1.6 Some Applications of Proportional Control Valves

There are many applications in which the precision of servo valves (discussed in Chapter 18) is not needed, but more accuracy is needed than can be obtained with conventional valves. Proportional valves were developed to fill this gap. Their use has increased significantly since the mid-1970s. The following sub-sections present few applications of proportional control valves.

1.6.1   Control of Actuators

Proportional control valves are commonly used in speed control of actuators such as counterbalancing circuits in broaching machine, three-axis CNC(computer numerically controlled) machines, etc.

1.6.1.1 Speed Control of Cylinders

The conventional speed control of a cylinder is by meter-in, meter-out or spill-off flow control valves. This sets a cylinder speed that can be manually varied. Alternatively, a cam drive can progressively close or open an adjustable orifice in accordance with a preset speed profile that is altered by changing the cam profile. Examples of these types of circuits are shown in Fig. 1.21. The acceleration and retardation of a cylinder can be controlled by the following:

- Relief valves limiting the maximum pressure available to accelerate the load.
- Using a two-stage directional control valve with a choke pack to control the speed of movement of the main spool.
- Using a variable displacement pump.
- Using internal cylinder cushions or external shock absorbers to decelerate the cylinder.
- Building brake, deceleration and counterbalance valves into the circuit to control the deceleration and sometimes the acceleration of the actuator.

All these manual methods are incapable of continuous variations whilst the system operates. A proportional control valve in the cylinder circuit enables continuous regulation of speed, acceleration and retardation. If a proportional control card is used to drive the valve, any adjustments to the maximum current ramp-up and ramp-down have to be carried out by adjusting potentiometers on the card. However, a microprocessor or minicomputer may be employed to control the proportional valve by varying the solenoid current over different parts of the cycle.
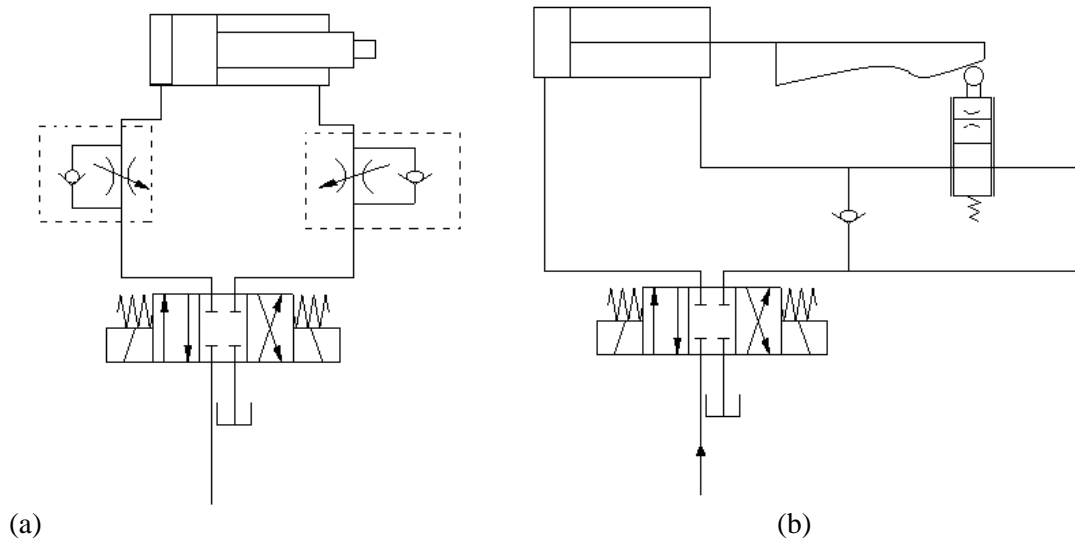
(a)   (b)

Figure 1.21 (a) Meter-out speed control. (b) Cam-operated speed control.

1.6.2 Speed Control of Hydraulic Motors

This is similar to the speed control of cylinders but it is relatively simple to monitor the motor speed and use a feedback system to control the proportional solenoid as seen in Fig. 1.13. The speed of response to changes in load or command limits the applications. For high response and accurate speeds, servo valves must be used.

