CHAPTER 60
HYDRAULIC SYSTEMS

Hugh R. Martin
University of Waterloo
Waterloo, Ontario, Canada

60.1 HYDRAULIC FLUIDS

One of the results of the study of fluid mechanics has been the development of the use of hydraulic oil, a so-called incompressible fluid, for performing useful work. Fluids have been used to transmit power for many centuries, the most available fluid being water. While water is cheap and usually readily available, it does have the distinct disadvantages of promoting rusting, of freezing to a solid, and of having relatively poor lubrication properties.

Mineral oils have provided superior properties. Much of the success of modern hydraulic oils is due to the relative ease with which their properties can be altered by the use of additives, such as rust and foam inhibitors, without significantly changing fluid characteristics.

Although hydraulic oil is used mainly to transmit fluid power, it must also 1) provide lubrication for moving parts, such as spool valves, 2) absorb and transfer heat generated within the system, and 3) remain stable, both in storage and in use, over a wide range of possible physical and chemical changes.

It is estimated that 75% of all hydraulic equipment problems are directly related to the improper use of oil in the system. Contamination control in the system is a very important aspect of circuit design.

In certain industries, such as mining and nuclear power, it is critically important to control the potential for fire hazards. Hence, fire-resistant fluids have been playing an ever-increasing role in these types of industry. The higher pressure levels in modern fluid power circuits have made fire hazards more serious when petroleum oil is used, since a fractured component or line will result in a fine mist of oil that can travel as far as 40 ft and is readily ignited. The term *fire-resistant fluid*

Mechanical Engineers' Handbook, 2nd ed., Edited by Myer Kutz.
ISBN 0-471-13007-9

(FRF) generally relates to those liquids that fall into two broad classes: a) those where water provides the fire resistance, and b) those where a fire retardant is inherent in the chemical structure.[1–4] Fluids in the first group are water/glycol mixtures, water in oil emulsions (40–50% water), and oil in water emulsions (5–15% water). The second group are synthetic materials, in particular chlorinated hydrocarbons and phosphate esters.

A disadvantage with water-based fluids is that they are limited to approximately 50–60°C operating temperature because of evaporation. The high vapor pressure indicates this group is more prone to cavitation than mineral oils. Synthetic fluids such as the phosphate esters do not have this problem and also have far superior lubrication properties. Some typical characteristics of these various types of fluids are shown in Table 60.1.

Of all the physical properties that can be listed for hydraulic fluids, the essential characteristics of immediate interest to a designer are 1) bulk modulus, to assess system rigidity and natural frequency, 2) viscosity, to assess pipe work and component pressure losses, 3) density, to measure flow and pressure drop calculations, and 4) lubricity, to determine threshold and control accuracy assessments. The first three items are discussed in separate sections, as they relate directly to circuit design. *Lubricity,* the final item, is difficult to define, as it is very much a qualitative judgment. Lubricity affects the performance of a system, since it is a major factor in determining the level of damping in the system, that is, *viscous* or velocity-dependent *damping.* It also affects the accuracy of operation of a system because of its influence on the other type of friction, *coulomb friction,* which is velocity-independent.

Oil film strength is often referred to as the *anti-wear value* of a lubricant, which is the ability of the fluid to maintain a film between moving parts and thus prevent metal-to-metal contact. These characteristics are important for the moving parts in valves, cylinders, and pumps.[5]

60.2 CONTAMINATION CONTROL

There is little doubt that component failure or damage due to fluid contamination is an area of major concern to both the designer and user of fluid power equipment. Sources of contamination in fluid power equipment are many. Although oil is refined and blended under relatively clean conditions, it does accumulate small particles of debris during storage and transportation. It is not unusual for hydraulic oil circulating in a well maintained hydraulic circuit to be cleaner than that from a newly purchased drum. New components and equipment invariably have a certain amount of debris left from the manufacturing process, in spite of rigorous post-production flushing of the unit.

The contaminant level in a system can be increased internally due to local burning (oxidation) of oil to create sludges. This can be a result of running the oil temperature too high (normally 40–60°C is recommended) or due to local cavitation in the fluid.

The trend towards the use of higher system pressures in hydraulics generally results in narrower clearances between mating components. Under such design conditions, quite small particles in the range of 2–20 microns can block moving surfaces.

Extensive work on contamination classification has been carried out by Fitch and his co-workers.[6]

To take a specific example, consider the piston pump shown in Fig. 60.1. Component parts of the pump are loaded towards each other by forces generated by the pressure, and this same pressure always tends to force oil through the adjacent clearance. The life of the pump is related to the rate at which a relatively small amount of material is being worn away from a few critical surfaces. It is logical to assume, therefore, if the fluid in a clearance is contaminated with particles, rapid degradation and eventual failure can occur.

Although the geometric clearances are fixed, the actual clearances vary with eccentricity due to load and viscosity variations. Some typical clearances between moving parts are shown in Table 60.2.

Contamination control is the job of filtration. System reliability and life are related not only to the contamination level but also to contaminant size ranges. To maintain contaminant levels at a magnitude compatible with component reliability requires both the correct filter specification and suitable placement in the circuit. Filters can be placed in the suction line, pressure line, return line,

Table 60.1 Comparison of Some Hydraulic Fluids

Property	Units	FRF (Ester)	Mineral Oil	Water in Oil
Density (38 C)	kg m^{-3}	1136.0	858.2	980.0
Viscosity (38 C)	$m^2\ s^{-1}$	4.6×10^{-6}	4.0×10^{-5}	0.15×10^{-5}
(99 C)		4.9×10^{-6}	5.8×10^{-6}	
Bulk modulus (38 C and 34.5 MPa)	N m^{-2}	2.25×10^9	1.38×10^9	2.18×10^9
Vapor pressure	kPa (abs)	6×10^{-5}	6×10^{-5}	1.0

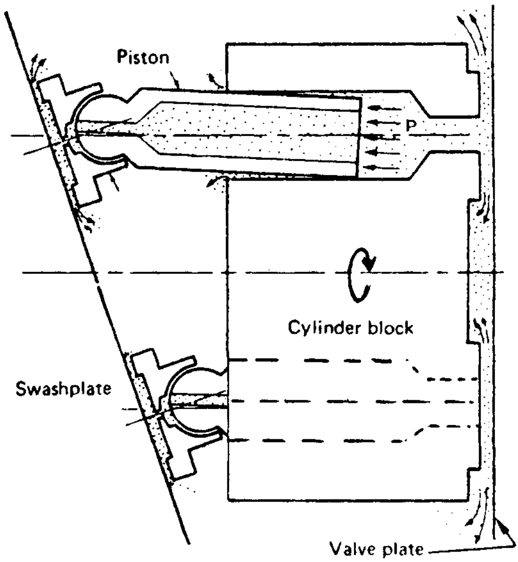


Fig. 60.1 Piston pump clearances.

or in a partial flow mode. To use a broad approach of just inserting a filter with a very low rating is unsatisfactory from the aspects of both cost and high pressure loss. The optimization of choice can be approached using simple computer modeling, as described by Foord.[7]

Dirt in hydraulic systems consists of many different types of material, ranging in size from less than 1 micron to greater than 100 microns. Since most general industrial hydraulics operating below 14 MPa are able to tolerate particles up to 25 microns, a 25-micron-rated filter is satisfactory. Equipment operating at pressures in the 14–21 MPa range should have 10–15-micron-rated filters, while high pressure pumps and precision servo valves need 5 micron-rated filtration. A good practical reference for filter selection has been written by Spencer.[8]

The size distribution of particles is of course random, and, generally speaking, the smaller the size range the greater the number of particles per 100 ml of fluid. Filters are not capable of removing all the contaminants, but for example, a 10-micron filter is one capable of removing about 98% of all particles exceeding 10 microns of a standard contaminant in a given concentration of prepared solution.

60.3 POSITIVE ASPECTS OF CONTAMINATION

Contamination buildup in a system can be used as a diagnostic tool. Regular sampling of the oil and examination of the particles can often give a clue to potential failure of components. In other words, this is a preventive maintenance tool. Many methods can be used for this type of examination, such as spectrochemical[9] or Ferrographic[10] methods. Sampling of the oil can be taken at any time and does not interfere with the operation of the equipment.

Table 60.3 shows the normally expected contaminant levels in parts per million (ppm); levels rising above these values and particularly rates of change of levels are indicative of potential failures.

Table 60.2 Typical Clearances in Pumps

Component	Clearance Range (micron)
Spool to sleeve in valve	1–10 diametrical
Gear pump tip to casing	0.5–5
Piston to bore	5–40
Valve plate to body of pump	0.5–5

Table 60.3 Some Typical Normal Contaminant Levels

Material	Source in System	Max Level (ppm)
Iron	Bearings, gears, or pipe rust. Pistons and valve wear.	20
Chromium	Alloyed with bearing steel	4
Aluminum	Air cooler equipment	10
Copper	Bronze or brass in bearings. Connectors. Oil temperature sensor bulb. Cooler core tubes.	30
Lead	Usually alloyed with copper or tin. Bearing cage metal.	20
Tin	Bearing cages and retainers	15
Silver	Cooling tube solder	3
Nickel	Bearing steel alloy	4
Silicon	Seals; dust and sand from poor filter or air leak	9
Sodium	Possible coolant leak into hydraulic oil	50

The Ferrographic technique allows the separation of wear debris and contaminants from the fluid and allows arrangement as a transparent substrate for examination. When wear particles are precipitated magnetically, virtually all nonmagnetic debris is eliminated. The deposited particles deposit according to size and may be individually examined. By this method it is possible to differentiate cutting wear, rubbing wear, erosion, and scuffing by the size and geometry of the particles. However, the Ferrographic method is expensive compared to other methods of analysis.[11]

60.4 DESIGN EQUATIONS—ORIFICES AND VALVES

The main controlling element in any hydraulic circuit is the orifice. The fluid equivalent of the electrical resistance, it can be fixed in size or can be variable, in the case of a spool valve. The orifice in its various configurations is also the main source of heat generation, resulting in the need for cooling techniques and a major source of noise.

The orifice equation is developed from Bernoulli's energy balance approach, which results in the following relationship:[12]

$$Q = \frac{C_c C_v A_o}{\sqrt{1 - \left(\frac{C_v A_o}{A_u}\right)^2}} \sqrt{\frac{2(p_u - p_{vc})}{\rho}} \tag{60.1}$$

where Q = volume flow rate, m^3/s
A_o = orifice area, m^2
A_u = upstream area, m^2
p_u = upstream static pressure, Pa
p_{vc} = static pressure at *Vena contracta,* Pa
C_c = contraction coefficient
C_v = velocity coefficient
ρ = mass density of hydraulic fluid, kg/m^3

These parameters are shown in Fig. 60.2, together with the static pressure distribution on either side of a sharp-edged orifice. Experimental measurements show that the actual flow is about 60% of that given by Bernoulli's equation. Hence, the need for the contraction and velocity coefficients. This results in the practical form of Eq. (60.1) for typical industrial hydraulic oil

$$Q = 3.12 \times 10^{-2} A_o \sqrt{p_u - p_d} \ \mathrm{m^3 s^{-1}} \tag{60.2}$$

The symbols have the same definition as those for Eq. (60.1). The adequacy of Eq. (60.2) is demonstrated in Table 60.4.

In the case of a variable orifice, such as that found in a spool-type valve, the orifice area is a variable. In fact, it can be seen from Fig. 60.3 that the exposed area available for oil flow is part of a circle. If the orifice, in this case called a *control orifice* or *port,* is of radius r and the spool displacement from the closed position is x, then the uncovered area is

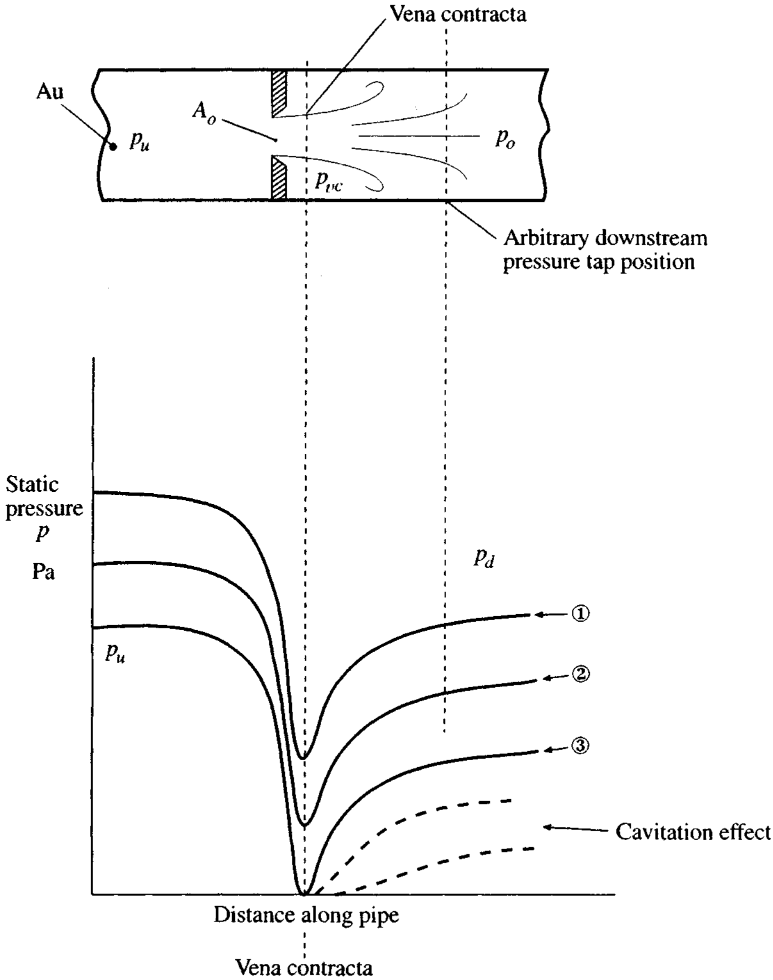


Fig. 60.2 Static pressure distribution.

$$A_o = [\theta - \cos(\theta/2)]\left(\frac{r^2}{2}\right) \tag{60.3}$$

$$\theta = 2\cos^{-1}[1 - (x/r)] \tag{60.4}$$

The area displacement characteristic plotted in Fig. 60.3 shows the nonlinear nature of the curve.

