Lecture 20

FLOW-CONTROL VALVES

Learning Objectives
Upon completion of this chapter, the student should be able to:

- Explain various functions of flow-control valves.
- Explain various classifications of pressure-control valves.
- Describe the working and construction of various non-compensated flow-control valves.
- Differentiate between compensated and non-compensated flow-control valves.
- Identify the graphic symbols for various types of flow-control valves.
- Explain different applications of flow-control valves.
- Explain the working principle of bleed-off circuits.
- Evaluate the performance of hydraulic systems using flow-control valves.

1.1 Introduction

Flow-control valves, as the name suggests, control the rate of flow of a fluid through a hydraulic circuit. Flow-control valves accurately limit the fluid volume rate from fixed displacement pump to or from branch circuits. Their function is to provide velocity control of linear actuators, or speed control of rotary actuators. Typical application include regulating cutting tool speeds, spindle speeds, surface grinder speeds, and the travel rate of vertically supported loads moved upward and downward by forklifts, and dump lifts. Flow-control valves also allow one fixed displacement pump to supply two or more branch circuits fluid at different flow rates on a priority basis. Typically, fixed displacement pumps are sized to supply maximum system volume flow rate demands. For industrial applications feeding two or more branch circuits from one pressurized manifold source, an oversupply of fluid in any circuit operated by itself is virtually assured. Mobile applications that supply branch circuits, such as the power steering and front end loader from one pump pose a similar situation. If left unrestricted, branch circuits receiving an oversupply of fluid would operate at greater than specified velocity, increasing the likelihood of damage to work, hydraulic system and operator.

1.1.1 Functions of Flow-Control Valves

Flow-controlvalves have several functions, some of which are listed below:

1. **Regulate the speed of linear and rotary actuators:**They control the speed of piston that is dependent on the flow rate and area of the piston:

$$\text{Velocity of piston } (V_\text{p}) \text{ (m/s)} = \frac{\text{Flow rate in the actuator (m}^3\text{/ s)}}{\text{Piston area (m}^2\text{)}} = \frac{Q}{A_\text{p}}$$

2. **Regulate the power available to the sub-circuits by controlling the flow to them:**

$$\text{Power (W)} = \text{Flow rate (m}^3\text{/s)} \times \text{Pressure (N/m}^2\text{)}$$
$$\Rightarrow P = Q \times p$$

3. **Proportionally divide or regulate the pump flow to various branches of the circuit:**It transfers the power developed by the main pump to different sectors of the circuit to manage multiple tasks, if necessary.

A partially closed orifice or flow-control valve in a hydraulic pressure line causes resistance to pump flow. This resistance raises the pressure upstream of the orifice to the level of the relief-valve setting and any excess pump flow passes via the relief valve to the tank (Fig. 1.1).

In order to understand the function and operation of flow-control devices, one must comprehend the various factors that determine the flow rate(Q) across an orifice or a restrictor. These are given as follows:

1. Cross-sectional area of orifice.
2. Shape of the orifice (round, square or triangular).
3. Length of the restriction.
4. Pressure difference across the orifice (Δp).
5. Viscosity of the fluid.

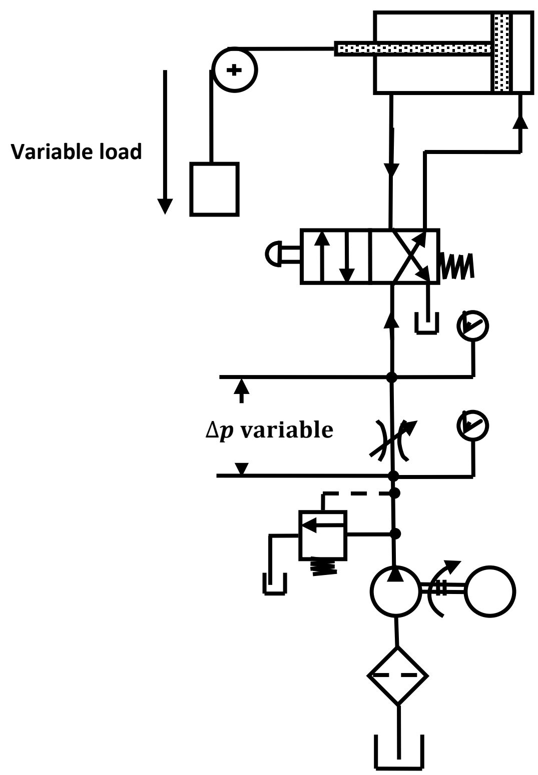

Figure 1.1 simple restrictor-type flow-control valves.

Thus, the law that governs the flow rate across a given orifice can be approximately defined as

$$Q^2 \propto \Delta p$$

This implies that any variation in the pressure upstream or downstream of the orifice changes the pressure differential Δp and thus the flow rate through the orifice (Fig. 1.2).

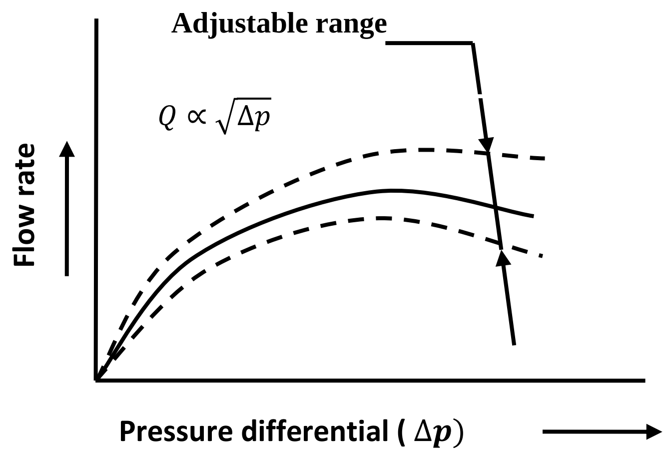

Figure 1.2 Variation of flow rate with pressure drop.

1.1.2Classification of Flow-Control Valves

Flow-control valves can be classified as follows:

1. Non-pressure compensated.
2. Pressure compensated.

1.1.2.1 Non-Pressure-Compensated Valves

Non-pressure-compensated flow-control valves are used when the system pressure is relatively constant and motoring speeds are not too critical. The operating principle behind these valves is that the flow through an orifice remains constant if the pressure drop across it remains the same. In other words, the rate of flow through an orifice depends on the pressure drop across it.

The disadvantage of these valves is discussed below. The inlet pressure is the pressure from the pump that remains constant. Therefore, the variation in pressure occurs at the outlet that is defined by the work load. This implies that the flow rate depends on the work load. Hence, the speed of the piston cannot be defined accurately using non-pressure-compensated flow-control valves when the working load varies. This is an extremely important problem to be addressed in hydraulic circuits where the load and pressure vary constantly.

