Learning Objectives

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

- Appreciate the history of proportional control valves.
- Explain the operation of proportional solenoids.
- Describe various design considerations for proportional control valves.
- Describe various proportional direction and proportional pressure control valves.
- Explain the working of two-stage proportional valves.
- Compare a proportional valve and a servo valve.
- Describe the various applications of proportional valves.

1.1 Introduction

Proportional control valves can be operated easily using a solenoid. Solenoid controls have a digital control system: A valve is opened when the solenoid is energized and is closed when it is de-energized or vice versa. They are very quick in their operation and thus give rise to pressure and flow surges in the fluid power control units. If the control valves can be gradually opened or closed as a manually operated house tap, it results in a gradual transition between a fully opened and a fully closed position. These valves are operated by the application of electronics rather than just electrical switching. The advantage of these valves is that they give greater flexibility in the system design and operation. They also decrease fluid power circuit complexity especially for processes requiring multiple speed or force outputs.

1.2. History of Proportional Control Valves

A proportional valve is a valve that produces an output (direction, pressure, flow) that is proportional to an electronic control input. The output force exerted by the armature of a DC solenoid depends on the current flowing through it. This can be utilized in the design of a proportional DC solenoid in which the force exerted by the armature is proportional to the current flowing and independent of the armature over the working range of the solenoid.

In earlier days, there were only two types of electrically operated valves – solenoid valves and servo valves – that had a huge performance gap. Solenoid valves were either actuated or unactuated (i.e., fully open or fully closed) and had no intermediate position; thus, solenoid valves facilitated very little control. These were simply ON–OFF valves and their maximum frequency was 5 Hz or less. Servo valves, in contrast, were continuously controlled, high-frequency response devices that received commands through their electronic control systems that provided a high degree of control over position, velocity, acceleration, etc. They had high accuracy. They could accept and accurately respond to command signals at frequencies exceeding 100 Hz. The continuous feedback from electronic transducers ensured high accuracy. Between these extremes, there was nothing just a huge gap in performance, control capability and cost.
With the advent of proportional control valves, the situation changed. The design of their actuating device allowed the spool to be stopped at intermediate positions rather than only at the ends of the solenoid stroke. The associated electronics controlled the spool positions and offered a high degree of flexibility compared with the operation of the solenoid valve. The new valves had a maximum frequency response of 10 Hz that was better than solenoids but less when compared to servo valves. Thus, these valves were an intermediate between solenoids and servo valves. There was no feedback from the circuit, so the controllability and control accuracy were poor as compared to the parameters of the servo valves but greatly exceeded anything that could be achieved by solenoid valves. The final result was a valve that stood comfortably between a solenoid and a servo valve in performance, cost and complexity.

With the evolution of performance and application of proportional control valves, the efficiency increased. First a spool position feedback loop was added; next came improvements in the design of spools and the electronics; then came the external feedback systems, high-frequency responses, better performance in accuracy, hysteresis, dead band, threshold and other parameters. In short, the proportional valves began to look more and more like servo valves in capability. This was accompanied by an increase in cost and thus blurred the distinction between servo valves and proportional valves. As a result, performance and control are no longer distinguishing criteria. Rather physical features such as design and manufacturing processes are the defining characteristics. For instance, proportional valves are operated by proportional solenoids whereas servo valves are operated by torque motors. The spools in proportional valves are almost entirely machine produced, while the spools in servo valves require a great deal of manual lapping and finishing. The clearances and tolerances in servo valves are much tighter than in proportional valves. These differences mean that servo valves are still more expensive than proportional valves and also that they outperform proportional valves in terms of accuracy, hysteresis, leakage, etc. It is fair to say that a proportional valve can be linked to a low-cost, low-performance-range servo valve. These valves are divided into three types – directional, pressure and flow controls.

1.3 Proportional Solenoids

A directional control valve is the most common electrohydraulic proportional control valves (EHPV). The general aspects of its operation can also be applied to pressure and throttle valves. Though they look like solenoid valves, there are significant differences between the two. Both types have solenoids, and both have a valve body with a movable spool port and other components. We will look at the differences beginning with solenoids.

