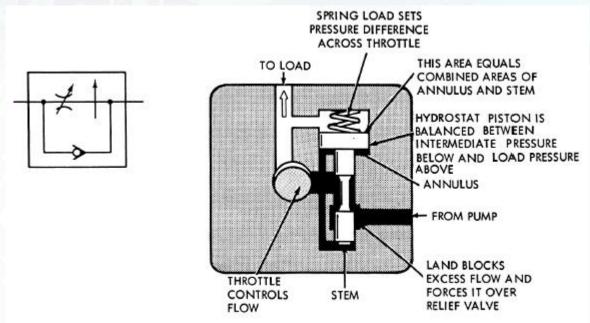
Figure 8-36. Operation of pressure-compensated flow control valve. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Mechanical-Type Servo Valves
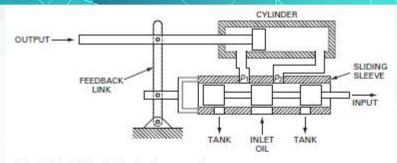
Figure 8-38. Mechanical-hydraulic servo valve.
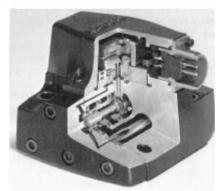
Figure 8-39. Electrohydraulic servo valve. (*Courtesy of Moog Inc., Industrial Division, East Aurora, New York.*)

mechanical-hydraulic servo valve is the hydraulic power steering system of automobiles and other transportation vehicles.

Electrohydraulic Servo Valves
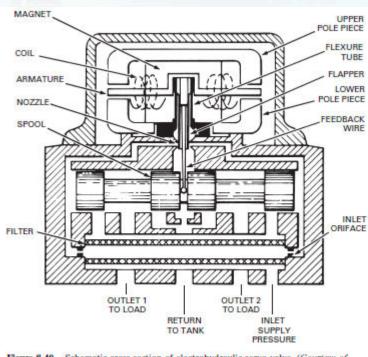
Figure 8-40. Schematic cross section of electrohydraulic servo valve. *(Courtesy of Moog Inc., Industrial Division, East Aurora, New York.)*
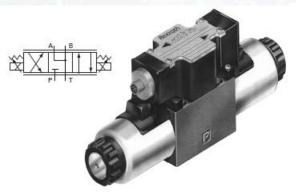
Figure 8-41. External view of four-way proportional directional control valve. *(Courtesy of Bosch Rexroth Corp., Bethlehem, Pennsylvania.)*
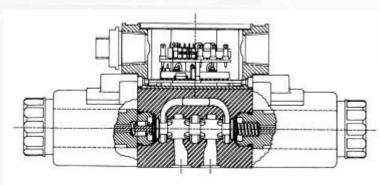
Figure 8-43. Pictorial cutaway view of four-way proportional directional control valve. *(Courtesy of Bosch Rexroth Corp., Bethlehem, Pennsylvania.)*

Figure 8-42. Schematic cutaway view of four-way proportional directional control valve. *(Courtesy of Bosch Rexroth Corp., Bethlehem, Pennsylvania.)*

Figure 8-44. Cutaway views of threaded cartridge valves. *(Courtesy of Parker Hannifin Corp., Elyria, Ohio.)*

Figure 8-46. Solenoid-operated directional control cartridge valves. *(Courtesy of Parker Hannifin Corp., Elyria, Ohio.)*

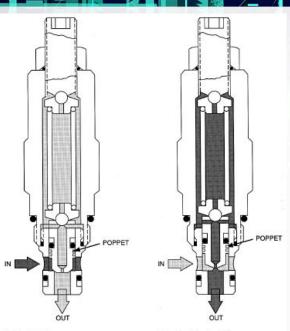**Figure 8-47.** Cartridge pressure relief valve. *(Courtesy of Parker Hannifin Corp., Elyria, Ohio.)*

Figure 8-45. Manifold block containing cartridge valves. *(Courtesy of Parker Hannifin Corp., Elyria, Ohio.)*

Figure 8-48. Cartridge solenoid-operated flow control valve. *(Courtesy of Parker Hannifin Corp., Elyria, Ohio.)*

POPPET

POPPET

IN → OUT

IN → OUT

No Flow Condition
- System pressure is lower than relief valve setting.
- Poppet is seated, held in position by spring force.
- Flow is blocked at inlet.

Throttled Flow Condition
- System pressure has reached relief valve setting.
- Pressure has moved Poppet away from Seat, allowing flow to pass through valve.
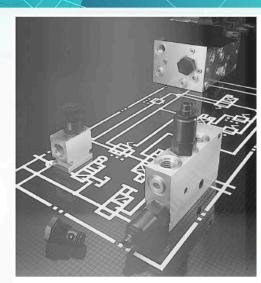- Valve is throttling flow to maintain relief pressure at inlet.

Figure 8-49. Cartridge pressure relief valve. *(Courtesy of Parker Hannifin Corp., Elyria, Ohio.)*

Figure 8-50. Integrated hydraulic circuit. *(Courtesy of Parker Hannifin Corp., Elyria, Ohio.)*

DRAIN TO TANK

INLET PRESSURE

THIN METAL DISK

HYDRAULIC FUSE

(a) SCHEMATIC DRAWING

(b) GRAPHIC SYMBOL IN PARTIAL CIRCUIT
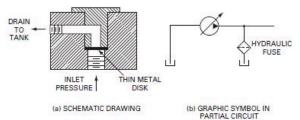
Figure 8-51. Hydraulic fuse.

STEEL PIPES

Size Designation

NOMINAL PIPE SIZE	PIPE OUTSIDE DIAMETER	PIPE INSIDE DIAMETER		
		SCHEDULE 40	SCHEDULE 80	SCHEDULE 160
1/8	0.405	0.269	0.215	–
1/4	0.540	0.364	0.302	–
3/8	0.675	0.493	0.423	–
1/2	0.840	0.622	0.546	0.466
3/4	1.050	0.824	0.742	0.614
1	1.315	1.049	0.957	0.815
1-1/4	1.660	1.380	1.278	1.160
1-1/2	1.900	1.610	1.500	1.338
2	2.375	2.067	1.939	1.689

Figure 10-2. Common pipe sizes.
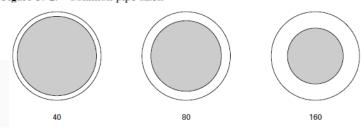
Figure 10-3. Relative size of the cross section of schedules 40, 80, and 160 pipe.