One of the significant differences between the theoretical valve and the practical valve is the *lap.* It is not economical to produce zero lapped valves, so that only at the center position is the flow through the valve zero. Normally, the valve is either overlapped or underlapped, as shown in Fig. 60.4. An overlapped valve saves fluid loss when the spool is central. This is fine for directional control valves, but it produces both accuracy and stability problems if the valve is a precision control valve within a closed-loop configuration.

An underlapped valve gives much better control and stability, at the expense of a higher leakage rate (power loss). Many more details of valve design can be found in Martin and McCloy.[12]

60.5 DESIGN EQUATIONS—PIPES AND FITTINGS

While orifices serve the important function of controlling flow in the system, pipes and fittings are necessary to transmit fluid power from the input (usually a pump) to the output (usually a ram or motor). It is important to minimize losses through these conductors as well as through other com-

Table 60.4 Comparing Experimental Data to Predictions of Eq. (60.2)

Supply Pressure = 13.78 MPa Valve overlap = ±0.0127 mm		
Valve Displacement (mm)	Calculated Flow (ml/sec)	Measured Flow (ml/sec)
+0.3810	104.140	—
+0.3048	—	79.540
+0.2540	56.088	59.860
+0.2032	40.180	45.264
+0.1270	20.172	24.272
+0.0508	4.920	6.560
+0.0254	1.968	0.820
Center	0	0
−0.0254	1.968	0.820
−0.0508	4.920	4.264
−0.1270	20.172	20.992
−0.2032	40.180	41.000
−0.2540	56.088	58.384
−0.3048	—	76.588
−0.3810	104.14	—

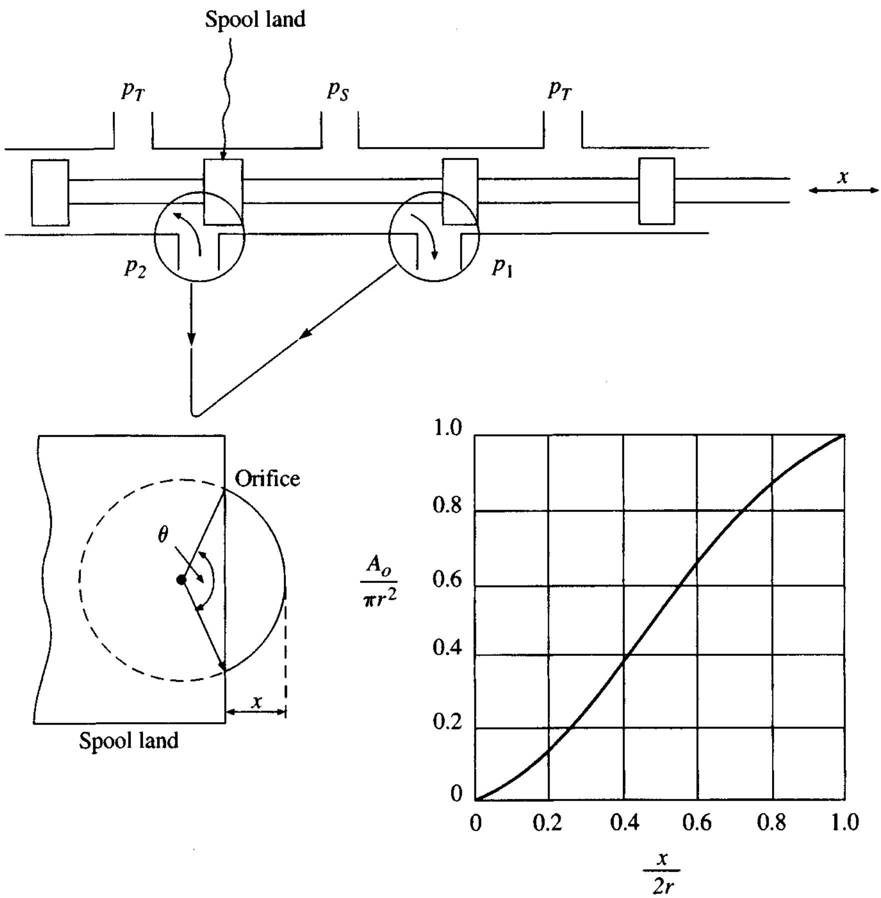


Fig. 60.3 Effective exposed orifice area for a spool-type valve.

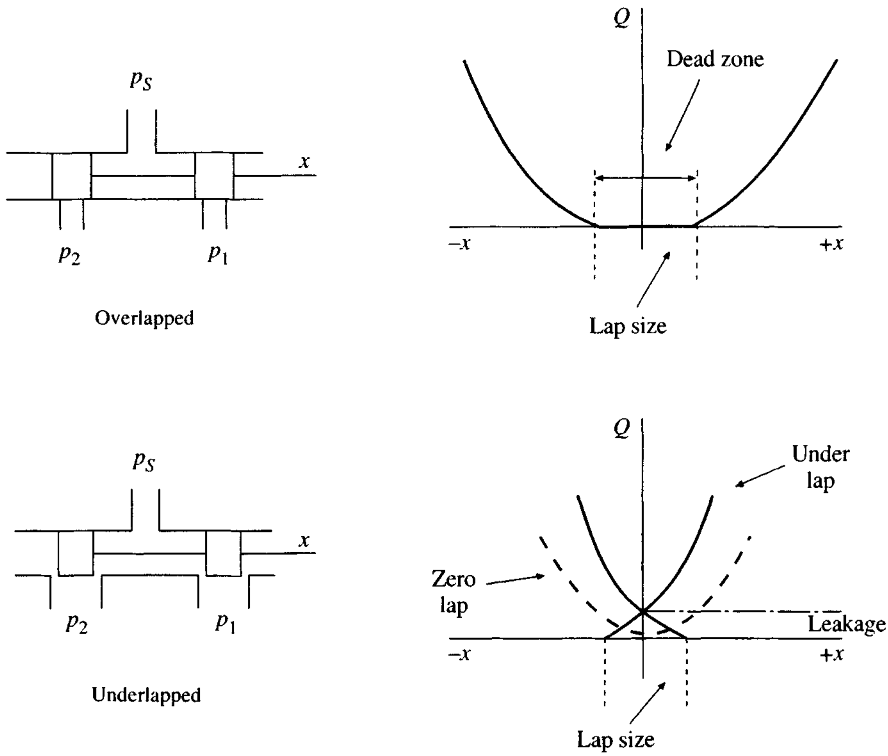


Fig. 60.4 Characteristics of valve lap.

ponents so that the maximum power is available for useful work at the circuit output. It is equally important to minimize component and piping cost. In some applications, it is also important to minimize weight and bulk size.

Pipe sizes are specified by nominal diameters, and the wall thickness by schedule number. The three schedules (or wall thicknesses) used in hydraulic piping are 40, 80, and 160, corresponding to *standard* pipe, *extra heavy,* and a little less than *double extra heavy.* The metric system of units has helped to complicate things for the designer during this transition period, more details can be found in Martin and McCloy.[12]

In the selection of piping for hydraulic circuits, the following are suggested:

- Suction lines to pumps should not carry fluid at velocities in excess of 1.5 m/sec in order to reduce the possibility of cavitation at the pump inlet.
- Delivery lines should not carry fluid at velocities in excess of 4.5 m/sec in order to prevent excessive shock loads in the pipework due to valve closure. Pressure loss due to friction in pipes should be limited to approximately 5% of the supply pressure and the recommendation also keeps heat generation to a reasonable level.
- Return lines should be of larger diameter than delivery lines to avoid back pressure buildup.

For typical industrial hydraulic oil, we can write

$$\Delta p = K_L / K_1 Q^2 \qquad (60.5)$$

where Δp = pressure drop along a straight pipe (kPa)
K_L = loss coefficient = $f\ell/d$
f = friction factor
ℓ = pipe length (m)
d = internal pipe diameter (m)
K_1 = see Table 60.5

Table 60.5 Coefficients for Eqs. 60.5 and 60.6

Nominal Bore		S.I. System		Old System		Pipe Area	
mm	in.	K_1	K_2	K_1	K_2	m^2	in.^{2}
8	¼	1.027×10^{-11}	138	12.64	10379.12	6.64×10^{-5}	0.1041
10	⅜	3.506×10^{-11}	102	42.26	7663.28	12.27×10^{-5}	0.1909
15	½	8.949×10^{-11}	81	108.08	6073.93	19.60×10^{-5}	0.3039
20	¾	2.740×10^{-10}	61	333.0	4584.95	34.30×10^{-5}	0.5333
25	1	7.195×10^{-10}	47	874.66	3601.52	55.57×10^{-5}	0.8643
32	1¼	2.181×10^{-9}	36	2620.49	2737.68	96.76×10^{-5}	1.496
40	1½	4.014×10^{-9}	31	4854.14	2340.66	13.13×10^{-4}	2.036
50	2	1.090×10^{-8}	24	13180.81	1823.64	21.65×10^{-4}	3.355

Q = flow rate (m^3/sec)

The friction factor f has been shown experimentally to be a function of Reynolds number (Re) and of pipe roughness. The Reynolds number for industrial hydraulic calculations can be calculated from

$$\mathrm{Re} = \frac{K_2}{v} Q \tag{60.6}$$

where the kinematic viscosity v has a typical value of 4.0×10^{-5} m^2/s, and K_2 can be found in Table 60.5.

Given the flow through a section of straight pipe the procedure to calculate the pressure loss is simple. Using Eq. 60.6 and v given above calculate the Reynolds number. Using the Reynolds number to calculate the appropriate value for f, calculate a value for K_L. Referring to Table 60.5 for K_1, the pressure drop can be calculated using Eq. 60.5.

Unfortunately, not all piping is in straight runs so when a bend occurs the loss of pressure will be greater. The effective bend loss can be estimated from Eq. 60.7 and Figs. 60.5 and 60.6. These results are from Ref. 13.

$$\Delta p = (K + cK_B)Q^2/K_1 \tag{60.7}$$

where c = correction factor for bend angle (Fig. 60.5)
K_B = resistance coefficient for 90° bends (Fig. 60.6)

Further useful information about circuit design can be found in Keller.[14]

60.6 HYDROSTATIC PUMPS AND MOTORS

The source of power in a hydraulic circuit is the result of *hydrostatic* flow under pressure with the energy being transmitted by static pressure. In another type of fluid power, termed *hydrokinetic,* the transmission of energy is related to the change in velocity of the hydraulic fluid. While hydrostatic systems use positive displacement pumps, hydrokinetic systems use centrifugal pumps.[15]

Positive displacement machines have been in existence for many years. The concept is simply a variable displacement volume which can take the form of a piston in a cylinder, gear teeth engaging, or the sweeping action of a vane with eccentric axis placement. All these configurations are positive displacement in the sense that for each revolution of the pump shaft, a nearly constant quantity of fluid is delivered. In addition, there is some form of valving which either takes the form of nonreturn valves or a porting arrangement on a valve plate.

Examples of different types of pumps are shown in Figs. 60.7, 60.8, and 60.9. While torque and speed are the input variables to a pump, the output variables are pressure and flow. The product of these variables will give the input and output power. The difference between these values is a measure of the fluid and mechanical losses through the machine. These factors should be taken into account even for a simple analysis.

The torque required to drive the pump at constant speed can be divided into five components:

$$T_p = T_i + T_v + T_f + T_c \tag{60.8}$$

where T_p = actual required input torque (Nm)
T_i = ideal torque due to pressure differential and physical dimensions only

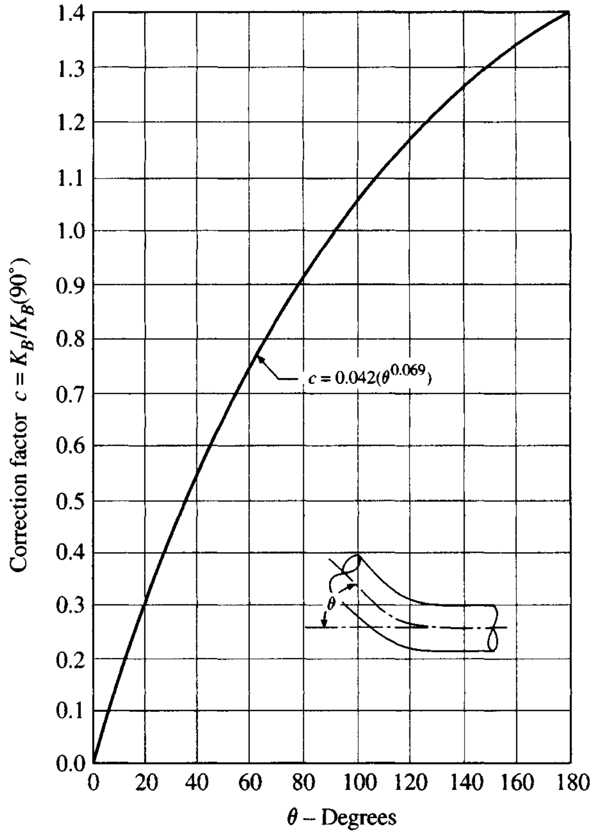


Fig. 60.5 Correction factor *c*. (Reproduced from AF Rocket Propulsion Lab., 1964.)

T_v = resisting torque due to viscous shearing of the fluid between stationary and moving parts of the pump, that is, viscous friction
T_f = resisting torque due to pressure and speed-dependent friction sources such as bearings and seals
T_c = remaining dry friction effects due to rubbing

The delivery from the pump can be expressed in a similar manner:

$$Q_p = Q_i - Q_l - Q_r \tag{60.9}$$

where Q_p = actual pump delivery (ml/sec)
Q_i = ideal delivery of a pump due to geometric shape only
Q_l = viscous leakage flow
Q_r = loss in delivery due to inlet restriction[16]

If the pump is well designed and operating under its working specification, the loss represented by Q_r should not occur.