1.3.1 Proportional Solenoids

All standard solenoids have no intermediate positions; rather they are always at one end or the other of the solenoid stroke. The magnetic flux attempts to drive the plunger to its fully closed position when the coil is energized. The force developed by the solenoid is a function of square of the solenoid current and inverse function of square of the air gap. The result is that the force increases as the air gap closes as well as when the current increases. A typical force–displacement curve is shown in Fig. 1.1.
The solenoid force is at its minimum when the plunger is at the maximum position. By incrementally increasing the current in a particular solenoid, we can generate a family of curves (Fig. 1.2).

If there is a spring, then it is an additional design requirement. Let us assume a spring whose force is a linear function of its compressed distance $(F = kx)$. If we plot the spring force versus air gap dimension, we get a graph shown in Fig. 1.3. From this plot, we can see that for a valve to operate at all, we must provide a current that ensures sufficient force to overcome the spring force throughout the plunger stroke. If the solenoid force ever drops below the spring force, the solenoid stops. The dependence of spring force on its compressed distance is shown in Fig. 1.3.
Figure 1.3 Solenoid force versus stroke with spring force.

A proportional solenoid differs from a standard solenoid in the design of the area near the end of the plunger stroke. In the design of a basic solenoid, the air gap is closed at a uniform rate as the plunger moves in. Because the square of the air gap dimension appears in the denominator of the force equation, the solenoid force increases exponentially as the gap closes. The design of a proportional solenoid eliminates the effect of diminishing air gap dimension at the end of the plunger stroke by utilizing constructional features that maintain a constant effective air gap or by using a magnetic impervious material to which the solenoid appears as a constant air gap. The results for any given current are a force curve similar to that in Fig. 1.4.

Figure 1.4 Proportional solenoid force versus stroke at constant current.

The flat portion of the curve occurs in the constant air-gap portion of the stroke. Figure 1.5 shows the variation of force versus stroke for various values of current.
Figure 1.5 Proportional solenoid force versus stroke for varying current.

Use of a carefully designed, calibrated spring to oppose the solenoid force results in a solenoid force versus spring force arrangement as shown in Fig. 1.6. The concept is to have the spring force curve intersect the solenoid force lines in the flat portion of the solenoid force lines. Thus, the solenoid plunger position (and, subsequently, the valve spool position) can be controlled by the current applied. A higher current produces a higher solenoid force that compresses the spring until the spring force balances the solenoid force. When this force is balanced, the plunger stops. The position at which the plunger (spool) stops determines the size of the flow path through the valve. This along with the pressure differential across the valve determines the fluid flow rate through the valve. The functional result is that flow direction and flow rate can be controlled with a single valve. This portion of the solenoid stroke is known as the control zone. The length of this zone is only about 0.06–0.08 inch (0.15–0.20 cm). The total plunger stroke is also small, usually about 0.120 inch (0.3 cm).

Figure 1.6 Proportional solenoid force versus stroke with spring force overlaid.

1.4 Design Considerations of Proportional Control Valves

The output force exerted by the armature of a DC solenoid depends on the current flowing through it (Fig. 1.7). This fundamental concept can be used in the design of a proportional DC
solenoid in which the force exerted by the armature is proportional to the current flowing through it and independent of the armature movement over the working range of the solenoid. A typical characteristic is shown in Fig. 1.7.

![Figure 1.7 Proportional solenoid characteristics.](image)

1.4.1 Force Position Control

The electrical control to the proportional valve normally uses a variable current rather than a variable voltage. If a voltage control system is adopted, any variation in coil resistance caused by temperature change will result in a change in current. This problem is eliminated by using a current control system. It is possible to control a force electrically. By applying the force to a compression spring, its deflection can be controlled. If the spool in a valve (as in Fig. 1.8) is acted on by a spring at one end and a proportional solenoid on the other, the orifice size can be varied along with the control current.

![Figure 1.8 Diagrammatic section of a proportional control valve.](image)

The flow from the valve is proportional to the current flowing through the solenoid. Because of the difficulties in manufacturing a zero lap spool, overlapped spools are used in proportional...
spool valves. This means that the spool has to move a distance equal to the overlap before any flow occurs through the valve, giving rise to a dead zone as shown in Fig. 1.9.