1.6.2.1 Position Control of Hydraulic Cylinders

In order to control the position of the piston rod, a transducer has to be used to monitor the actual position. The output of the transducer is compared with the desired piston rod position and the difference is fed to the current amplifier and then to the proportional solenoid. The output of the current amplifier has to be biased so that any error signal that it receives provides a sufficiently large output to drive the proportional valve out of the dead zone. Otherwise the system will become unstable.

1.6.2.2 Pump Control Systems

In an ideal hydraulic system, the output of the power pack in terms of quantity flowing and maximum pressure is matched exactly to the system demand but this situation is rarely ever achieved. A proportional relief valve can be used as the main relief valve to set the maximum pressure and the setting remotely varies with the system sequence. A secondary relief valve should be fitted into the system as a safety feature in case of failure of the proportional relief valve. Supply pressure is matched to the system demand by controlling the proportional relief valve. It is only useful if the flow requirement of the system is constant.

In order to match the pump delivery to a system demand, a variable displacement pump has to be used. One possibility is to use a pressure-compensated pump with a proportional relief valve

acting as a compensator (Fig. 1.22). The circuit has to include flow control valves. This method has limited versatility and can be adopted when there is a series of actuators demanding different pressures and the flows are operated sequentially.
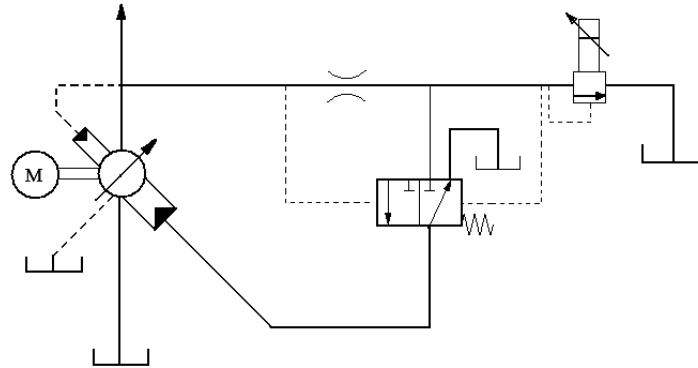

Figure 1.22 Pressure-compensated pump with proportional pressure control.

In any system using flow control valves, there are associated pressure drops that can be eliminated by removing the valves. In this case, the full pump output has to be utilized at all times. Wherever the flow demand varies within different parts of the sequence, this can be achieved by varying the pump displacement by electric motor drives and cam drives. But both have limitations – the motor drive in response speed and cam in versatility. An alternative is a proportional flow control valve and a proportional pressure control (Fig. 1.23). The pump output can be exactly matched to the system demand. This gives a power efficient system with little heat generation. This is shown in Fig. 1.23.

With valve A in the closed position, the directional valve C is piloted to the right which opens the large pump control piston to the tank. The small piston causes the pump to move to zero displacement. When a current is applied to the proportional solenoid on valve A, the orifice partially opens and the fluid flows and provides the system pressure that is below the set pressure of valve B. The pressures occurring across the orifice formed are applied to the pilots of valve C. Valve C spool centralizes locking the displacement of the pump. The system pressure is set by the proportional relief valve B. If the system pressure is greater than the setting pressure of B, the valve opens causing a reduced pressure on the right-hand pilot of valve C that moves to the right, thereby opening the large pump control piston to the tank. Pump displacement reduces until the system pressure matches that set by valve B that balances valve C locking the displacement. Thus, in this system the pump delivery and pressure can be matched to the system by remotely operating the proportional flow and proportional pressure control valves A and B.

The response time is of the order of 50–100 ms. If a pressure surge occurs in the system, the pump may not respond quickly enough to reduce the surge; therefore, the conventional pressure relief valve D is fitted to cater for pressure surges. This valve should be set 20% above the maximum operating pressure of the system.

