CHAPTER 43
FLUID POWER SYSTEMS AND CIRCUIT DESIGN

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GLOSSARY OF SYMBOLS

a	Acceleration
A	Area
C	Coefficient
e	Efficiency
F	Force or load
g	Acceleration due to gravity
H	Power
L_N	Normal force component
m	Mass
N	Normal force, speed
p	Pressure
Q	Flow rate

S	Actuator stroke
t	Time
T	Torque
v	Velocity
V	Volume
W	Weight
γ	Specific weight
$\dot{\theta}$	Angular velocity
μ	Coefficient of friction
ω	Angular velocity

Fluid power technology is one of the three primary *energy transmission technologies* used throughout industry, the defense community, agriculture, and all aspects of productive activity in the industrialized world. In addition to fluid power, there are electric and mechanical systems. This chapter deals with the design of hydraulic and pneumatic systems by using the cycle-profile plotting techniques discussed in Chap. 42.

43.1 PRESSURE PLOTS

The pressure plots are the same as the load plots discussed in Sec. 42.4 except for the introduction of a constant reflecting the interface area over which the load is distributed. The equation is

$$p = F_1\left(\frac{1}{A_p}\right) \tag{43.1}$$

Note that the constant $1/A_p$ is different for each size of cylinder.

Once the load and pressure plots have been completed, the *level* of energy transfer occurring throughout the machine cycle has been defined.

43.2 FLOW PLOTS

The next step is to define the *rate* of energy transfer in the machine. This is a function of the velocities of the various piston rods (or motor shafts, if rotary motors are used).

To determine the required rod velocities, the designer can go back to the sequence chart. The time scale along the bottom of the chart indicates how much time is available to complete each part of the machine cycle. Time (Ref. [43.1], pp. 14, 15) has two basic implications in the design of a cycle:

1. It determines the flow rate requirement relative to the actuator motion pattern.
2. It determines the horsepower requirement of the circuit or branch.

After establishing time increments, the engineer must turn to the machine-element layouts to determine the length of stroke necessary to complete each motion. Next, the steady-state velocity required of the piston can be calculated, allowing for acceleration and deceleration. At this point the designer must make a choice of velocity patterns (Ref. [43.1], pp. 16, 19, 22, 26, 29, 347, 355, 358).

The superimposition of the flow plots for individual actuators has an important implication. If study of the sequence diagram indicates simultaneous operation of two or more cylinders or motors, their flow plots must be superimposed, as shown in Fig. 43.1. Such superimposition of the plots will give the designer an insight into the maximum flow rate required. This affects the selection of the pump or pumps and influences separation of circuit branches.

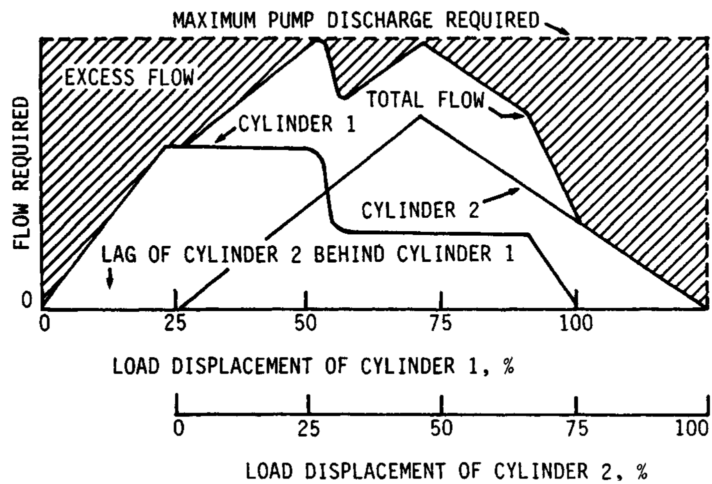


FIGURE 43.1 Flow plot of two actuators superimposed to indicate total flow rates. *(From Ref. [43.1].)*

43.3 POWER PLOTS

With the information developed for the pressure and flow plots, the designer can now make power plots. This is a necessary preliminary to selection of the prime mover. It is particularly useful in pointing out *power peaks* which might otherwise be hidden in averaged calculations. Such peak power demands, occurring when unexpected, could be great enough to stall an undersized prime mover. The fluid horsepower can be calculated from

$$H_f = \frac{pQ}{1714} \tag{43.2}$$

where p is in pounds per square inch (psi) and Q is in gallons per minute (gpm). The input horsepower of the prime mover is then

$$H_i = \frac{H_f}{e_o} \tag{43.3}$$

where e_o = overall efficiency of the pump.

43.4 CYCLE PROFILE

A complete *cycle profile* for a single actuator might look something like that shown in Fig. 43.2. Note that the cycle must be plotted for both directions of motion.

If the circuit designer has intelligently followed the cycle-profile procedure, a complete graphic portrayal of what should happen at any point in the cycle of operation of the machine is the result. The designer should be able to communicate any information necessary to an understanding of the operational capabilities and limitations of the equipment. Even more important, the designer should be able to spot any malfunctions of the machine much more quickly and surely than if she or he had to guess what combinations of events were supposed to transpire and compare them with what was observed (Ref. [43.1], Chap. 26).

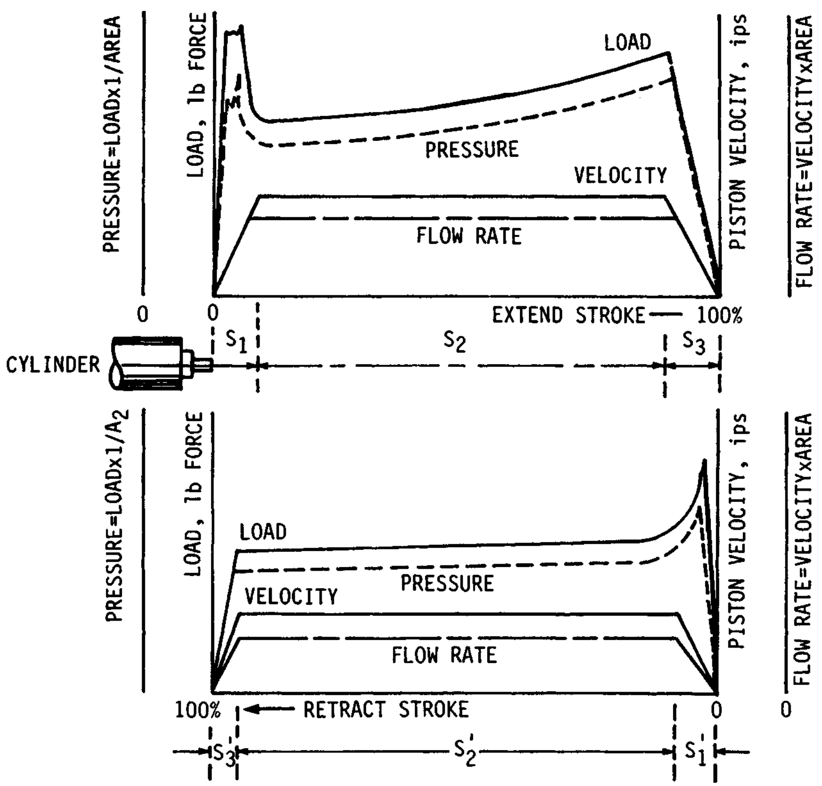


FIGURE 43.2 Cycle profile for a single actuator. *(From Ref. [43.1].)*

43.5 CIRCUIT DESIGN

Fluid power circuits can be thought of as consisting of four sections, as shown in Fig. 43.3. Section I represents energy output, where energy is transferred to the load across the hydromechanical interface. Section II is the control area; fluid switching

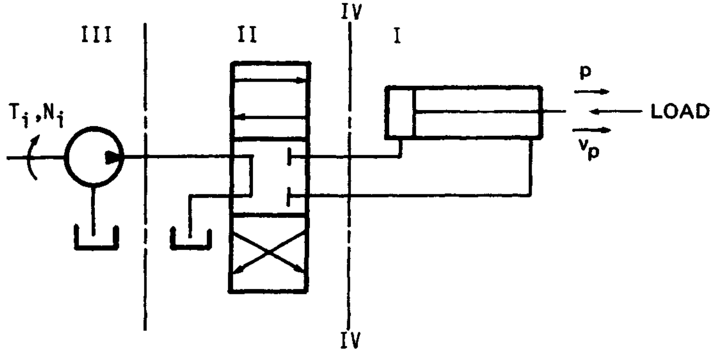


FIGURE 43.3 The four sections of a fluid power circuit. *(From Ref. [43.1].)*

and energy modulation are effected in this section. Section III is the energy input section; this is where the pumps are involved. Section IV is the auxiliaries area; this consists of piping and fittings and all the other components and equipment necessary to make a circuit work, including the fluid.

The pattern of Fig. 43.3 suggests a logical method for solving a circuit design problem. A format like that of Fig. 43.4 is helpful. Divide the sketch sheet into three areas by drawing vertical lines. The left-hand column is reserved for energy input devices, i.e., pumps. The right-hand column is for energy output devices, i.e., motors or actuators. The middle area is for control devices.

Next, sketch the symbols (Ref. [43.5]) for the output components in the right-hand column, in vertical array, as shown in Fig. 43.4. Then divide the page into rows by drawing horizontal lines which separate each actuator from its neighbors. We now have a matrix of sorts, with the columns representing circuit functions (energy output, energy control, and input), and the rows representing machine functions, as typified by the actuators.

The circuit designer can now select functions to match the requirements of the machine functions.

43.6 OPEN-LOOP AND CLOSED-LOOP CIRCUITS

Fluid power systems can be divided into two major groups: open-loop and closed-loop.

In a *closed-loop* system, a *feedback* mechanism continually monitors system output, generating a signal proportional to this output and comparing it to an input or command signal. If the two match, there is no adjustment, and the system continues to operate as programmed. If there is a difference between the input command signal and the feedback signal, the output is adjusted automatically to match command requirements.

There is no feedback mechanism in an *open-loop* system. The performance characteristics of the circuit are determined entirely by the characteristics of the individual components and their interaction in the circuit. A typical open-loop circuit is illustrated in Fig. 43.5*a*. Most industrial circuits fall into this category.

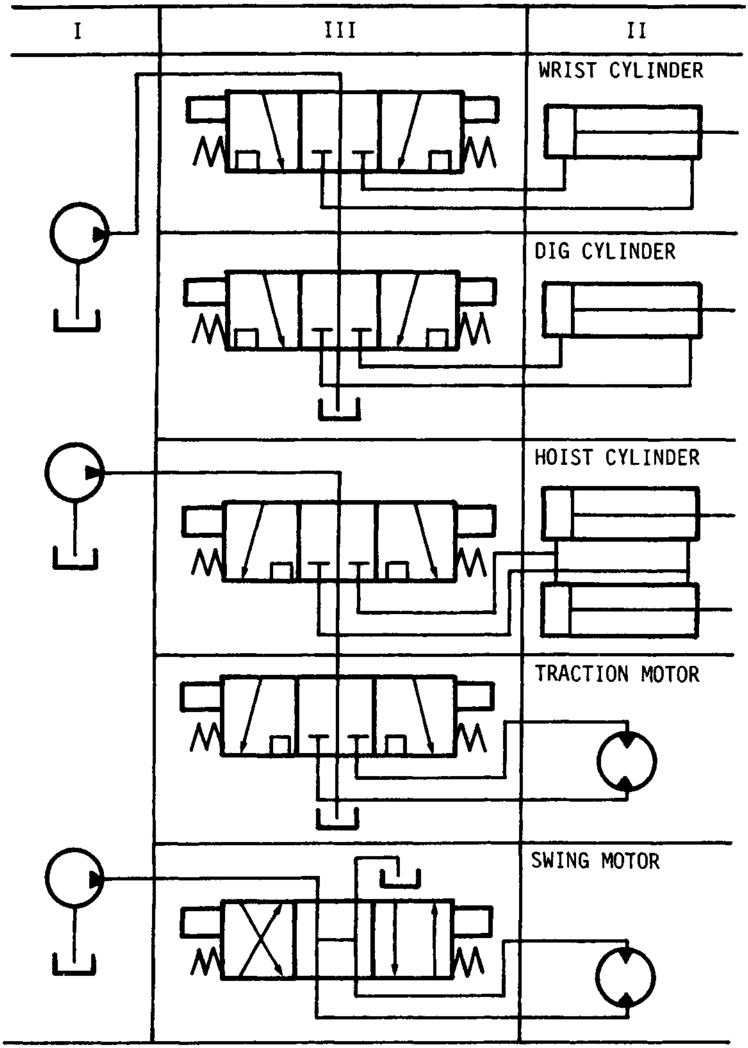


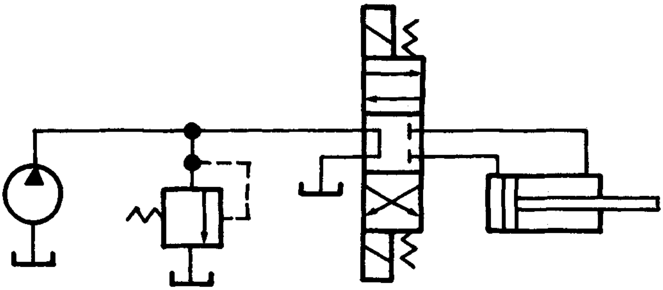
FIGURE 43.4 Graphical layout of a circuit design problem. *(From Ref. [43.1].)*

An *electrohydraulic servo system* is a feedback system in which the output is a mechanical position or function thereof; see Fig. 43.5*b*.

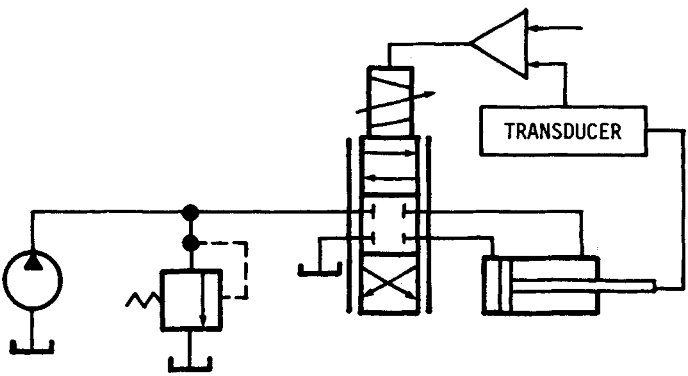
Open-loop circuits can be grouped by the functions performed or by the control methods.

43.6.1 Functions Performed

Classification of open-loop circuits by function is related to the basic areas of control used in a fluid power system:



(a)



(b)

FIGURE 43.5 (*a*) Typical open-loop circuit; (*b*) typical closed-loop circuit. *(From Ref. [43.1].)*

1. *Directional* controls regulate the distribution of energy (Ref. [43.4], Chap. 12, pp. 79–91, 151–164).
2. *Flow* controls regulate the rate at which energy is transferred by adjusting the flow rate in a circuit or branch of circuit (Ref. [43.4], Chaps. 10, 11, pp. 65–75, 164–168).
3. *Pressure* controls regulate energy transfer by adjusting the pressure level or by using a specific pressure level as a signal to initiate a secondary action (Ref. [43.4], Chaps. 8, 9, pp. 47–60, 143–151).

43.6.2 Control Methods

Directional Control. *Valve controls* make use of one of many types of directional control valves to regulate the distribution of energy throughout the circuit. These valves switch flow streams entering and leaving the valve.

Pump control is limited to reversal of the direction of flow from a variable-displacement reversible pump. *Fluid motor control* is similar to pump control; it uses reversible, variable-displacement motors.