For a hydraulic motor, the procedure is reversed in the sense that flow and pressure are the input variables, and torque with angular velocity appears at the output. The corresponding equations are therefore

$$T_p = T_i - T_v - T_f - T_c \tag{60.10}$$

$$Q_p = Q_i + Q_l \tag{60.11}$$

Q_r is not a factor in motor performance.

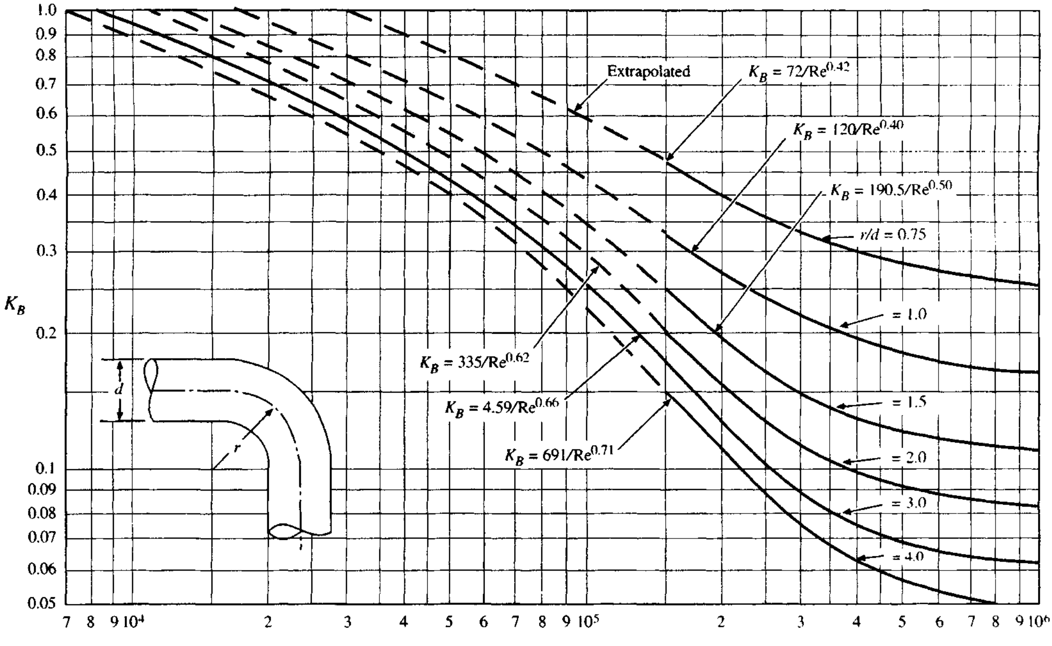


Fig. 60.6 Correction factor K_B for pressure loss in pipe bends.

The ideal positive-displacement machine displaces a given volume of fluid for every revolution of the input shaft. This value is given the name *displacement* of the pump or motor and is extensively used by manufacturers to label the pump size. Some typical characteristics for a hydraulic radial piston motor are shown in Fig. 60.10 and Table 60.6.

If the pump or motor rotates at N rpm, then

$$Q_i = D_p N \tag{60.12}$$

where D_p = swept volume per revolution = nV
V = swept volume per cylinder per revolution

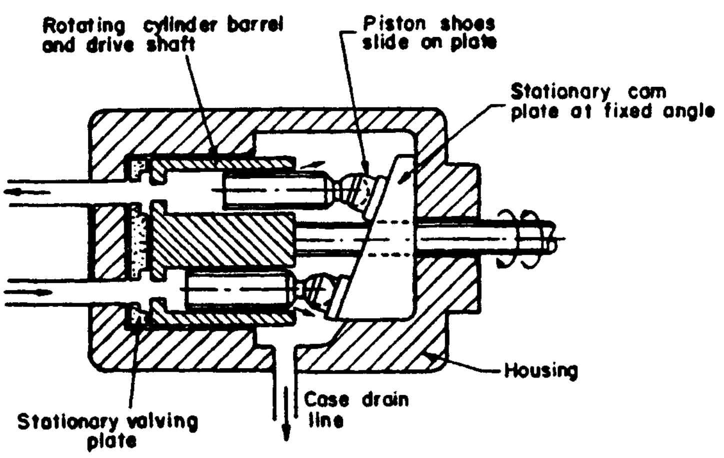


Fig. 60.7 Axial piston pump.

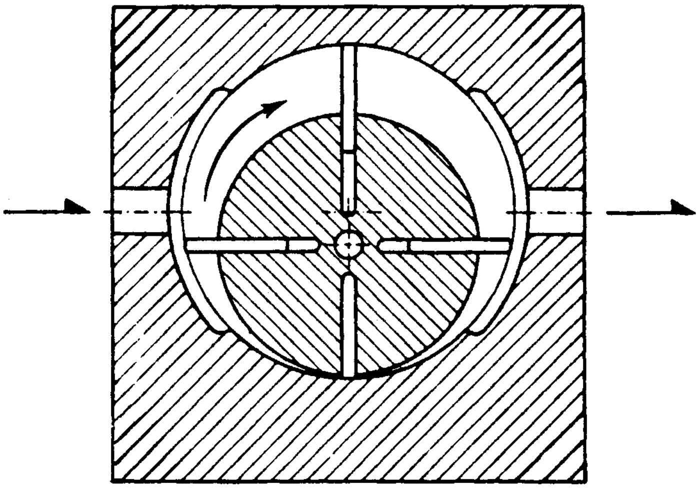


Fig. 60.8 Schematic cross section through a vane pump. (From J. Thoma, *Modern Oil Hydraulic Engineering.* © 1970 Technical and Trade Press. Reprinted with permission.)

n = number of cylinders in the pump or motor

The leakage term Q_l can be expressed in terms of a leakage coefficient C_s which is sometimes called the slip coefficient:

$$Q_l = \frac{\Delta p D_p C_s}{\mu} \tag{60.13}$$

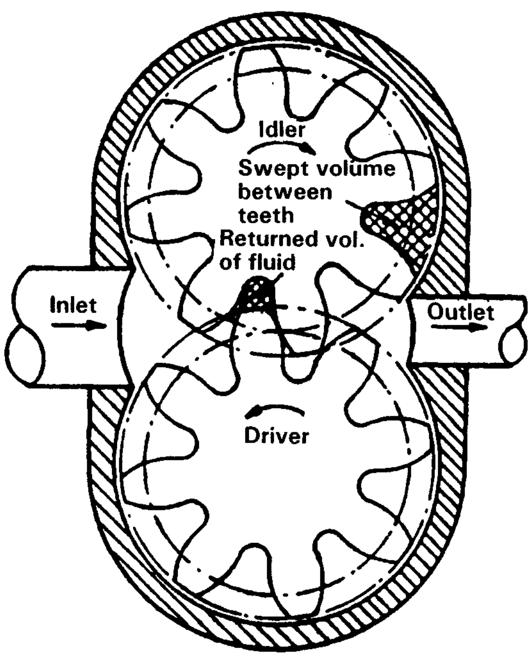


Fig. 60.9 Gear pump construction.

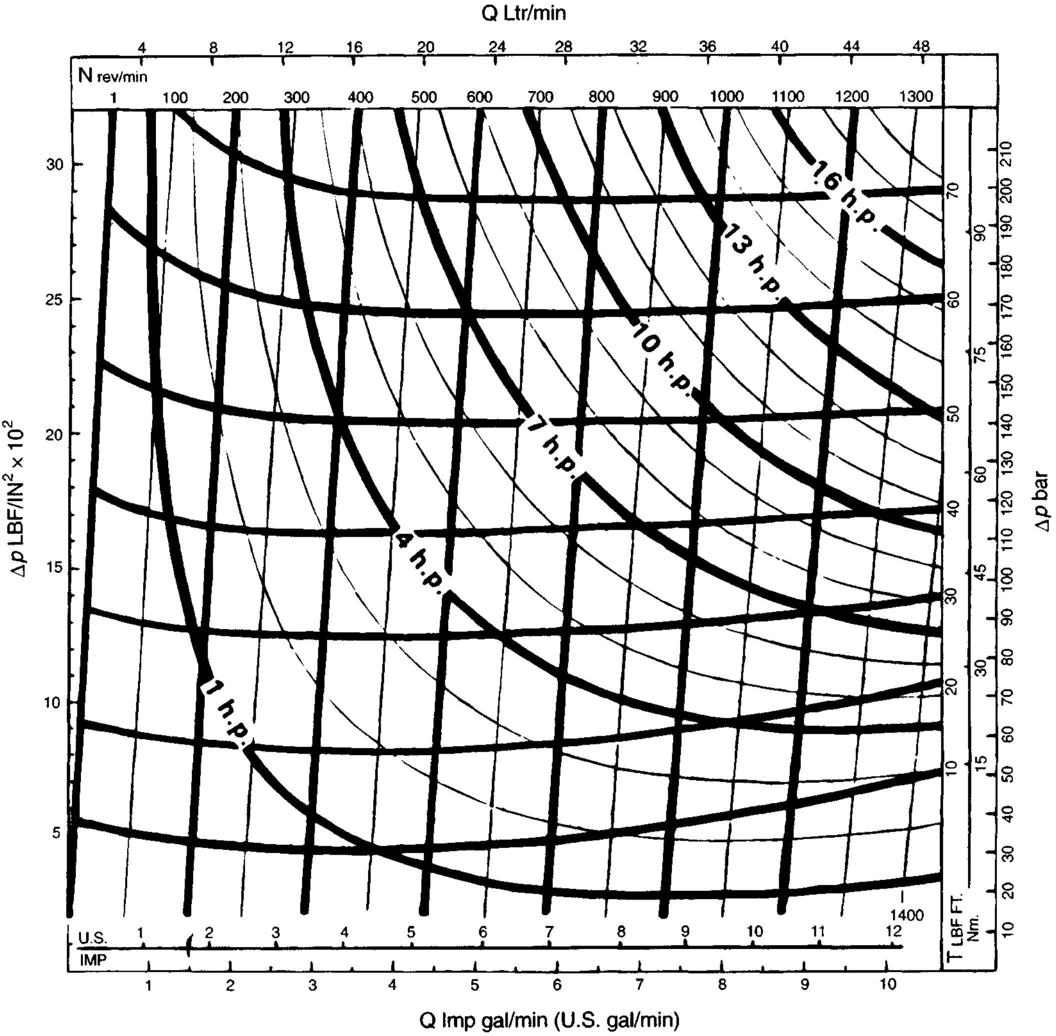


Fig. 60.10 Typical performance range for a hydraulic motor, specifications appear in Table 60.6. (Courtesy of Kontak Manufacturing, Lincolnshire, England.)

For most designs, the slip coefficient is proportional to the cube of typical clearances within the machine.[17]

While the volume of fluid theoretically pumped per revolution can be calculated from the geometry of the design, in practice, a pump does not deliver that amount. The *volumetric efficiency* η_V is used to assess this characteristic and is essentially a measure of the quality of machining or of wear in a pump.

$$\eta_V = \frac{Q_p}{Q_i} = \frac{Q_i - Q_l}{Q_i}$$

$$= 1 - \frac{Q_l}{Q_i} = 1 - \frac{\Delta p}{\mu N} C_s \tag{60.14}$$

The value of η_V can vary from about 75% up to 97%. In general, the cheaper the pump, the lower the volumetric efficiency.

The theoretical applied torque to a pump is given by

Table 60.6 Typical Performance Specification for a Radial Piston Motor[a]

Displacement	in.^{3}/rev (cm^3/rev)	1.99 (32.6)
Max. torque	lbf · ft (Nm)	74 (100)
Max. recommend speed	rev/min	1360
Max. output	hp	18
Max. recommended flow rate		10 imp.gal/min (45 ltr/min) (12 U.S.gal/min)
Max. pressure		3000 lbf/in.^{2} (207 bar)
Approx. overall efficiency		85%
Max. back pressure (reversible)		3000 lbf/in.^{2} (207 bar)
Max. drain line pressure (reversible)		50 lbf/in.^{2} (3.5 bar)
Max. back pressure (uni-directional)		50 lbf/in.^{2} (3.5 bar)
Weight		15 lb (6.8 kg)
Max. permissible shaft end load		2000 lbf (907 kgf)
Max. permissible shaft side load (¾ in. (19 mm) from shaft end)		2000 lbf (907 kgf)

[a]Acknowledgments to Kontak Manufacturing, Lincolnshire, England.

$$T_i = \Delta p D_p / 2\pi \tag{60.15}$$

In this case, the losses are assessed by the viscous drag coefficient C_d which is inversely proportional to the typical pump clearances, and by the drag coefficient C_f, which is proportional to the size of the pump. Referring to Eq. 60.10, T_c in a well-designed pump is normally small enough to ignore.

$$T_v = C_d D_p \mu N \tag{60.16}$$

$$T_f = C_f \frac{\Delta_p D_p}{2\pi} \tag{60.17}$$

Wilson[15] gives guidance as to the magnitude of these coefficients. His figures are given in Table 60.7.

The *mechanical efficiency* η_m of the device is a measure of the power wasted in friction. A reduction in the mechanical efficiency could, for example, be an indicator of bearing failure due to lack of lubrication.

$$\eta_m = \frac{T_i}{T_p} = \frac{T_i}{T_i + T_v + T_f}$$

$$\eta_m = \frac{1}{1 + \dfrac{2\pi C_d \mu N}{\Delta p} + C_f} \tag{60.18}$$

Finally, the *overall efficiency* of a pump or motor is the product of the volumetric and mechanical efficiencies. In general, gear pumps are suitable for pressures up to 17 MPa and have overall efficiencies of approximately 80%. A good-quality piston pump has an overall efficiency of 95% and is capable of operating with pressures up to 68.9 MPa.