The precise continuous regulation of flow pressure and displacement with consequential control of speed, thrust, position, etc., achievable from modern servo and proportional systems has made hydraulics indispensable in the field of modern drive and control techniques. The special characteristics of this sophisticated equipment involve electronic circuitry matched to the

individual components. Therefore, system design and construction are naturally much more complex than electrohydraulic digital control. Here commissioning a prototype may be time consuming than in the case with conventional hydraulic equipment. Nevertheless, these developments have presented the hydraulic design engineer with many exciting opportunities and extended the application of the subject to new areas.
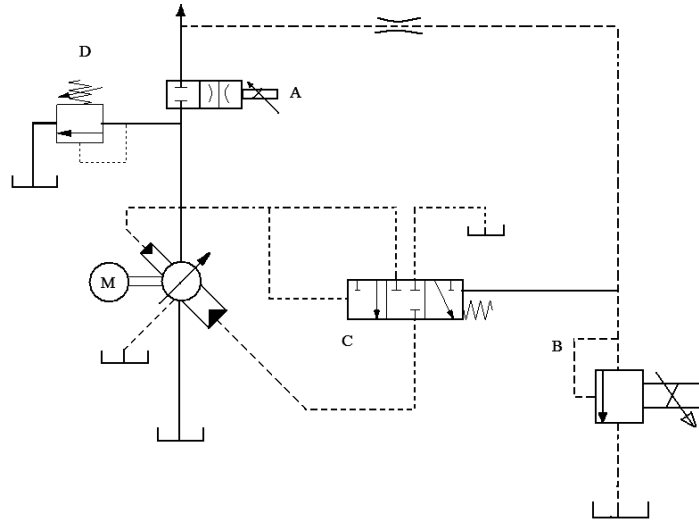

Figure 1.23 Pump with proportional pressure and flow control.

1.7 Analysis of Proportional Valves
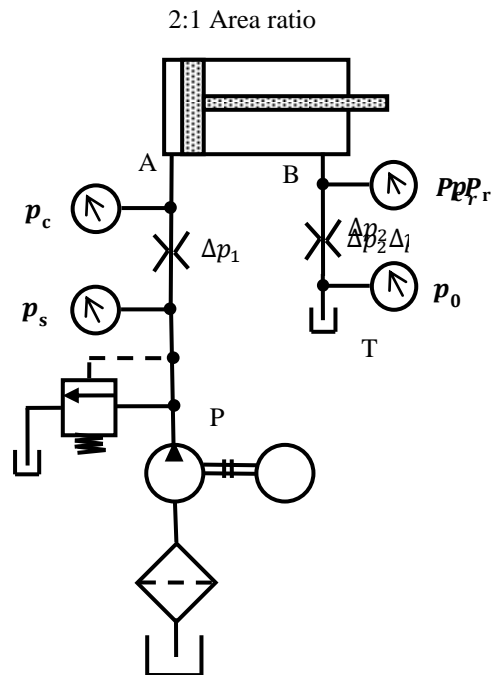
2:1 Area ratio

Figure 1.24 Cylinder circuit with a proportional valve.

The diagram of a proportional valve in a simple cylinder circuit is shown in Fig. 1.24. Let Δp_1 be the pressure drop between ports A and P, Δp_2 the pressure drop between ports B and T, as shown in Fig. 1.24. Q_1 the flow corresponding to the pressure drop Δp_1 and Q_2 the flow corresponding to the pressure drop Δp_2. The cylinder has an area ratio of 2:1 and proportional valve with 2:1 area ratio is used.

Using the orifice equations, we can write the expression for flow as

$$Q_1 = C_d A_1 \sqrt{\Delta p_1} \qquad\qquad\qquad (1.1)$$
$$Q_2 = C_d A_2 \sqrt{\Delta p_2} \qquad\qquad\qquad (1.2)$$

where Q_1 is the flow into the cap end of the cylinder, Q_2 is the flow out of the rod end of the cylinder, A_1 is the area of the orifice between ports P and A and A_2 is the area of the orifice between ports B and T.

The orifices have the same shape on both sides of the spool land. The number of grooves on one side is twice the other side for a 2:1 area ratio proportional valve. The same orifice coefficient for both sides shall be used.

The area ratio of the cylinder is 2:1; therefore, during the extension,

$$Q_2 = \frac{Q_1}{2}$$

Also

$$A_2 = \frac{A_1}{2}$$

Using the flow equations, we get

$$Q_1 = C_d A_1 \sqrt{\Delta p_1} \qquad\qquad\qquad (1.3)$$
$$Q_2 = C_d A_2 \sqrt{\Delta p_2} = \frac{Q_1}{2} = C_d \left\{ \frac{A_1}{2} \right\} \sqrt{\Delta p_2} \qquad\qquad (1.4)$$

or we can write Q_1 in terms of Δp_2 as

$$Q_1 = C_d A_1 \sqrt{\Delta p_2}$$

If $A_1 = A_2$, then we can show that

$$\Delta p_2 = \frac{\Delta p_1}{4}$$

(It is left as an exercise to the students.)