Thread Design

1. THE TAPERED MALE THREAD ON THE SECTION OF PIPE . . .

2. SCREWS INTO THE FEMALE THREAD IN THE FITTING OR HYDRAULIC COMPONENT. THIS THREAD ALSO IS TAPERED.

3. AS THE JOINT IS TIGHTENED, AN INTERFERENCE OCCURS BETWEEN THE THREADS, SEALING THE JOINTS.

4. IN STANDARD PIPE THREADS, THE FLANKS COME IN CONTACT FIRST.

5. THERE CAN BE A SPIRAL CLEARANCE AROUND THE THREADS.

6. IN DRY-SEAL THREADS, THE ROOTS AND CRESTS ENGAGE FIRST, ELIMINATING SPIRAL CLEARANCE.

Figure 10-4. Hydraulic pipe threads are the dry-seal tapered type. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)

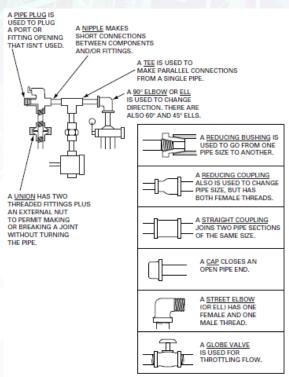## STEEL PIPES (FITTINGS)

Figure 10-5. Fittings make the connections between pipes and components. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)
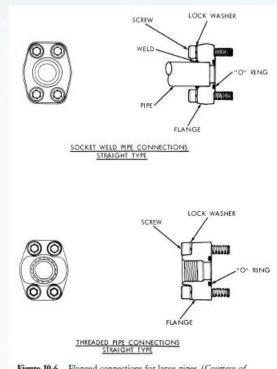

Figure 10-6. Flanged connections for large pipes. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)

STEEL TUBING

Size Designation

TUBE OD (in)	WALL THICKNESS (in)	TUBE ID (in)	TUBE OD (in)	WALL THICKNESS (in)	TUBE ID (in)	TUBE OD (in)	WALL THICKNESS (in)	TUBE ID (in)
1/8	0.035	0.055	1/2	0.035	0.430	7/8	0.049	0.777
				0.049	0.402		0.065	0.745
3/16	0.035	0.118		0.065	0.370		0.109	0.657
				0.095	0.310			
1/4	0.035	0.180				1	0.049	0.902
	0.049	0.152	5/8	0.035	0.555		0.065	0.870
	0.065	0.120		0.049	0.527		0.120	0.760
				0.065	0.495			
5/16	0.035	0.243		0.095	0.435	1-1/4	0.065	1.120
	0.049	0.215					0.095	1.060
	0.065	0.183	3/4	0.049	0.652		0.120	1.010
				0.065	0.620			
3/8	0.035	0.305		0.109	0.532	1-1/2	0.065	1.370
	0.049	0.277					0.095	1.310
	0.065	0.245						

Figure 10-7. Common tube sizes.

Tube Fittings
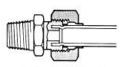

(a) 37° FLARE FITTING
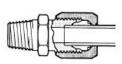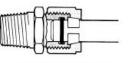
(b) 45° FLARE FITTING

(c) STRAIGHT THREAD "O" RING CONNECTOR

(d) FERRULE COMPRESSION FITTING

(e) "O" RING COMPRESSION FITTING

(f) SLEEVE COMPRESSION FITTING

Figure 10-8. Threaded fittings and connectors used with tubing. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)

Figure 10-9. Swagelok tube fitting. (Courtesy of Swagelok Co., Solon, Ohio.)

Figure 10-10. The 45° flare fitting. (Courtesy of Gould, Inc., Valve and Fittings Division, Chicago, Illinois.)

Figure 10-11. Various steel tube fittings. (a) Union elbow. (b) union tee, (c) union, (d) 45° male elbow. (Courtesy of Gould, Inc., Valve and Fittings Division, Chicago, Illinois.)
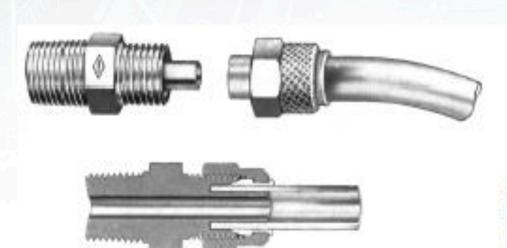
PLASTIC TUBING

Figure 10-12. Poly-Flo Flareless Plastic Tube Fitting. (Courtesy of Gould, Inc., Valve and Fittings Division, Chicago, Illinois.)

FLEXIBLE HOSES

Design and Size Designation

HOSE SIZE	OD TUBE SIZE (In)	SINGLE-WIRE BRAID			DOUBLE-WIRE BRAID		
		HOSE ID (In)	HOSE OD (In)	MINIMUM BEND RADIUS (In)	HOSE ID (In)	HOSE OD (In)	MINIMUM BEND RADIUS (In)
4	1/4	3/16	33/64	1-15/16	1/4	11/16	4
6	3/8	5/16	43/64	2-3/4	3/8	27/32	5
8	1/2	13/32	49/64	4-5/8	1/2	31/32	7
12	3/4	5/8	1-5/64	6-9/16	3/4	1-1/4	9-1/2
16	1	7/8	1-15/64	7-3/8	1	1-9/16	11
20	1-1/4	1-1/8	1-1/2	9	1-1/4	2	16

Figure 10-14. Typical hose sizes.