Flow Control. *Valve controls* use one of several types of pressure-compensated or noncompensated flow control valves (Ref. [43.3], Chap. 9, pp. 91–98). The position of the flow control valve in the circuit determines the appropriate type to use. These are as follows:

1. *Meter-in* The flow control valve is in the supply line to the actuator and controls the energy transfer by limiting the rate of flow out of that actuator; see Fig. 43.6*a*.
2. *Meter-out* The flow control valve is in the return line from the actuator and controls the energy transfer by limiting the rate of flow out of that actuator; see Fig. 43.6*b*.

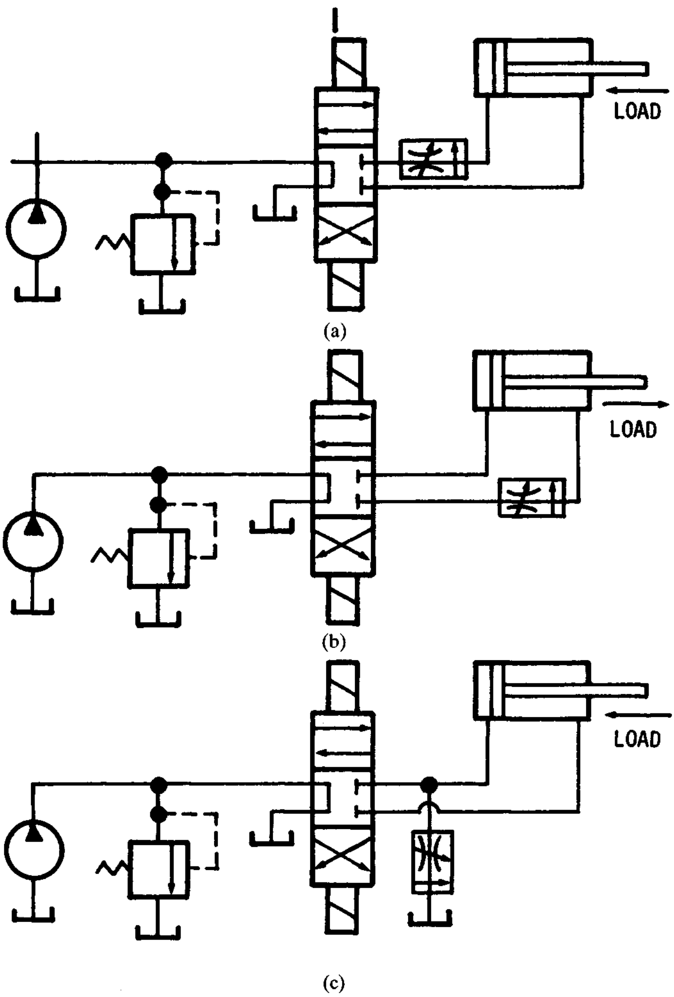


FIGURE 43.6 Valve controls for open-loop circuits. (*a*) Meter-in; (*b*) meter-out; (*c*) bleed-off. *(From Ref. [43.1].)*

3. *Bleed-off* The flow control valve is in parallel with the actuator. It limits the rate of energy transfer to the actuator by controlling the amount of fluid bypassed through the parallel circuit; see Fig. 43.6*c*.

Pump control involves the use of one of two methods, depending on the type of pump used. *Multiple* pumps provide a step variation in flow (Fig. 43.7*a*); *variable*-displacement pumps deliver infinitely (from zero to maximum) variable flows (Fig. 43.7*b*).

Fluid motor controls use techniques similar to pump controls, and this involves the use of multiple motors as in Fig. 43.8*a* for step variation or variable-displacement motors as in Fig. 43.8*b* for infinite variation in output speeds.

Pressure Control. *Valve controls* use one or more of six types of pressure control valves:

1. *Relief* valves limit the maximum energy level of the system by limiting the maximum operating pressure; see Fig. 43.9.
2. *Unloading* valves regulate the pressure level by bypassing the supply fluid to the tank at a low energy level. Unloading valves shift when the system pressure reaches a preset level; see Fig. 43.10.

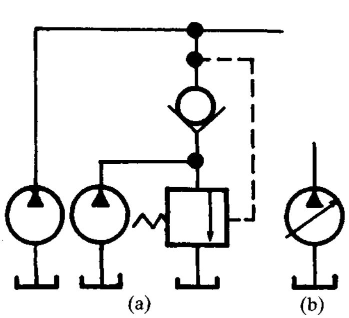


FIGURE 43.7 Pump controls for open-loop circuits. (*a*) Multiple pumps; (*b*) variable displacement. *(From Ref. [43.1].)*

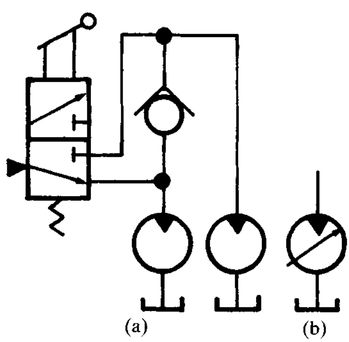


FIGURE 43.8 Actuator controls for open-loop circuits. (*a*) Multiple-fluid motors; (*b*) variable displacement. *(From Ref. [43.1].)*

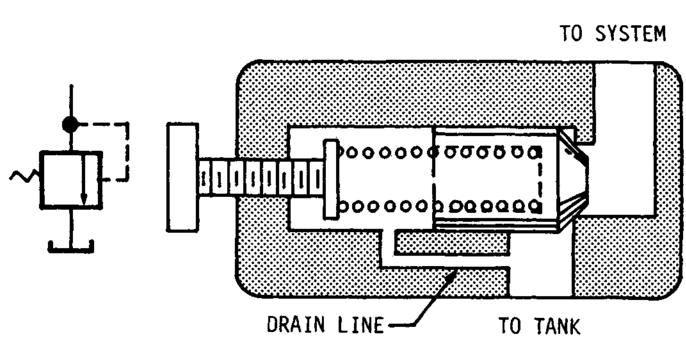


FIGURE 43.9 Pressure relief valve regulates system output fluid pressure. *(From Ref. [43.1].)*

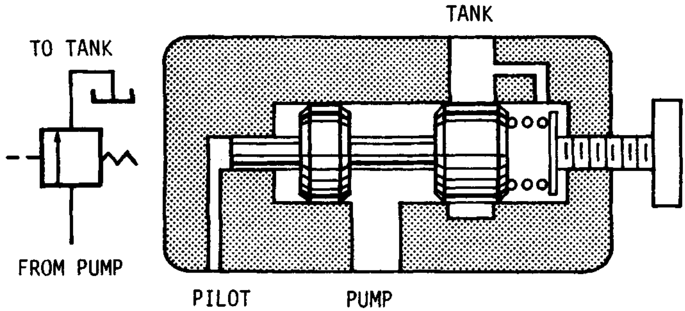


FIGURE 43.10 Pressure unloading valve unloads pump output to the tank at low pressure when high-pressure flow is not required. *(From Ref. [43.1].)*

3. *Sequence* valves react to a pressure signal to divert energy from a primary circuit to a secondary circuit; see Fig. 43.11.
4. *Reducing* valves react to a pressure signal to throttle flow to a secondary circuit, thus delivering energy at a lower level to the secondary than to the primary circuit; see Fig. 43.12.

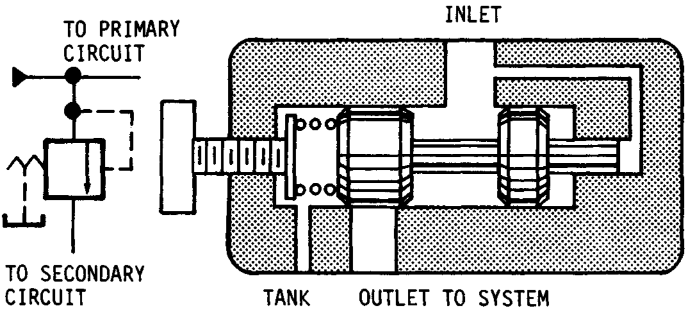


FIGURE 43.11 Sequence valve prevents fluid from entering one branch of a circuit before a preset pressure is reached in the main circuit. *(From Ref. [43.1].)*

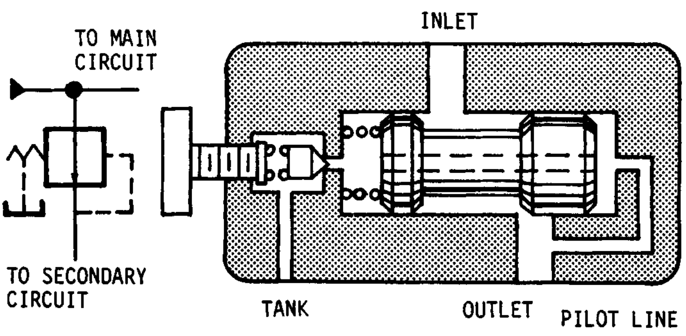


FIGURE 43.12 Pressure-reducing valve allows one branch of a circuit to operate at a lower pressure than the main system. *(From Ref. [43.1].)*

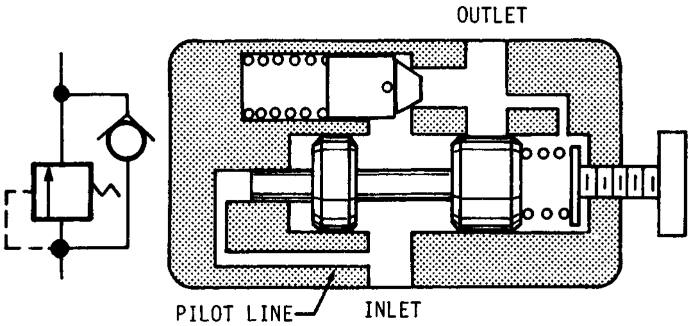


FIGURE 43.13 Counterbalance valve holds fluid pressure in part of a circuit to counterbalance weight on the external force. *(From Ref. [43.1].)*

5. *Counterbalance* valves control the potential energy differential across an actuator by maintaining a preset backpressure in the return line; see Fig. 43.13. Their purpose is to prevent a load from drifting.

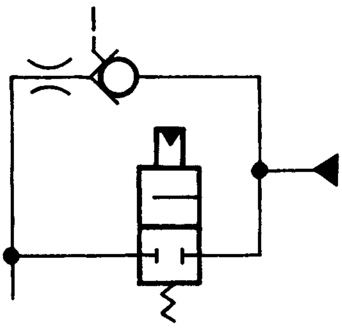


FIGURE 43.14 Decompression valve releases fluid at controlled rate to release energy stored in high-pressure system. *(From Ref. [43.1].)*

6. *Decompression* valves provide controlled release of energy stored in high-pressure systems, because of the elasticity in the system; see Fig. 43.14.

Pump control of pressure fluid in open-loop circuits is generally achieved with pressure-compensated variable-displacement pumps. Energy transfer is controlled by varying the flow from the pump in response to a pressure-level signal across the compensator; see Fig. 43.15.

Rotary actuator control of fluid pressure is not generally used.

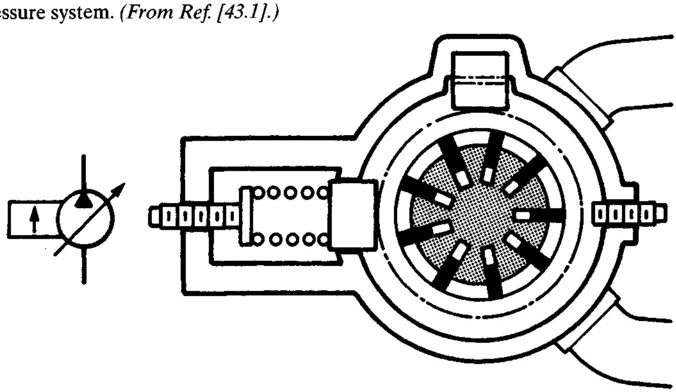


FIGURE 43.15 Pressure-compensated variable-displacement pump. Governor spring loads pump toward full-displacement position. As output pressure rises, it supplies the required force to stroke the cam ring toward deadhead position. *(From Ref. [43.1].)*

43.7 CONSTANT-FLOW VERSUS DEMAND-FLOW CIRCUITS—OPEN LOOP

The next step in implementing the circuit shown in Fig. 43.4 is to decide which basic type of circuit to use. An understanding of the characteristics of each is required. Fluid power circuits are broadly categorized as open-loop or closed-loop, as we have seen. Open-loop circuits are further subdivided into *constant-flow* and *demand-flow* circuits. Figure 43.16 illustrates the constant-flow principle in a simple circuit that has only one directional control valve. Figure 43.17 illustrates this for a multiple- or stack-valve installation where pump flow is returned directly to the tank only when *both* valves are in the neutral position. If either valve is shifted, normal four-way valve directional control will start.

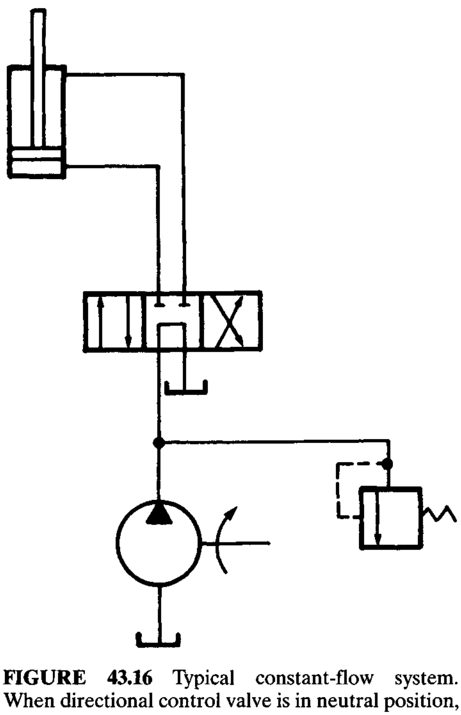


FIGURE 43.16 Typical constant-flow system. When directional control valve is in neutral position, pump output bypasses to tank through tandem center. *(From Ref. [43.1].)*

43.7.1 Pump Discharge Pressure

In constant-flow circuits, the pressure at which the pump discharges fluid is a function of the load resistance encountered by and reflected across the actuator; see Fig. 43.4. The system operating pressure generated by the load is a function of the actuator geometry. If the prime mover can satisfy the energy demand, it will do so. If not, the prime mover will stall; or, as is more likely to happen in actual practice, the relief valve will open to bypass fluid to tank—this wastes energy.

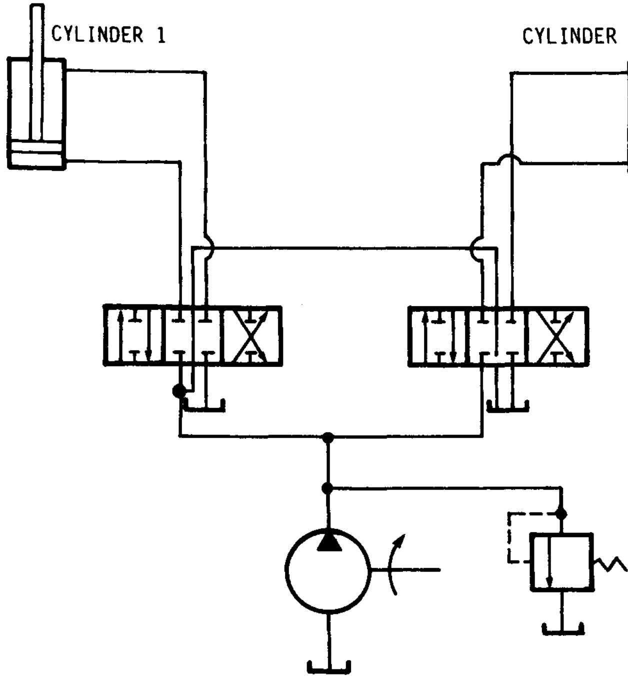


FIGURE 43.17 Constant-flow multiple-valve system. Pump output bypasses to tank only when *both* directional valves are in neutral position. *(From Ref. [43.1].)*

43.7.2 Relation of Pump Discharge to Actuator Speed

In constant-flow circuits, the pump output is *not* determined by the actuator's instantaneous speed requirements. The discharge rate is a function of pump displacement and its speed of rotation. Pump output and actuator displacement jointly determine a steady-state speed, according to the equations

$$v = \frac{Q_p}{A_p} \quad \text{(for a cylinder)} \qquad N_o = \frac{Q_p}{V_m} \quad \text{(for a motor)} \tag{43.4}$$

where v = velocity
Q_p = pump output
A_p = actuator area
N = output speed
V_m = motor displacement per revolution

Load inertia may preclude rapid acceleration to this steady-state speed; if it does, excess flow from the pump must return to the tank through the relief valve; see Fig. 43.18. At time t_0 the control valve shifts, porting pressure fluid to the actuator. There is a slight time lag caused by such factors as the compressibility of the oil in the system and throttling while the valve spool is shifting. Actuator flow Q_a then increases to full rated output $Q_a = Q_p$; see the vertical dotted line in Fig. 43.18.