1.7.1 Overrunning Load

Suppose that the circuit shown in Fig. 1.24 has overrunning load during forward stroke. We can write the force balance on the cylinder as

$$p_c A_c = F_f + F_L + p_r A_r \qquad (1.5)$$

where $F_L = W =$ load on the cylinder (N), F_f is the frictional force (N), p_r is pressure at port B(Fig. 1.24). In this case, F_L is negative, since the load is overrunning, that is, it is acting in the direction of the movement of the cylinder. Solving for p_r we get

$$p_r = \frac{p_c A_c - F_f + F_L}{A_r} \qquad (1.6)$$

The pressure drop across port P to a orifice in the proportional valve is

$$\Delta p_1 = p_s - p_c \qquad (1.7)$$

Neglecting any drop between the proportional valve outlet and the tank, we can write $p_o = 0$. Then pressure drop from ports A to T is

$$\Delta p_2 = p_r - p_o = p_r \qquad (1.8)$$

If the area ratio is unity (i.e., $A_1 = A_2 = A$), the orifice equation becomes

$$Q_1 = C_d A \sqrt{\Delta p_1} \qquad (1.9)$$
$$Q_2 = C_d A \sqrt{\Delta p_2} \qquad (1.10)$$

Squaring both sides and eliminating $C_d A$ we get

$$\frac{Q_1^2}{Q_2^2} = \frac{\Delta p_1}{\Delta p_2} \qquad (1.11)$$

$$\Rightarrow \Delta p_2 = \Delta p_1 \times \frac{Q_2^2}{Q_1^2} \qquad (1.12)$$

Using Eqs. (1.7), (1.8) and (1.12) we can write

$$p_r = (p_s - p_c) \times \frac{Q_2^2}{Q_1^2} \qquad (1.13)$$

Equating Eqs.(1.6) and (1.13), we can now solve for the pressure at the cap end of the cylinder as

$$p_r = \frac{p_c A_c - F_f + F_L}{A_r} = (p_s - p_c) \times \frac{Q_2^2}{Q_1^2}$$

Rearranging we get relationship between p_c and other system parameters as

$$p_c = \cfrac{p_s\left[\dfrac{Q_2^2}{Q_1^2}\right] - \dfrac{[F_f - F_L]}{A_r}}{\left[\dfrac{A_c}{A_r}\right] + \left[\dfrac{Q_2^2}{Q_1^2}\right]}$$

Under certain conditions, p_c can be negative, which means that a vacuum exists in the cap end of the cylinder and the cylinder is not completely filled with oil. When this condition develops, the positive control of the load is lost during extension.

Example 1.1

Consider the hydraulic system shown in Fig. 1.24 The cylinder ratio is 2:1, pressure $p_s = 100\,\text{bar}, A_c = 0.002032\,\text{m}^2$, $A_r = 0.001070\,\text{m}^2$, $F_f = 290\,\text{N}$.

(a) Find the load that will cause negative pressure at the cap end of the cylinder and corresponding pressure at the rod end and pressure drop across port P to A and port B to T.

(b) If the load is 4450 N, find the pressure drop across port P to port A and corresponding pressure at the rod end. Is it possible to obtain this pressure drop valve area ratio 1:1?

(c) If the valve area ratio is 2:1, what is the overrunning load? Comment on the result.