1. THE OUTER LAYER IS SYNTHETIC RUBBER ... USED TO PROTECT ...

2. THE SECOND LAYER OF WIRE OR CLOTH BRAID.

3. FOR HIGHER PRESSURE, ADDITIONAL BRAIDED LAYERS ARE USED.

4. THE INNER LAYER IS A MATERIAL COMPATIBLE WITH THE HYDRAULIC FLUID.

Figure 10-13. Flexible hose is constructed in layers. (Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)

AEROQUIP FC 194 AQP (a)

AEROQUIP FC 195 AQP (b)

AEROQUIP FC 300 AQP (c)

AEROQUIP 1525 (d)

AEROQUIP 2791 (e)

Figure 10-15. Various flexible hose designs. (a) FC 194: single-wire braid; (b) FC 195: double-wire braid; (c) FC 300: single-wire braid, polyester inner braid; (d) 1525: single-textile braid; (e) 2791: four heavy spiral wires, partial textile braid. (Courtesy of Aeroquip Corp., Jackson, Michigan.)

FLEXIBLE HOSES
Hose Fittings

Hose Routing and Installation

Figure 10-16. Flexible hoses with permanently attached end fittings. (a) Straight fitting, (b) 45° elbow fitting, (c) 90° elbow fitting. (Courtesy of Aeroquip Corp., Jackson, Michigan.)

Figure 10-17. Flexible hose reusable-type end fittings. (a) Straight fitting, (b) elbow fitting, (c) 90° elbow fitting. (Courtesy of Aeroquip Corp., Jackson, Michigan.)

Hose routing and installation

Under pressure, a hose may change in length. The range is from −4% to +2%. Always provide some slack in the hose to allow for this shrinkage or expansion. (However, excessive slack in hose lines is one of the most common causes of poor appearance.)

If a hose is installed with a twist in it, high operating pressures tend to force it straight. This can loosen the fitting nut or even burst the hose at the point of strain.

When hose lines pass near an exhaust manifold, or other heat source, they should be insulated by a heat resistant boot, firesleeve or a metal baffle. In any application, brackets and clamps keep hoses in place and reduce abrasion.

At bends, provide enough hose for a wide radius curve. Too tight a bend pinches the hose and restricts the flow. The line could even kink and close entirely. In many cases, use of the right fittings or adapters can eliminate bends or kinks.

In applications where there is considerable vibration or flexing, allow additional hose length. The metal hose fittings, of course, are not flexible, and proper installation protects metal parts from un- due stress, and avoids kinks in the hose.

When 90° adapters were used, this assembly became neater-looking and easier to inspect and maintain. It uses less hose, too!

Four basic adapter functions

1. To join a hose to a component. Example, a valve might have a ½" female pipe thread and a hose a ¾" S.A.E. 37° swivel nut. The right Aeroquip Adapter fits both.

2. To connect two or more pieces of hose and tubing. Here, a T-shaped adapter connects two hoses with a length of tubing. Each end of the adapter may have a different thread.

3. To provide both connection and anchor at a bulkhead. In this example, it provides an anchor in addition to connecting a hose to a tube.

4. To eliminate the need for a bushing. Example, one end of the adapter is ¼" pipe thread, connected to the assembly, and the other is ½" S.A.E. 37° flare, which connects to an S.A.E. 37° swivel fitting. The adapter itself replaces the bushing.

Figure 10-18. Hose routing and installation information. (Courtesy of Aeroquip Corp., Jackson, Michigan.)

QUICK DISCONNECT COUPLINGS

Figure 10-19. Quick disconnect coupling (cross-sectional view). (*Courtesy of Hansen Manufacturing Co., Cleveland, Ohio.*)

Ancilary hidraulic devices

- **RESERVOIRS**
- **ACCUMULATORS**
- **PRESSURE INTENSIFIERS**
- **SEALING DEVICES**
- **HEAT EXCHANGERS**
- **PRESSURE GAGES**
- **FLOWMETERS**
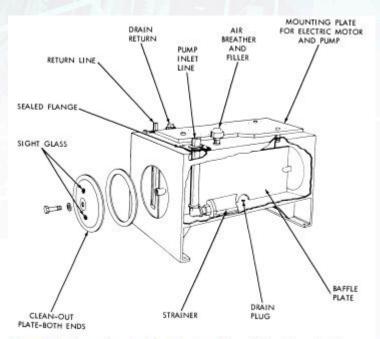
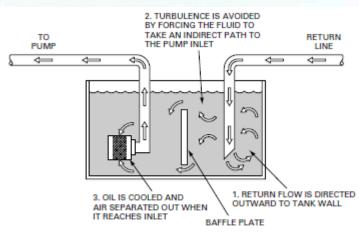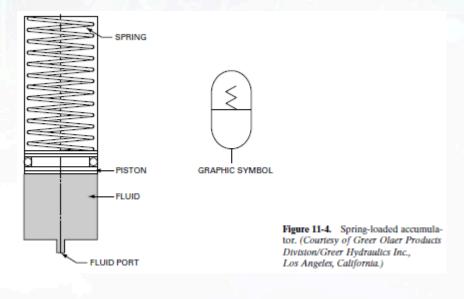**Figure 11-1.** Reservoir construction. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Figure 11-2. Baffle plate controls direction of flow in reservoir. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Figure 11-4. Spring-loaded accumulator. *(Courtesy of Greer Olaer Products Division/Greer Hydraulics Inc., Los Angeles, California.)*
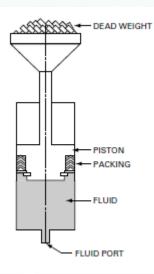
Figure 11-3. Weight-loaded accumulator. *(Courtesy of Greer Olaer Products Division/Greer Hydraulics Inc., Los Angeles, California.)*
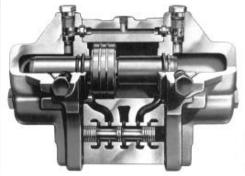
Figure 11-17. Cutaway view of pressure intensifier. *(Courtesy of Rexnord Inc., Hydraulic Components Division, Racine, Wisconsin.)*

EXHAUST OIL

LOW PRESSURE OIL

HIGH PRESSURE OIL
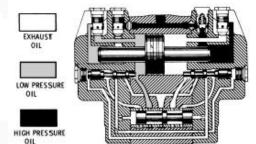
Figure 11-18. Oil flow paths of pressure intensifier. *(Courtesy of Rexnord Inc., Hydraulic Components Division, Racine, Wisconsin.)*
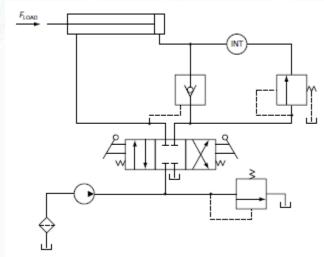
Figure 11-19. Pressure intensifier circuit.