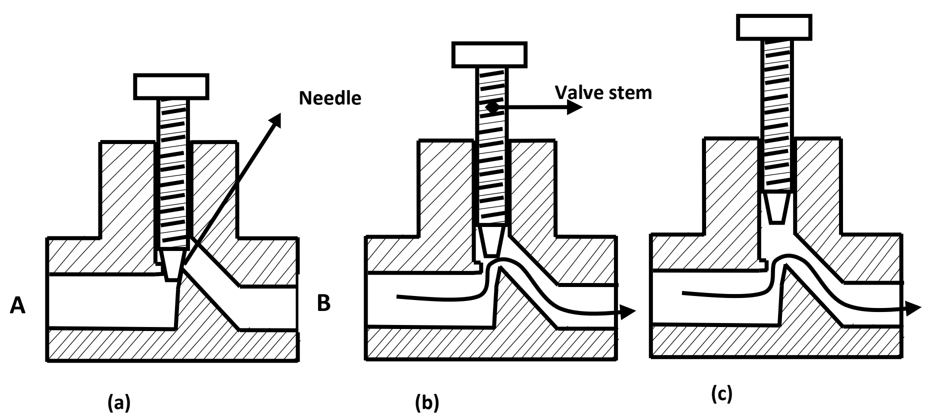

Figure 1.3 Non-pressure-compensated needle-type flow-control valve. (a) Fully closed; (b) partially opened; (c) fully opened.

Schematic diagram of non-pressure-compensated needle-type flow-control valve is shown in Fig. 1.3. It is the simplest type of flow-control valve. It consists of a screw (and needle) inside a tube-like structure. It has an adjustable orifice that can be used to reduce the flow in a circuit. The size of the orifice is adjusted by turning the adjustment screw that raises or lowers the needle. For a given opening position, a needle valve behaves as an orifice. Usually, charts are available that allow quick determination of the controlled flow rate for given valve settings and pressure drops.
Sometimes needle valves come with an integrated check valve for controlling the flow in one direction only. The check valve permits easy flow in the opposite direction without any restrictions. As shown in Fig. 1.4, only the flow from A to B is controlled using the needle. In the other direction (B to A), the check valve permits unrestricted fluid flow.

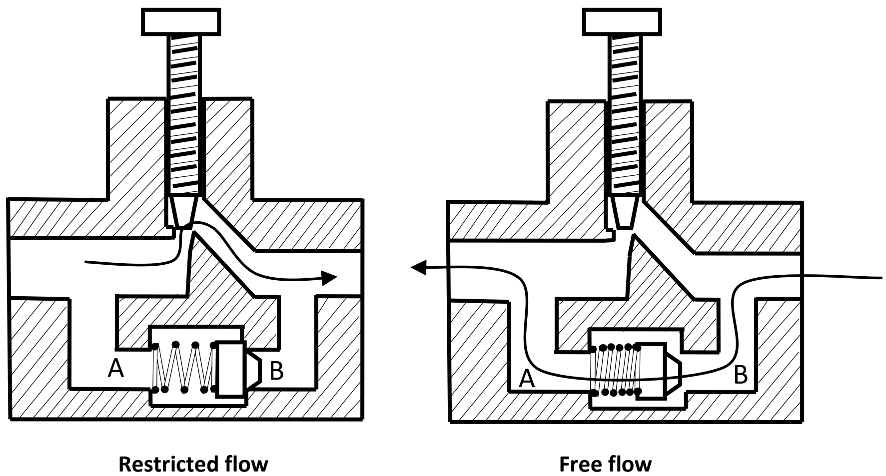

Figure 1.4 Flow-controlvalve with an integrated check valve.

1.1.2.2Pressure-Compensated Valves

Pressure-compensated flow-control valvesovercome the difficulty causedby non-pressure-compensated valves by changing the size of the orifice in relation to the changes in the system pressure. This is accomplished through a spring-loaded compensator spool that reduces the size of the orifice when pressure drop increases. Once the valve is set, the pressure compensator acts to keep the pressure drop nearly constant. It works on a kind of feedback mechanism from the outlet pressure. This keeps the flow through the orifice nearly constant.

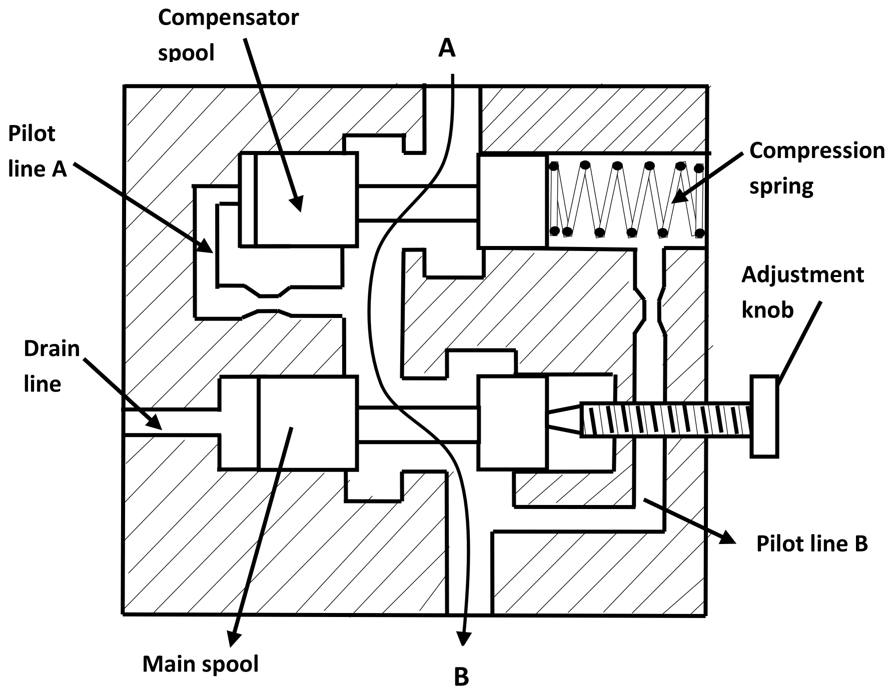

Figure 1.5 Sectional view of a pressure-compensated flow-control valve.

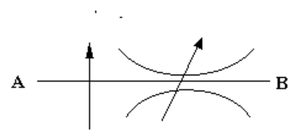

Figure 1.6 Graphic symbol of a pressure-compensated flow-control valve.

Schematic diagram of a pressure compensated flow-control valve is shown in Fig. 1.5 and its graphical symbol in Fig. 1.6. A pressure-compensated flow-control valve consists of a main spool and a compensator spool. The adjustment knob controls the main spool's position, which controls the orifice size at the outlet. The upstream pressure is delivered to the valve by the pilot line A. Similarly, the downstream pressure is ported to the right side of the compensator spool through the pilot line B. The compensator spring biases the spool so that it tends toward the fully open position. If the pressure drop across the valve increases, that is, the upstream pressure increases relative to the downstream pressure, the compensator spool moves to the right against the force of the spring. This reduces the flow that in turn reduces the pressure drop and tries to attain an equilibrium position as far as the flow is concerned.

In the static condition, the hydraulic forces hold the compensator spool in balance, but the bias spring forces it to the far right, thus holding the compensator orifice fully open. In the flow condition, any pressure drop less than the bias spring force does not affect the fully open compensator orifice,but any pressure drop greater than the bias spring force reduces the compensator orifice. Any change in pressure on either side of the control orifice, without a corresponding pressure change on the opposite side of the control orifice, moves the compensator spool. Thus, a fixed differential across the control orifice is maintained at all times. It blocks all flow in excess of the throttle setting. As a result, flow exceeding the preset amount can be used by other parts of the circuit or return to the tank via a pressure-relief valve.