Solution

(a) Because the cylinder area is 2:1, we have

$$Q_2 = \frac{Q_1}{2} \;\; \text{or} \;\; \left[\frac{Q_1^2}{Q_2^2}\right] = 0.25$$

To find the load that causes the negative pressure on the cap end, set $p_c = 0$. We know that

$$p_c = \cfrac{p_s\left[\dfrac{Q_2^2}{Q_1^2}\right] - \dfrac{[F_f - F_L]}{A_r}}{\left[\dfrac{A_c}{A_r}\right] + \left[\dfrac{Q_2^2}{Q_1^2}\right]}$$

$$\Rightarrow 0 = p_s\left[\frac{Q_1^2}{Q_2^2}\right] - \frac{F_f - F_L}{A_r}$$

$$\Rightarrow F_L = p_s\left[\frac{Q_1^2}{Q_2^2}\right]A_r + [F_f]$$

Substituting the values we get $F_L = 100 \times 10^5[0.25] \times 0.001070 + [290] = 2965\,\text{N}$

Any load greater than 2965 N will cause a negative pressure at the cap end of the cylinder. If the overrunning load is 2965 N, the pressure at the rod end is given by

$$p_r = \frac{p_c A_c - F_f + F_L}{A_r}$$
$$= \frac{0 - 290 + 2965}{0.001070} = 25\,\text{bar}$$

The pressure drop across the port P to port A orifice is

$$\Delta p_1 = p_s - p_c = 100 - 0 = 100\,\text{bar}$$
$$\Delta p_2 = p_r - p_o = p_r = 25\,\text{bar}$$

For any overrunning load greater than 2965 N, the valve will not create enough pressure drop across port B to port A orifice to maintain the control of load.

(b) When the load $F_L = 4450\,\text{N}$, we have

$$p_c = \frac{p_s\left[\dfrac{Q_2^2}{Q_1^2}\right] - \dfrac{[F_f - F_L]}{A_r}}{\left[\dfrac{A_c}{A_r}\right] + \left[\dfrac{Q_2^2}{Q_1^2}\right]}$$

$$= \frac{100 \times 10^5[0.25] - \dfrac{[290 - 4450]}{0.001070}}{\left[\dfrac{0.002032}{0.001070}\right] + [0.25]}$$

$$= -\frac{13.8785 \times 10^5}{2.14907} = -6.4579 \times 10^5 = -6.4579\,\text{bar}$$

To create $p_c = -6.67$ bar, the pressure drop across port P to port A orifice must be

$$\Delta p_1 = p_s - p_c$$
$$= 100 - (-6.4579)$$
$$= 106.4579\,\text{bar}$$

which is not possible. Using Eq.(1.6) the required pressure at the rod end is

$$p_r = \frac{p A_c - F_f + F_L}{A_r}$$
$$= \frac{-6.4579 \times 10^5 \times 0.002032 - 290 + 4450}{0.001070} = 26.61\,\text{bar}$$

(c) Let us use the valve with the area ratio of 2:1. We have $A_1 = 2 A_2$

$$Q_1 = C_d A_1 \sqrt{\Delta p_1} \tag{1.14}$$

$$Q_2 = C_d A_2 \sqrt{\Delta p_2} = C_d \frac{A_1}{2} \sqrt{\Delta p_2} \tag{1.15}$$

SolvingEqs. (1.14) and (1.15) we get

$$\frac{Q_1^2}{4Q_2^2} = \frac{\Delta p_1}{\Delta p_2}$$

$$\Rightarrow \Delta p_1 = \frac{Q_1^2}{4Q_2^2} \Delta p_2 \tag{1.16}$$

Using Eqs.(1.7) and (1.8) and substituting in Eq. (1.16). Solving for the pressure at the end of the cylinder, we can get

$$p_r = (p_s - p_c) \times \frac{4Q_2^2}{Q_1^2} \tag{1.17}$$

Equating Eqs.(1.6) and (1.17) , we can now solve for pressure at the cap end of the cylinder as

$$p_c = \frac{p_s \left[\dfrac{4Q_2^2}{Q_1^2} \right] - \dfrac{[F_f - F_L]}{A_r}}{\left[\dfrac{A_c}{A_r} \right] + \left[\dfrac{4Q_2^2}{Q_1^2} \right]}$$

Under certain conditions p_c can be negative. This means a vacuum will exist in the cap end of the cylinder; the cylinder will not be completely filled with oil. When this condition develops, positive control of load is lost during the extension. It is instructive to determine what load will cause p_c to go negative,