SEALING DEVICES

BASIC FLANGE JOINTS

GASKET

METAL-TO-METAL JOINTS

Figure 11-22. Static seal flange joint applications. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*
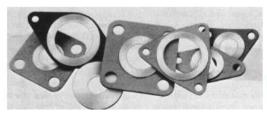
Figure 11-23. Die-cut gaskets used for flanged joints. *(Courtesy of Crane Packing Co., Morton Grove, Illinois.)*
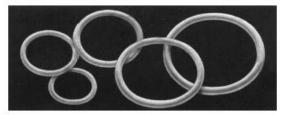
Figure 11-24. Several different-sized O-rings. *(Courtesy of Crane Packing Co., Morton Grove, Illinois.)*

NOTE: CLEARANCES ARE GREATLY EXAGGERATED FOR EXPLANATION

1. THE O-RING IS INSTALLED IN AN ANNULAR GROOVE AND COMPRESSED AT BOTH DIAMETERS.

2. WHEN PRESSURE IS APPLIED, THE O-RING IS FORCED AGAINST A THIRD SURFACE CREATING A POSITIVE SEAL.
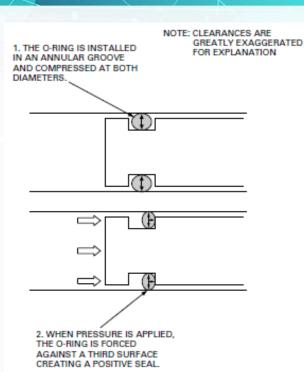
Figure 11-25. O-ring operation. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Figure 11-35. Air-cooled heat exchanger. *(Courtesy of American Standard, Heat Transfer Division, Buffalo, New York.)*

Figure 11-36. Water-cooled shell and tube heat exchanger. *(Courtesy of American Standard, Heat Transfer Division, Buffalo, New York.)*

PRESSURE GAGES
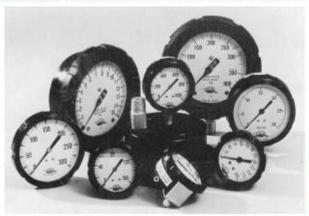
Figure 11-37. Bourdon gages with different pressure ranges. (*Courtesy of Span Instruments, Inc., Plano, Texas.*)

TUBE TENDS TO STRAIGHTEN UNDER PRESSURE CAUSING POINTER TO ROTATE.

BOURDON TUBE

PRESSURE INLET

Figure 11-38. Operation of Bourdon gage. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

INLET

CONNECTING LINK

psig

PIVOT

PISTON

SLEEVE MOVES LINKAGE WHEN PRESSURE IS APPLIED
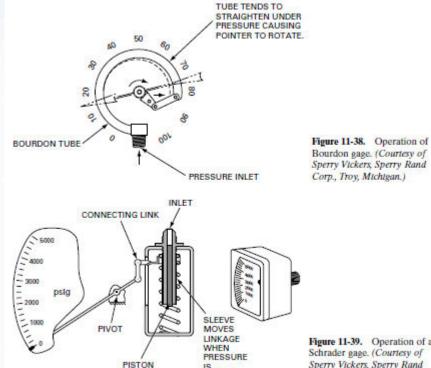
Figure 11-39. Operation of a Schrader gage. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Figure 11-40. Rotameter. *(Courtesy of Fischer & Porter Co., Worminster, Pennsylvania.)*
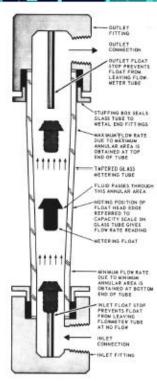
Figure 11-41. Operation of rotameter. *(Courtesy of Fischer & Porter Co., Worminster, Pennsylvania.)*

Figure 11-42. Sight flow indicator. *(Courtesy of Fischer & Porter Co., Worminster, Pennsylvania.)*

Figure 11-45. Digital Electronic Readout. *(Courtesy of Flo-Tech, Inc., Mundelein, Illinois.)*

Minggu 4

Newton's law of motion: $F = ma$

Work: $W = FS$

Mechanical power (linear): $power = Fv$

Mechanical horsepower (linear): $HP = \dfrac{F(lb) \times v(ft/s)}{550}$

Torque: $T = FR$

Mechanical power (rotary)

Brake horsepower: $HP = \dfrac{T(in \cdot lb) \times N(rpm)}{63,000}$

Brake power (metric units): $Power(kW) = \dfrac{T(N \cdot m) \times \omega(rad/s)}{1000}$
$$= \dfrac{T(N \cdot m) \times N(rpm)}{9550}$$

Efficiency: $\eta = \dfrac{output\ power}{input\ power}$

Force multiplication ratio in hydraulic jack: $\dfrac{F_2}{F_1} = \dfrac{A_2}{A_1}$

Potential energy due to elevation: $EPE = WZ$

Potential energy due to pressure: $PPE = W\dfrac{P}{\gamma}$

Kinetic energy: $KE = \dfrac{1}{2}\dfrac{W}{g}v^2$

Continuity equation: $Q_1 = A_1 v_1 = A_2 v_2 = Q_2$

Hydraulic power

English units: $hydraulic\ power\ (ft \cdot lb/s) = p(lb/ft^2) \times Q(ft^3/s)$

Metric units: $hydraulic\ power\ (W) = p(N/m^2) \times Q(m^3/s)$

Hydraulic horsepower: $HHP = \dfrac{p(psi) \times Q(gpm)}{1714}$

Bernoulli's equation: $Z_1 + \dfrac{p_1}{\gamma} + \dfrac{v_1^2}{2g} = Z_2 + \dfrac{p_2}{\gamma} + \dfrac{v_2^2}{2g}$

Energy equation (modified Bernoulli's equation): $Z_1 + \dfrac{p_1}{\gamma} + \dfrac{v_1^2}{2g} + H_p - H_m - H_L = Z_2 + \dfrac{p_2}{\gamma} + \dfrac{v_2^2}{2g}$