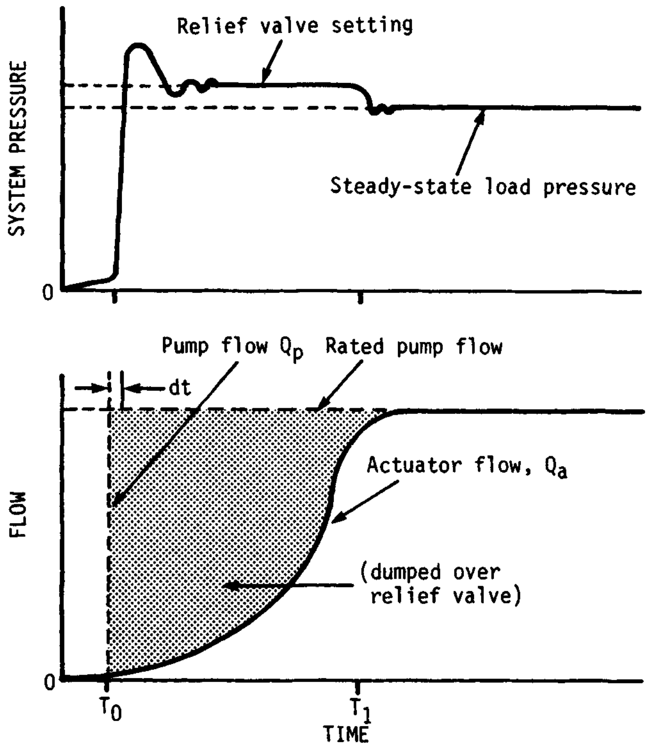


FIGURE 43.18 Should load inertia prevent acceleration to steady-state speed, excess flow from the pump returns to the tank through the relief valve. *(From Ref. [43.1].)*

At time t_0 the actuator velocity is zero. At time $t_0 + dt_0$ (the time that corresponds to full buildup of pump output), the actuator has not yet started to move. Therefore, actuator flow Q_a is zero at that instant. It could be demonstrated mathematically that for these conditions of finite pump output Q_p and zero actuator flow Q_a to coexist, the instantaneous acceleration of the actuator would have to be infinite. An infinitely powerful driving force would be required for this.

If we examine the pressure plot in Fig. 43.18*a* and compare it with the flow plot of Fig. 43.18*b*, we see that the fluid pressure rises rapidly and peaks at some level above the relief valve setting. This level depends on the response time of the relief valve; in addition, it depends on the internal slip in the pump, valve leakage, and actuator slip. Once the relief valve opens, fluid pressure in the system levels out at the relief valve setting.

Now consider the plot of the actuator flow rate Q_a. In any well-designed system consisting of one pump and one actuator, the pump output just matches the actuator input requirement at design speed. Thus, under steady-state conditions,

$$Q_p = Q_a \tag{43.5}$$

At the time dt_0, however, Q_p is equal to rated flow and Q_a is zero.

The actuator and load must accelerate from zero to design velocity. This takes a finite interval of time, from t_0 to t_1. During this time interval, Q_a increases until $Q_a = Q_p$ at time t_1, which is when the actuator reaches design speed. Note that at that time, the system pressure drops to the steady-state design level, and the relief valve closes. The shaded area between the two flow curves (Fig. 43.18*b*) represents the volume of oil returned to the tank through the relief valve during the acceleration period.

Because this complex sequence of events takes place in a fraction of a second, it is difficult to observe under normal operating conditions. And in most constant-flow circuit applications, it is not even a matter of concern. The designer would analyze this sequence only when dealing with applications that have high performance requirements or when an operating malfunction cannot otherwise be explained. Such a malfunction might occur if the pump's output flow rate and the actuator flow rate were badly matched.

For this reason, the designer must make sure that these two quantities are properly matched, especially when designing multibranch circuits. If a pump must be sized for multibranch circuit operation, as is frequently the case, the designer must choose a pump with a capacity that equals peak flow requirements. Note that the capacity of such a pump exceeds the fluid needs of a single actuator.

Sizing the Actuator. In constant-flow circuits, the designer tries to size actuators to meet speed requirements as a function of pump output. For example, a cylinder might be selected so that

$$\frac{A_p S}{t} = Q_p \tag{43.6}$$

where A_p = piston area
S = cylinder stroke
t = time
Q_p = pump flow rate

In some instances this formula may call for a cylinder with a capacity larger than that required for force output alone.

The designer would ordinarily select a fluid motor with a capacity (at desired operating speed) equal to the rated pump output.

$$Q_p = V_a N \tag{43.7}$$

Unloading the Pump. In a constant-flow circuit, the directional control valve unloads the pump when the valve is in its neutral position. This is an advantage in that auxiliary controls are not required. By unloading the pump, the designer reduces unnecessary energy dissipation during dwell or passive intervals in the cycle, thus minimizing the generation of heat. Care must be taken that the directional control valve selected has enough capacity to bypass the *full* pump output *without* causing excessive pressure drop.

Output Speed Control. One way to control actuator speed in a constant-flow circuit is to restrict flow with a metering or flow control device. The most common metering approach uses one of the many types of flow control valves in combination with one of the basic methods of flow control described previously. Another approach takes advantage of the throttling characteristics of the directional control valve. This approach is frequently adopted in circuits equipped with manually operated and proportional control valves. The designer must remember that any flow control method that uses throttling creates energy losses with attendant heat generation.

Application Problem. Let us consider a typical example of constant-flow analysis. As previously discussed, a thorough analysis of system objectives is fundamental to good circuit design. The cycle-profile technique was suggested as one approach to orderly design. Remember that this approach divides the circuit into sections; see Fig. 43.19. Note that the load is primarily a resistive one. Therefore, under steady-state conditions,

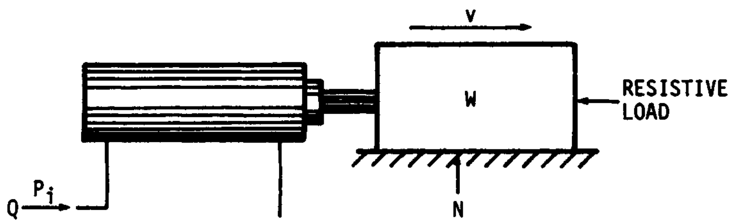


FIGURE 43.19 Example of a typical resistive load system. *(From Ref. [43.1].)*

$$F_a = ma = \frac{W}{g}\left(\frac{v_2 - v_1}{t}\right) \tag{43.8}$$

where F_a = force required to accelerate load
t = time
v_1 = initial velocity
v_2 = final velocity
W = weight of load, actuator elements, machine-tool carriage, etc.

Since at startup $v_1 = 0$, we have

$$F_a = \frac{Wv_2}{gt}$$

The equation

$$F_R = p_i A_p \tag{43.9}$$

states a relationship among the resistive force F_R required to overcome the resistive load, the cylinder piston area A_p, and an initial system pressure p_i. However, the equation is not complete because we must consider several other factors: the frictional component of the resistive load F_f, the breakaway friction F_{fb}, and the running friction F_{fr}.

The frictional component of the resistive load is given by

$$F_f = \mu N$$

where μ = coefficient of friction and N = normal force. Since

$$N = W + L_N$$

where W = weight of the load, actuator elements, etc., and L_N = normal component of any applied force (such as cable tension or cutter reaction), we may write

$$F_f = \mu(W + L_N)$$

We have noted that breakaway and running friction also enter into the relationship. We must distinguish between these two quantities because the coefficient of friction varies between the static condition μ_s and the dynamic condition μ_d, so that $\mu_s > \mu_d$. The breakaway friction is

$$F_{fb} = \mu_s(W + L_N)$$

and the running friction is

$$F_{fr} = \mu_d(W + L_N)$$

Note that breakaway friction force F_{fb} may vary during the cycle because of a variable normal component force L_N. (There is another frequently neglected component of the total resistance energy requirement which, in some cases, cannot be overlooked—the energy needed to accelerate the mass of oil within the system. We do not discuss this component here, since it is beyond the scope of this presentation. See Ref. [43.1], Chap. 27.)

Thus the total resistive force at *breakaway* is

$$F_{Rb} = p_i A_p + \mu_s(W + L_N) \tag{43.10}$$

and the total resistive force at *running speed* is

$$F_{Rr} = p_i A_p + \mu_d(W + L_N) \tag{43.11}$$

We can now complete the equations for the total load reflected at the actuator. We distinguish three cases as follows:

1. The load at breakaway:

$$F_{ab} = \frac{W v_2}{g t} + p_i A_p + \mu_s(W + L_N) \tag{43.12}$$

2. The load while running and while the system is accelerating to a constant speed:

$$F_{ar} = \frac{W v_2}{g t} + p_i A_p + \mu_d(W + L_N) \tag{43.13}$$

3. The load while running at steady-state velocity:

$$F_{ar} = p_i A_p + \mu_d(W + L_N) \tag{43.14}$$

Figure 43.20 shows a typical load-cycle plot for the application. Note that $0 < t < dt$ represents the short interval during which breakaway from zero velocity takes place. This is a transient state, and it would be difficult to plot without use of an analytical instrument such as an oscilloscope.

Qualitatively, however, load components are functions of static friction and the resistive load itself. In the interval $dt < t < \Delta t$ (also of short duration), the load and actuator masses are accelerated. Again, this is a transient state. The components are dynamic friction, resistive load, and the load due to the acceleration of a mass.

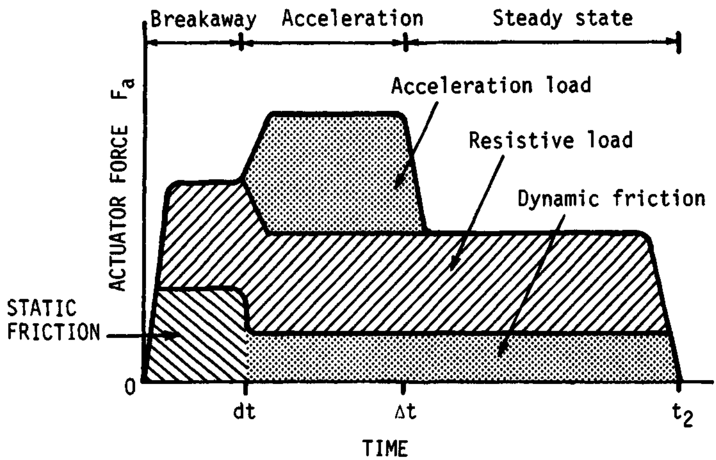


FIGURE 43.20 Typical load-cycle plot for the system illustrated in Fig. 43.19. Note the nonlinear time scale of the abscissa. *(From Ref. [43.1].)*

Beyond $\Delta t < t$, acceleration of the load is essentially a steady-state quantity; at least we usually assume this, even if it is not quite true in actual practice. In this time interval, the important components are resistive load and dynamic friction.

Figure 43.21 shows the shape of a typical system pressure plot determined from the load-cycle plot shown in Fig. 43.20. Note that the highest pressures appear when $0 < dt_1 < \Delta t$. These are the familiar transients, frequently seen on oscilloscopes, caused by the breakaway phenomenon and the superimposition of acceleration forces on normal load resistance. Figure 43.21 also shows that a steady-state pressure p_{ss} is achieved when the load speed corresponds to the actuator design speed.

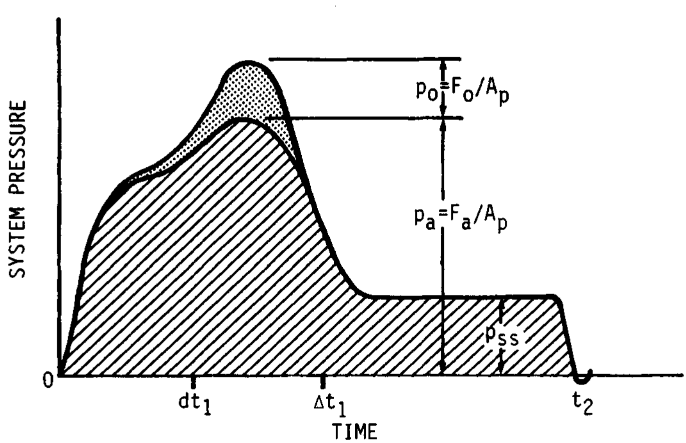


FIGURE 43.21 Shape of typical pressure plot determined from the load-cycle plot shown in Fig. 43.20. Note that the highest pressures appear in the interval $0 < dt_1 < \Delta t_1$. *(From Ref. [43.1].)*

The relief valve setting at this point is ordinarily between p_{ss} and $p_a + p_o$. If Δt_1 is brief in comparison with t_2, the relief valve setting can be close to p_{ss}, because the pressure transient will be so short that the relief valve cannot respond—or even if it could, the relatively small quantity of oil bypassed would not affect circuit operation significantly. If, however, Δt_1 is large in comparison with t_2, the relief valve will have to be set higher.

When interpreting such pressure plots, you should bear in mind that the system will develop a fluid pressure p_o caused by the acceleration of the oil column in the line that connects the pump and the actuator. This pressure is superimposed on the other pressures reflected by the load at the actuator; this relationship develops because p_o is *not* load-reflective and occurs only in the oil in the line to the actuator.

Characteristics of Constant Flow. In light of this discussion, remember these characteristics of constant-flow circuits:

1. The pump discharge pressure is a function of the load resistance and must build from zero.
2. The pump output is not determined by the actuator speed requirements.
3. Actuators are sized to meet speed requirements as a function of the pump output.

Statements 2 and 3 make sense only if we assume that the pump in a multibranch circuit is sized to accommodate maximum system demand, no matter when or where it occurs. During parts of the cycle when demand is not at peak, the pump output will exceed that required by one actuator. In the simple example of Fig. 43.19, the pump and actuator displacements would have to be matched. Let us analyze the basic circuit of Fig. 43.22 required for the application.