Setting $p_c = 0$, we can find the maximum overrunning load that can be controlled as

$$F_L = p_s A_r \left[\frac{4Q_2^2}{Q_1^2} \right] + [F_f]$$

For 2:1 area ratio,

$$Q_2 = \frac{Q_1}{2}$$

Therefore

$$\left[\frac{4Q_2^2}{Q_1^2} \right] = 1$$

So

$$F_{\mathrm{L}} = p_{\mathrm{s}} A_{\mathrm{r}} \left[\frac{4Q_2^2}{Q_1^2} \right] + [F_{\mathrm{f}}]$$

$$= 100 \times 10^5 \times 0.001070 + 290$$

$$= 10990\,\mathrm{N}$$

Therefore, the 2:1 area ratio valve can control the overrunning load more than three times the size load controlled with a 1:1 area ratio valve. Now the cap end pressure can be calculated using the equation

$$p_{\mathrm{c}} = \frac{p_{\mathrm{s}} \left[\dfrac{4Q_2^2}{Q_1^2} \right] - \dfrac{[F_{\mathrm{f}} - F_{\mathrm{L}}]}{A_{\mathrm{r}}}}{\left[\dfrac{A_{\mathrm{c}}}{A_{\mathrm{r}}} \right] + \left[\dfrac{4Q_2^2}{Q_1^2} \right]}$$

$$p_{\mathrm{c}} = \frac{100 \times 10^5 [1] - [290 - 4450]/0.001070}{\left[\dfrac{0.002032}{0.001070} \right] + [1]}$$

$$= \frac{61.12 \times 10^5}{2.8990}$$

$$= 21.0826\,\mathrm{bar}$$

Also

$$p_{\mathrm{r}} = \frac{p_{\mathrm{c}} A_{\mathrm{c}} - F_{\mathrm{f}} + F_{\mathrm{L}}}{A_{\mathrm{r}}}$$

$$= \frac{21.0826 \times 10^5 [0.002032] - 290 + 4450}{0.001070}$$

$$= 78.91\,\mathrm{bar}$$

The pressure drop across the valve is

$$\Delta p_1 = p_{\mathrm{s}} - p_{\mathrm{c}}$$

$$= 100 - (21.0826)$$

$$= 78.91\,\mathrm{bar}$$

$$\Delta p_2 = p_{\mathrm{r}} - 0 = 78.91\,\mathrm{bar}$$

The total pressure drop across the valve is

$$\Delta p_1 + \Delta p_2 = 78.91 + 78.91 = 157.82\,\mathrm{bar}$$

Example 1.2

Consider the hydraulic circuit with resistive load shown in Fig. 1.25. The cylinder ratio is 2:1, pressure $P_s = 100\,\text{bar}$, $A_c = 0.002032\,\text{m}^2$, $A_r = 0.001070\,\text{m}^2$, $F_f = 290\,\text{N}$, $F_L = 4450\,\text{N}$. If the valve has 2:1 area ratio. Determine p_r, p_c and total pressure drop.
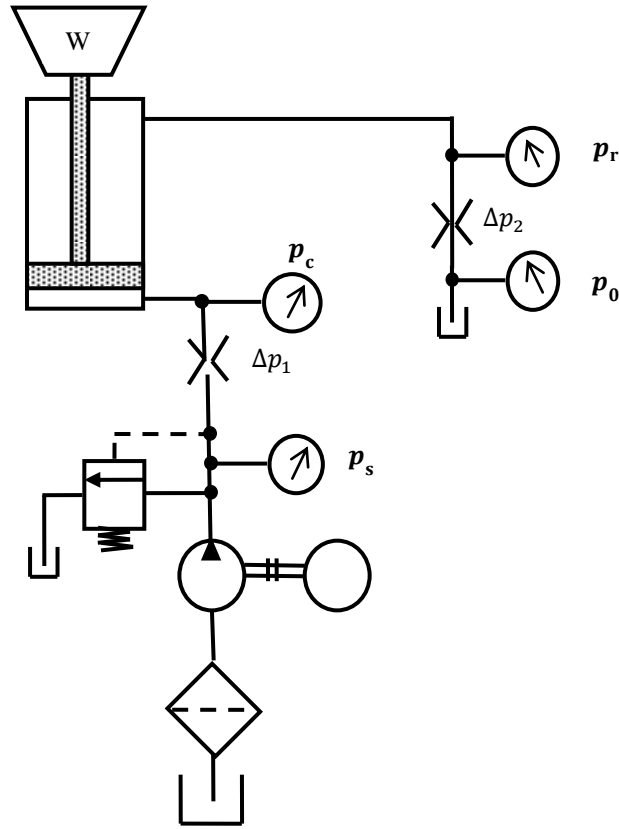

Figure 1.25 Control of resistive load.