Pump head

Special English units: $H_p(ft) = \dfrac{3950 \times (HHP)}{Q(gpm) \times (SG)}$

Metric units: $H_p(m) = \dfrac{pump\ hydraulic\ power(W)}{\gamma(N/m^3) \times Q(m^3/s)}$

Fluid velocity: $v = \dfrac{Q}{A}$ (3-22 an

Energy: $Energy = FS$
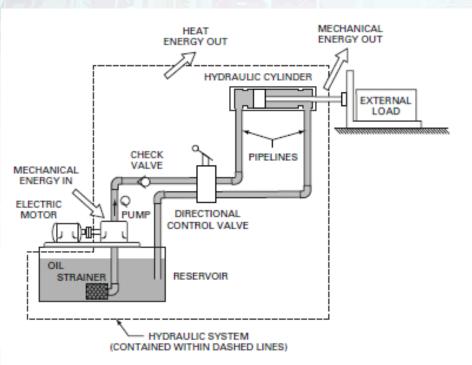
Figure 4-1. Energy transfer into and out of a hydraulic system includes energy loss (in the form of heat) due to frictional fluid flow through pipes, valves, and fittings.

Darcy's equation:

$$H_L = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$

Laminar friction factor:

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

Hagen-Poiseuille equation:

$$H_L = \frac{64}{N_R}\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$

Relative roughness:

$$\text{relative roughness} = \frac{\varepsilon}{D}$$

Head loss in valves and fittings:

$$H_L = K\left(\frac{v^2}{2g}\right)$$

Equivalent length of valves and fittings:

$$L_e = \frac{KD}{f}$$

Figure 4-3. Velocity profile in pipe.
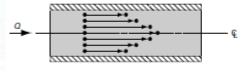
Figure 4-4. Straight-line path of fluid particles in laminar flow.
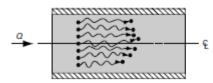
Figure 4-5. Random fluctuation of fluid particles in turbulent flow.

(a)   (b)
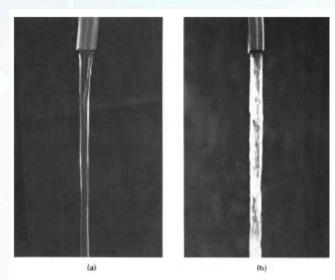
Figure 4-6. Flow patterns from water faucet. (a) Laminar flow, (b) turbulent flow. (*Reprinted from Introduction to Fluid Mechanics by J. E. John and W. L. Haberman, Prentice Hall, Englewood Cliffs, NJ, 1988.*)

Reynolds number (N_R) and (as Reynolds discovered from his tests) has the following significance:

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 critical zone between laminar and turbulent flow.

$$N_R = \frac{v\,(\text{ft/s}) \times D\,(\text{ft}) \times \rho\,(\text{slugs/ft}^3)}{\mu\,(\text{lb}\cdot\text{s/ft}^2)} = \frac{v\,(\text{ft/s}) \times D\,(\text{ft})}{\nu\,(\text{ft}^2/\text{s})}$$

$$N_R = \frac{v\,(\text{m/s}) \times D\,(\text{m}) \times \rho\,(\text{kg/m}^3)}{\mu\,(\text{N}\cdot\text{s/m}^2)} = \frac{v\,(\text{m/s}) \times D\,(\text{m})}{\nu\,(\text{m}^2/\text{s})}$$

Darcy's equation:

$$H_L = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$

Laminar friction factor:

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

Hagen-Poiseuille equation:

$$H_L = \frac{64}{N_R}\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$

$$\text{relative roughness} = \frac{\varepsilon}{D}$$
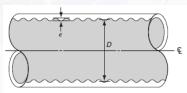
Figure 4-8. Pipe absolute roughness (ε).

TYPE OF PIPE	ABSOLUTE ROUGHNESS	
	ε (in)	ε (mm)
GLASS OR PLASTIC	SMOOTH	SMOOTH
DRAWN TUBING	0.00006	0.0015
COMMERCIAL STEEL OR WROUGHT IRON	0.0018	0.046
ASPHALTED CAST IRON	0.0048	0.12
GALVANIZED IRON	0.006	0.15
CAST IRON	0.0102	0.26
RIVETED STEEL	0.072	1.8

Figure 4-9. Typical values of absolute roughness.

Darcy's equation:

$$H_L = f\left(\frac{L}{D}\right)\left(\frac{v^2}{2g}\right)$$
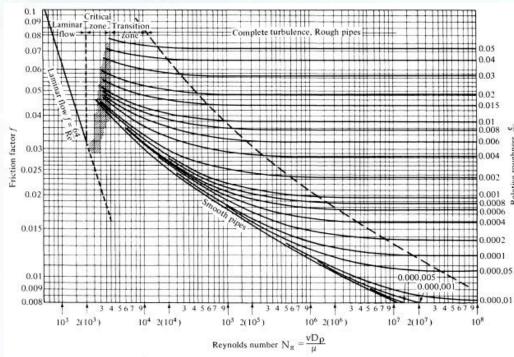
Figure 4-10. The Moody diagram. *(Reprinted from Introduction to Fluid Mechanics by J. E. John and W. L. Haberman, Prentice Hall, Englewood Cliffs, NJ, 1988.)*

$$H_L = K\left(\frac{v^2}{2g}\right) \qquad L_e = \frac{KD}{f}$$

Figure 4-12. Globe valve. (Courtesy of Crane Co., New York, New York.)

Figure 4-13. Gate valve. (Courtesy of Crane Co., New York, New York.)

Figure 4-14. 45° elbow. (Courtesy of Crane Co., New York, New York.)

Figure 4-15. 90° elbow. (Courtesy of Crane Co., New York, New York.)

Figure 4-16. Tee. (Courtesy of Crane Co., New York, New York.)

Figure 4-17. Return bend. (Courtesy of Crane Co., New York, New York.)

Figure 4-18. Ball check valve. (Courtesy of Crane Co., New York, New York.)