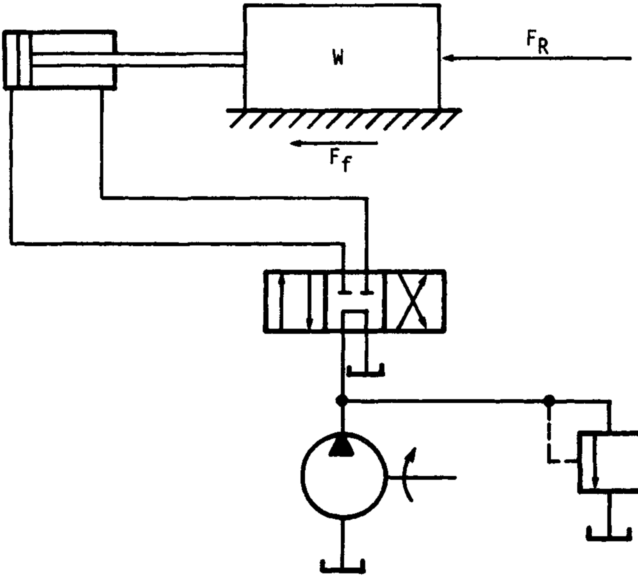


FIGURE 43.22 Basic circuit for application illustrated in Fig. 43.19. *(From Ref. [43.1].)*

Assume that the load resistance of the circuit calls for a piston area A_p for a design pressure p_i and that the load must be moved through a stroke S. The cylinder will displace a volume

$$V_a = A_p S \tag{43.15}$$

If the job to be done requires that the load be moved in t seconds, then the necessary flow rate to the cylinder is

$$\frac{V_d}{t} = \frac{A_p S}{t} = Q \tag{43.16}$$

where Q is in cubic inches per second.

(At this point in the analysis, the designer must check the columnar strength of the piston rod, which may turn out to be the critical factor. If a larger rod is needed, the cylinder bore may have to be increased accordingly. Such a change would, in turn, require adjustment of the pump-displacement calculation.)

Assuming that we have satisfactorily calculated the required pump output Q_p, we can complete the input segment of the circuit we are using as an example; see Fig. 43.22. By the very nature of this circuit, we must use the tandem-center four-way valve shown. Also, a constant-flow circuit requires a relief valve with a fixed-displacement pump.

The main functional element still missing from the simple circuit is a method for speed control. Since it was indicated that speed control could be accomplished only by throttling, the manually operated directional control or proportional valve can be used to throttle flow. In all circuits, the pressure drop across the valve has the effect of reducing the pressure available at the actuator. Consequently, the force available to accelerate the load and overcome friction is reduced.

A flow control valve used in a meter-in circuit (Fig. 43.23) would have essentially the same effect, unless it were used in a bleed-off circuit. In this case, the flow to the cylinder would actually be reduced. One could use a bleed-off circuit (Fig. 43.24) for minor speed adjustment, but a valve in a meter-in circuit would be preferred for adjustment over a wide flow range. A meter-out circuit could be used, but since the load in this example is a resistive one, this alternative would have little advantage over a meter-in circuit.

43.8 DEMAND-FLOW CIRCUITS

A closed-center circuit is one in which the port from the pump to the directional control valve is blocked when the valve is in its neutral position (see Fig. 43.25). Typically, demand-flow circuits are equipped with a fixed-displacement pump, an unloading valve, and an accumulator, as shown in Fig. 43.26, or a variable-displacement, pressure-compensated pump, as shown in Fig. 43.27. Closed-center circuits are more accurately characterized as demand-flow circuits.

43.8.1 Fixed-Displacement Pump Circuits

In demand-flow circuits that use a fixed-displacement pump, unloading valve, and accumulator, fluid pressure from the pump is not directly determined by actuator force requirements. As Fig. 43.26 illustrates, the pump charges the accumulator to design pressure when the directional control valve is centered.

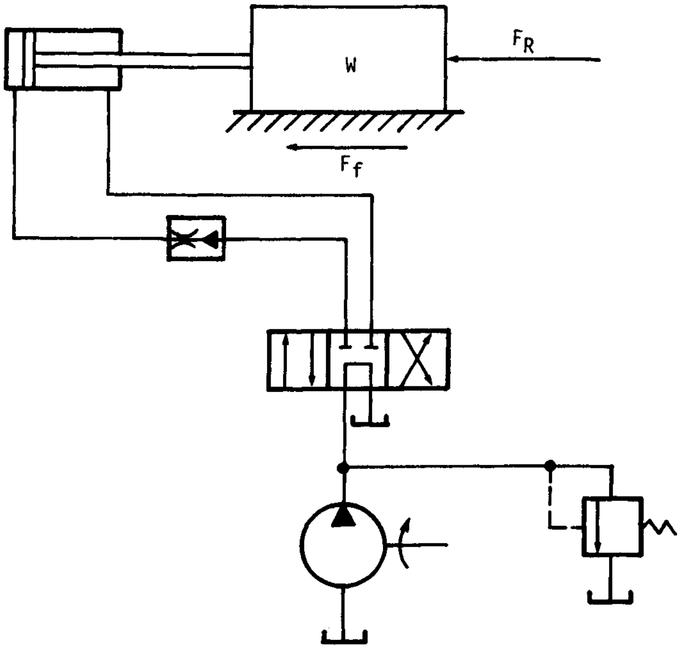


FIGURE 43.23 Flow control valve in meter-in circuit provides speed control for circuit in Fig. 43.22. *(From Ref. [43.1].)*

Maximum design pressure in the circuits is controlled by the spring setting of an unloading valve. When this setting is reached, the valve opens and bypasses oil to the tank, at low pressure. Note that the pilot signal to the relief valve is sensed downstream of a check valve placed between the pump and the accumulator. The check valve prevents the unloading of the accumulator as well as the pump.

When the directional control valve is shifted so that it ports oil to the actuator, the full design pressure (as stored in the accumulator) is immediately available to the system. As the cylinder moves, oil is forced from the accumulator by the compressed gas charge behind the oil. After a time interval, system pressure drops because of the expansion of the gas charge in the accumulator.

At some pressure level for which it has been designed, the unloading valve closes and causes output from the pump to reenter the system rather than bypass to the tank. At this time, the pump will do one of two things:

1. Add its output to that from the accumulator at the lower pressure level.
2. Recharge the accumulator to a higher pressure.

Which event occurs is a function of many other factors.

Some accumulator circuits are designed so that the accumulator supplies all the oil used during the active part of the cycle. It cannot do so at constant pressure, because the pressure of the gas charge drops as the gas expands when the oil flows out of the accumulator. The load cycle must be designed so that the system can still function at

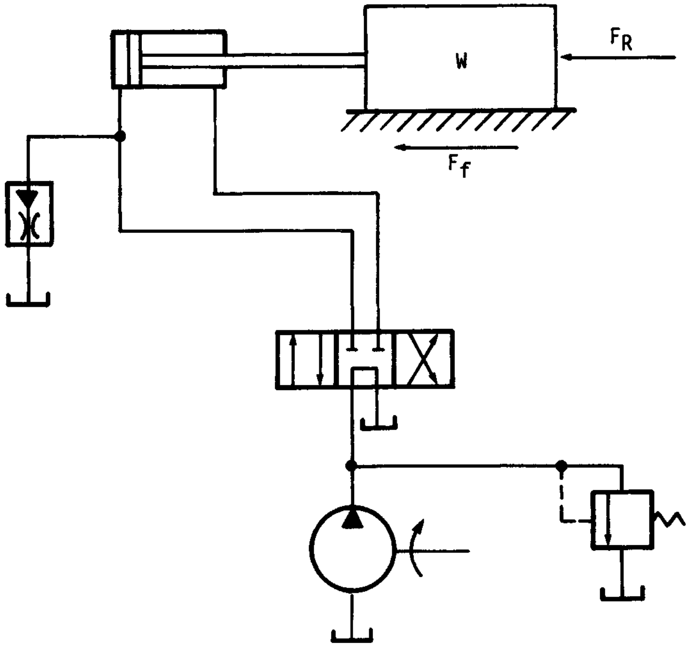


FIGURE 43.24 Bleed-off circuit could be used to control flow to cap end of cylinder, but meter-in circuit is preferred for fine adjustments. *(From Ref. [43.1].)*

the lowest pressure level delivered by the accumulator. This design feature is used where the active, or work, segment of the cycle is rather short and is followed by a relatively long passive, or dwell, segment during which the pump recharges the accumulator. In such circuits, the pump is sized to charge the accumulator during the work-cycle dwell segment; see Fig. 43.28.

Example 1. Assume a circuit similar to that in Fig. 43.26, in which an accumulator supplies 924 cubic inches (in^3) of oil to the circuit in 10 seconds (s). What is the required pump output rate if the dwell time between work periods is 50 s?

Solution. The required pump discharge, in gallons per minute, is

$$Q = \frac{924}{50}\ (\text{in}^3/\text{s}) = \frac{18.5(60)}{231}\ (\text{gpm}) = 4.8\ \text{gpm}$$

The horsepower required to drive the pump is

$$H_1 = \frac{pQ}{1714} = \frac{1000(4.8)}{1714} = 2.8\ \text{hp}$$

In circuits that differ from the one illustrated in this example, the accumulator is frequently used to supplement the pump during brief periods when high-rate flows are needed. Thus, if the design calls for a high flow rate for a short time in the active part of the work cycle, the engineer can use a smaller pump in conjunction with an

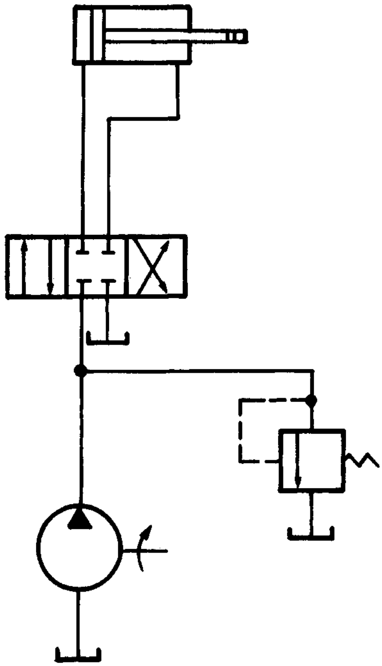


FIGURE 43.25 In a simple demand-flow circuit, the line from the fixed-displacement pump to the valve is blocked when the directional control valve is in the neutral position. *(From Ref. [43.1].)*

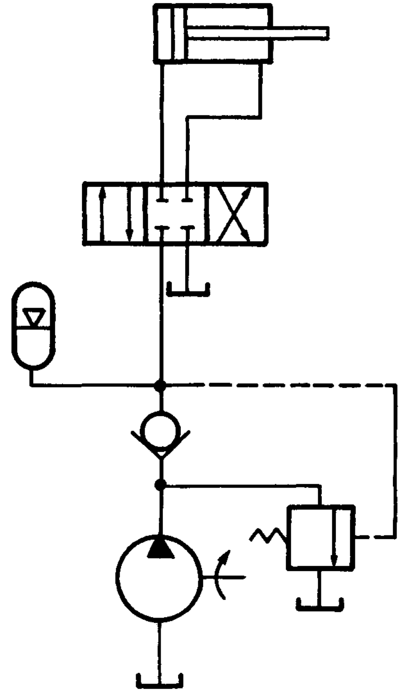


FIGURE 43.26 In a simple demand-flow circuit powered by a fixed-displacement pump, an accumulator is added to supply full design pressure immediately. *(From Ref. [43.1].)*

accumulator which it charges during the passive part of the work cycle. When the operator shifts the directional control valve, the accumulator output flow is *added* to the pump flow. Note that the combined flows may exceed several times the output of the pump alone. However, this condition will exist for only a very short time. In designs where a peak flow of short duration may be desirable, this configuration may be much more economical than one that relies on one large pump; see Fig. 43.29.

Example 2. Assume that a pump is used instead of an accumulator to supply the required oil in Example 1. If the operating pressure is 1000 psi, what is the difference in horsepower required to drive the pump in these two examples?

Solution. The total flow to the system is

$$Q_t = Q_p + Q_a = 4.8 + \frac{924}{10}\,\frac{60}{231} = 28.8 \text{ gpm}$$

The power required to drive the pump in Example 1 was 2.8 hp. In this example, the power required is

$$H_2 = \frac{pQ}{1714} = \frac{1000(28.8)}{1714} = 16.8 \text{ hp}$$

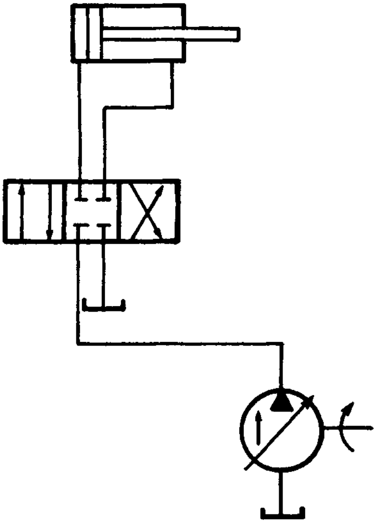


FIGURE 43.27 A variable-displacement, pressure-compensated pump supplies pressure fluid to this demand-flow circuit. Note the absence of a relief valve. *(From Ref. [43.1].)*

Therefore, using a pump instead of an accumulator to supply the required short-duration, high-volume flows increases the horsepower requirement by 600 percent.

In these types of circuits, actuators are sized to meet the force requirements based on load-cycle analysis. Frequently actuators, particularly cylinders, can be sized smaller than in comparable, constant-flow circuits. This is true because in constant-flow circuits, the cylinders must be sized to provide the required speed based on available pump output. In demand circuits, however, a given force is available that accelerates the load at a rate proportional to the mass. Thus, the cylinder demands oil from the accumulator in proportion to its instantaneous velocity. The accumulator delivers only on demand because, unlike a pump, it is not a positive-displacement device.

43.8.2 Pressure-Compensated Pumps

We discussed the simplest form of pressure-compensated pump in a demand-flow circuit in Fig. 43.27. If the demand-flow circuit uses a pressure-compensated pump

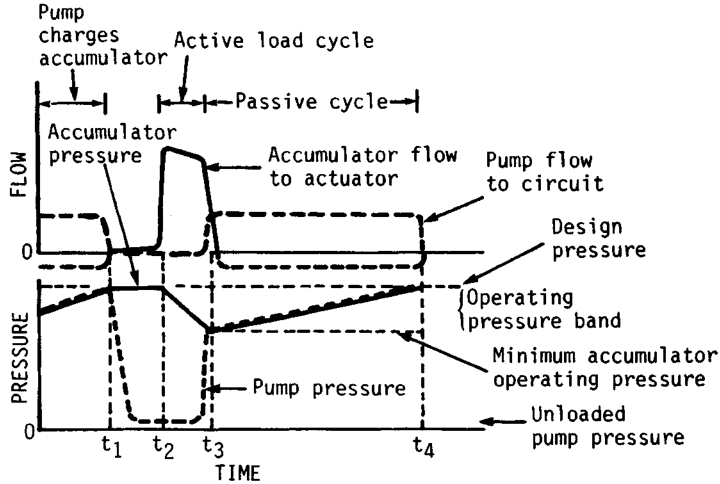


FIGURE 43.28 In systems with cycles that have short work segments and long dwell segments, the pump is sized to charge the accumulator during dwell. *(From Ref. [43.1].)*

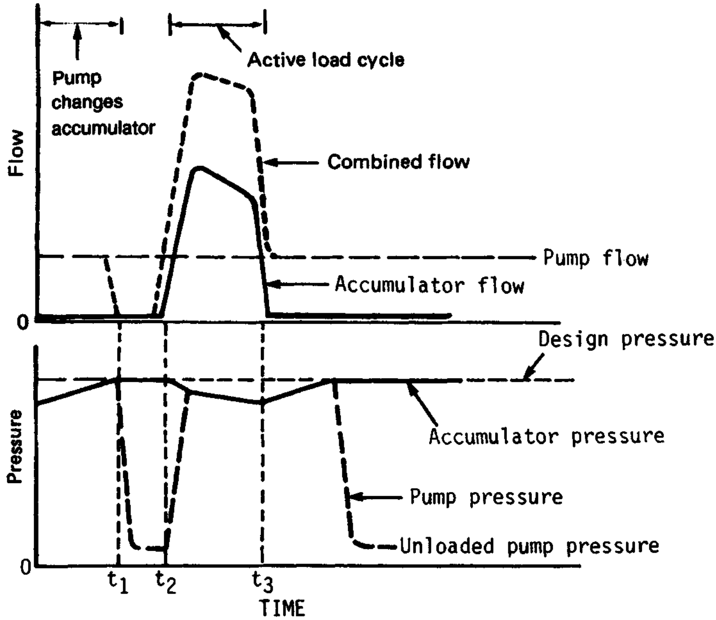


FIGURE 43.29 In systems where short-duration peak flows are needed, an accumulator can often supplement pump output during short periods of high-flow needs. *(From Ref. [43.1].)*

(Ref. [43.1], Chap. 15, pp. 116–122), then the compensator setting determines the maximum circuit pressure; see Fig. 43.30. Pump output is constant until the system reaches a given pressure, called the *cutoff pressure.* At this point the force acting on the compensator begins to exceed the force of the control spring that holds the pump on stroke.