Performance of flow-control valve is also affected by temperature changes which changes the viscosity of the fluid. Therefore, often flow-control valves have temperature compensation. Graphical symbol for pressure and temperature compensated flow-control valve is shown in Fig. 1.7.

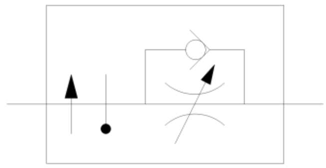

Figure 1.7 Pressure- and temperature-compensated flow-control valve.

1.2Speed-Controlling Circuits

In hydraulic operations, it is necessary to control the speed of the actuator so as to control the force, power, timing and other factors of the operation. Actuator speed control is achieved by controlling the rate of flow into or out of the cylinder.

Speed control by controlling the rate of flow into the cylinder is called meter-in control.Speed control by controlling the rate of flow out of the cylinder is called meter-out control.

1.2.1 Meter-In Circuit

Figure 1.8 shows a meter-in circuit with control of extend stroke. The inlet flow into the cylinder is controlled using a flow-control valve. In the return stroke, however, the fluid can bypass the needle valve and flow through the check valve and hence the return speed is not controlled. This implies that the extending speed of the cylinder is controlled whereas the retracing speed is not.

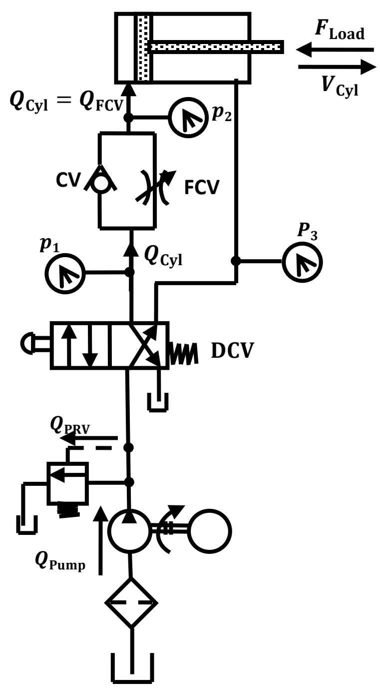

Figure 1.8 Meter-in circuit.

1.2.2 Meter-Out Circuit

Figure 1.9 shows a meter-out circuit for flow control during the extend stroke. When the cylinder extends, the flow coming from the pump into the cylinder is not controlled directly. However, the flow out of the cylinder is controlled using the flow-control valve (metering orifice). On the other hand, when the cylinder retracts, the flow passes through the check valve unopposed, bypassing the needle valve. Thus, only the speed during the extend stroke is controlled.

Both the meter-in and meter-out circuits mentioned above perform the same operation (control the speed of the extending stroke of the piston), even though the processes are exactly opposite to one another.

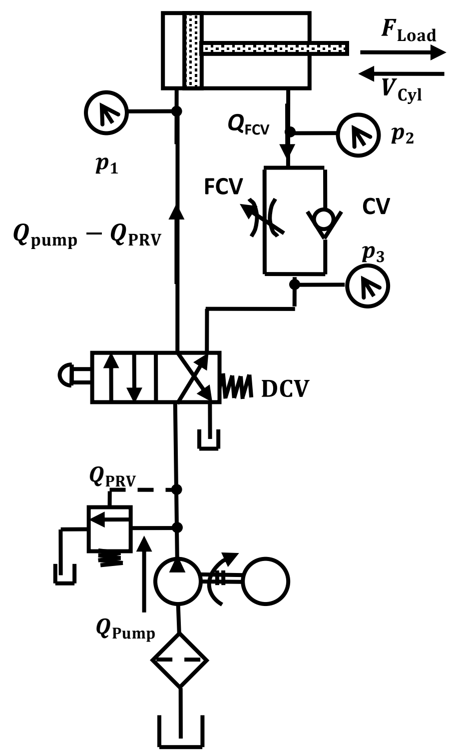

Figure 1.9 Meter-Out circuit.

1.2.3 Bleed-Off Circuit

Compared to meter-in and meter-out circuits, a bleed-off circuit is less commonly used. Figure 1.10 shows a bleed-off circuit with extend stroke control. In this type of flow control, an additional line is run through a flow-control valve back to the tank. To slow down the actuator, some of the flow is bledoff through the flow-control valve into the tank before it reaches the actuator. This reduces the flow into the actuator, thereby reducing the speed of the extend stroke.

The main difference between a bleed-off circuit and a meter-in/meter-out circuit is that in a bleed-off circuit, opening the flow-control valve decreases the speed of the actuator, whereas in the case of a meter-in/meter-out circuit, it is the other way around.

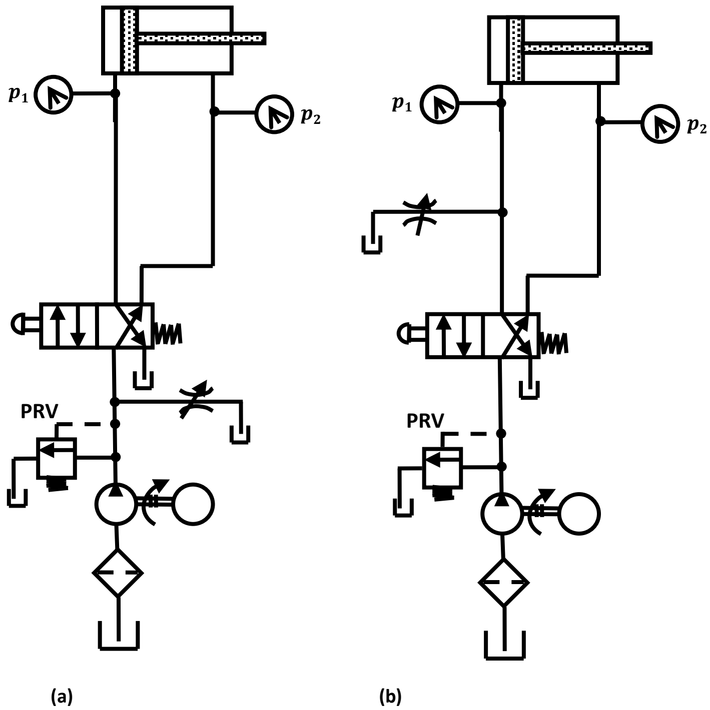

Figure 1.10 Bleed-off circuits:(a) Bleed-off for both directions and (b) bleed-off for inlet to the cylinder or motor.

Example 1.1

A 55-mm diameter sharp-edged orifice is placed in a pipeline to measure the flow rate. If the measured pressure drop is 300 kPa and the fluid specific gravity is 0.90, find the flow rate in units of m^3/s.