VALVE OR FITTING		K FACTOR
GLOBE VALVE:	WIDE OPEN	10.0
	1/2 OPEN	12.5
GATE VALVE:	WIDE OPEN	0.19
	3/4 OPEN	0.90
	1/2 OPEN	4.5
	1/4 OPEN	24.0
RETURN BEND		2.2
STANDARD TEE		1.8
STANDARD ELBOW?		0.9
45° ELBOW		0.42
90° ELBOW		0.75
BALL CHECK VALVE		4.0

Figure 4-11. K factors of common valves and fittings.

Minggu 5
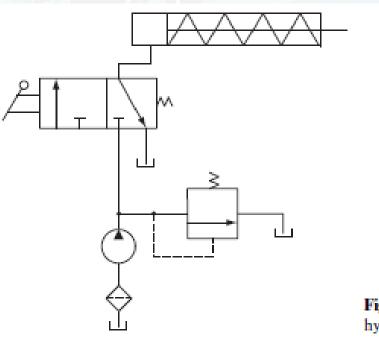
Figure 9-2. Control of single-acting hydraulic cylinder.

Figure 9-3. Control of a double-acting hydraulic cylinder. (This circuit is simulated on the CD included with the textbook.)

(a) Complete circuit.

(b) Partial circuit showing flow paths during cylinder extension stroke.
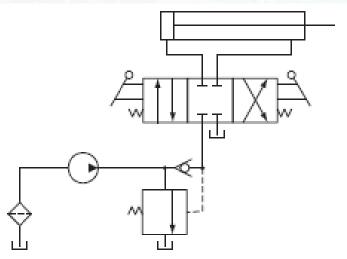
Figure 9-4. Regenerative cylinder circuit.

Figure 9-6. Pump-unloading circuit.

Figure 9-7. Double-pump hydraulic system.

Figure 9-8. Counterbalance valve application.

Figure 9-9. Hydraulic cylinder sequence circuit. (This circuit is simulated on the CD included with this textbook.)

Figure 9-10. Automatic cylinder reciprocating system.

Figure 9-11. Locked cylinder using pilot check valves.

Minggu 6
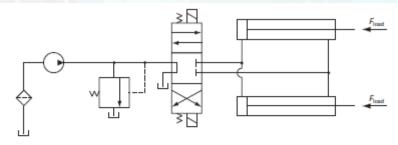
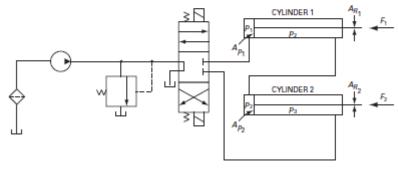**Figure 9-12.** Cylinders hooked in parallel will not operate in synchronization.

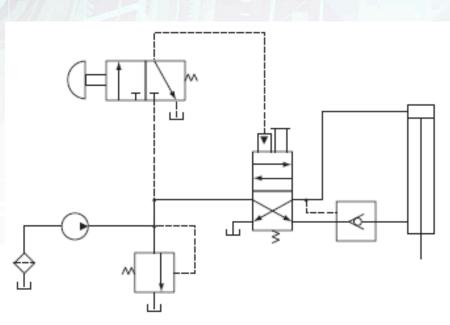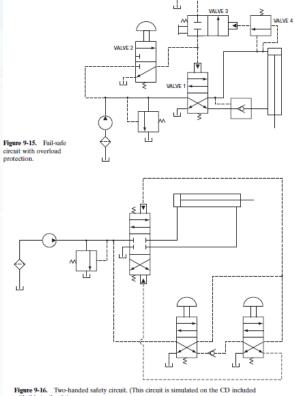**Figure 9-13.** Cylinders hooked in series will operate in synchronization.

Figure 9-14. Fail-safe circuit.

Figure 9-15. Fail-safe circuit with overload protection.

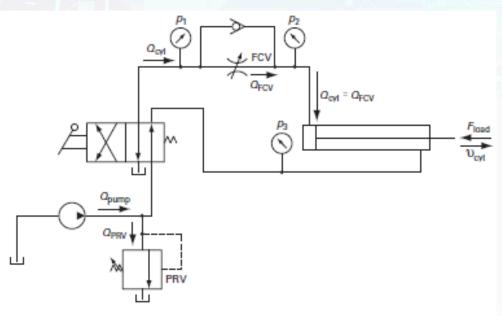**Figure 9-16.** Two-handed safety circuit. (This circuit is simulated on the CD included with this textbook.)

Figure 9-17. Meter-in speed control of hydraulic cylinder during extending stroke using flow control valve. (*DCV is in manually actuated position.*)
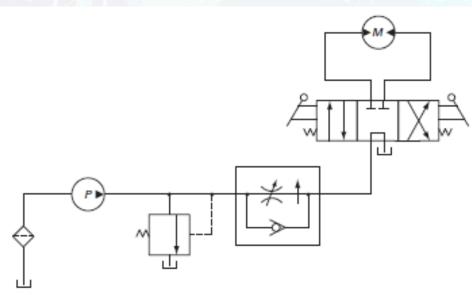
Figure 9-20. Speed control of hydraulic motor using pressure-compensated flow control valve.

Figure 9-21. Hydraulic motor braking system.

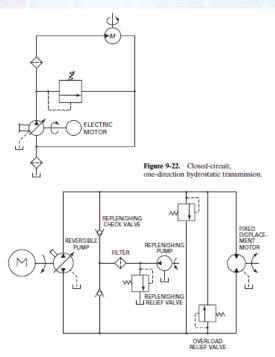**Figure 9-22.** Closed-circuit, one-direction hydrostatic transmission.

Figure 9-23. Closed-circuit, reversible-direction hydrostatic transmission. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*
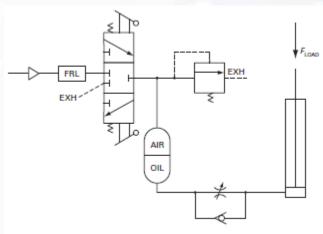
Figure 9-25. Air-over-oil circuit.