Now, as pressure increases, the pump starts to move off stroke to reduce displacement. The slope of the curve of this decreasing displacement can be controlled by the design of the compensator. Thus, the designer can specify a sharp or gradual cutoff, as required. When fluid pressure in the system reaches the level known as *deadhead pressure,* the pump output flow is zero. The only power consumed by the pump at deadhead is the relatively small amount required to overcome mechanical losses and compensate for internal leakage. The pump maintains full deadhead pressure in the system at this low power input.

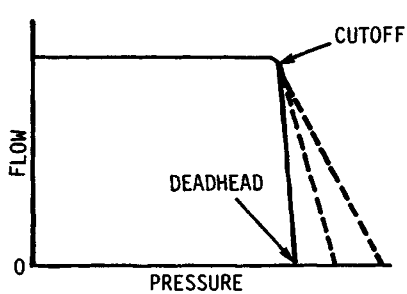


FIGURE 43.30 In a demand-flow circuit equipped with a pressure-compensated, variable-displacement pump, the compensator setting determines maximum circuit pressure. *(From Ref. [43.1].)*

As indicated, the compensator setting determines the upper limit of system pressure. Up to its maximum capability, pump output is a function of the ability of the actuator and load to respond to the force exerted on both.

In our discussion of load response in systems equipped with a fixed-displacement pump, we stated that it would require infinite acceleration for the load to absorb the entire pump output the instant it delivers fluid to the actuator. With a pressure-compensated pump, full force can act on the actuator and load, but there will be no flow until the load starts to accelerate. Thus, a system that uses a pressure-compensated pump is a demand system, as is the case with an accumulator.

A pressure-compensated pump functions as its own relief valve, shifting to deadhead conditions if and when an excessive load is applied. If a designer decides to use a relief valve for fail-safe protection, its pressure setting must be approximately 250 to 300 psi higher than the pressure of the pump at deadhead to minimize system instability. A rupture disk can also be used to provide fail-safe protection.

In a demand-flow circuit equipped with a pressure-compensated pump, pump delivery is related to the actuator speed requirement, as illustrated in Fig. 43.31. From time zero to time t_0, the control valve is in neutral position and the pump at deadhead—that is, the pump maintains maximum pressure at zero delivery. At time t_0 the control valve shifts, porting pressure oil to the actuator; see Fig. 43.27. Thus, full deadhead pressure acts on the actuator.

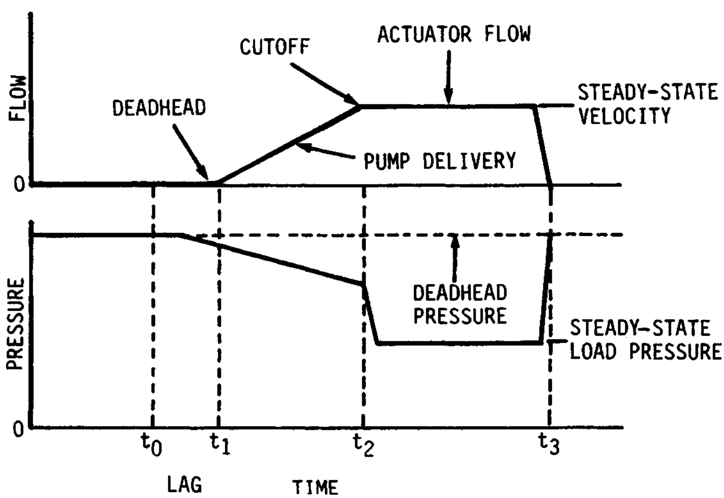


FIGURE 43.31 In a demand-flow circuit with a pressure-compensated pump, the pump output is related to actuator speed requirements. *(From Ref. [43.1].)*

Because the actuator cannot accelerate instantaneously, the pump output remains at zero for a short time $t_0 < t < t_1$; see Fig. 43.31. During this interval, the actuator begins to move. The pressure drops to some level below deadhead—required to accelerate the load level. Simultaneously, the pump moves on stroke.

If the acceleration force requires a pressure greater than the cutoff pressure, the pump will compensate by reducing its output flow rate. This new output flow rate will be lower than that corresponding to the cutoff pressure, but higher than the flow rate corresponding to deadhead pressure. In this sense, a demand-flow circuit with a pressure-compensated pump is a self-regulating system.

Between times t_1 and t_2 in Fig. 43.28, the load accelerates to steady-state speed. By time t_2, the pump has been stroked to full displacement, the load stops accelerating, and the system pressure drops to some value corresponding to the steady-state resistive load. At time t_3, the actuator hits a mechanical stop, or the end of its stroke, and the fluid pressure rises immediately. The pump is destroked, and its output drops to zero; it is at deadhead until the control valve shifts to retract the cylinder. In this circuit no pressurized oil flows over a relief valve; the pump supplies precisely what the system demands.

43.8.3 Flow-Compensated Pumps

The simplest form of a flow-compensated pump is shown schematically in Fig. 43.32*a*. In these pumps, a control orifice senses the flow rate, with the pressure drop across the orifice being proportional to flow rate (Ref. [43.3], pp. 75ff) according to the equation

$$Q = C_d A_o \sqrt{\frac{2g\,\Delta p}{\gamma}} \tag{43.17}$$

where C_d = discharge coefficient
A_o = cross-sectional area of orifice opening
g = acceleration due to gravity
Δp = pressure differential
γ = specific weight of fluid

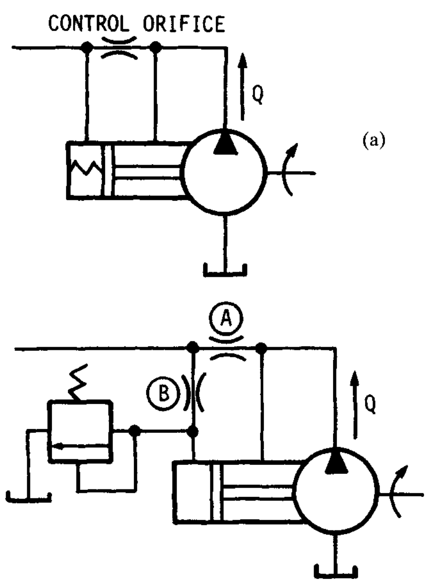


FIGURE 43.32 (*a*) Schematic of simple flow-compensated pump. (*b*) Diagram of flow- and pressure-compensating control. Fixed orifice *A* senses flow; fixed orifice *B* is in line to compensator. *(From Ref. [43.1].)*

This equation indicates that pressure drop is a function of the square of the flow. The induced pressure drop is felt by the compensator control piston, which adjusts pump output in proportion to flow.

Figure 43.32*b* illustrates a flow rate and pressure-compensating control. This configuration also uses a fixed orifice *A* to sense the flow rate. In addition, it has a second fixed orifice *B* in the line to the spring end of the compensator. A pressure control valve regulates the pressure in the spring-chamber end of the compensator. When pressure in this chamber matches that of the valve setting, the valve opens and bypasses oil to the tank, creating a pressure drop across orifice *B*. Thus the total pressure differential imposed across the compensator piston is the sum of the two pressure drops. This value will exceed the pressure drop induced across orifice *A* by flow alone.

43.9 HYDRAULIC VERSUS PNEUMATIC SYSTEMS

The material discussed previously dealt with fluid power systems, that is, energy transmission systems using a fluid as the transfer medium. The technology covered has been *hydraulics* based on incompressible-fluid liquid transfer media. *Pneumatics* is the second area of fluid power technology in which a compressible fluid—gas—is used as the transfer medium.

Functionally, hydraulics and pneumatics are similar. What has been said about functional design of hydraulic systems is applicable for pneumatic systems as well. The major differences lie in the areas of

1. Hardware used to implement the functions
2. Energy levels usually involved in the applications of each
3. The vastly different characteristics of liquids and gases

Additional information is contained in Ref. [43.1].

43.10 PNEUMATIC CIRCUITS

Pneumatic systems can be classified in two primary functional categories:

1. *Power* systems
2. *Logic* systems

Pneumatic power systems can be further classified as

1. Sequencing circuits, which encompass the majority of industrial applications and are usually open-loop systems
2. *Servo* systems, which are closed-loop, proportional systems

43.11 EFFECT OF FLUID CHARACTERISTICS ON ACTUATOR PERFORMANCE

The performance of the fluid in a hydraulic or pneumatic system is related primarily to compressibility. Figure 43.33 illustrates this point.

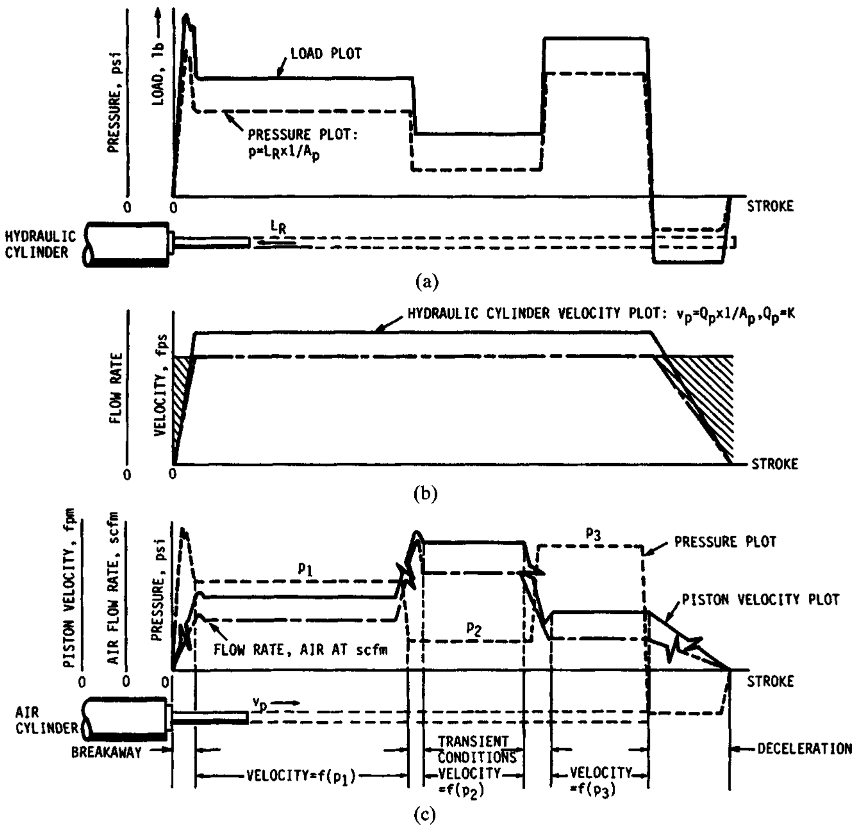


FIGURE 43.33 Performance comparison of hydraulic and pneumatic cylinders.

The load-cycle plot of Fig. 43.33*a* is identical for both the hydraulic and the pneumatic power system. The pressure plot of Fig. 43.33*a* is determined from the load plot. If the hydraulic cylinder is driven by a fixed-displacement pump, then the volumetric flow rate to the actuator is constant. This is represented by the horizontal flow curve in Fig. 43.33*b*. A constant input to the cylinder results in a constant output velocity (of the piston rod). Because of the relative incompressibility of the hydraulic fluid, fairly accurate velocity control is possible.

It is very difficult to determine what the air flow rate to the cylinder really is. For instance, with an initial load pressure of p_1, there is a corresponding initial flow rate of air. The density of the air is a function of the pressure p_1 and the temperature T_1. The source of the compressed air is the central compressor station; the air is delivered through a pressure regulator, which is a throttling device. Since the compressor is assumed to be capable of delivering an unlimited quantity of air to the cylinder, the factors which limit flow rate to the cylinder include load, resistance of the connecting pipes to flow, valve orifices, the regulator, etc. It is important to realize that *all* these factors contribute pressure drops of one sort or another, and that every change in pressure brings about a corresponding change in the volume of the gas.

Consider the change in load pressure from p_1 to p_2, as indicated in Fig. 43.33*c*. This pressure difference is caused by a change in the load reaction on the piston rod, not by a change in the pressure of the air in the cylinder. At the instant the load pressure drops to p_2, a force imbalance in the cylinder results. This occurs because there is air in the cylinder at pressure p_1. The gas does the only thing it can—it expands until it reaches a new equilibrium pressure p_2. This sudden expansion of the gas in the cylinder will be evident from the lunging forward, or jerking, of the piston rod.

The load pressure curve of Fig. 43.33*c* indicates that there will next be a pressure increase from p_2 to p_3. The reverse of the process just described occurs now. That is, the piston rod slows down, or even stops momentarily, to allow incoming air to recompress the gas already in the cylinder to the new equilibrium pressure p_3.

In a pneumatic system, these momentary changes in piston-rod velocity occur every time there is a change in load pressure. Thus we cannot speak in terms of a flow rate to the actuator in the same sense as we do when we are discussing a hydraulic system. Instead, we must deal with instantaneous piston velocities.

The point in this discussion is that it is difficult to obtain controlled output velocities with pneumatic systems under varying load conditions. It is also difficult to maintain accurate position with a pneumatic system because of the compressibility of the gas medium (see Fig. 43.34). As the load changes from F_1 to F_2, the gas in the cylinder is reduced in volume. This causes a change in the position of the piston rod.

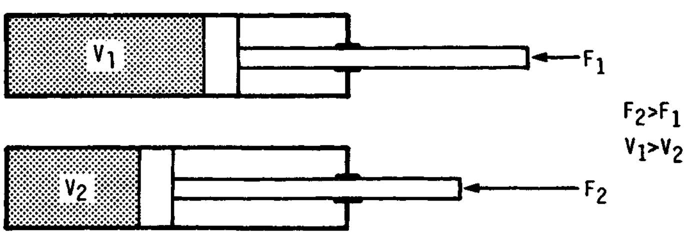


FIGURE 43.34 Explanation of why position control is difficult with pneumatic systems.

It is easy to deduce why hydraulic systems have taken the lead over pneumatic systems in those applications requiring accurate control of position or velocity. Up to the present, pneumatic systems have been used mainly for sequential types of circuits where the end conditions are those of prime importance, i.e., circuits in which the important thing is whether the rod is fully extended or fully retracted. Transfer, clamping, and press circuits are typical of such applications. That the relative importance of pneumatic systems is changing will become apparent in later discussions of logic-circuit design.

Where the economics of the situation dictate the use of a pneumatic power system, yet control requirements are greater than those attainable with a purely pneumatic system, an air-oil system might be used. In air-oil systems (Ref. [43.5], Part II, pp. 149–200), compressed air provides the source of potential energy, and hydraulic oil provides the incompressible-fluid characteristics necessary to achieve the desired degree of control. Figure 43.35 illustrates the use of a tandem air-oil cylinder for such a purpose.