Solution: For a sharp-edged orifice, we can write

$$Q = 0.0851 A C_{\text{V}} \sqrt{\frac{\Delta p}{\text{SG}}}$$

where Q is the volume flow rate in LPM, C_{V} is the capacity coefficient = 0.80 for the sharp-edge orifice, c = 0.6 for a square-edged orifice, A is the area of orifice opening in mm^2, Δp is the pressure drop across the orifice (kPa) and SG is the specific gravity of the flowing fluid = 0.9. Now,

$$A_{\text{orifice}} = \frac{\pi}{4}(D_{\text{orifice}}{}^2) = \frac{\pi}{4}(55^2) = 2376 \text{ mm}^2$$

Using the orifice equation we can find the flow rate as

$$Q\text{ (LPM)} = 0.0851 \times 2376 \times 0.80 \sqrt{\frac{300}{0.9}}$$

$$= 2953.3 \text{ LPM} = 0.0492 \ \text{m}^3/\text{s}$$

Example 1.2

For a given orifice and fluid, a graph can be generated showing a Δp versus Q relationship. For the orifice and fluid in Example 1.1, plot the curves and check the answers obtained mathematically. What advantage does the graph have over the equation? What is the disadvantage of the graph?

Solution: From Example 1.1, we have

$$Q\ (\text{LPM}) = 0.0851 \times 2376 \times 0.80 \sqrt{\frac{300}{0.9}}$$

We can write the general expression as

$$Q\ (\text{LPM}) = 0.0851 \times 2376 \times 0.80 \sqrt{\frac{\Delta p}{0.9}} = 161.76 \sqrt{\frac{\Delta p}{0.9}} = 170.5\ \sqrt{\Delta p}$$

Using Excel, the graph shown in Fig. 1.11 is obtained.

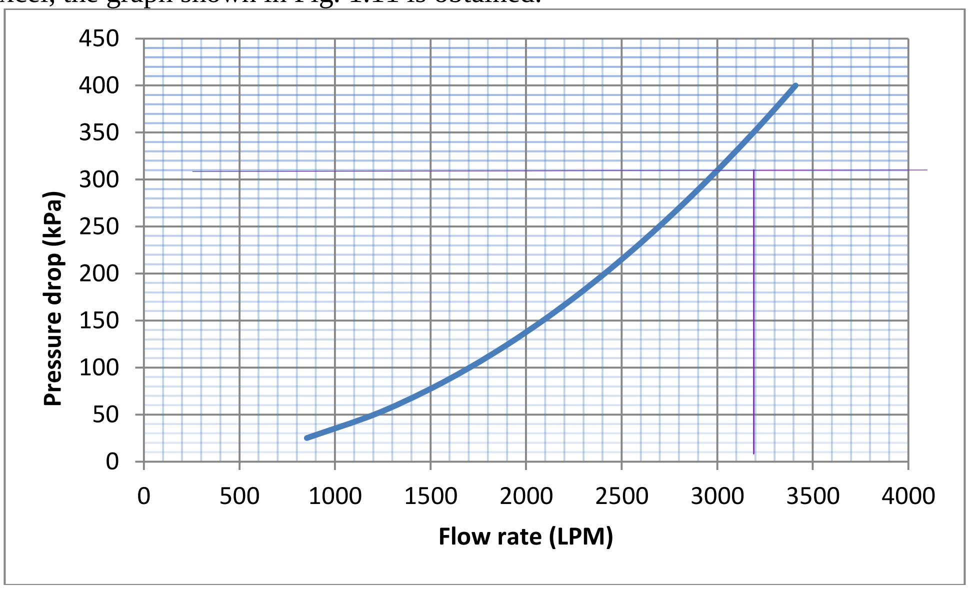

Figure 1.11 Pressure drop versus flow rate.

From the graph, corresponding to $\Delta p = 300$ kPa, we get Q = 2950 LPM which is close to 2953.3 LPM.A graph is quicker to use but is not as accurate as the equation. A pressure gauge can be calibrated (according to this relationship) to read Q directly rather than Δp .

Example 1.3

Determine the flow rate through a flow-control valve that has a capacity coefficient of $2.2\,\text{LPM}/\sqrt{\text{kPa}}$ and a pressure drop of 687 kPa. The fluid is hydraulic oil with a specific gravity of 0.90.

Solution: For a sharp-edged orifice, we can write

$$Q = 2.2\sqrt{\frac{\Delta p}{\text{SG}}} = 2.2\sqrt{\frac{687}{0.9}} = 60.8\,\text{LPM}$$

Example 1.4

The system shown in Fig. 1.12 has a hydraulic cylinder with a suspended load *W*. The cylinder piston and rod diameters are 50.8 and 25.4 mm, respectively. The pressure-relief valve setting is 5150 kPa. Determine the pressure p_2 for a constant cylinder speed:

(a) $W = 8890$ N
(b) $W = 0$ (load is removed)
(c) Determine the cylinder speeds for parts (a) and (b) if the flow-control valve has a capacity coefficient of $0.72\,\text{LPM}/\sqrt{\text{kPa}}$. The fluid is hydraulic oil with a specific gravity of 0.90.

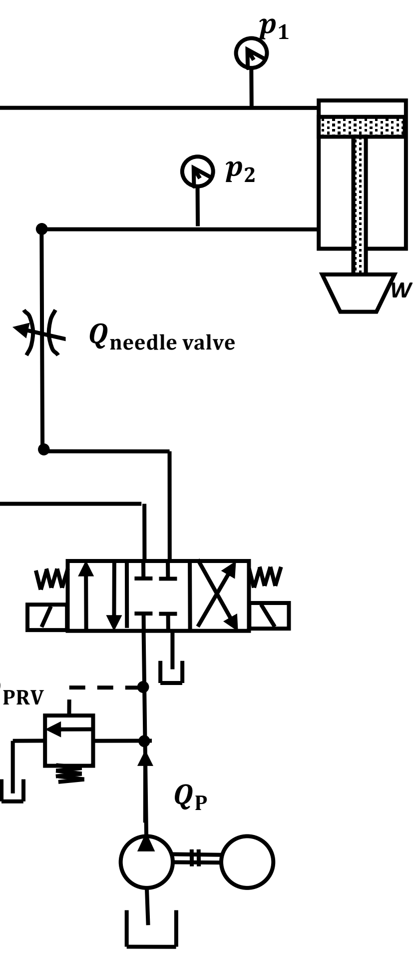

Figure 1.12 Hydraulic cylinder with a suspended weight.

Solution:
For a constant cylinder speed, the summation of the forces on the hydraulic cylinder must be equal to zero. Thus, we have

$$-W - p_1 A_p + p_2 (A_p - A_r) = 0$$

where $p_1 =$ pressure-relief valve setting $= 5150\,\text{kPa}$. Now

$$A_{\mathrm{p}} = \frac{\pi}{4}(D_{\mathrm{P}}{}^2) = \frac{\pi}{4}(0.0508^2) = 0.00203\,\mathrm{m}^2$$

$$A_{\mathrm{r}} = \frac{\pi}{4}(D_{\mathrm{R}}{}^2) = \frac{\pi}{4}(0.0254^2) = 0.000506\,\mathrm{m}^2$$

So

$$A_{\mathrm{p}} - A_{\mathrm{r}} = 0.00152\ \mathrm{m}^2$$

Case 1: If W = 8890 N.

$$-W - p_1 A_{\mathrm{p}} + p_2(A_{\mathrm{p}} - A_{\mathrm{r}}) = 0$$

$$\Rightarrow -8890 - 5150\times10^3\,\mathrm{N/m^2}\times2.03\times10^{-3}\ \mathrm{m}^2 + p_2(0.00152\ \mathrm{m}^2) = 0$$

$$\Rightarrow -8890 - 10450\,\mathrm{m}^2 + p_2(0.00152) = 0$$

$$\Rightarrow p_2 = 12700\,\mathrm{kPa}$$

Case 2: If W = 0.