60.7 STIFFNESS IN HYDRAULIC SYSTEMS

One important and often neglected aspect of hydraulic circuit design is the fact that in practice hydraulic oil is compressible. So far, only steady flow through the circuit has been discussed. How-

Table 60.7 Typical Hydraulic Pump Coefficients

Pump Type	D_p, in.^{3}/rev	C_d	C_s	C_f
Piston	3.600	16.8×10^4	0.15×10^{-7}	0.045
Vane	2.865	7.3×10^4	0.477×10^{-7}	0.212
Spur gear	2.965	10.25×10^4	0.48×10^{-7}	0.179
Internal gear	2.965	9.77×10^4	1.02×10^{-7}	0.045

ever, when a demand for flow is changed or a valve is shut, flow and pressure in the system become subject to the rates of change. Under these conditions, natural modes of resonance can be excited, which can result in seemingly endless problems, ranging from excessive noise to fatigue failures.

Referring to Fig. 60.11, an increase in the applied force F to the piston will cause the volume of trapped oil to compress according to the relationship

$$-\Delta V = \frac{(p_2 - p_1)V_0}{\beta} \tag{60.19}$$

where $p_1 = F1/A$ = steady initial pressure
$p_2 = F2/A$ = steady final pressure
V_0 = original volume
β = bulk modulus of oil

The negative sign is to indicate that the oil volume reduces as the applied pressure increases. It is assumed that the walls of the container are rigid.

Although the change in volume is small, with a value of about 0.5% per 7 MPa applied pressure, it does result in high transient flow rates. As a comparison, air compresses about 50% for a pressure change of 0.1 MPa (1 atmosphere). The transient flow rates due to oil compressibility effects can be estimated from the first derivative of Eq. 60.19:

$$Q_c = \frac{V_0}{\beta}\frac{d\Delta p}{dt} \tag{60.20}$$

The actual value of oil bulk modulus is strongly dependent on the amount of air present in the form of bubbles. In practice, it is impossible not to have some level of air entrainment. The effective bulk modulus can then be estimated using

$$\beta_e = \frac{1}{\left[\dfrac{1}{\beta_0} + \dfrac{\alpha}{p}\right]} \tag{60.21}$$

where β_0 = oil bulk modulus with no air present
α = ratio of air volume to oil volume (typically 0.5%)
p = operating oil pressure

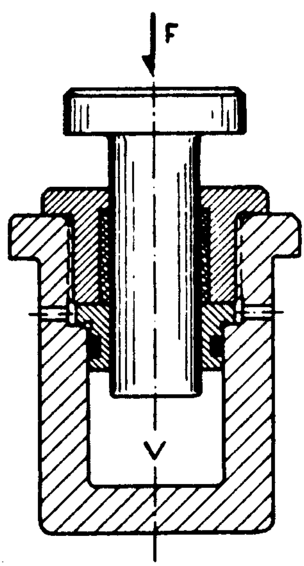


Fig. 60.11 Pressure chamber for measuring compressibility. (From J. Thoma, *Modern Oil Hydraulic Engineering.* © 1970 Technical and Trade Press. Reprinted with permission.)

These effects are illustrated in Fig. 60.12. The bulk modulus of the oil and the entrained air contribute to the effective spring a hydraulic system exhibits. For example, the hydraulic braking system of a vehicle feels spongy if there is air in the brake fluid, as a result of the circuit not being *bled* correctly.

The third factor in the system *stiffness* is the contribution from the containment vessel, which in this case is the steel pipework or reinforced rubber hose.[18] For a thin-walled metal pipe, the effective bulk modulus is estimated from[19]

$$\beta_c = \frac{TE}{D} \tag{60.22}$$

where T = wall thickness, m
E = modulus of elasticity, Pa
D = pipe diameter, m

When the pipeline is a hydraulic hose, there is some difficulty in obtaining design information. Values for β_c in the range of 6.8×10^7 to 7.7×10^8 Pa have been quoted. Some further guidance is given in Ref. 18.

The total effective system bulk modulus taking all these effects into account can be calculated from

$$\beta_e = \left[\frac{1}{\beta_0} + \frac{\alpha}{p} + \frac{D}{TE}\right]^{-1} \tag{60.23}$$

The effective stiffness of a system is important when the designer is concerned with reverse loading. For example, consider a hydraulic ram controlling a metal cutting tool. The loads on the tool can vary as it cuts through metal. If the hydraulic system is not very stiff, the tool will move about, giving a poor finish. Obviously there will be some movement, as it is not possible to design an infinitely stiff system. However, a high stiffness will make any such tendency to move very little.

The second problem due to system stiffness is related to dynamic behavior. Since hydraulic machines have moving parts, there will be masses and inertias to accelerate. The interaction of mass with stiffness results in natural resonant modes. These natural frequencies are normally passive, but, if excited by a power source of comparable frequency, the result can be significant noise and vibration or, in the extreme, structural failure. It is very important, therefore, for the designer to estimate these passive modes at the design stage.

Consider the case of a simple ram shown in Fig. 60.13, which is used to position a mass M. It can be shown in Martin and McCloy[12] that the flow into the ram is

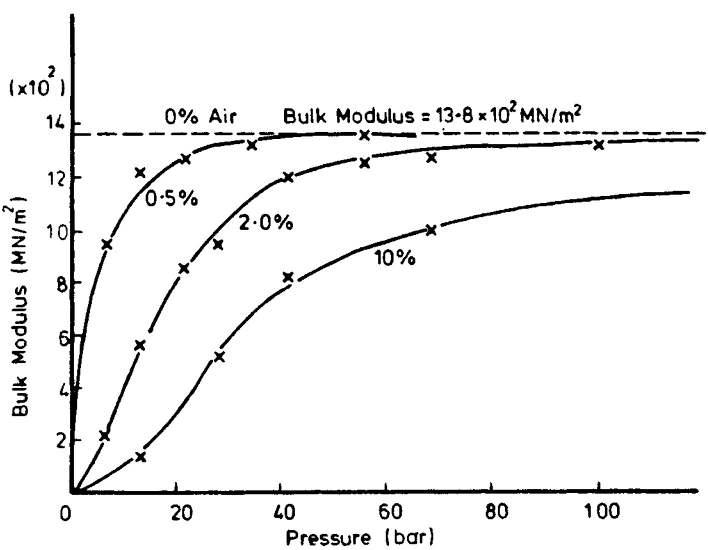


Fig. 60.12 Bulk modulus for a typical hydraulic oil including the effect of free air.

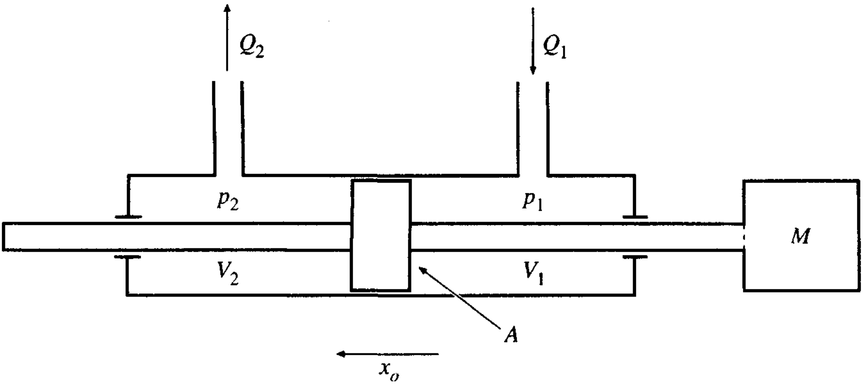


Fig. 60.13 Compressibility effects in a cylinder.

$$Q_1 = \frac{V_1}{\beta_1}\frac{dp_1}{dt} + A\frac{dx_0}{dt} \tag{60.24}$$

In other words, the first term on the right-hand side of Eq. (60.24) represents the contribution of compressibility to the total oil flow. If the flow is steady, this term disappears. The second term is the more commonly recognized flow into the ram as the piston bore volume geometry changes.

A similar argument can be applied to the left-hand side of the ram where the oil is being pushed out:

$$Q_2 = -\frac{V_2}{\beta_2}\frac{dp_2}{dt} + A\frac{dx_0}{dt} \tag{60.25}$$

The sign change is to differentiate between oil that is being compressed and oil that is expanding. The average flow through the ram can now be estimated by combining Eqs. (60.24) and (60.25). In practice, it is unlikely that there are two different fluids in the ram unless it is an air–oil system. Therefore, $\beta_1 = \beta_2 = \beta$. If V is the swept volume of the ram, then $V_1 = V_2 = V/2$ for the piston control. This results in the load flow equation

$$Q = \frac{V}{4\beta}\frac{d(p_1 - p_2)}{dt} + A\frac{dx_0}{dt} \tag{60.26}$$

Now, the pressure drop across the piston $(p_1 - p_2)$ is, in this case, used to accelerate the mass attached to this piston rod.

$$p_1 - p_2 = \frac{M}{A}\frac{d^2x_0}{dt^2} \tag{60.27}$$

Combining Eqs. (60.26) and (60.27) gives

$$Q = \frac{VM}{4\beta A}\frac{d^3x_0}{dt^3} + A\frac{dx_0}{dt}$$

Operating on both sides with the Laplace operator yields

$$\int_0^\infty Q(t)e^{-st}\,dt = \frac{VM}{4\beta A}\int_0^\infty e^{-st}\frac{d^3x_0}{dt}\,dt + A\int_0^\infty \frac{dx_0}{dt}e^{-st}\,dt$$

With the appropriate initial conditions, namely $x_0 = dx_0/dt = 0$ at $t = 0$, the result is

$$\overline{Q}(s) = A\left[\frac{VM}{4\beta A^2}s^2 + 1\right]s\overline{x}_0(s) \tag{60.28}$$

The bar over the symbol denotes the Laplace transform of the function, and s is the Laplace transform independent variable. The term inside the square brackets of Eq. (60.28) can be compared directly with the general equation for a second order system with zero damping ratio. Hence, the term $VM/4\beta A^2$ is the reciprocal of the system natural frequency squared.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{\text{Stiffness}}{\text{mass}}} = \frac{1}{2\pi}\sqrt{\frac{4\beta A^2}{V} \times \frac{1}{m}} \tag{60.29}$$

If this system is excited, say by an impact on the mass, it will oscillate at the frequency defined by Eq. (60.29). Since there is no damping term, the mass would oscillate continuously. In practice, there will always be some damping, however small, from seal rubbing and oil film shear. The difficulty for the designer is to make some meaningful guesses as to what value to use. For example, one might use 15% of the stall load on the ram divided by the maximum velocity of the piston as a first guess.

60.8 SYSTEM CLASSIFICATIONS

Most hydraulic circuits, regardless of the application, fall into one or two general classifications.[20] The two major divisions are constant flow and constant pressure, depending on whether the output is mainly a function of flow (i.e., velocity, displacement, or acceleration) or mainly related to pressure (i.e., force or torque).

The simplest hydraulic circuits fall into the constant flow classification with open center valving. A simple example of this is shown in Fig. 60.14*a*. It is open center in the sense that when the control valve is centered the fluid is circulated directly back to tank. This method ensures minimum power loss and fluid heating in the quiescent periods. Compare this with Fig. 60.14*b*, which is a simple constant flow circuit in which oil is dumped through the relief valve at the end of the ram stroke. The restriction of the orifice in the relief valve generates high levels of heat and noise, thereby wasting power.

Figure 60.14*a* introduces the use of standard symbols for circuit design. The pressure relief valve is represented by a square with an offset arrow indicating that this valve is normally closed until sufficient pressure is developed in the pilot line (dotted) to push the valve open against the mechanical spring. This symbol is quite different from the physical drawing of a relief valve shown in Fig. 60.15, but it certainly conveys to the reader how the device is expected to operate.[21]

In the case of a directional control valve, it is always shown in its normally closed position. The reader is expected to visualize what happens when the valve is moved into its other two operating positions, as shown in Fig. 60.14*c*. In Europe, these symbols are standardized by the International Organization of Standards (ISO)[22] and also by the British Standards Institution (BSI).[23] In the United States, the American National Standards Institute develops the standards for the fluid power industry.[24]

The simplest form of flow control uses an adjustable orifice. These circuit configurations are shown in Fig. 60.16. When the orifice control is placed in the supply line upstream of the hydraulic cylinder, the system is said to be under *meter-in* flow control. Flow control is operative when flow is directed to the large-area side of the piston in Fig. 60.16*a*, while in the reverse direction, flow is dumped freely through the check valve. This type of control is best suited for resistive-type loads that the piston rod pushes against, and not for overrunning type loads.

When the orifice control is placed in the return line (Fig. 60.16*b*), the system is said to be under *meter-out* flow control. This type of control is best suited over running loads that are moving in the same direction as the motion of the actuator.

In *bleed-off* control (Fig. 60.16*c*), the flow control parallels the cylinder feed line. This approach can be used to adjust the cylinder speed over a range that is less than the maximum speed available. It has the advantage of using a small control valve, only large enough to handle the bleed flow and not the total flow. It also does not introduce a pressure drop in the main delivery line to the ram.

There are many pitfalls in designing fluid power circuits for the inexperienced, some of which are due to lack of design information. Some of these problems are reviewed in an excellent article by Achariga.[25]

60.9 PUMP SETS AND ACCUMULATORS

Any hydraulic circuit is useless unless there is a unit to provide the fluid power. Hence, the pump set is an important component of the system. Depending on the application, this can be designed and constructed by the user or the decision may be to purchase one of the many commercially available complete packages.