EXAMPLE 9-6

The system shown in Figure 9-26 contains a pump delivering high-pressure oil to a hydraulic motor, which drives an external load via a rotating shaft. The following data are given:
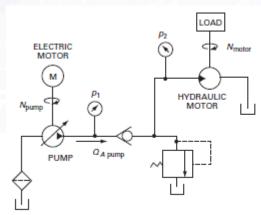

Figure 9-26. System for Example 9-6.

Pump:

$$\eta_m = 94\%$$
$$\eta_v = 92\%$$
$$V_D = 10 \text{ in}^3$$
$$N = 1000 \text{ rpm}$$
$$\text{inlet pressure} = -4 \text{ psi}$$

Hydraulic motor:

$$\eta_m = 92\%$$
$$\eta_v = 90\%$$
$$V_D = 8 \text{ in}^3$$
$$\text{inlet pressure } p_2 \text{ required to drive load} = 500 \text{ psi}$$
$$\text{motor discharge pressure} = 5 \text{ psi}$$

Pump discharge pipeline:

Pipe: 1-in schedule 40, 50 ft long (point 1 to point 2)

Fittings: two 90° elbows ($K = 0.75$ for each elbow), one check valve ($K = 4.0$)

Oil:

$$\text{viscosity} = 125 \text{ cS}$$
$$\text{specific gravity} = 0.9$$

If the hydraulic motor is 20 ft above the pump, determine the

a. Pump flow rate
b. Pump discharge pressure p_1
c. Input hp required to drive the pump
d. Motor speed
e. Motor output hp
f. Motor output torque
g. Overall efficiency of system

Solution

a. To determine the pump's actual flow rate, we first calculate the pump's theoretical flow rate.

$$(Q_T)_{pump} = \frac{(V_D)_{pump} \times N_p}{231} = \frac{10 \text{ in}^3 \times 1000 \text{ rpm}}{231} = 43.3 \text{ gpm}$$

$$(Q_A)_{pump} = (Q_T)_{pump} \times (\eta_v)_{pump} = 43.3 \times 0.94 = 40.7 \text{ gpm}$$

b. To obtain the pump discharge pressure p_1 we need to calculate the frictional pressure loss $(p_1 - p_2)$ in the pump discharge line. Writing the energy equation between points 1 and 2, we have

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

Since there is no hydraulic motor or pump between stations 1 and 2, $H_m = H_p = 0$. Also, $v_1 = v_2$ and $Z_2 - Z_1 = 20$ ft. Also, per Figure 10-2 for 1-in schedule 40 pipe, the inside diameter equals 1.040 in. We next solve for the Reynolds number, friction factor, and head loss due to friction.

$$v = \frac{(Q_A)_{pump}}{A} = \frac{40.7/449 \text{ ft}^3/s}{\frac{\pi}{4}\left(\frac{1.040}{12}\text{ft}\right)^2} = \frac{0.0908 \text{ ft}^3/s}{0.00590 \text{ ft}^2} = 15.4 \text{ ft/s}$$

$$N_R = \frac{7740v\,(\text{ft/s}) \times D(\text{in})}{v(\text{cS})} = \frac{7740 \times 15.4 \times 1.040}{125} = 992$$

Since the flow is laminar, the friction factor can be found directly from the Reynolds number.

$$f = \frac{64}{N_R} = \frac{64}{992} = 0.0645$$

Also,

$$H_L = f\left(\frac{L_{eTOT}}{D}\right)\frac{v^2}{2g} \quad \text{where}$$

$$L_{eTOT} = 50 + 2\left(\frac{KD}{f}\right)_{90^\circ \text{ elbow}} + \left(\frac{KD}{f}\right)_{check\ valve}$$

$$= 50 + \frac{2 \times 0.75 \times 1.040/12}{0.0645} + \frac{4.0 \times 1.040/12}{0.0645} = 50 + 2.02 + 5.37 = 57.39 \text{ ft}$$

Thus,

$$H_L = 0.0645 \times \frac{57.39}{1.040/12} \times \frac{(15.4)^2}{2 \times 32.2} = 157.3 \text{ ft}$$

Next, we substitute into the energy equation to solve for $(p_1 - p_2)/\gamma$.

$$\frac{p_1 - p_2}{\gamma} = (Z_2 - Z_1) + H_L = 20 \text{ ft} + 157.3 \text{ ft} = 177.3 \text{ ft}$$

Hence,

$$p_1 - p_2 = 177.3 \text{ ft} \times \gamma\left(\frac{\text{lb}}{\text{ft}^3}\right) = 177.3 \text{ ft} \times 0.9 \times 62.4\frac{\text{lb}}{\text{ft}^3} = 9960\frac{\text{lb}}{\text{ft}^2} = 69 \text{ psi}$$

$$p_1 = p_2 + 69 = 500 + 69 = 569 \text{ psi}$$

c. $$\text{hp delivered to pump} = \frac{\text{pump hydraulic horsepower}}{(\eta_o)_{pump}}$$

$$= \frac{(569 + 4)\text{psi} \times 40.7 \text{ gpm}}{1714 \times 0.94 \times 0.92} = 15.7 \text{ hp}$$

d. To obtain the motor speed we first need to determine the motor theoretical flow rate

$$(Q_T)_{motor} = (Q_A)_{pump} \times (\eta_v)_{motor} = 40.7 \times 0.90 = 36.6 \text{ gpm}$$

$$N_{motor} = \frac{(Q_T)_{motor} \times 231}{(V_D)_{motor}} = \frac{36.6 \times 231}{8} = 1057 \text{ rpm}$$

e. To obtain the motor output hp we first need to determine the motor input hp.

$$\text{motor input hp} = \frac{(500 - 5)\text{psi} \times 40.7 \text{ gpm}}{1714} = 11.8 \text{ hp}$$

Thus,

$$\text{motor output hp} = \text{motor input hp} \times (\eta_o)_{motor} = 11.8 \times 0.92 \times 0.90 = 9.77 \text{ hp}$$

f. $$\text{motor output torque} = \frac{\text{motor output hp} \times 63,000}{N_{motor}}$$

$$= \frac{9.77 \times 63,000}{1057} = 582 \text{ in} \cdot \text{lb}$$

g. The overall efficiency of the system is

$$(\eta_o)_{overall} = \frac{\text{motor output hp}}{\text{pump input hp}} = \frac{9.77}{15.7} = 0.622 = 62.2\%$$

Thus, 62.2% of the power delivered to the pump by the electric motor is delivered to the load by the hydraulic motor.