$$0 - 5150\times10^3\ \mathrm{N/m^2}\times2.03\times10^{-3}\ \mathrm{m}^2 + p_2(0.00152\,\mathrm{m}^2) = 0$$

$$\Rightarrow p_2 = 6880\,\mathrm{kPa}$$

Case 3: Cylinder speed for case 1: For a sharp-edged orifice, we can write

$$Q = C_{\mathrm{V}}\sqrt{\frac{\Delta p}{\mathrm{SG}}} = 0.72\sqrt{\frac{12700}{0.9}} = 85.5\,\mathrm{LPM}$$

where $\Delta p = p_2$ because the flow-control valve discharges directly to the oil tank. This is the flow rate through the flow-control valve and thus the flow rate of the fluid leaving the hydraulic cylinder. Thus, we have

$$v_{\mathrm{p}}(A_{\mathrm{p}} - A_{\mathrm{r}}) = Q$$

$$\Rightarrow v_{\mathrm{p}}\ (\mathrm{m/s})(0.00152)\ \mathrm{m}^2 = 85.5\ \ \mathrm{L/min}\times\frac{1\,\mathrm{m}^3}{10^3\ \mathrm{L}}\times\frac{1\,\mathrm{min}}{60\,\mathrm{s}}$$

$$\Rightarrow v_{\mathrm{p}} = 0.938\ \mathrm{m/s}$$

Case 4: Cylinder speed for case 2. We have

$$Q = C_{\mathrm{V}}\sqrt{\frac{\Delta p}{\mathrm{SG}}} = 0.72\sqrt{\frac{6880}{0.9}} = 63\mathrm{LPM}{=}63\mathrm{L/min}\times\frac{1\,\mathrm{m}^3}{10^3\ \mathrm{L}}\times\frac{1\,\mathrm{min}}{60\,\mathrm{s}} = 0.00105\frac{\mathrm{m}^3}{s}$$

Also we can write

$$\begin{aligned} Q &= \mathrm{Velocity}\times\mathrm{Area} \\ &= v_{\mathrm{p}}\times A \\ &= 0.00105\frac{\mathrm{m}^3}{s} \end{aligned}$$

$$\Rightarrow v_{\mathrm{p}}\ (\mathrm{m/s})(0.00152)\ \mathrm{m}^2 = 0.00105$$

$$\Rightarrow v_{\mathrm{p}} = 0.691\ \mathrm{m/s}$$

Example 1.5

A cylinder has to exert a forward thrust of 100 kN and a reverse thrust of 10 kN. The effects of using various methods of regulating the extend speed will be considered. In all the cases, the retract speed should be approximately 5 m/min utilizing full pump flow. Assume that the

maximum pump pressure is 160 bar and the pressure drops over the following components and their associated pipe work (where they are used):
Filter = 3 bar
Directional control valve (DCV) = 2 bar
Flow-control valve (controlled flow) = 10 bar
Flow-control valve (check valve) = 3 bar
Determine the following:

(a) The cylinder size (assume the piston-to-rod area ratio to be 2:1).
(b) Pump size.
(c) Circuit efficiency when using the following:

Case 1: No flow controls (calculate the extend speed).
Case 2: Meter-in flow control for extend speed 0.5 m/min.
Case 3: Meter-out flow control for extend speed 0.05 m/min.

Solution:

Case 1: No flow controls (Fig. 1.13)
Part (a) No flow controls
Maximum available pressure at the full bore end of cylinder = 160 − 3 − 2 = 155 bar
Back pressure at the annulus side of cylinder = 2 bar.
This is equivalent to 1 bar at the full bore end because of the 2:1 area ratio. Therefore, the maximum pressure available to overcome load at the full bore end is 155 − 1 = 154 bar

Full bore area = Load/Pressure = $\dfrac{100 \times 103}{154 \times 105}$ =0.00649 m^2

Piston diameter = $\left(\dfrac{4 \times 0.00649}{\pi}\right)^{1/2}$

Select a standard cylinder, say with 100-mm bore and 70-mm rod diameter. Then

Full bore area = 7.85×10^{-3} m^2

Annulus area = 4.00×10^{-3} m^2

This is approximately a 2:1 ratio.

Part (b) No flow controls
Flow rate for a return speed of 5 m/min is given by

$$\text{Area} \times \text{Velocity} = 4.00 \times 10^{-3} \times 5 \text{ m}^3\text{/min} = 20 \text{ LPM}$$

Extend speed = $\dfrac{20 \times 10^{-3}}{7.85 \times 10^{-3}}$ = 2.55 m/min

Pressure to overcome load on extend = $\dfrac{100 \times 10^{3}}{7.85 \times 10^{-3}}$ = 12.7 MPa = 127 bar

Pressure to overcome load on retract = $\dfrac{10 \times 10^{3}}{4.00 \times 10^{-3}}$ = 2.5 MPa = 25 bar

(i) Pressure at pump on extend (working back from the DCV tank port)

Pressure drop over DCV B to T	2 × (1/2)	1
Load-induced pressure		127
Pressure drop over DCV P to A		2
Pressure drop over filter		3

Therefore, pressure drop required at the pump during extend stroke = 133 bar

Relief-valve setting = 133 + 10% = 146 bar

(ii) Pressure required at the pump on retract (working from the DCV port as before) is

$$(2 \times 2) + 25 + 2 + 3 = 34 \text{ bar}$$

Note: The relief valve will not be working other than at the extremities of the cylinder stroke. Also, when movement is not required, pump flow can be discharged to the tank at low pressure through the center condition of the DCV.

Part (c) No flow controls

System efficiency:

$$\frac{\text{Energy required to overcome load on the cylinder}}{\text{Total energy into fluid}} = \frac{\text{Flow to the cylinder} \times \text{Pressure owing to load}}{\text{Flow from the pump} \times \text{Pressure at the pump}}$$

Efficiency on extend stroke = $\frac{20 \times 127}{20 \times 133} \times 100 = 95.5\ \%$

Efficiency on retract stroke = $\frac{20 \times 25}{20 \times 34} \times 100 = 73.5\ \%$

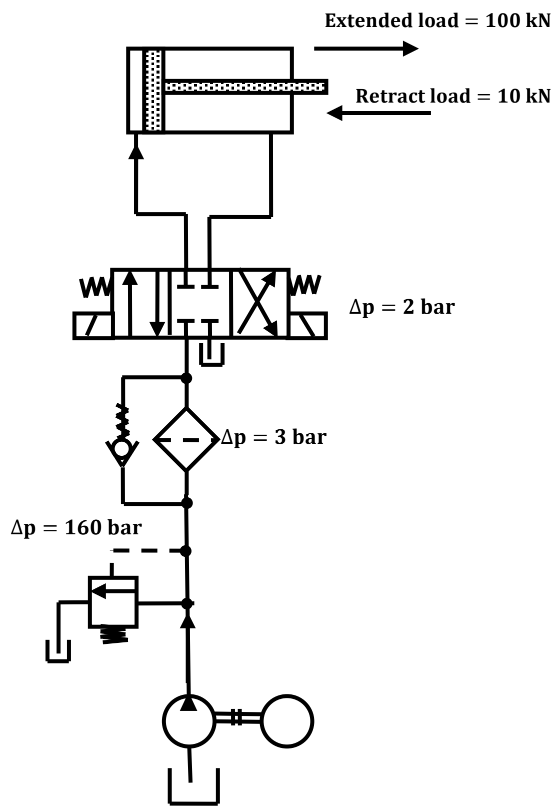

Figure 1.13 Hydraulic cylinder with no control.