A typical design[12,26] is shown in Fig. 60.17. The motor driving the pump is usually electric for industrial application and runs at 1740 rpm. Other prime movers, such as diesel engines, could also

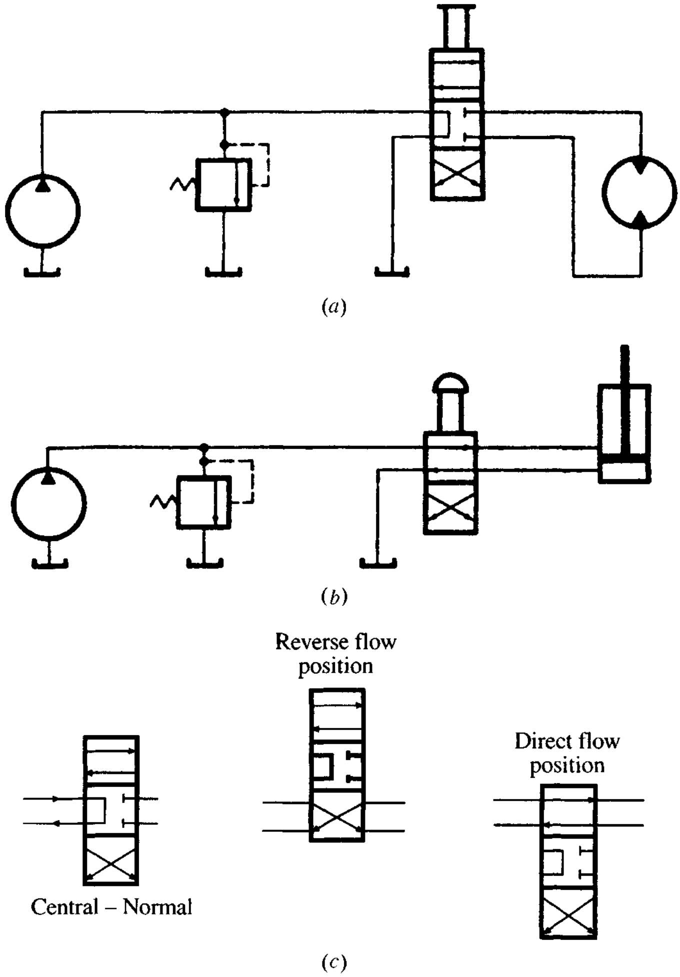


Fig. 60.14 Simple hydraulic circuits and valve symbols. (*a*) Open-center valve control; (*b*) simple constant flow circuit; (*c*) valve symbol meanings.

be used. The outlet from the fixed displacement pump is piped to a relief valve. This allows the working pressure to be selected. The nonreturn valve *N* prevents flow being forced back into the pump and also helps to stiffen the hydraulic circuit. An accumulator is included to smooth the pressure pulses developed in the pump. It will also provide additional flow for short-time high transient demands.

It is wise to include a shutoff cock, C1, to prevent oil spillage when the delivery line is disconnected and also to include a shutoff cock, C2, so that the accumulator can be discharged safely.

Good reservoir design, as in Fig. 60.17 is probably the most important aspect of preventive maintenance. It fulfills many functions besides containing sufficient oil to meet the demands of the complete system. It acts as a cooler and allows time for contaminants such as foam and dirt to settle out. The tank capacity is usually at least three times the maximum delivery of the pump in one minute and may be as large as six times if there are numerous valves in the circuit to generate heat. The inlet strainer removes larger debris, but care must be taken to ensure that it or any other component does not create significant pressure loss in the inlet (suction) side of the pump.

In starting a new hydraulic circuit, the following steps should be followed:

1. Make certain the pump is being driven in the correct direction for its design.
2. Make sure all nonreturn valves are located in the correct flow direction.

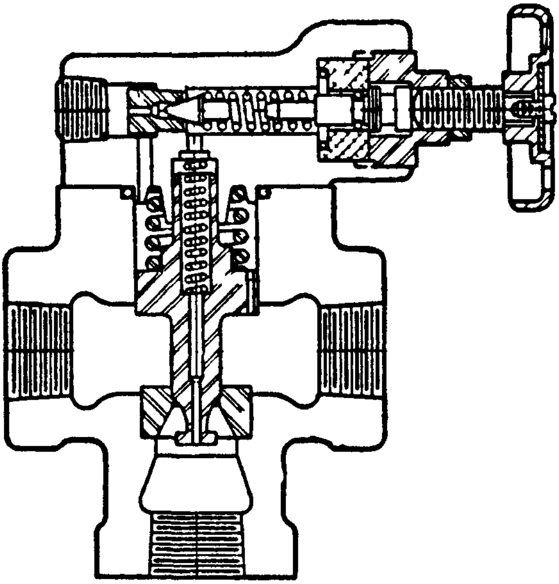
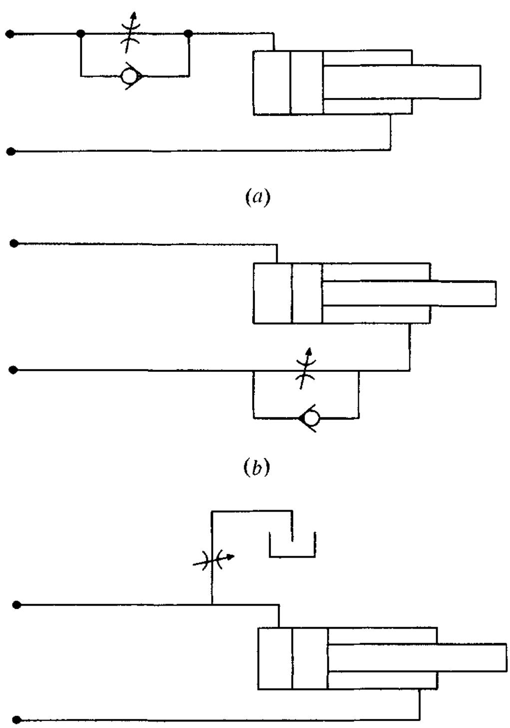


Fig. 60.15 Cross section through a pilot-operated pressure relief valve. (Courtesy of Vickers.)



(*a*)

(*b*)

(*c*)

Fig. 60.16 Means of speed control. (*a*) Meter-in flow control; (*b*) meter-out flow control; (*c*) bleed-off.

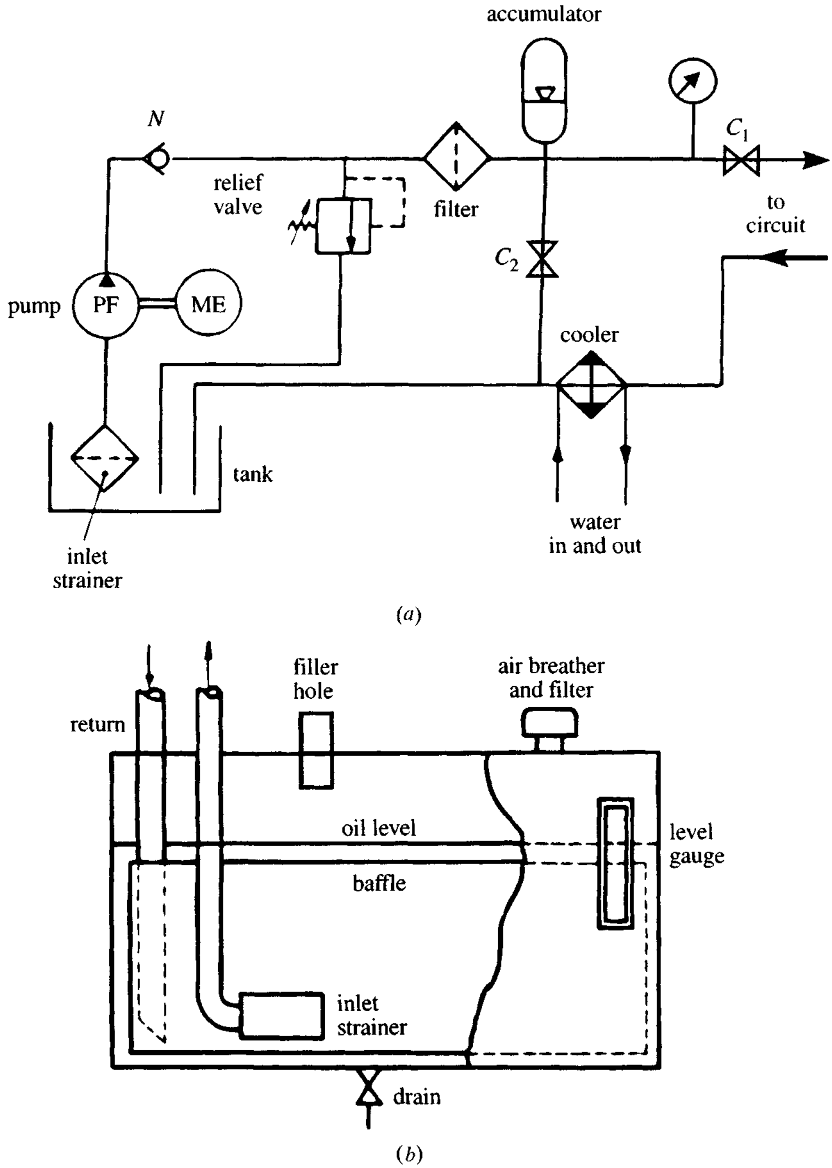


Fig. 60.17 Pump set design. (*a*) The basic pump circuit; (*b*) layout of a typical hydraulic reservoir.

3. Jog the pump drive motor two or three times watching for leaks and note system pressure for an indication of pump priming.
4. Check oil level in the tank and top off if a new circuit has used a significant amount of oil in filling.
5. Check that the accumulator is charged correctly.
6. After 5–10 hours of running, check the filter and strainer for debris left in the new system from parts manufacture.

The accumulator is an energy storage element into which hydraulic oil is pumped by the system to compress the contained gas. The accumulator can be used as a low pass filter to smooth out pump delivery fluctuations, or it can be used to supply small amounts of additional power for transient demands. It is sometimes cost-effective to use a small pump in a circuit where the duty cycle calls for low power requirements most of the time and large demands for relatively short periods. In this case, a larger accumulator can be used to store energy during the low-level part of the duty cycle and release it when high demand is required. A third application is to provide a stand-by short-term power source in case of failure of the pump set. This is particularly important in aerospace applications.

Accumulation size estimates are normally based on Boyle's law and assume that the charging gas temperature remains constant. The gas precharge in the accumulator is selected at about ⅓ of the final maximum pressure required in the oil. It is advisable, of course, to use nitrogen as the gas. The usual stages in the operation of an accumulator are shown in Fig. 60.18, where

p_1, V_1 = gas precharge pressure and volume

p_2, V_2 = gas charge pressure when the pump is turned on and will correspond to the system maximum pressure in the oil

p_3, V_3 = minimum pressure required in the circuit

If the volume of oil delivered from the accumulator is

$$V_0 = V_3 - V_2 \tag{60.30}$$

then for gas compressed isothermally,

$$V_1 = \frac{p_2/p_1 V_0}{(p_2/p_3 - 1)} \tag{60.31}$$

If there is shock loading present, the process is closer to adiabatic, and Eq. 60.31 is modified to

$$V_1 = \frac{(p_2/p_1)^{1/\gamma} V_0}{[(p_2/p_3)^{1/\gamma} - 1]} \tag{60.32}$$

where γ is the ratio of specific heats for the gas, which normally has a value $\gamma = 1.4$.

60.10 HYDROSTATIC TRANSMISSIONS

The hydrostatic transmission is usually a closed-circuit system in which the pump output flow is sent directly to a hydraulic motor. This package finds extensive use in many industries, such as steel and paper manufacturing. The electromechanical version is the Ward–Leonard drive, which consists of a dc generator supplying current to a dc motor, often used in locomotives. However, this unit tends to be very bulky in comparison to the hydraulic transmission.

Because of the ease of control of speed, torque, and direction of rotation, the hydrostatic transmission has become a popular choice in industrial equipment design. As can be seen from Fig. 60.19,

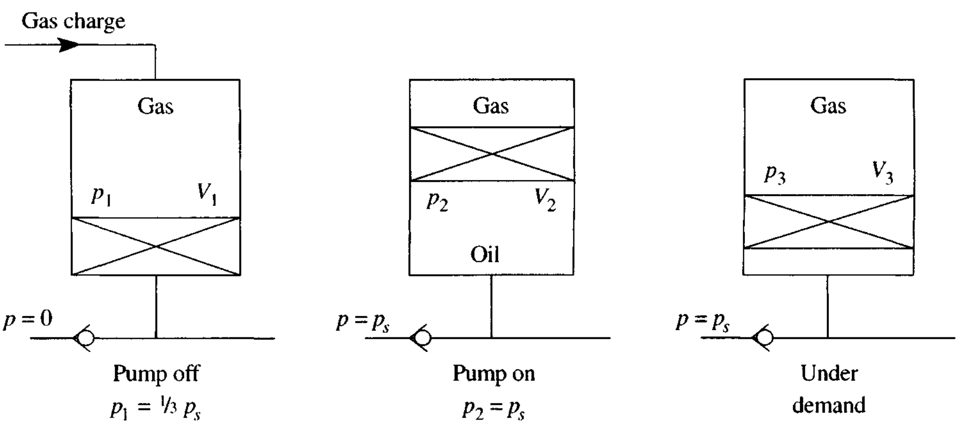


Fig. 60.18 Accumulator operation.

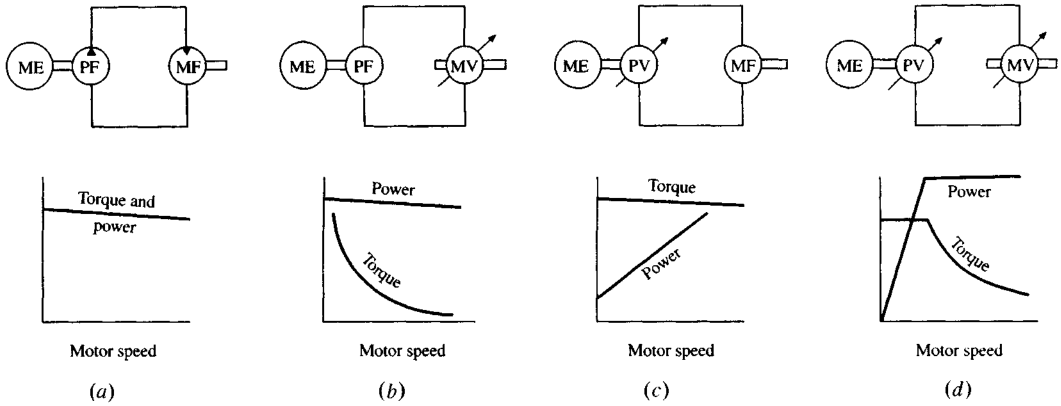


Fig. 60.19 Pump and motor combinations. (*a*) Fixed displacement pump and fixed displacement motor; (*b*) variable displacement motor and fixed displacement pump; (*c*) variable displacement pump and fixed displacement motor; (*d*) variable displacement pump and variable displacement motor.

there are several combinations whereby a positive displacement pump can be used to drive a motor. The choice will depend on the application.