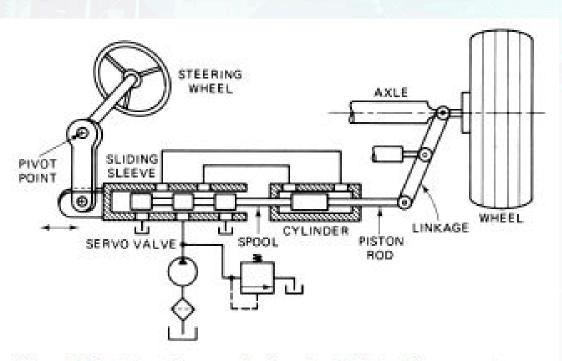**Figure 9-27.** Automotive example of mechanical-hydraulic servo system.

Minggu 7

The following is a list of the most common causes of hydraulic system breakdown:

1. Clogged or dirty oil filters

2. Inadequate supply of oil in the reservoir

3. Leaking seals

4. Loose inlet lines that cause the pump to take in air

5. Incorrect type of oil

6. Excessive oil temperature

7. Excessive oil pressure

Sampling and testing of the fluid

Figure 12-1. Hydraulic fluid test kit. *(Courtesy of Gulf Oil Corp., Houston, Texas.)*

Figure 12-2. Rust caused by moisture in the oil. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Figure 12-3. Corrosion caused by acid formation in the hydraulic oil. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

The following are the usual characteristics tested for in order to determine the flammability of a hydraulic fluid:

Flash point. the temperature at which the oil surface gives off sufficient vapors to ignite when a flame is passed over the surface

Fire point. the temperature at which the oil will release sufficient vapor to support combustion continuously for five seconds when a flame is passed over the surface

Autogenous ignition temperature (AIT). the temperature at which ignition occurs spontaneously

There are basically four different types of fire-resistant hydraulic fluids in common use:

1. Water-glycol solutions.

2. Water-in-oil emulsions.

3. Straight synthetics.

4. High-water-content fluids.

1. MICROSCOPIC IMPERFECTIONS OF THE MATING PARTS ARE SEPARATED ...

× 100

2. BY A FILM OF FLUID ...

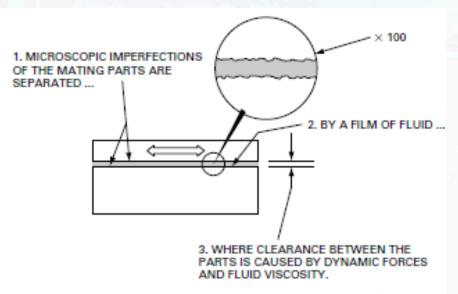3. WHERE CLEARANCE BETWEEN THE PARTS IS CAUSED BY DYNAMIC FORCES AND FLUID VISCOSITY.

Figure 12-4. Lubricating film prevents metal-to-metal contact. *(Courtesy Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

The following recommendations should be adhered to for properly maintaining and disposing of hydraulic fluids:

1. Select the optimum fluid for the application involved.This includes consideration of the system operating pressures and temperatures, as well as the desired fluid properties of specific gravity, viscosity, lubricity, oxidation resistance, and bulk modulus.

2. Use a well-designed filtration system to reduce contamination and increase the useful life of the fluid. Filtration should be continuous, and filters should be changed at regular intervals.

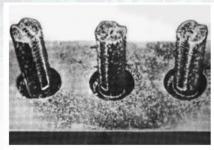**3.** Follow proper storage procedures for the unused fluid supply.

Figure 12-5. Magnetic plugs trap iron and steel particles. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)
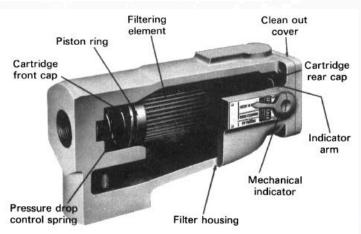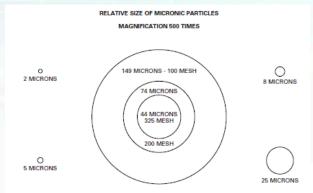

Figure 12-8. Cutaway view of a Tell-Tale filter. (*Courtesy of Parker Hannifin Corp., Hazel Park, Michigan.*)

Figure 12-7. Relative and absolute sizes of micronic particles. (*Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.*)

Code No.	No. of Particles per Milliliter	Code No.	No. of Particles per Milliliter
30	10,000,000	14	160
29	5,000,000	13	80
28	2,500,000	12	40
27	1,300,000	11	20
26	640,000	10	10
25	320,000	9	5
24	160,000	8	2.5
23	80,000	7	1.3
22	40,000	6	0.64
21	20,000	5	0.32
20	10,000	4	0.16
19	5,000	3	0.08
18	2,500	2	0.04
17	1,300	1	0.02
16	640	0.9	0.01
15	320	0.8	0.005

Figure 12-10. ISO code numbers for fluid cleanliness levels.

Component	ISO Code
Servo Valves	14/11
Vane and Piston Pumps/Motors	16/13
Directional and Pressure Control Valves	16/13
Gear Pumps/Motors	17/14
Flow Control Valves and Cylinders	18/15

Figure 12-11. Typical fluid cleanliness levels required for hydraulic components.
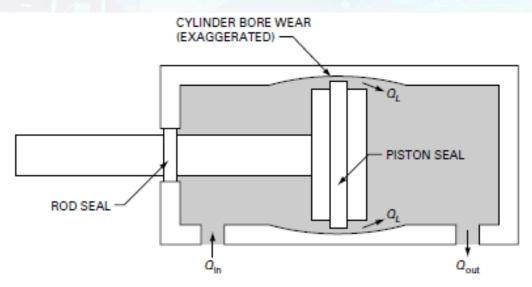
Figure 12-12. Hydraulic cylinder with a bore that is worn due to solid-particle contamination of the fluid.

The End

SISTEM HIDROLIK DAN PNEUMATIK