Case 2: Meter-in flow control for the extend speed of 0.5 m/min(Fig. 1.14)

Part (a) meter in controls
From case 1,
Select a standard cylinder, say with 100-mm bore and 70-mm rod diameter.

Cylinder 100-mm bore diameter × 70-mm rod diameter
Full bore area 7.85×10^{-3} m^2
Annulus area = 4.00×10^{-3} m^2
Load-induced pressure on extend = 127 bar
Load-induced pressure on retract = 25 bar
Pump flow rate = 20 L/min
Part (b) meter in controls
Flow rate required for the extend speed of 0.5 m/min is

$$7.85 \times 10^{-3} \times 0.5 = 3.93 \times 10^{-3}\ \text{m}^3/\text{min} = 3.93\ \text{L/min}$$

Working back from the DCV tank port:
Pressure required at the pump on retract is

$$(2 \times 2) + (2 \times 3) + 25 + 2 + 3 = 40\ \text{bar}$$

Pressure required on the pump at extend is

$$2 \times (1/2) + 127 + 10 + 2 + 3 = 143\ \text{bar}$$

Relief-valve setting = 143 + 10% = 157 bar

This is close to the maximum working pressure of the pump (160 bar). In practice, it would be advisable to select either a pump with a higher working pressure (210 bar) or use the next standard size of the cylinder. In the latter case, the working pressure would be lower but a higher flow rate pump would be necessary to meet the speed requirements.

Part (c) meter in controls
Now that a flow-control valve has been introduced when the cylinder is on the extend stroke, the excess fluid will be discharged over the relief valve.

$$\text{System efficiency on extend} = \frac{3.93 \times 127}{20 \times 157} \times 100 = 15.9\%$$

$$\text{System efficiency on retract} = \frac{20 \times 25}{20 \times 40} \times 100 = 62.5\%$$

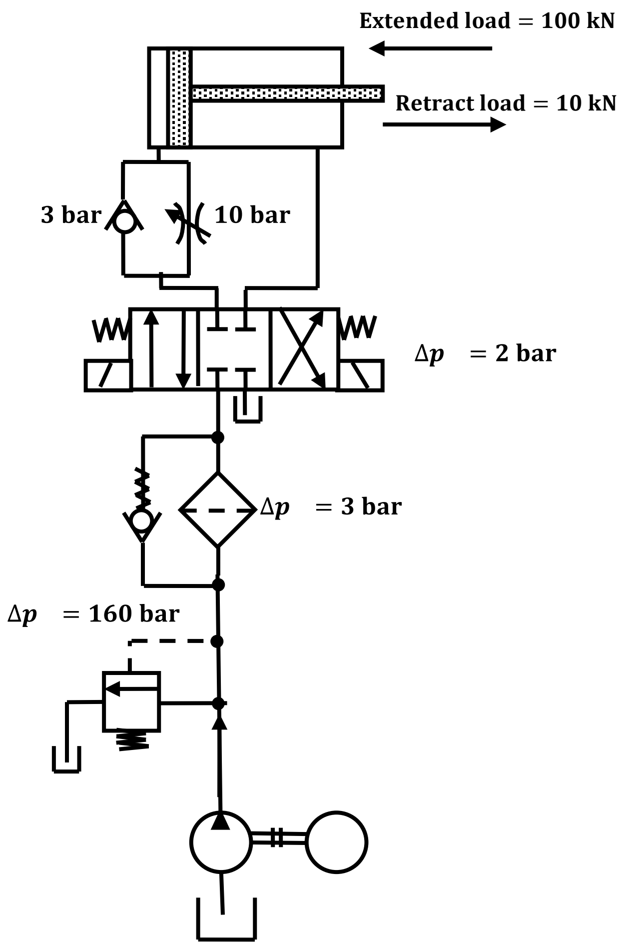

Figure 1.14 Hydraulic cylinder with meter-in control

Case 3: Meter-out flow control for the extend speed of 0.5 m/min(Fig. 1.15)
Cylinder, load, flow rate and pump details are as before (partsa and b of meter in control).

Part (c) meter out controls
Working back from the DCV tank port:

Pressure required at the pump on retract is

$$(2 \times 2) + 25 + 3 + 2 + 3 = 37 \text{ bar}$$

Pressure required at the pump on extend is

$$[2 \times (1/2)] + [10 \times (1/2)] + 127 + 2 + 3 = 138 \text{ bar}$$

Relief-valve setting = 38 + 10 % = 152 bar

System efficiency on extend = $\dfrac{3.93 \times 127}{20 \times 152} \times 100 = 16.4\%$

System efficiency on retract = $\dfrac{20 \times 25}{20 \times 37} \times 100 = 67.6\%$

Discussion of results of all three cases: No control, meter-in and meter-out.
As can be seen, meter-out is marginally more efficient than meter-in owing to the ratio of piston to piston rod area. Both systems are equally efficient when used with through-rod cylinders or hydraulic motors. It must be remembered that meter-out should prevent any tendency of the load to run away.

In both cases, if the system runs light, that is, extends against a low load, excessive heat is generated over the flow controls in addition to the heat generated over the relief valve. Consequently, there is further reduction in the efficiency. Also, in these circumstances, with meter-out flow control, very high pressure intensification can occur on the annulus side of the cylinder and within the pipe work between the cylinder and the flow-control valve. Take a situation where meter-out circuit is just considered. The load on extend is reduced to 5 kN without any corresponding reduction in the relief-valve settings.

Flow into the full bore end is 3.91 L/min.
Therefore, excess flow from the pump is

$$20 - 3.93 = 16.07 \text{ L/min}$$

that passes over the relief valve at 152 bar.
The pressure at the full bore end of the cylinder is = 152 – 3 – 2 = 147 bar
This exerts a force that is resisted by the load and the reactive back pressure on the annulus side:

$$147 - \left(\frac{5 \times 10^{3}}{7.85 \times 10^{-3} \times 10^{5}} \right) = (2 + 10 + p) \times \frac{4.00}{7.85}$$

where*p* is the pressure within the annulus side of the cylinder and between the cylinder ant the flow-control valve. So

$$p = [(147 - 6.4) \times 7.85/4.00] - 12 = 264 \text{ bar}$$

The system efficiency on extend is

$$\frac{3.93 \times 6.4}{20 \times 152} \times 100 = 0.83\%$$

Almost all of the input power is wasted and dissipates as heat into the fluid, mainly across the relief and flow-control valves.