The combination of a fixed displacement motor with a fixed displacement pump, Fig. 60.19*a* gives a fixed drive ratio. This is the simplest configuration where the output speed can only be controlled by altering the speed of the prime mover. For speed control, an alternative approach would be to include a bleed flow control valve from the main delivery line.[27,28]

When the motor is variable displacement (Fig. 60.19*b*), a fixed ratio drive at any given motor setting is obtained. The torque decreases with speed increase, making the characteristics compatible with winding machines. As a roll diameter increases, the rotational speed must decrease to hold a constant linear velocity of the material being wound while at the same time mass is increasing, requiring more drive torque.

By making the pump variable displacement instead of the motor, as in Fig. 60.19*c*, the torque remains constant over the speed range. With the pump at zero output, an idle condition is produced that is similar to a disengaged clutch.

The final configuration, shown in Fig. 60.19*d* introduces variability for both the pump and the motor. This combination can produce either a constant power or a constant torque drive. The combination has great flexibility—at a cost, of course—and can have both pump and motor adjusted together or separately. For example, where two separated parts of the same machine are to be driven at different speeds, a variable pump with two variable displacement motors can be used.

60.11 CONCEPT OF FEEDBACK CONTROL IN HYDRAULICS

A simple open-loop hydraulic servomechanism is shown in Fig. 60.20*a*. It consists of a spool valve, moving in a sleeve, so as to uncover two sets of control orifices and, therefore, allowing flow to and from the ram. Details of the valve design are discussed in Section 60.4.

A displacement of the spool x_v to the right will allow the supply pressure p_s and flow Q to pass into the ram, causing the piston to move to the left. Consequently, the oil flow $Q2$ in the left-hand ram chamber will be exhausted through one set of control orifices to the tank. Three important facts should be noted about this device. First, a displacement of the spool causes the piston to move at a constant velocity. Hence, a simple hydraulic servomechanism has the characteristics of an integrator

$$y = K \int x_v \, dt \tag{60.33}$$

Second, it only requires a relatively small effort in displacing the spool valve to make considerable force available at the output of the ram. Third, the system is nonlinear and difficult to analyze accurately, except under certain simple loading conditions. This is mainly due to the fact that the valve flow is a fraction formed from two variables, orifice area and pressure drop [see Eq. (60.1)].

It is shown in Martin and McCloy[12] that if the load pressure is defined as $p_L = p_1 - p_2$, then the load flow is given by

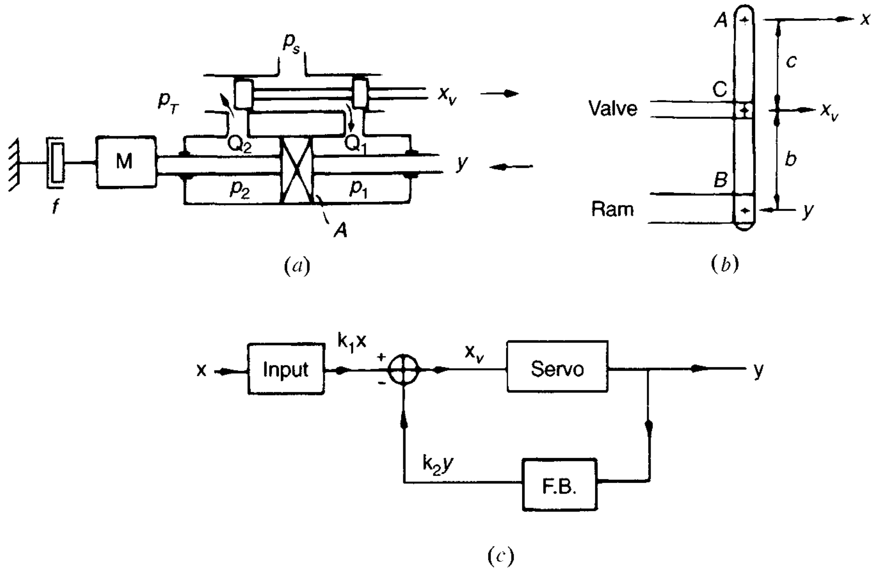


Fig. 60.20 Simple open loop hydraulic servomechanism. (*a*) Open loop configuration; (*b*) feedback linkage; (*c*) closed loop block diagram.

$$Q_L = C_d A_0 \sqrt{\frac{2}{\rho}} \sqrt{\frac{p_s - p_L}{2}} \tag{60.34}$$

If the mass M and damping f are very small (see Fig. 60.20*a*), then $p_1 = p_2 \approx \frac{1}{2}p_s$, and Eq. (60.34) becomes

$$Q_L = K_1 x_v \sqrt{\frac{p_s}{2}} = A \frac{dy}{dt} \tag{60.35}$$

where

$$K_1 x_v = C_d A_0 \sqrt{\frac{2}{\rho}}$$

and finally,

$$y = \frac{K_1}{A} \sqrt{\frac{p_s}{2}} \int x_v, dt \tag{60.36}$$

Consider now the lever system shown in Fig. 60.20*b*. When this follow-up mechanism is attached to the valve and ram rod, a closed loop configuration results. A displacement of the input x causes a movement of the valve x_v, also to the right from a central closed position, since initially the lever $(c + b)$ pivots about B. The contribution to the valve displacement is now initially

$$x_v = \frac{b}{c + b} x = k_1 x \tag{60.37}$$

Once the spool valve opens, flow passes to the ram and the pivot B starts to move to the left. The top pivot point A is now held fixed by the input, hence the original valve displacement, Eq. 60.37, is now closed. The control equation is

$$x_v = \frac{b}{c+b} \times x - \frac{c}{c+b} \times y = k_1 x - k_2 y \tag{60.38}$$

The block diagram in Fig. 60.20*c* should clarify the arrangement.

Combining Eq. (60.35) with Eq. (60.38) results in the closed loop transfer function for this configuration

$$y(s)/x = \frac{b/c}{(1 + Ts)} \tag{60.39}$$

where the gain equals b/c, and the time constant T is

$$T = \frac{c+b}{c} \frac{A}{K_1 \sqrt{p_s/2}} \tag{60.40}$$

Equation (60.39) means that the closed-loop servo operates as a simple exponential type lag, instead of an integrator, as in the open-loop configuration.

The performance of the unit can be assessed in several ways, depending on the type of information needed and the test equipment available. The transient response is a plot of the output movement y against time, for a defined magnitude of step input. Mathematically, the solution to Eq. (60.39) is

$$y = \frac{b}{c}(1 - e^{-t/T})x \tag{60.41}$$

The plot of this equation is shown in Fig. 60.21*a*, where the time constant can be found from 63.2% of the final steady-state point. In theory, the steady state given by

$$y = bx/c \tag{60.42}$$

will only be reached when t reaches infinity. However, it is normal in practice to use $4T$ = 98% of final steady state as the practical steady-state value. Transient response testing is less costly and is easy to perform. However, the information is limited in that spectral information is difficult to interpret especially phase shift between the input and output. Some nonlinearities, such as dead zone, clipping, and small amplitude parasitic oscillations, can be identified.

A simple production test for assessing the level of friction in the moving parts of the servomechanism is to apply the ramp test depicted in Fig. 60.21*b*. In this case, the input is suddenly subjected to a constant velocity of ω_i rad/sec. The output tries to follow but lags by a steady state error of $T\omega_i$ where T is the system time constant. If the system were modeled by Eq. (60.39), this error could be reduced by changing the lever ratio $(c + b)/c$, for example.

60.12 IMPROVED MODEL

Experimental testing of an actual hydraulic servo system immediately reveals that the first order model discussed in the previous section does not adequately represent the actual performance. This is especially true at the prototype testing phase, where the system response has not been optimized. Instead of the smooth exponential type behavior shown in Fig. 60.21*a*, the response is likely to be quite oscillatory. The main reason for this is the fact that compressibility of the oil and the mass of the moving parts were considered negligible in the simple first-order model.

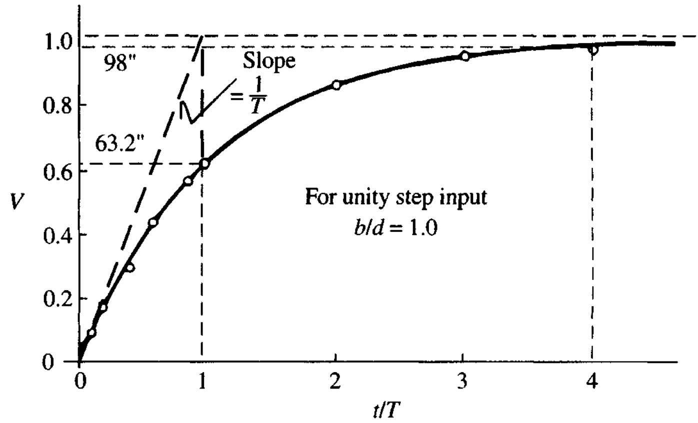
In addition, if mass is to be included, the load pressure p_L cannot be ignored as it was in Eq. (60.35). This means that the valve equation becomes a function of two variables: pressure drop and valve displacement.

Early attempts to solve this problem are recorded in a classic paper by Harpur[29] using a small perturbation method. This had the disadvantage that the valve characteristics at any instant in the motion of the servo were defined by the instantaneous values of the slopes of two nonlinear curves. In order to improve this situation, an alternative approach was suggested[30] that minimizes the average error and to a large extent overcomes this problem. Referring once more to Fig. 60.20, the flow into the actuator is given by

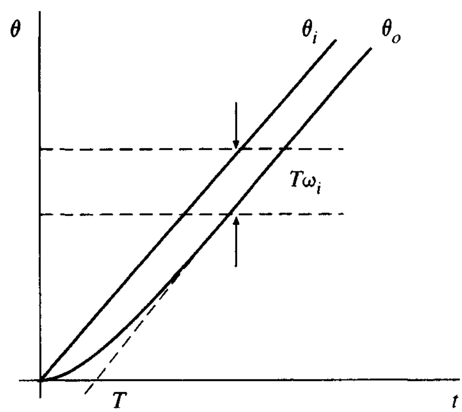
$$Q_1 = KA_{01}\sqrt{p_s - p_1} \tag{60.43}$$

and the flow out is given by

$$Q_2 = KA_{02}\sqrt{p_2 - p_T} \tag{60.44}$$



(a)



(b)

Fig. 60.21 First order performance–time domain. (*a*) Step response; (*b*) ramp input; $\theta = \omega_i t$ where *t* is time.

where $K = C_d\sqrt{2\phi}$ and the valve is symmetrical $A_{01} = A_{02} = A_0$. It can be shown (see Ref. 12) that the load flow through the valve is

$$Q_L = KA_0\sqrt{\frac{p_s - p_L}{2}} \tag{60.45}$$

If compressibility of the oil is now included [see Eq. (60.26) of Section 60.7], then the load flow can be equated to the flow through the actuator so that

$$\begin{aligned} Q_L &= \text{(flow due to piston movement)} + \text{(flow due to compressibility)} \\ &= A\frac{dy}{dt} + \frac{V}{4\beta}\frac{d}{dt}(p_L) \end{aligned} \tag{60.46}$$

By equating Eqs. (60.45) and (60.46), the relationship between valve displacement x_v and output movement y can be obtained, and it is much more complex than in the previous model shown in Eq. (60.36).

$$K_v x_v \sqrt{\frac{p_s - p_L}{2}} = A\frac{dy}{dt} + \frac{V}{4\beta}\frac{d(p_L)}{dt} \tag{60.47}$$

where a linear relationship between valve displacement and uncovered orifice area has been assumed.

The worst case for loading a system is when the output load is pure mass. It is the load pressure p_L that is used to accelerate the mass.

$$p_L = \frac{M}{A}\frac{d^2y}{dt^2} \tag{60.48}$$

Introduce this into Eq. (60.47):

$$\frac{K_v x_v}{A}\sqrt{\frac{p_s}{2}\left(1 - \frac{M}{p_s A}\frac{d^2y}{dt^2}\right)} = \frac{dy}{dt} + \frac{VM}{4\beta A^2}\frac{d^3y}{dt^3} \tag{60.49}$$

If the constants are lumped together so that $K_r = K_v/A\sqrt{p_s/2}$, and if the square root term is expanded by the binomial theorem, neglecting terms greater than first order, Eq. (60.49) becomes

$$K_r x_v\left[1 - \frac{1}{2}\left(\frac{M}{p_s A}\right)\frac{d^2y}{dt^2}\right] = \frac{dy}{dt} + \frac{VM}{4\beta A^2}\frac{d^3y}{dt^3} \tag{60.50}$$

Rearranging and transforming to Laplace domain, as was done previously with Eq. (60.28)

$$\left[\frac{VM}{4\beta A^2}s^2 + \frac{x_v}{2}\left(\frac{MK_r}{p_s A}\right)s + 1\right]s = K_r x_v \tag{60.51}$$

The equation within the square brackets is equivalent to the general equation for a spring-mass-damper system, hence Eq. (60.51) can be written

$$\frac{1}{\omega_n^2}s^2 + \frac{2\xi}{\omega_n}s + s = K_r x_v \tag{60.52}$$

It can now be seen that the damping contributed from the valve is partially determined by the valve displacement; the symbol ξ is the damping ratio. This explains why, in practical test results, the frequency response curves of a hydraulic servo change depending on the size of the input amplitude. Keating and Martin[30] discuss a further refinement to this model that results in an even better estimate of the dynamic response of this type of system.

60.13 ELECTROHYDRAULIC SYSTEMS—ANALOG

In the arrangements discussed in the previous section, signal transfer for feedback was done using mechanical linkages. It is much more convenient to employ electrical means to achieve these loops. However, this does require the use of transducers to convert mechanical and fluid signals into an electrical form.