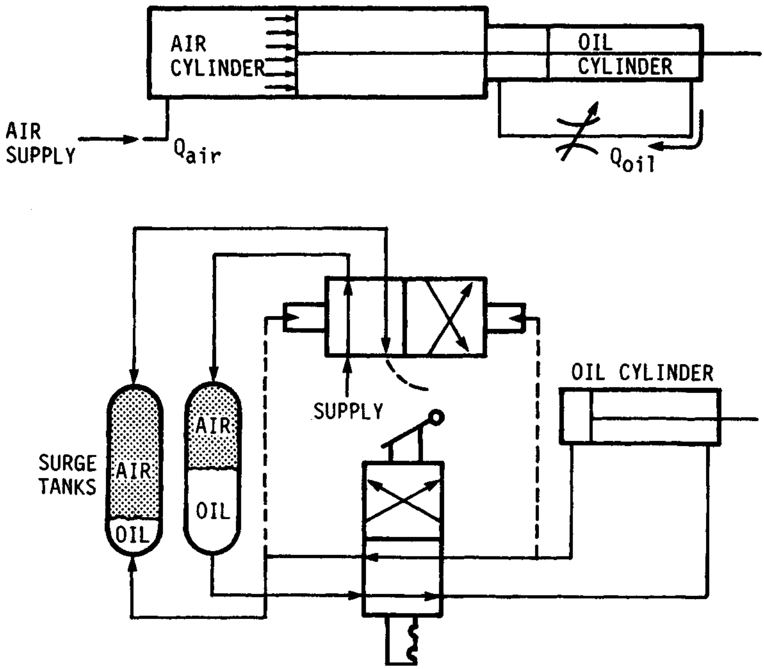


FIGURE 43.35 Example of an air-oil system.

43.12 EFFECT OF FLUID CHARACTERISTICS ON CONTROL-VALVE PERFORMANCE

The compressibility of the gaseous medium makes performance prediction for pneumatic valves more difficult than for their hydraulic counterparts. Pneumatic direction control valves are very similar in function to their hydraulic counterparts. Thus the functional designations—two-way, three-way, four-way, etc.—are applicable to both.

A major difference in the design and construction of pneumatic and hydraulic components reflects the vast difference between them in pressure level. Hydraulic valves, which must operate at pressures from 1000 up to 10 000 psi, are made from heavy castings or bar stock. On the other hand, pneumatic valves seldom encounter pressures over 150 psi and can be die-cast or otherwise fabricated from aluminum, brass, or even zinc alloys.

The types and functions of the valve operators are the same as those previously indicated for hydraulic valves. That is, pneumatic power valves can be operated by solenoids; by pilot controls; by hand, foot, cam, or palm button manual operators, etc. The major difference between hydraulic and pneumatic valve operators reflects the lower operating pressures of the latter. Since lower forces are encountered, the operators are usually smaller.

Pneumatic power valves have shorter response times than hydraulic valves. For example, a pneumatic valve of a given size will probably shift from 3 to 4 times as rapidly as its hydraulic counterpart. A solenoid-operated hydraulic valve of a given size might shift in 30 to 40 milliseconds (ms). A pneumatic valve of the same size might require 5 to 10 ms.

Another difference between pneumatic and hydraulic valves reflects the fact that one is designed to handle "incompressible" fluids (hydraulic), whereas the other handles "compressible" fluids (pneumatic).

Increasingly, valve manufacturers are turning to C_v factors to rate hydraulic and pneumatic valves. For hydraulic valves, this factor is given by

$$C_v = \frac{Q}{(\Delta p/S_G)^{1/2}} \tag{43.18}$$

which is a special form of the classical equation expressing the relation between orifice flow Q and pressure change Δp. This classical equation is usually written as

$$Q = \frac{C_d A_0}{(2g\ \Delta p/\gamma)^{1/2}} \tag{43.19}$$

Essentially C_v is a measure of the ability of the valve to conduct fluid and can be used to select or compare similar control valves. For pneumatic valves, the formula is

$$C_v = \frac{Q}{22.67\{S_G T/[K(p_1 - p_2)]\}^{1/2}} \tag{43.20}$$

where S_G = specific gravity (1 for air)
T = absolute temperature (460+°F)
p_1 = absolute inlet pressure (psig + 14.7)
p_2 = absolute outlet pressure (psig + 14.7)
K = constant K_1, K_2, or K_3
$K_1 = p_2$ when Δp is no more than $0.1p_1$
$K_2 = p_1$ when $\Delta p \geq 0.25p_1$
$K_3 = (Cp_1 + p_2)/2$ when $0.1p_1 < \Delta p < 0.25p_1$
Q = flow rate in standard cubic feet per minute

The flow coefficient C_v indicates the ability of a valve to conduct a compressible fluid (gas); the pressure p_2 must be greater than $0.53p_1$ to assure subsonic flow through the valve orifice. If Δp across the orifice is greater, flow becomes supersonic and the equation is invalid.

43.13 BASIC PNEUMATIC POWER CIRCUIT

The basic direction control circuit for a single-acting cylinder, shown in Fig. 43.36, illustrates some of the differences between such a circuit and its hydraulic counterpart. Note the absence of an input device, or pump. Most pneumatic circuits use main plant compressors as their source of energy. To provide the input to the circuit, one simply "hooks into" the air manifold at a convenient location. This does not, however, preclude the use of an individual air compressor located at the machine. Conversely, for most hydraulic systems, the use of individual power units does not preclude the use of a central system.

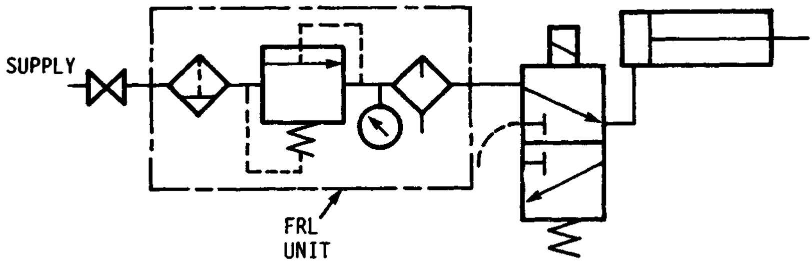


FIGURE 43.36 A pneumatic direction control circuit.

A peculiarity of pneumatic circuits is the use of a *filter-regulator-lubricator* (FRL) unit at the source (the point where the air manifold is tapped). This unit provides clean air at regulated pressure and adds enough lubricant to the air to minimize wear of component parts.

Because of the complexity of handling thermodynamic considerations associated with compressible-fluid flow, much of industrial pneumatics technology has been reduced to empiricism (see Ref. [43.6]).

In theory, all the circuits should work equally well. However, when objectives such as low operating cost, circuit simplicity, energy efficiency, and high productivity are added to the analysis, only one or two of the circuits will actually be feasible.

This winnowing out of impractical designs can be a long and tedious procedure with many pitfalls. A more efficient design procedure (adaptable to the computer) has been developed that focuses on the optimum circuit quickly. Based on experience with the computer technique, a number of guidelines have been developed that help shorten the time from first proposal to final design.

43.13.1 The Basics

The optimization procedure is based on the flow capability (or conductance) of pneumatic devices. Flow capability, the reciprocal of flow resistance, provides a convenient means of evaluating components to optimize operating pressures, valve and line sizes, compressor loads, etc.

Capability is expressed as C_v, a dimensionless number. The National Fluid Power Association (NFPA) and the International Standards Organization (ISO) have prescribed carefully controlled tests for determining C_v values for valves and other fixed-orifice devices.

In general, a C_v value cannot be assigned to a valve; it must be related to a port size—specifically, to the inner diameter (ID) of the smallest conductor or fitting entering the port. For instance, a direct solenoid-actuated valve has a C_v of 3.5 with a ½ NPT port and 3.2 with a ⅜ NPT port.

For conductors, C_v is related to internal diameter, length, and friction factor. Typically, a friction factor of 0.02 is used for smooth-wall conductors and of 0.03 for rough-wall conductors. Ideally, a Reynolds number should be computed to determine the friction factor, but because relatively small conductors are used with compressed air, 0.02 and 0.03 are reasonable values regardless of diameter and air velocity.

Fittings can be converted to equivalent conductor lengths, and the C_v can be calculated in the same manner as for conductors. Some manufacturers of quick connections and fittings publish C_v values for their products.

The C_v of a cylinder is related to the ID of its smallest fitting or conductor, multiplied by a constant. The constant depends on the flow of compressed air through the end caps. The smoother the air passage, the higher the C_v.

Once flow capability has been determined for each circuit component, individual capabilities can be combined to determine an overall capability for the entire circuit. Circuit capability C_{vs}, for components in series, is found from the equation

$$\frac{1}{C_{vs}^2} = \frac{1}{C_{v1}^2} + \frac{1}{C_{v2}^2} + \frac{1}{C_{v3}^2} + \cdots + \frac{1}{C_{vn}^2} \qquad (43.21)$$

For components in parallel, individual flow capabilities are simply added.

Maximum flow possible through a pneumatic circuit is limited by the size of the orifices in the system. The succession of valves, lines, and cylinders obstructs flow, with the restriction being inversely proportional to flow capability. Thus the total system capability is always less than individual component capabilities. With Eq. (43.21) and the instructions for parallel devices, it is not difficult to determine the total flow or system capability.

43.13.2 Analyzing a System

The procedure for determining and combining the flow capabilities of any fixed-orifice pneumatic device provides the means for predicting the response time of an actuator. Furthermore, air usage, productivity, valve size, or a combination of several objectives can be optimized easily by optimizing system C_{vs}.

In a typical analysis, several potential system designs are generated, depending on the design objective. With this information, compromises and tradeoffs can be made until the system is fine-tuned to meet specific needs.

The analysis assesses valve sizes, cylinder sizes, fittings, piping, and operating pressures to minimize energy usage. This analysis is based on several considerations that remain fixed, regardless of design objectives:

- Conductor lengths should be as short as possible.
- Conductor paths from valve to cylinder should be as straight as possible and have as few fittings as possible. A machine may look better cosmetically if the conductors follow a natural contour of the machine, but such piping wastes air.
- Cylinder bore sizes should be selected to handle the expected load plus a reasonable safety factor.
- Cylinder stroke length should be no more than required. A longer stroke than necessary wastes energy.
- Air valves can be oversized without wasting energy.
- Overpressurizing a circuit beyond a certain point does not increase cylinder speed but does waste air.
- If the application calls for two different loads or times for the extension and retraction portions of the cycle, two different pressures should be used.
- Increasing conductor diameter increases C_v but also increases the volume that must be filled and evacuated each cycle. Therefore, each application has an optimum conductor diameter.
- Taking all these points into consideration, required cylinder bore size for the given load conditions and pressures is calculated from the equation

$$D = 1.13\left(\frac{LS_f}{P_d}\right)^{1/2}$$

where L = cylinder load
S_f = factor of safety
P_d = design pressure

The highest load acting on the cylinder is used in this calculation, and the safety factor can range from 15 to 50 percent, depending on the service. Also, the design pressure used in this calculation is 80 percent of the minimum available pressure. The actual cylinder bore specified is the next larger standard bore above that calculated.

Once the required bore size is determined, cylinder performance graphs are used to specify target C_v's for both extension and retraction strokes to meet the load and time requirements.

The required system flow capability can be approximated from cylinder performance graphs such as those shown in Fig. 43.37. This is an important step in the analysis because it provides a check on subsequent steps.

To establish target C_v's for the extension and retraction strokes, plot the intersection of lines from the required stroke time scale and read the C_v values. If the flow capabilities calculated at the end of the analysis are lower than the target C_v's, the cylinder will not extend or retract in the required time, and a larger value of conductor must be selected. If the calculated capacities are higher than the target C_v's, the system will operate satisfactorily.

Determining Conductor Flow Capability. * The conductor size equal to the smallest port size of the selected valve is used to determine the conductor C_v from piping performance graphs, as in Figs. 43.38 and 43.39. The flow capability of pneumatic pipe, tube, and hose depends on the inside diameter and length. Capability can be determined from the two graphs shown in the figures by drawing a horizontal line from the length scale to the conductor size plot and then drawing a vertical line to the capability scale. These graphs can also be used for fittings by converting the fittings to an equivalent conductor length.

The system flow capability is calculated by combining all the component capabilities including the cylinder C_v, which is

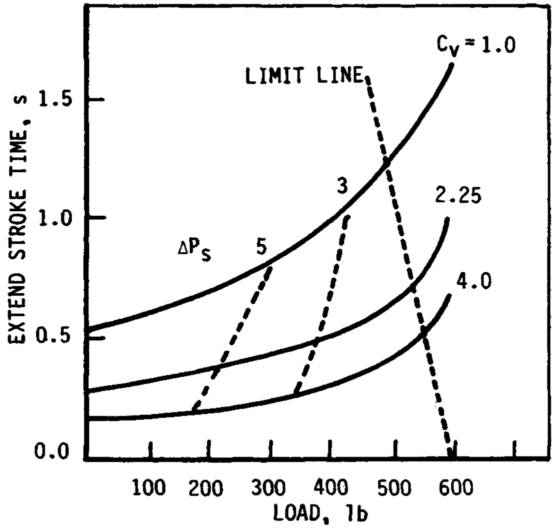
$$C_{vc} = 18d^2 \tag{43.23}$$

If the calculated system flow capability is larger than both target C_v's, the system is properly designed. However, if the calculated capability is lower than either target C_v, then a valve with a higher C_v must be selected and the calculations repeated.

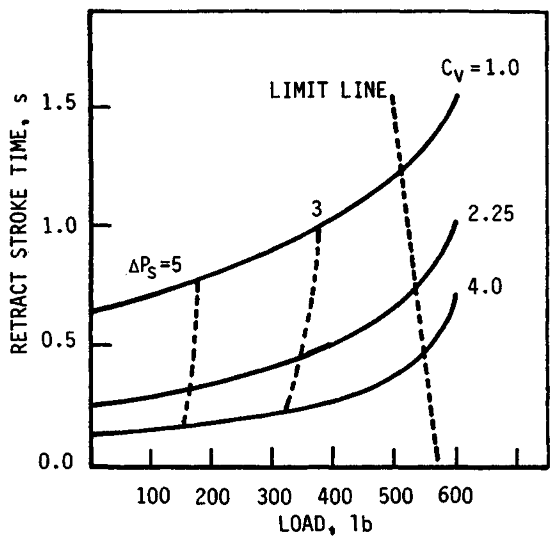
The final step in the procedure is to calculate possible extension and retraction times from

$$t_a = \frac{C_{vt}}{C_{vs}t_r} \tag{43.24}$$

* This discussion of pneumatic power system design is by the courtesy of Numatics Inc., Highland, Michigan, and is from the publication "Practical Air Valve Sizing" by Henry Fleischer, Director of Engineering.



(a)



(b)

FIGURE 43.37 Specifying target C_v's using typical cylinder performance charts. Plots shown are for a 4-in-bore, 6-in-stroke, double-acting air cylinder. Supply pressure is 60 psig, initial pressure is 60 psig, and ΔP_s is the mean system pressure drop. (*a*) Extension stroke; (*b*) retraction stroke.