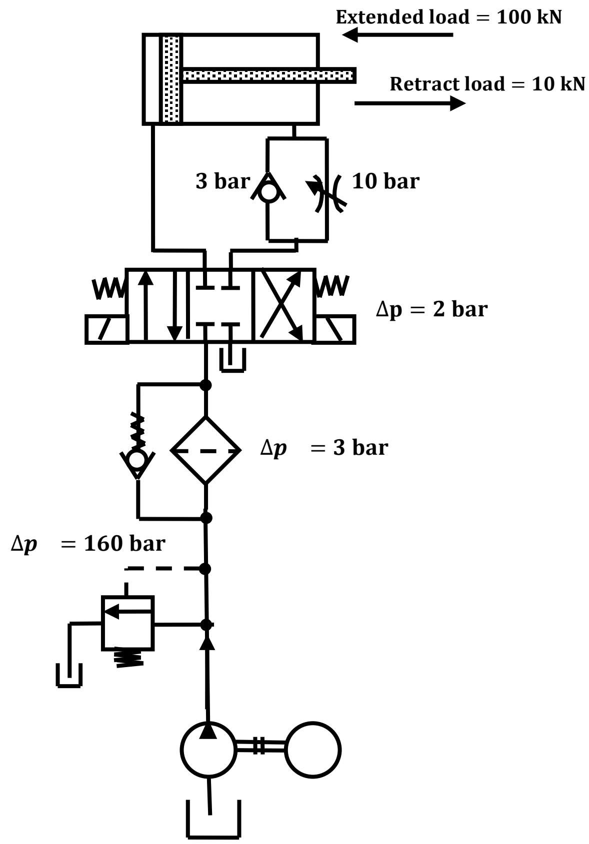

Figure 1.15 Hydrauliccylinder with meter-out control.

Example 1.6

Figure 1.16 shows a hydraulic circuit where the actuator speed is controlled by a meter-in system employing a series pressure-compensated valve. Determine the power input to the pump under a steady-state condition. If the series compensation is replaced by parallel compensation, and the load and speed of the actuator remain unchanged, determine the change of overall efficiency of the circuit.

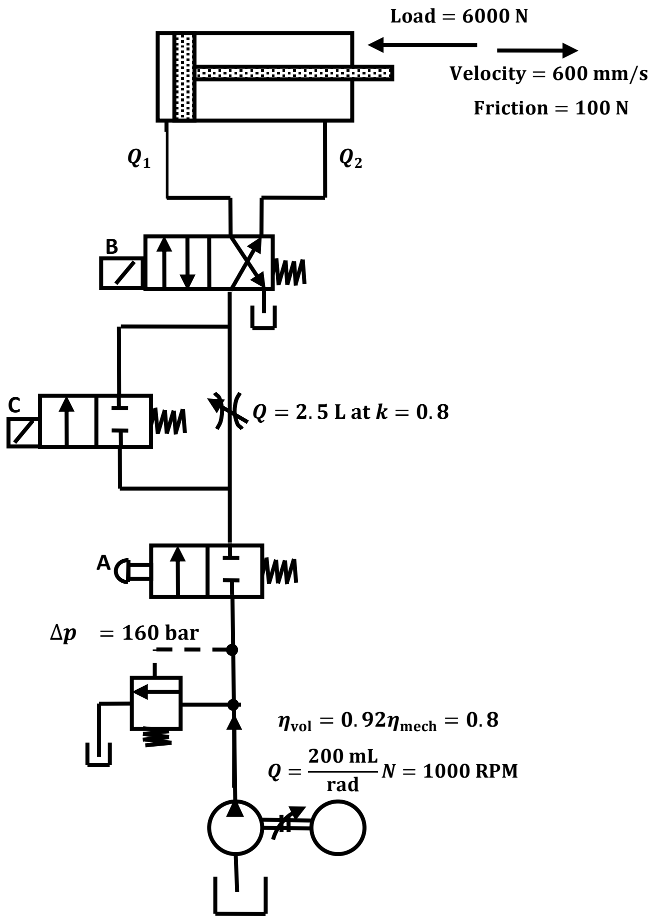

Figure 1.16 Hydraulic cylinder with a pressure-compensated valve.

Solution:

For valve A we have $Q = 0.5\sqrt{\Delta p}$

For valve B we have $Q = 0.4\sqrt{\Delta p}$

For valve C we have $Q = 0.3\sqrt{\Delta p}$

Now referring to Fig. 1.16, let us calculate the flow to piston side of the cylinder:

$$Q_1 = \frac{\pi}{4} \times 60^2 \times 600 = 1.7 \text{ L/s}$$

Flow from the return side of the cylinder is

$$Q_2 = 1.4 \text{ L/s}$$

Pump flow is given by

$$Q_p = 2\pi \times 1000/60 \times 20 = 2.1 \text{ L/s}$$

$$Q_3 = \eta_{vol} \times Q_p = 1.93 \text{ L/s}$$

Power input to the pump = $150 \times 10^5 \times 2.1 \times 10^{-3} \times 1/0.8$ W = 39.4 kW

Power output to the actuator = $6100/1000 \times 600/1000 = 3.6$ kW

Therefore, system efficiency = $3.66/39.4 \times 100 = 9\%$

Pressure loss at valve B due to $Q_2 = 1 \times (1.4/0.4)^2 = 12.2$ bar

Pressure at the head end of actuator

$$p \times \frac{\pi}{4} \times 60^2 \times 10^{-6} = 6100 + 12.2 \times 10^5 \times \frac{\pi}{4} \times (602 - 252) \times 10^{-6}$$

$$\Rightarrow p = 29 \text{ bar}$$

Pressure losses at B due to $Q_1 = (1.7/0.4)^2 = 18$ bar
Pressure losses at valve A = $(1.7/0.5)^2 = 11.6$ bar
Therefore, the total pressure, excluding that lost in the pressure-compensated valve if it is of series type, is

$$29 + 18 + 11.6 + 4 = 62.6 \text{ bar}$$

Hence, 150 – 62.6 = 87.4 bar is dropped in the pressure-compensated valve if it is of series type.
For a parallel pressure-compensated valve, the excess oil $Q - Q_1$ would bypass at

$$62.6 \text{ bar} - 11.6 \text{ bar} = 51 \text{ bar}$$

The pump delivery would be at 62.6 bar and hence the total power consumption is

$$62.6 \times 10^5 \times 2.1 \times 10^{-3} \times 1/0.8 \text{ W} = 16.5 \text{ kW}$$

System efficiency = 22.2%

Example 1.7

A flow-control valve is used to control the speed of the actuator as shown in Fig. 1.17 and the characteristics of the system are given in Table 1.1. Determine the variable flow area A_v, the pressure downstream of the valve fixed orifice p_2, the valve displacement x and the spring preload F for the given motor operating conditions.