Typical electrohydraulic servo valves use an electrical torque motor to move the spool arrangement. The most famous of these types of valves is the Moog 1500 series two-stage valve shown in Fig. 60.22*a*. It is also possible to have a single stage spool, as shown in Fig. 60.22*b*. In the example shown, the first stage is a double nozzle pilot valve controlling a second-stage spool valve. The torque motor is really a limited movement electric motor, arranged so that the flapper extends between two nozzles. This allows differential pressure to be applied across the sliding valve, whose movement in term meters fluid out of the valve.

While the flexibility of combining electrics with hydraulics is a major advantage, it should be noted that the torque motor does introduce an additional transfer function into the system. This in itself need not be a problem, provided that care is taken in the design process that the overall phase shift is not increased. As with any series type system, the overall performance is determined by the component with the poorest dynamic characteristics. It is important, therefore, to ensure that the torque motor valve assembly is not the weak link in the chain.

Feedback can now be achieved with the use of electrical transducers such as precision potentiometer, pressure devices, and accelerometers. A typical arrangement is shown in Fig. 60.23. For this particular arrangement, the minor loop is closed with position of the second-stage spool valve and the major loop is closed with position of the driving ram which controls the inertia type load. All

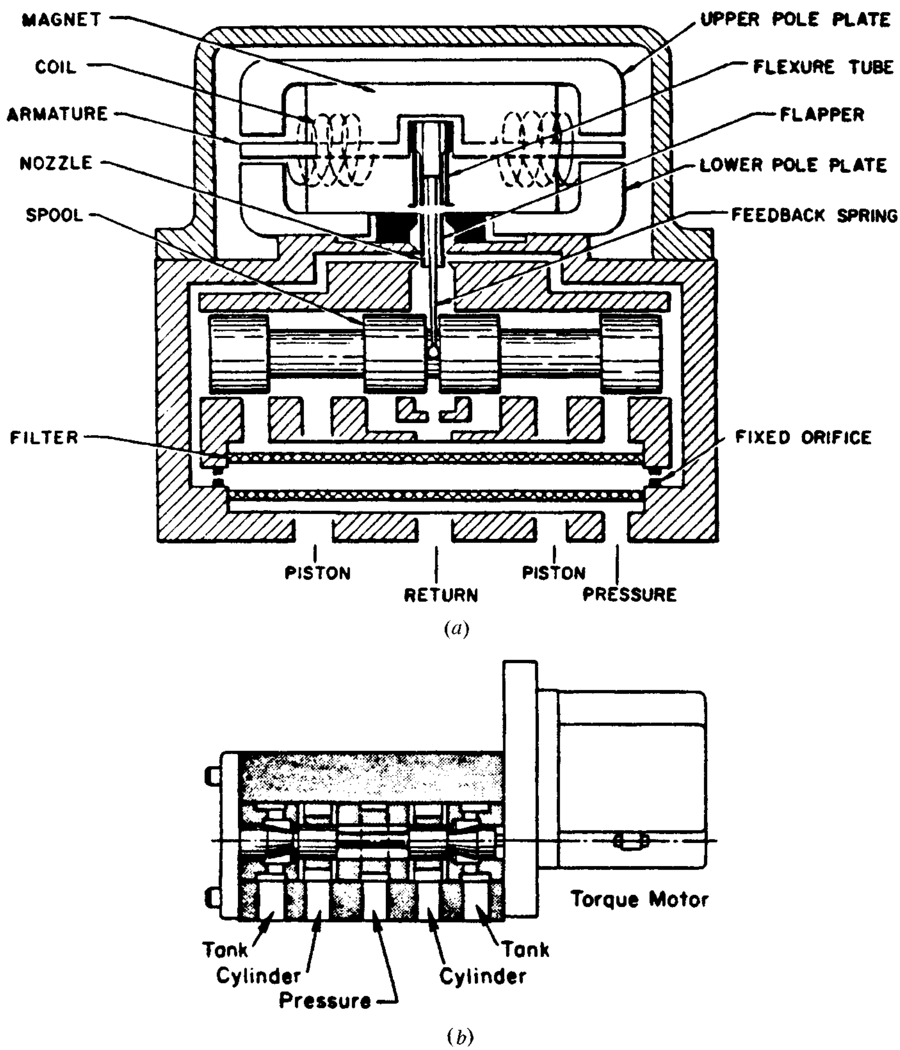


Fig. 60.22 Typical electrohydraulic valves. (*a*) Two-stage valve (Moog Inc., East Aurora, NY); (*b*) single-stage valve.

the signals around the servo are now electrical and easy to adjust. The amplifier is used both as an easy method of gain adjustment and as a summing junction.

Potentiometers are the cheapest and simplest devices for converting linear and angular displacement information to electrical signals, but unless their performance, especially when coupled to other parts of the circuit, is understood, there can be real practical problems in getting an electrohydraulic servo to function correctly. The selection of a suitable instrument potentiometer requires the following specification items to be addressed.[31–33] The most commonly occurring terms are

1. *Linearity.* The deviation of the output voltage from a potentiometer from a linear law related to shaft rotation, Fig. 60.24*a*.
2. *Conformity.* Similar to linearity, but used in relation to potentiometers designed to follow a nonlinear law, Fig. 60.24*b*.
3. *Deviation.* Some suppliers of potentiometers quote deviation instead of linearity. The deviation is defined as the maximum permissible offset from the best straight line that can be

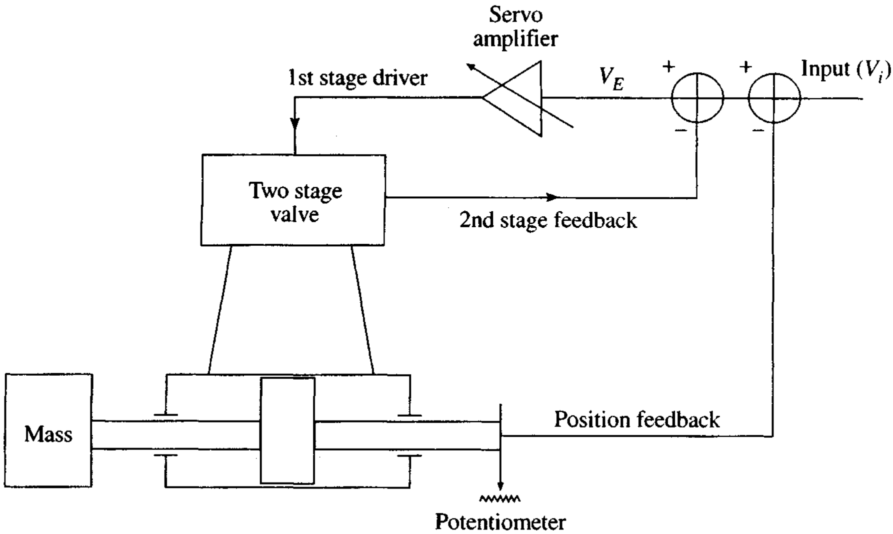


Fig. 60.23 Two-stage valve–electrohydraulic servo system.

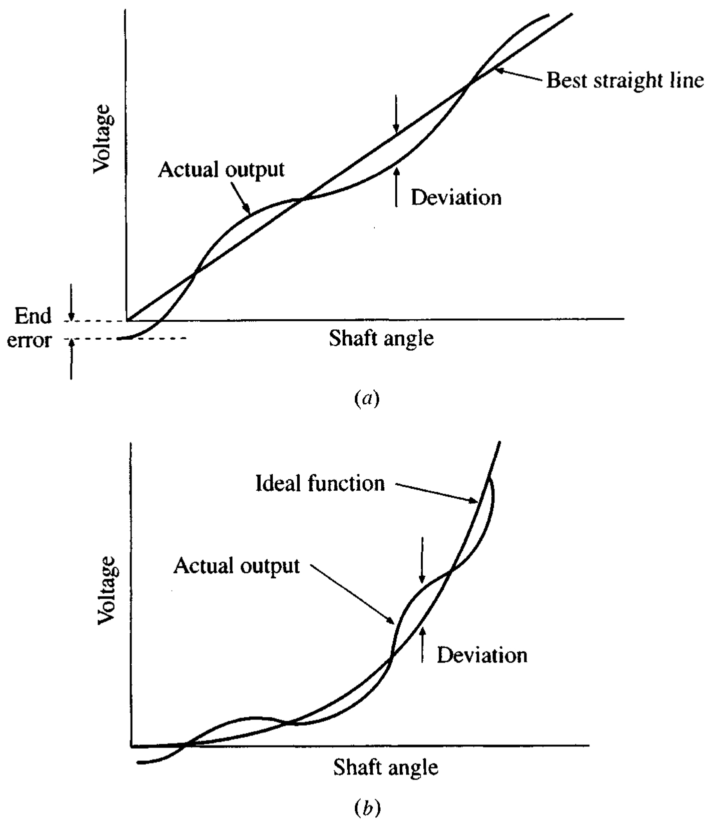


Fig. 60.24 Potentiometer linearity and conformity. (*a*) Linearity; (*b*) conformity.

drawn through the experimental points of measured resistance against rotation of the potentiometer. This tolerance is expressed as a percentage of the total resistance.

4. *Resolution.* The incremental rotation of the shaft necessary to produce the smallest incremental change of output voltage.
5. *Power rating.* The maximum continuous power that can be safely dissipated by the resistance at a specific temperature.
6. *Operating torque.* The torque required to start the wiper moving from rest. For small, general-purpose wire-wound potentiometers, this is in the range of 1–5 oz/in., while special-purpose units can be obtained having torques as low as 0.005 oz in.
7. *Electrical and mechanical angle.* The mechanical angle is the angle through which the potentiometer shaft can be rotated freely, while the electrical angle is that angle of rotation from which a voltage can be recorded.

A wide range of potentiometers are available, such as those using wire-wound cores, carbon tracks, and film deposits. Their mode of operation can be linear or nonlinear. Multiturn potentiometers are formed using a helix configuration and give much better resolution and accuracy than single-turn units.

Wire-wound potentiometers have the disadvantage that, when used in high-gain systems, the spacing between the wires produces a staircase effect in the output voltage. This results in a distinct roughness of motion in the servo output shaft. This problem can be resolved by using the more expensive film potentiometer. Carbon track versions tend to leave a deposit of carbon particles as the potentiometer wears in use. This also makes the servo output motion rough. These units need to be cleaned regularly. Operating torques, especially the starting torque for a potentiometer, become important when the unit is being driven from a low power source, such as the first or second stage of a spool valve. Another critical problem in electrohydraulic applications is to minimize loading errors on potentiometers, or in fact, any other transducer associated with the circuit.

Consider the arrangement shown in Fig. 60.25. In this circuit, a potentiometer is loaded by a circuit of resistance R_L, which represents the input impedance of the next stage. For the input circuit (Fig. 60.25*a*), $V_i = RI$, while for the output circuit, $V_o = kRI$, assuming the ideal case of $R_L = \infty$.

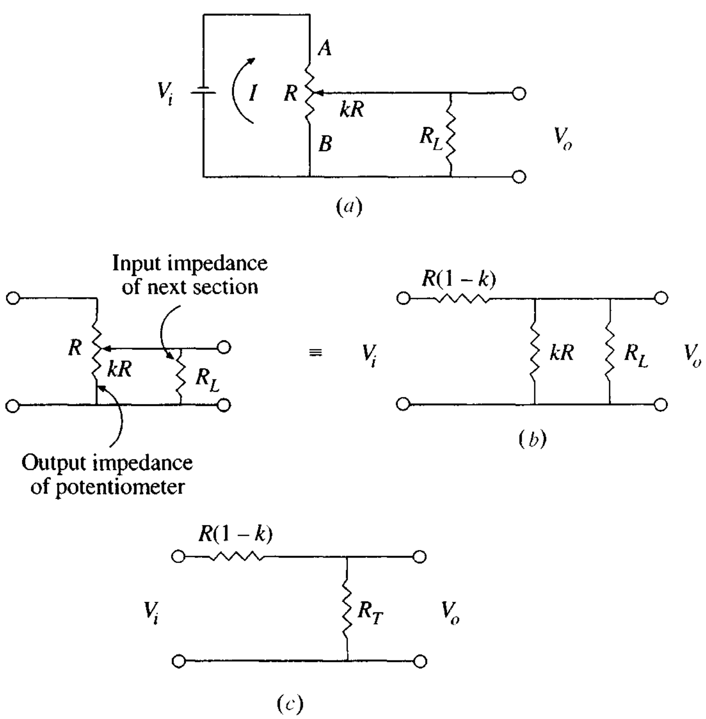


Fig. 60.25 Potentiometer loading. (*a*) Actual circuit; (*b*) equivalent circuit; (*c*) least complex equivalent circuit.

Since the current I in the circuit is common $V_i/V_o = 1/k$, which satisfies the conditions: $V_o = 0$ when wiper is at B, $k = 0$ and $V_o = V_i$ when wiper is at A, $k = 1$.

If now the practical case is considered, where R_L is finite in value, the circuit can be interpreted as shown in Fig. 60.25*b*. The parallel resistances can be replaced by

$$\frac{1}{R_T} = \frac{1}{kR} + \frac{1}{R_L}$$
$$R_T = \frac{kRR_L}{R_L + kR} \tag{60.53}$$

as shown in Fig. 60.25*c*. The total resistance as seen by V_i is now

$$R_i = R(1 - k) + \frac{kRR_L}{R_L + kR}$$

The current in the circuit must now be

$$I = \left(\frac{R_L + kR}{kR^2(1 - k) + RR_L}\right) V_i$$

But

$$V_o = R_T I = \frac{kRR_L}{R_L + kR} I$$

hence

$$\frac{V_o}{V_i} = \frac{k}{k(1 - k)\dfrac{R}{R_L} + 1} \tag{60.54}$$

The error in measurement due to potentiometer loading can be expressed as

$$\text{error \%} = \frac{V_o(\text{ideal}) - V_o(\text{actual})}{V_i} \times 100$$
$$= \frac{k^2(1 - k)}{k(1 - k) + \dfrac{R_L}{R}} \times 100 \tag{60.55}$$

This equation shows that the error is variable over the range of potentiometer shaft angle positions. The shaft angle yielding maximum error can be found from $\partial(\text{error})/\partial k = 0$ and is a function of R_L/R. The shaft angle with maximum error occurs near $k^* = 2/3$ for $0.2 < R_L/R$. As R_L/R becomes large, k^* becomes precisely $2/3$. If the error is to be less than 2% at this position, then $R_L/R \geq 7.2$.