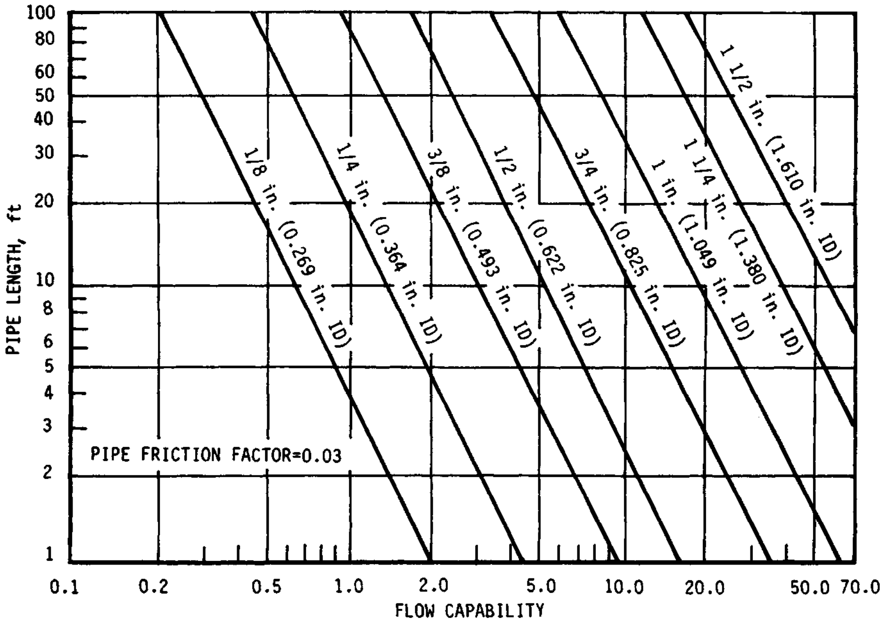


FIGURE 43.38 Determination of conductor flow capability for steel pipe; friction factor = 0.03.

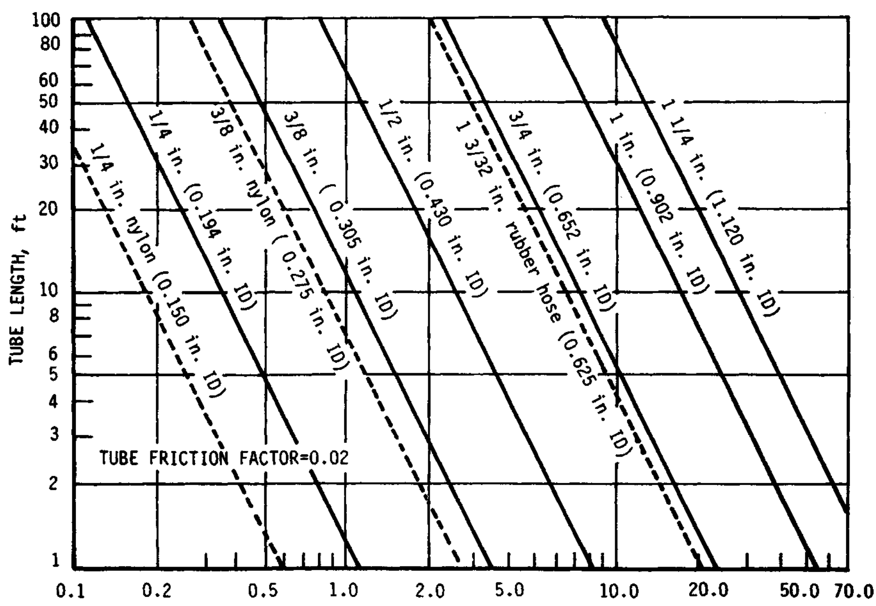


FIGURE 43.39 Determination of conductor flow capability for smooth-wall tubing; tube friction factor = 0.02.

If these times are slower than required, the conductor size must be increased and the calculations repeated. These calculation steps can be programmed easily, and a computer program is available* to make the necessary calculations and evaluate design tradeoffs.

Each application has only one optimum conductor size, because even though flow capability increases with diameter, so does volume. Thus, beyond a certain point, increased volume negates the C_v advantage of larger-diameter conductors.

The notation used in the discussion above is as follows:

P_d = design pressure, psig	C_v = flow capability
S_f = safety factor	D = cylinder bore diameter, in
t_a = attainable time, s	d = conductor ID, in
t_r = required time, s	L = cylinder load, lb

The subscripts are c for cylinder, s for system, t for target, and $1, \ldots, n$ for the system components.

43.14 FLUID LOGIC SYSTEMS†

Fluid logic systems had their genesis in the non-moving-part *fluidics technology* announced in the late 1950s and were all but eclipsed during the 1970s by microelectronics. Today, *fluidics* survives primarily in specialized applications requiring its special characteristics. Typical are medical, food processing, high-temperature, high-radiation, and other environments where all-fluid systems are advantageous.

The residual of two decades of evolution is *moving-part logic* (MPL). MPL utilizes miniature to small-size pneumatic devices which combine logic with power-handling functions. MPL devices include spool, poppet, diaphragm, floating diaphragm, and various proprietary designs of valves. Fluid sensors are used to input system-variable status to the MPL system. Fluid indicators and readouts are also available to provide visual monitoring of system status.

Transistor and microelectronic control technology is having an impact on MPL, as it did on fluidics. Figure 43.40 illustrates the situation wherein MPL and electronic control can be interposed between the *load-sensing* stages and the *fluid power* stage in a typical application.

Pneumatic control circuits are used to provide a prescribed sequence of events. Usually, arriving at that sequence is not a particularly difficult design task. However, problems frequently arise when the analysis stops after this step and "secondary" objectives that help ensure trouble-free operation are not considered.

Typical secondary goals include the following:

- Ensuring that the circuit does not respond to false input signals. Overlooking this detail can cause equipment damage or operator injury. These actions are prevented by including interlocks in the system.
- Providing a circuit that operates over a wide range of conditions. Potential problems arising from pressure variations, contamination, actuating speed, and sensitive adjustments must be recognized and eliminated.

* From Numatics Inc., Highland, Michigan.

† This discussion of fluid logic systems is by the courtesy of ARO Corporation, Bryan, Ohio; Bruce F. McCord, Manager, Control Systems.

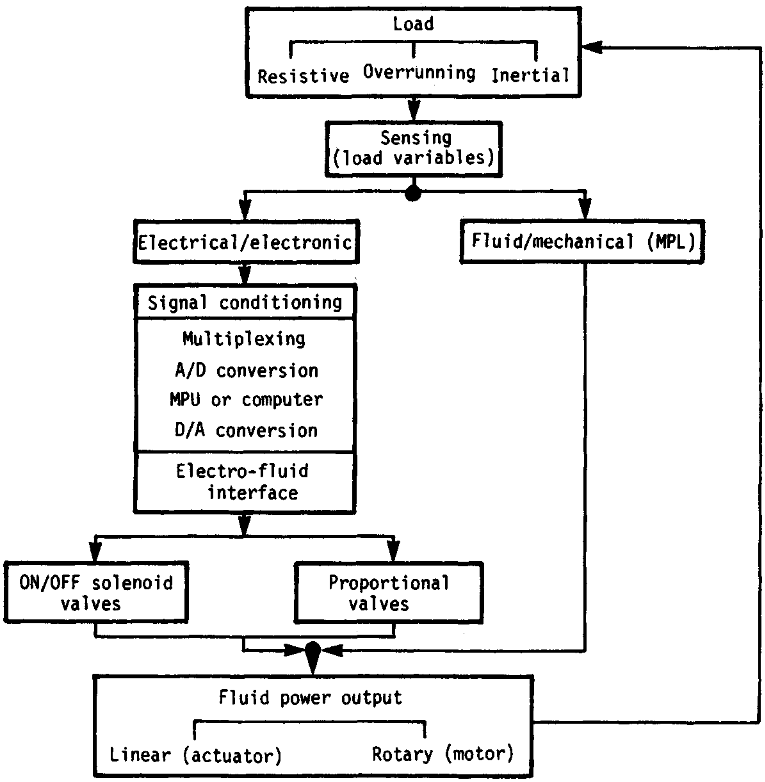


FIGURE 43.40 Comparison between electrical/electronic and MPL control options for a typical industrial application.

- Designing a circuit that assists in troubleshooting and diagnosis of problems. This objective is met in two phases. First, components that can mask problems, such as pulse or single-shot valves, are avoided. Second, display devices, such as pneumatic indicators, are included in the circuit, permitting the cause of failure to be traced quickly.

Meeting all these objectives can be a tall order, but a design technique has been developed that automatically avoids the common pitfalls and meets the design objectives easily. Based on the theory developed for electronic controllers, the method inherently incorporates the interlocks necessary to prevent improper operation. It also produces a circuit that is easy to troubleshoot. Finally, it includes a review procedure that ensures that the system is the least expensive possible.

43.14.1 Circuit Design

Pneumatic controls are used in two types of circuits: combinational and sequential. Combinational circuits monitor an operation and react to certain combinations of inputs regardless of the order of occurrence. Sequential circuits, however, accept inputs only in a prescribed order. Sequential circuits make up about 90 percent of the pneumatic control circuits used, and the method presented here deals only with these circuits.

The first step in the design of a pneumatic circuit is to build a stage register. The purpose of the stage register is to collect signals from input devices (pushbuttons and limit valves) and create a series of maintained output signals, one for each step (or stage) in the sequence. These signals are further manipulated to create the output signals necessary to shift the power valves that actually do the work on a machine. These are the basic rules used in building the stage register:

1. Any input signal that starts an action (or stage) and does not remain on until the end of the cycle is connected to the set port of a flip-flop element. Thus, if an input is not maintained throughout the rest of the cycle (once it is used to start a stage), the flip-flop output can be used as the maintained stage signal.

2. Any input signal that is maintained from the time it starts a stage until the end of the cycle is connected to an AND element input. The other input to the AND element comes from the previous-stage signal.

A maintained signal connected to an AND element acts as an interlock to the previous stage. This prevents the signal from affecting the circuit until that circuit is ready to receive the signal, as indicated by the presence of an output from the previous step.

3. A stage signal is developed for each step in the sequence plus one additional signal, called a *reset,* which returns all circuit elements to their original positions when the cycle is completed. The reset signal is connected to the reset port of the first flip-flop in the circuit. Then the reset output of this first flip-flop is connected to the reset input of the second flip-flop, the second to the third, and so on down the line. This connection interlocks the flip-flops so that they cannot be shifted out of sequence.

The second step in designing a circuit is to convert the maintained stage signals to the on-off signals required to shift the power valves. NOT elements are used to create the power valve signals. A NOT element is similar to a normally passing valve in that a signal applied to the *b* port is transmitted to the *c* port until a second signal is applied to the *a* port. See Fig. 43.41. Thus, to create the signal required by a power valve, the signal from a maintained stage must be applied to the *b* port of a NOT element. Because this signal is transmitted from the *b* to the *c* port, the *c* port signal becomes the power valve pilot signal.

To remove this signal at the beginning of a later stage, the signal from that stage is applied to the *a* port of the NOT element. When the *a* port is pressurized, the output signal from the NOT goes off. This sequence is followed for all power valves in the circuit.

43.14.2 Logic Elements

Pneumatic controls are used in combinational and sequential circuits, as we have learned. Normally, combinational circuits use AND, OR, and NOT elements with an occasional delay or memory (flip-flop) to delay or hold the signals from the other elements (Fig. 43.41). The OR element is sometimes used to add additional functions, such as emergency stops and manual overrides, to a sequential circuit. Also, delay elements are occasionally used to replace limit valves, while amplifiers increase low-pressure signals to the system operating pressure.

In the AND element, both input *A* and input *B* must be on for the output to be on. Conversely, if either input is off, the output is off.

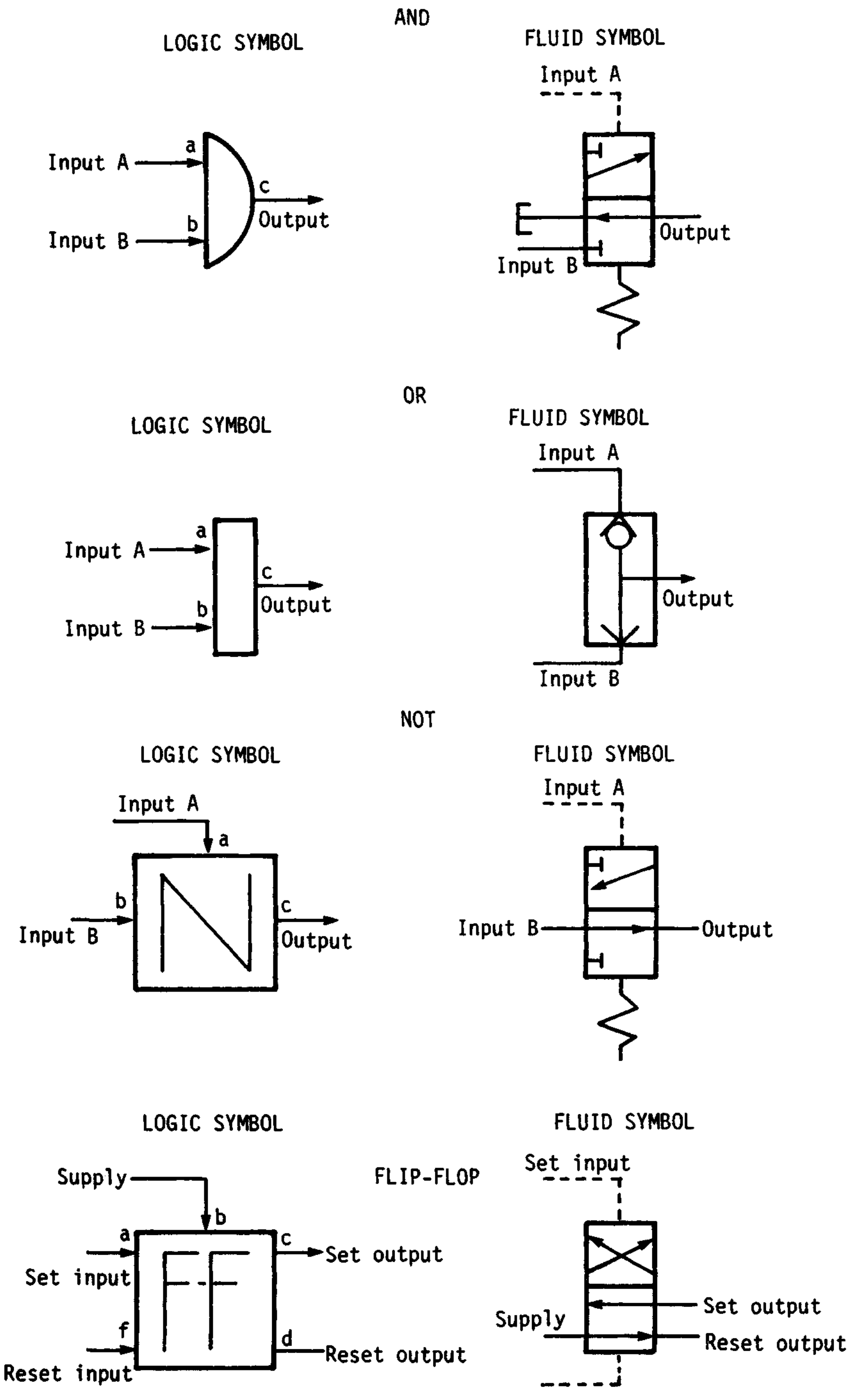


FIGURE 43.41 Logic elements.

In the OR element, the output is on if either input *A* or input *B* is on.

The output of the NOT element is on if input *A* is off and input *B* is on. Conversely, the output is off if

- Input *B* is off.
- Input *A* is on.
- Inputs *A* and *B* are both on.

With the supply to the flip-flop element on, only one of the two outputs (set or reset) will be on and the other will be off. A signal to the set port of the element turns the set output on and the reset output off. The flip-flop remains in this state even when the signal to the set input is removed. To reverse the output condition, a signal must be applied to the reset input port. The outputs are then reset on and set off. Again the element remains in this state until a new signal is applied to the set input port.

Note that to reverse the state of a flip-flop, first the opposite input must be removed. For example, if the set input is maintained on, the output state cannot be reversed by applying a reset signal. The reverse is also true.