Table 1.1

Parameters	**Value**
Valve flow constant (C_d)	0.6
Length h	7.8 mm
Valve area gradient for flow area (A_v), b	1.25 mm^2/mm
Fixed orifice flow area (A_o)	4.9 mm^2
Valve face area (A)	125 mm^2
Spring constant	57 kN/m
Motor displacement (D_m)	40 cm^3/rev
Motor torque	60 Nm
Motor speed	350 RPM
Motor volumetric efficiency (η_v)	96 %
Motor mechanical efficiency (η_m)	97 5

System pressure (p_1)	14.5 MPa
Return pressure (p_4)	1 MPa
Fluid density	840 kg/m^3

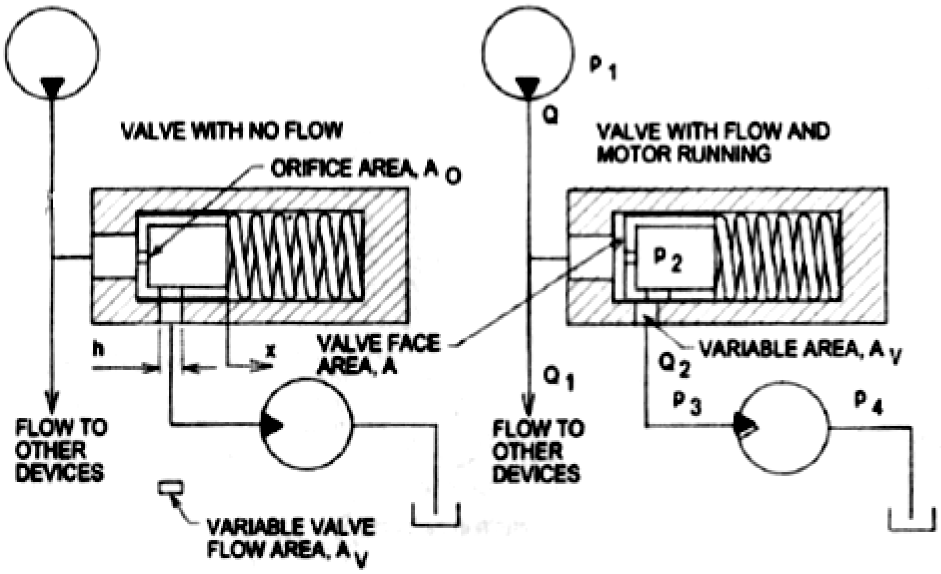

Figure 1.17

Solution: Refer to Fig. 1.17, flow from the pump divides as Q_1 and Q_2 . The pressure drop $p_1 - p_2$ occurs across orifice A_o. This makes the valve to move to right against the spring force F. The area of orifice A_v then adjusts to control the flow to the motor:

Let us first convert all the given variables to appropriate units

$$h = \frac{7.8}{1000}\text{ m},\; w = \frac{1.25}{1000}\text{ mm},\; k = 57000\text{ N/m},\; A_o = 4.9\times10^{-6}\text{ m}^2,\; A = 125\times10^{-6}\text{ m}^2$$

$$D_m = \frac{\frac{40}{100^3}\text{ m}^3}{\text{rev}},\; T = 60\text{ N m},\; n = 350\,\text{RPM},\; \rho = 840\text{ kg/m}^3,\; p_1 = 145000\times1000\text{ N/m}^2$$

$$\eta_V = 96\%,\; \eta_m = 97\%,\; p_4 = 1000\times1000\text{ Pa},\; \theta = \frac{2\times\pi\times350}{60} = 36.652\,\text{rad/s}$$

First let us calculate the discharge Q_2 through valve

$$Q_2 = \frac{\text{Motor displacement}}{\text{Revolution}} \times \frac{\text{Speed}}{\text{Volumetric efficiency}}$$
$$= \frac{(40/100^3)\ \text{m}^3}{0.96} \times \frac{350}{60}$$
$$= 2.431 \times 10^{-4}\ \text{m}^3/\text{s}$$

Pressure drop across motor is $p_3 - p_4$. So

$$p_3 - p_4 = \frac{\text{Torque}}{\text{Flow rate} \times \text{Mechanical efficiency}}$$

$$p_3 = \frac{T}{0.97} \times \frac{2\pi}{(40/100^3)\ \text{m}^3} + p_4$$
$$= \frac{60}{0.97} \times \frac{2\pi}{(40/100^3)\ \text{m}^3} + 10^6$$
$$= 1.072 \times 10^7\ \text{Pa}$$

Also flow through orifice from motor side is given by

$$Q_2 = c_d A_o \sqrt{\frac{2}{840}} \times \sqrt{p_2 - p_1}$$
$$2.431 \times 10^{-4} = 0.6 \times 4.9 \times 10^{-6} \sqrt{\frac{2}{840}} \times \sqrt{p_2 - 14500000}$$

Substituting all the known values we can get p_2

$$p_2 = 1.163 \times 10^7\ \text{Pa}$$

Also flow through orifice from pump side is given by

$$Q_2 = c_d A_V \sqrt{\frac{2}{840}} \times \sqrt{p_2 - p_3} = 0.6 \times A_V \sqrt{\frac{2}{840}} \times \sqrt{1.163 \times 10^7 - 1.072 \times 10^7}$$

Solving we get the value of A_V as

$$A_V = 8.688 \times 10^{-6}\ \text{m}^2 \text{ or } 8.688\,\text{mm}^2$$

But area of the valve is

$$A_V = (h - x)w$$

Solving we get

$$x = \frac{h \times w - A_V}{w} = 8.499 \times 10^{-4}\ \text{m}$$

Now let us write the force balance we get (see Fig. 1.18)

$$p_1 A = p_2 A + (K x + F)$$

Substituting all the values we get

$$14500 \times 1000 \text{ N/m}^2 \times 125 \times 10^{-6} = 1.163 \times 10^7 \times 125 \times 10^{-6} + 57000 \times 8.499 \times 10^{-4} + F$$

$$F = 310.4 \text{ N}$$

Figure 1.18

Objective-Type Questions

Fill in the Blanks

1. Non-pressure-compensated flow-control valves are used when the system pressure is relatively ________ and motoring speeds are not too critical.
2. A pressure compensator acts to keep the ________ nearly constant.
3. In ________ flow-control valve consists of a main spool and a compensator spool.
4. Speed control by controlling the rate of flow ________ the cylinder is called meter-in control.
5. In a meter-________ circuit, only the speed during the extend stroke is controlled.

State True or False

1. The speed of a piston can be defined accurately using non-pressure-compensated flow-control valves when the working load varies.
2. Speed control by controlling the rate of flow out of the cylinder is called meter-in control.
3. In a meter-in circuit, the extending speed of the cylinder is controlled whereas the retracing speed is not.
4. Compared to the meter-in and meter-out circuits, the bleed-off circuit is more commonly used.

5. In a meter-in circuit, flow coming from the pump into the cylinder is not controlled directly. However, the flow out of the cylinder is controlled using a flow-control valve.

Review Questions

1. What is the function of a flow-control valve?
2. What are the three ways of applying flow-control valves?
3. What is meant when a flow-control valve is said to be pressure compensated?
4. What is a meter-in circuit and where is it used?
5. What is a meter-out circuit and where is it used?
6. What are the advantages of a meter-in circuit?
7. What are the disadvantages of a meter-in circuit?
8. What are the advantages of a meter-out circuit?
9. What are the disadvantages of a meter-out circuit?
10. What are the advantages of a by-pass or bleed-off circuit?
11.What are the disadvantages of a by-pass or bleed-off circuit?
12. What is a modular valve and what are its benefits?
13. What is a hydraulic fuse?
14.What is the need for temperature compensation in a flow-control valve?
15. What is the difference between a hydraulic fuse and a pressure-relief valve?

Answers
Fill in the Blanks

1.Constant
2.Pressure drop
3.Pressure compensated
4.Into
5.Out

State True or False

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