Solution: We can write the force balance on the cylinder as

$$p_c A_c = F_f + F_L + p_r A_r$$

where $F_L = W$ is the load on the cylinder (N) and F_f is the frictional force (N). Solving for p_c we get

$$p_c = \frac{p_r A_r + F_f + F_L}{A_c} \qquad\qquad (1.18)$$

If the area ratio is unity (i.e., $A_1 = A_2 = A$), the orifice equation becomes

$$Q_1 = C_d A \sqrt{\Delta p_1}$$
$$Q_2 = C_d A \sqrt{\Delta p_2}$$

Solving we get

$$\frac{Q_1^2}{Q_2^2} = \frac{\Delta p_1}{\Delta p_2}$$

$$\Rightarrow \Delta p_1 = \frac{Q_1^2}{Q_2^2} \Delta p_2$$

Also also using Eqs.(1.7) and (1.8) in Eq. (1.12) we get

$$p_c = p_s - p_r \times \frac{Q_1^2}{Q_2^2} \qquad (1.19)$$

Equating Eqs.(1.18) and (1.19) and solving for p_r we get

$$p_r = \frac{p_s - [F_f + F_L]/A_c}{\left[\dfrac{A_r}{A_1}\right] + \left[\dfrac{Q_1^2}{Q_2^2}\right]}$$

If $F_L = 4450\,\text{N}$ and other parameters as the same as Example 1.1, then

$$p_r = \frac{p_s - [F_f + F_L]/A_c}{\left[\dfrac{A_r}{A_1}\right] + \left[\dfrac{Q_1^2}{Q_2^2}\right]} \qquad (1.20)$$

$$= \frac{100 \times 10^5 - [290 + 4450]/0.002032}{\left[\dfrac{0.001070}{0.002032}\right] + 4}$$

$$= 16.94\,\text{bar}$$

Now substituting back into Eq.(1.18) we get

$$p_c = \frac{p_r A_r + F_f + F_L}{A_c}$$

$$= \frac{16.94 \times 10^5 \times 0.001070 + 290 + 4450}{0.002032} = 32.25\,\text{bar}$$

The pressure drop across the valve are

$$\Delta p_1 = p_s - p_c = 100 - 32.25 = 67.75\,\text{bar}$$

$$\Delta p_2 = p_r = 16.94\,\text{bar}$$

So

$$\Delta p_{\text{total}} = \Delta p_1 + \Delta p_2$$
$$= 67.75 + 16.94$$
$$= 84.69\,\text{bar}$$

If the valve has a 2:1 area ratio, we have

$$\Delta p_1 = \frac{Q_1^2}{4Q_2^2} \Delta p_2$$

Equation (1.20) becomes

$$p_r = \frac{p_s - [F_f + F_L]/A_c}{\left[\dfrac{A_r}{A_1}\right] + \left[\dfrac{Q_1^2}{4Q_2^2}\right]}$$

$$= \frac{100 \times 10^5 - [290 + 4450]/0.002032}{\left[\dfrac{0.001074}{0.002032}\right] + [1]}$$

$$= 50.16 \, \text{bar}$$

Substituting back into Eq.(1.18), we get

$$p_c = \frac{p_r A_r + F_f + F_L}{A_c}$$

$$= \frac{50.16 \times 10^5 \times 0.001074 + 290 + 4450}{0.002032} = 49.84 \text{ bar}$$

The pressure drop across the valve are

$$\Delta p_1 = p_s - p_c = 100 - 49.84 = 50.16 \, \text{bar}$$

$$\Delta p_2 = p_r = 50.16 \, \text{bar}$$

So

$$\Delta p_{\text{total}} = \Delta p_1 + \Delta p_2$$
$$= 50.16 + 50.16$$
$$= 100.32 \text{ bar}$$