The term *noise* in electrohydraulic circuits refers to any undesirable electrical circuit signal that is superimposed on the desired command signals. The noise signal will cause roughness in actuator movement or can cause mechanical parts to *buzz*. Such signals are often random in nature. Sources of circuit noise are many, including poorly soldered connections (dry joints), voltaic effects arising from an electrolyte in the presence of two dissimilar metals, unshielded wiring, and resistances of high impedance. Probably the major offender is the potentiometer. Noise due to vibration occurs when the wiper jumps away from the track; this can be controlled by careful adjustment of the contact arm pressure. Another cause of noise can be excessive rotational speed of the potentiometer shaft causing the wiper to bounce along the track. Noise will also be generated from dirt on the track and, in more unusual circumstances, by chemical action due to moisture, oil, or other liquids that may have accidentally penetrated the equipment. Amplification of such noise signals in the servo amplifier will result in transient spikes that can cause the servo amplifier or another instrument amplifier to saturate.

60.14 ELECTROHYDRAULIC SYSTEMS—DIGITAL

The electrohydraulic stepping motor is used as a high-torque, high-speed drive whose output motion is precise and repeatable. Some of the unique advantages are summarized in Benson[34] as: 1) position

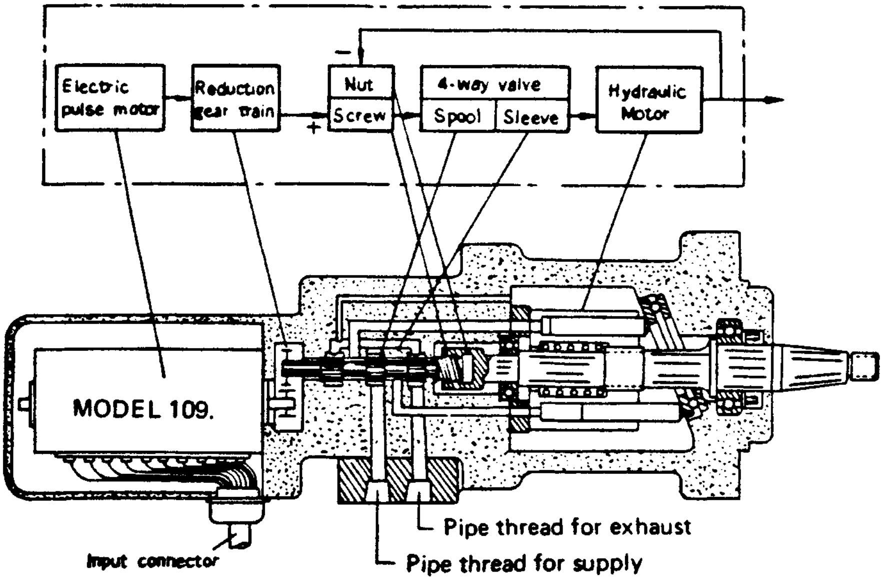


Fig. 60.26 Typical electrohydraulic stepping motor. (Courtesy of Fujitsu Ltd.)

feedback is not needed for positional control, unlike the electrohydraulic servo, 2) the operation of the unit is such that a microprocessor can often be interfaced directly (hence A/D and D/A converters are not needed), and 3) electrical tuning is not required, other than input of the command profile.

The electrohydraulic stepping motor has three main components: an electric stepping motor, a servo valve, and a rotary or linear actuator. A typical arrangement is shown in Fig. 60.26. The block diagram describes the function of the unit. The electric pulse motor controls the rate of flow and direction of flow of oil to the hydraulic motor.[35]

Command pulses are directed to the drive circuit amplifier. The pulses are fed in a phased sequence to the electric pulse motor, which, in turn, is connected through gears to a four-way spool valve. The gear on the stepping motor shaft has wide teeth so that the gear on the spool valve can move axially without disengaging. The other end of the spool has a lead screw, which is engaged with a nut on the hydraulic motor shaft. Thus, rotary motion of the stepping motor is transformed into axial motion of the spool, which in turn opens the four-way valve. This allows pressurized oil to flow to the hydraulic motor and cause it to rotate. As the hydraulic motor rotates, the nut on the motor shaft

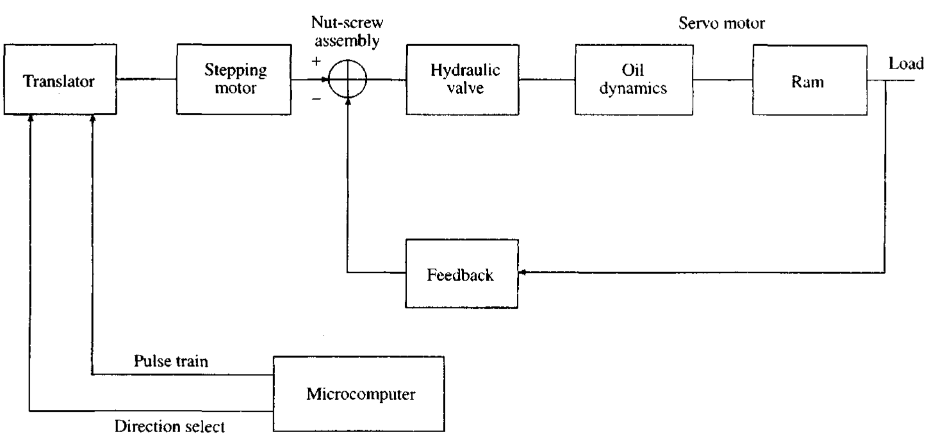


Fig. 60.27 Digital control system.

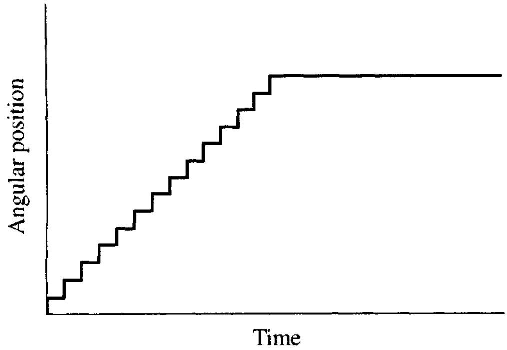


Fig. 60.28 Electrical stepping motor output.

rotates and moves in a direction opposite to the motion of the spool, returning the spool to its original position. Hence, the mechanical coupling of the spool valve and the hydraulic motor through the lead screw and nut form a negative feedback loop.

The electrical stepping motor is a fractional horsepower device that can be based on the concepts of variable reluctance, permanent magnet, rotating disc, or flexspline for the method of operation. In the case of the permanent magnet type, the magnetic rotor aligns itself with the magnetic orientation of the stator. When the stepping motor windings are correctly energized, the rotor will rotate one unit of angular displacement, typically $2\pi/200$ degrees/revolution, and then stop. Thus, any magnitude of motor rotation can be equated into the summation of some number of steps, as a result of a string of pulses from a microcomputer, applied at the input. Since the output position is not verified, the accuracy is solely a function of the ability of the motor to step through the exact number of steps commanded by the input. Hence, one important characteristic of a stepping motor is the maximum rate of the input pulses that can be followed.

The quantization of the motion into discrete steps is particularly well suited for a digital control device such as a microprocessor. The stepping motor drive (translator) accepts position and velocity profile commands in the form of a variable frequency pulse train and directional signal from the microprocessor. A complete block diagram is shown in Fig. 60.27.

The position of the electrical stepping motor shaft will be a sharply defined *staircase,* as shown in Fig. 60.28. The velocity of the shaft is, therefore, a function of the input pulse rate. The size of each step, both time- and position-wise, is determined by the pulse train and may also in practice have oscillatory overshoot if not well damped.

At frequencies exceeding 100 pulses/sec, the hydraulic valve in Fig. 60.26 does not have time to close. However, the number of pulses that the hydraulic motor lags behind the electric stepping motor is the amount of oil that should have been admitted into the hydraulic motor. This is remembered by the net and lead screw summing point at the valve shaft. Each stored pulse threads the screw into the nut a specific distance. When each stored pulse is used, the screw threads out the nut a specified distance, closing the four-way hydraulic valve a similar distance. This delay is a function of motor speed and is analogous to loop gain in a conventional analog electrohydraulic servo.

Since 100 pulses/sec is normally faster than the valve can respond, it remains open and follows the hydraulic motor smoothly. Therefore, the steps in Fig. 60.28 are not transmitted through the system.

REFERENCES

1. C. Stanley, "Fire Resistant Fluids," Paper F.2, B.H.R.A., 2nd Fluid Power Conf., University of Surrey, 1971.
2. A. Dalibert, "Fire Resistant Fluids," Paper F.3, B.H.R.A., 2nd Fluid Power Conf., University of Surrey, 1971.
3. E. S. Kelly, "Erosive Wear of Hydraulic Valves Operating in Fire Resistant Fluids," Paper F.4, B.H.R.A., 2nd Fluid Power Conf., University of Surrey, 1971.
4. J. Louie, R. T. Burton, and P. R. Ukrainetz, "Fire Resistant Hydraulic Fluids—State of the Art Review," *Proc. 37th Nat. Conf. on Fluid Power,* Chicago, IL, 1981, p. 285.
5. Anon., "Viscosity Considerations in the Selection of Hydraulic Fluids," *Hydraulics & Pneumatics,* Penton, 1980, pp. 23–26.
6. E. C. Fitch, L. E. Bensch, and R. K. Tessmann, *Contamination Control for the Fluid Power Industry,* Pacific Scientific, Montclair, CA, 1978.

7. B. A. Foord, "Specifying Fluid Filtration through Computer Simulation," in *Proc. 34th Nat. Conf. on Fluid Power,* Philadelphia, PA, 1978, p. 197.
8. J. Spencer, *Effective Contamination Control in Fluid Power Systems,* Sperry Vickers Publications, Troy, MI, 1980.
9. E. J. Forgeron and N. McCormack, "Controlled Maintenance System for Hydraulics through Used Oil Analysis," in *Proc. 34th Nat. Conf. on Fluid Power,* Philadelphia, PA, 1978, p. 187.
10. R. K. Tessman, "Ferrographic Measurements of Contaminant Wear in Gear Pumps," in *Proc. 34th Nat. Conf. on Fluid Power,* Philadelphia, PA, 1978, p. 179.
11. R. A. Collacott, *Mechanical Fault Diagnosis,* Chapman-Hall, 1977.
12. H. R. Martin and D. McCloy, *Control of Fluid Power,* 2nd ed., Ellis Horwood (John Wiley & Sons), New York, 1980.
13. *Aerospace Fluid Component Designer's Handbook,* RPL-TDR-64-25. AF Rocket Propulsion Lab., Edwards, CA, 1964.
14. G. Keller, "Hydraulic System Analysis," in *Hydraulics and Pneumatics,* Penton/IPC, Cleveland, OH, 1974.
15. J. Thoma, *Modern Oil Hydraulic Engineering,* Technical and Trade Press, 1970.
16. W. E. Wilson, *Positive Displacement Pumps and Fluid Motors,* Pitman Publications (and ASME Paper 48-SA-14), 1950.
17. D. E. Turnball, *Fluid Power Engineering,* Newnes–Butterworths, London, 1976.
18. H. R. Martin, "Effects of Pipes and Hoses on Hydraulic Circuit Noise and Performance," in *37th Nat. Conf. Fluid Power,* Chicago, IL, 1981, pp. 71–76.
19. H. E. Merritt, *Hydraulic Control Systems,* Wiley, New York, 1967.
20. *Fluid Power Designers Handbook,* Parker-Hannifin Corp., Cleveland, OH.
21. A. Esposito, *Fluid Power with Applications,* Prentice-Hall, Englewood Cliffs, NJ, 1980.
22. ISO 1219: 1976 Fluid Power Systems and Components—Graphical Symbols, International Standards Organization, 1976.
23. *BS 2917: 1977 Specification for Graphical Symbols Used on Diagrams for Fluid Power Systems and Components,* British Standards Institute, London, 1977.
24. USA SY32-10-1967, *Standard Symbols for Fluid Power Diagrams,* American National Standards Institute, New York, 1967.
25. R. Achariga, "Pitfalls the Inexperienced Fluid Power System Designer Should Avoid," *Hydraulic and Pneumatics,* Penton, 1982, pp. 152–168.
26. H. R. Martin, *The Design of Hydraulic Components and Systems,* Ellis Horwood (John Wiley & Sons), New York, 1995.
27. F. D. Yeaples, *Hydraulic and Pneumatic Power and Control,* McGraw-Hill, New York, 1966.
28. J. Kern, *Hydrostatic Transmission Systems,* Intertext Books, London, 1969.
29. N. F. Harpur, "Some Design Considerations of Hydraulic Servos of the Jack Type," in *Proc. Conf. on Hydraulic Seromechanisms, Int. Mech. Eng.,* London, 1953.
30. T. Keating and H. R. Martin, "Mathematical Models for the Design of Hydraulic Actuators," *ISA Transactions* **12**(2) (1973).
31. G. W. Dummer, *Variable Resistors and Potentiometers,* Pitman, 1963.
32. S. A. Davis and B. K. Ledgerwood, *Electromechanical Components for Servomechanisms,* McGraw-Hill, New York, 1961.
33. H. Neubert, *Instrument Transducers,* Oxford University Press, 1963.
34. B. Benson, "Rotary and Linear Electrohydraulic Stepping Actuators," in *Proc. 38th Nat. Conf. on Fluid Power,* Houston, TX, 1982.
35. T. Samar, "Electrohydraulic Stepping Motor Drives Machine Tools Without Feedback," in *Hydraulics & Pneumatics,* Penton, 1970.