43.14.3 Design Example

To illustrate the use of the technique, consider a simple clamp and punch machine. The power devices and input devices for the machine are shown in Fig. 43.42, and the automatic sequence of operation is as follows:

1. Operator presses start pushbutton—clamp cylinder extends.
2. Clamp cylinder actuates a limit valve—punch cylinder extends.
3. Punch cylinder actuates second limit valve—punch cylinder retracts.
4. Punch cylinder actuates third limit valve—clamp cylinder retracts.
5. Clamp cylinder actuates fourth limit valve—cycle is complete.

This is a single-cycle machine, and the operator need only operate the start pushbutton momentarily.

Figure 43.42 also shows the signals to the circuit and how they come on and go off through a normal cycle. Signals are not normally graphed in this manner, but they were in this case to better illustrate the difference between maintained and nonmaintained signals.

Stage 1 is initiated by start pushbutton PB-1. Since the operator need not hold the button down, this signal is not maintained until the end of the cycle. Therefore, a flip-flop must be used to maintain this output signal.

Stage 2 is initiated by the signal from limit valve LV-1. This signal is lost shortly after the beginning of stage 4. Thus another flip-flop is used to maintain the stage 2 signal.

Stage 3 is initiated by LV-2. Again, the signal is lost before the end of the cycle, and a flip-flop is used to maintain the stage 3 output.

Stage 4 and the reset stage are initiated by LV-3 and LV-4. Both signals remain on from the point where they initiate their stages to the end of the cycle. Therefore, an AND element is used to interlock these limit valves into the sequence and provide the stage 4 outputs and reset.

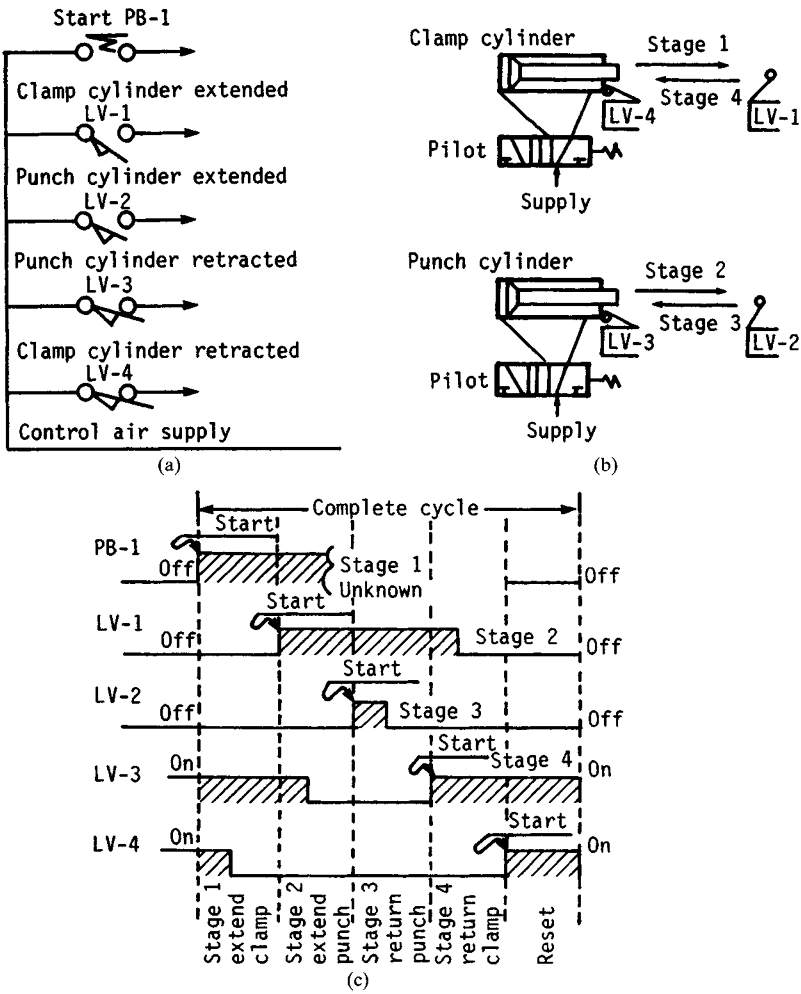


FIGURE 43.42 Basic elements of example circuit. Sketches show the starting positions of (*a*) input and (*b*) power devices for clamp and punch machine. (*c*) The chart shows the sequence of signals through one machine cycle. Such diagrams indicate the type of logic elements needed in the control circuit.

Figure 43.43, the resulting stage register circuit, shows that the stage indicators must come on in numerical order. For example, in the rest condition (all indicators off), the only input this circuit accepts is the input from PB-1. If PB-1 has not been actuated (stage 1 output off) and the clamp extended limit valve, LV-1, is accidentally actuated, then stage 2 output does not go on. The reason is that the reset output coming from the *d* port of flip-flop 1 holds flip-flop 2 in the reset position (pressure at *f* port). Therefore, to initiate stage 2, stage 1 must be actuated first, turning on the stage 1 indicator and releasing the pressure at the reset port of flip-flop 2.

Also, although the punch-retracted limit valve LV-3 is actuated and the signal for the circuit is *on,* the stage 4 indicator light is off because AND element 4 must have two inputs to create an output. Since stage 3 output is off, the *b* port of AND 4 is off and the stage 4 output cannot be on. This type of signal (on at the beginning of the cycle) often seems to indicate that a pulse, or "single-shot," element should be used in the circuit because this signal must be removed before the sequence can start. However, single-shot, or pulse, elements are not required because the effect of the signals is locked out once the system is reset.

The top section of Fig. 43.44 shows the stage signals already developed, and the lower section shows the signals required for the power valves. NOT elements are used to create the power valve signal patterns. To create the proper signal for the clamp-cylinder power valve, the stage 1 signal must be applied to the *b* port of a NOT element. Since this signal is transmitted to the *c* port, the *c* port becomes the clamp-

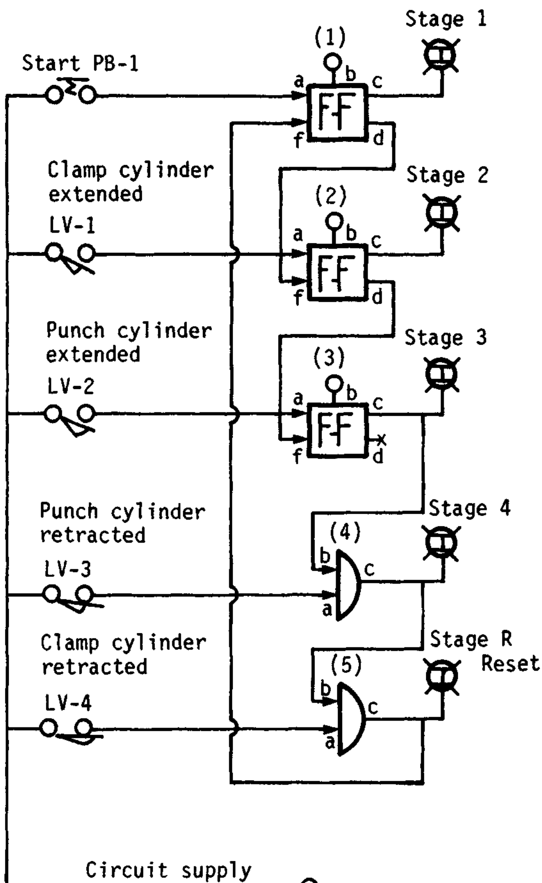


FIGURE 43.43 Stage register for example circuit. Basic control circuit consists of three flip-flops and two AND elements. Indicator lights show when each stage has been completed.

extend pilot signal. To remove this signal at the beginning of stage 4, the stage 4 signal is connected to the *a* port of this NOT element. When the *a* port is pressurized, the output signal from the NOT element goes off. This duplicates the signal pattern shown on the lower portion of Fig. 43.44. The signal for the punch valve is developed by using the same method, applying the stage 2 signal to the *b* port and the stage 3 signal to the *a* port of another NOT element. Figure 43.45 shows the completed circuit.

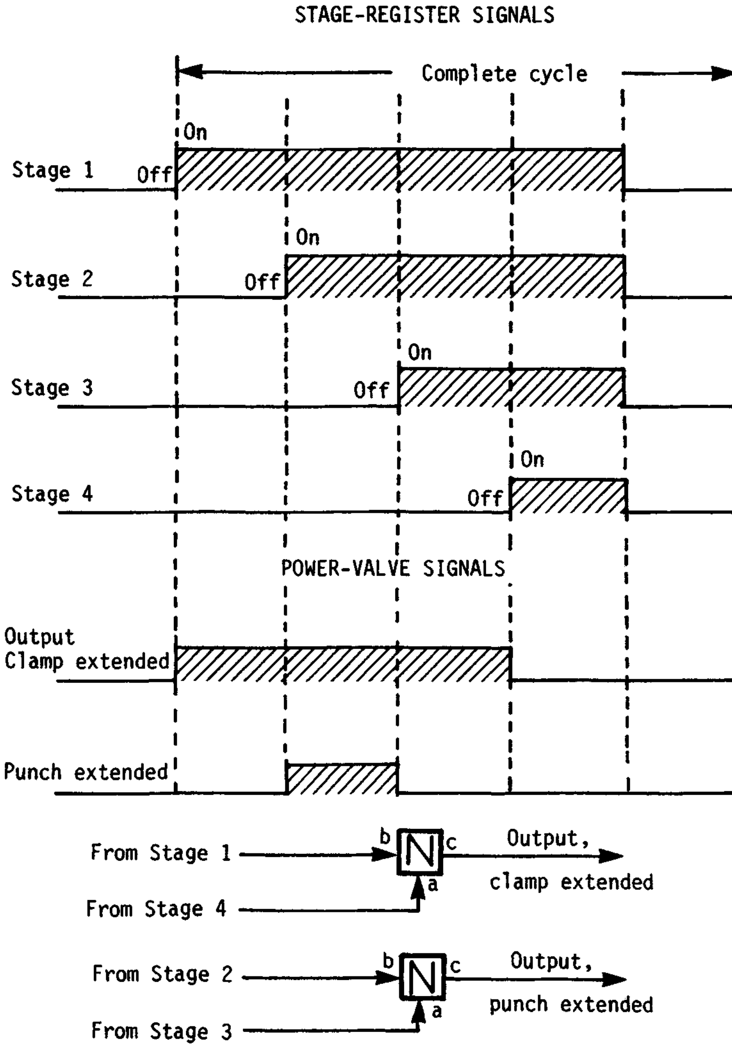


FIGURE 43.44 Circuit signals for example machine. Top portion of graph shows the signals produced by the stage register elements, while bottom portion shows the signals required to actuate the power valves. NOT elements transform the stage register signals to power valve pilot signals.

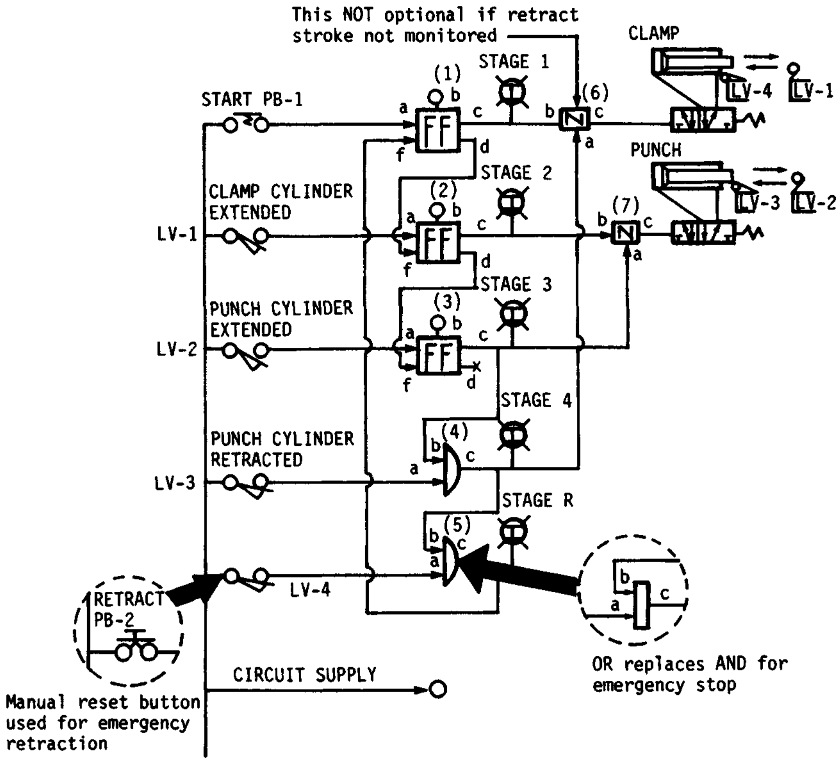


FIGURE 43.45 Completed example circuit prior to final analysis. The control circuit consists of seven logic elements. The circuit can now be evaluated to determine the effect of removing or adding elements. Here, the retraction stroke need not be monitored; therefore AND 5, LV-4, and NOT 6 can be removed. An emergency stop function is provided by adding PB-2 and OR 5.

43.14.4 Circuit Evaluation

The stage register provides a means of reviewing the design objectives.

- By comparing the circuit to the original sequence of operation, it can be determined that the circuit functions as required.
- The circuit also prevents false input signals from causing machine action. For instance, in the normal sequence of operation, the punch cylinder extends when the clamp-cylinder-extended limit valve LV-1 is inadvertently actuated by the operator while loading or unloading a part in the machine; the stage register interlock prevents the punch cylinder from extending. Such interlocks are included automatically.
- Pulse devices and momentary signals that could be affected by the speed at which the limit valves are actuated, length of tubing, pressure changes, leaks, etc., are excluded from the circuit. Therefore, this circuit should operate over as wide a range of conditions as the components themselves can tolerate. Also, no adjustments are required that could affect the operation of this system.

- By including the indicators as shown in Fig. 43.45, the cause of most problems can be traced. Figure 43.46 shows a simple troubleshooting chart that could accompany the machine for use by a repair technician. The bottom three lines of this chart illustrate that the circuit also detects a malfunction in the first three input signals if they happen to remain on. For instance, if a cycle is completed and limit valve 1 is not released, a new cycle cannot be initiated.

This is an important bonus interlock because, on the next cycle, LV-1 would indicate that the clamp was actuated prematurely. However, since the LV-1 signal is applied to the set port of flip-flop 2, it cannot reset and the circuit will not start a new cycle. Instead, the indicators will remain in the position shown in the next-to-last line of Fig. 43.46 and indicate that, in all likelihood, limit valve 1 has failed; if not, a failure in flip-flop 2 is the only other possibility.

By carefully analyzing and removing elements that are not necessary, circuit cost can be reduced considerably. However, this evaluation cannot be made until the circuit has been developed.

If the machine fails to operate and the indicators are in this condition					Follow these steps to locate problem and check in the order listed
1	2	3	4	R	
○	○	○	○	○	Air supply, PB-1, flip-flop-1
●	○	○	○	○	Clamp cylinder (should be extended) LV-1, flip-flop-2
●	●	○	○	○	Punch cylinder (should be extended). LV-2, flip-flop-2
●	●	●	○	○	Punch cylinder (should be retracted) LV-3,AND-4, NOT-7
●	●	●	●	○	Clamp cylinder (should be retracted) LV-4, AND-5, NOT-6
●	●	●	●	●	PB-1, flip-flop-1
○	●	●	●	●	LV-1, flip-flop-2
○	○	●	●	●	LV-2, flip-flop-3

FIGURE 43.46 Troubleshooting chart. The chart matches the condition of the indicator lamps to the probable cause of failure. Such a chart should accompany each machine to allow malfunctions to be traced quickly.

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