To select a valve for this application, the designer must look in manufacturer's literature and choose a valve with 2:1 area ratio spool having an operating curve for ~100 bar pressure drop. The best control is achieved if full spool stroke, or almost full spool stroke, is used to obtain the desired flow at the desired pressure drop. For large pressure drop like 100 bar, here, we may have to use less than the full spool stroke. Sample data for 2:1 spool area ration rated for 27 GPM with 10bar pressure drop is given in Fig. 1.26.
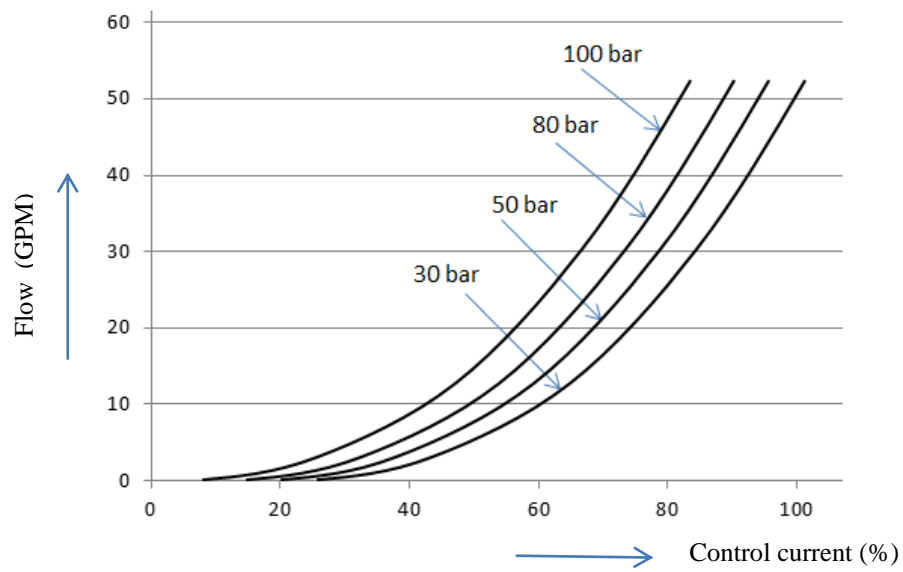
Figure 1.26

From Fig. 1.26 it is clear that, a 65% current will give 30 GPM flow at 100 bar pressure drop.

Objective-type questions

Fill in the Blanks

1. A proportional valve is a valve that produces an output (direction, pressure, flow) that is _____ to an electronic control input.

2. The performance of a proportional valve is a compromise between a conventional solenoid valve and _____ valve.

3. Proportional valves are operated by proportional _____, whereas servo valves are operated by torque motors.

4. A proportional valve has a maximum frequency response of _____.

5. In a conventional pressure control valve, a spring is used to control the pressure at which the valve operates. The spring is replaced by a _____ in the case of proportional valves.

State True or False

1. The response time for a proportional valve spool to move fully over is around 50 ms.

2. Notched spools give a better control of the flow rate.

3. All standard solenoids have no intermediate positions; rather they are always at one end or the otherof the solenoid stroke.

4. A proportional solenoid maintains a diminishing air gap dimension at the end of the plunger.

5. An electrical control to a proportional valve normally uses a variable voltage rather than a variable current.

Review Questions

1. Compare electrohydraulic servo valves with proportional hydraulic valves.
2. Where are proportional valves preferred?
3. Explain the principle of a proportional pressure-reducing valve.
4. With the help of a neat sketch, explain how the speed of a cylinder can be controlled using a proportional valve.
5. Discuss the various controls of proportional valves.
6. What is the difference between a standard solenoid and a proportional solenoid?
7. Explain the concept of operation of a proportional solenoid.
8. What is the purpose of dither in a proportional circuit?
9. Explain the difference between force control and position control in proportional control valves.
10. What is a proportional valve?
11. What is a proportional solenoid?
12. Draw the symbols of proportional 3/2-way DCV.
13. Define resolution, accuracy and repeatability as applied to proportional valves.
14. Name three applications of proportional valves.
15. What is the difference between force-controlled and stroke-controlled proportional valves?

Answers

Fill in the Blanks

1.   Proportional
2.   Servo
3.   Solenoids
4.   100 Hz
5.   DC solenoid

State True or False

1. True
2. True
3. True
4. False
5. True