![Dead zone diagram](image)

**Figure 1.9** Flow current characteristics of a proportional control valve.

Notched spools gives better control of the flow rate because the orifice is progressively opened. Notch shape determines the amount of maximum flow. A diagrammatic representation of the notched spool valve is shown in Fig. 1.10(a) together with an electrical control diagram in Fig. 1.10(b). A proportional directional control valve with a double solenoid and spring centered is similar to the notched spool valve except that it has a solenoid at each end of the spool and spring-centering device. The symbol for such a valve is shown in Fig. 1.11 as either a five-position [Fig. 1.11(a)] or a three-position valve [Fig. 1.11(b)]; both symbols are in common use. The extremes on a five position valve represent fully operated conditions.

![Notched spool valve diagram](image)

(a)

![Electrical control diagram](image)

(b)
1.4.2 Spool Positional Control

In order to increase the accuracy and extend the range of applications of proportional control valves, a linear transducer may be fitted to measure the spool position. The output from the transducer is a voltage that is proportional to the spool displacement and it continuously varies through the total spool movement. The actual position of the spool is fed back via the transducer to the electrical control system and then compared with the required position, the control current being adjusted accordingly. Such a system is shown in Fig. 1.12.

In such valves, spool opening and the flow rate is controlled in both forward and return direction. The transducer used for the position feedback of the spool does not monitor the quantity of fluid flowing through the valve. So it is an open-loop control system. If additional accuracy is required, it is possible to use a transducer to measure the system output and feed this back to the control circuit. In the speed control circuit of a hydraulic motor shown in Fig. 1.13, a tachogenerator or a similar device is used to measure the speed, in which case the effect of the “dead zone” must be considered. This will be more critical in the case of position control rather than speed control.
1.4.3 Proportional Pressure Control

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 DC solenoid in the case of proportional valves; the force set up by the solenoid is controlled by being dependent on the current flowing through it.

1.4.3.1 Single-Stage Proportional Relief Valves

Direct-acting proportional relief valves are shown in Fig. 1.14. The proportional solenoid exerts a force on the poppet keeping the valve closed, until the hydraulic pressure at port P overcomes this force and opens the valve. In the design of the relief valve, the proportional solenoid acts directly on the valve poppet.

Figure 1.14 Direct-acting proportional relief valve.

1.4.3.2 Proportional Pressure-Reducing Valves

This operates in a manner similar to a conventional pressure-regulating valve, with the control spring being replaced by a proportional solenoid. When this solenoid is not energized, the proportional valve is closed unlike the conventional pressure reducing, which is normally open.
The output pressure of the valve, shown diagrammatically in Fig. 1.15 is proportional to the current flowing through the solenoid.

**Figure 1.15** Proportional pressure-reducing valve.

When the solenoid is energized, it moves the spool to the right, the control orifice A opens and allows fluid to flow to the output port X. As the aperture of orifice A increases, the aperture of orifice B reduces; the pressure at the control output X is dependent upon the openings of control orifices A and B. This is shown in Fig. 1.16.

**Figure 1.16** Principle of a pressure-reducing valve.

Let the supply pressure be \( p_1 \). The pressure drops across the control orifices A and B are \( \Delta p_A \) and \( \Delta p_B \), respectively, and the output pressure is \( p_x \). Then

\[
p = \Delta p_A + \Delta p_B \quad \text{and} \quad p_x = \Delta p_B
\]

If the control orifice B is fully closed, \( p_x \) equals the supply pressure \( p_1 \). The output pressure is applied to the right-hand end of the spool and if this is greater than the equivalent pressure exerted by the proportional solenoid, the spool moves to the left. This increases the opening of orifice B and reduces that of orifice A, reducing the output \( p_x a = F \). The output pressure is proportional to the current flowing in the proportional solenoid. There is always a flow to the tank.
from this type of valve if the output pressure $p_x$ is less than the supply pressure $p_1$. It is essential that there is no back pressure in the tank line if the valve is to function properly.

1.4.4 Two-Stage Proportional Valves

The valves already discussed have a maximum flow capacity of 5 LPM; to obtain higher flow rates in valves, two-stage versions are available. A single-stage proportional pressure control valve is used to pilot the main valve. These operate in a manner similar to conventional two-stage valves.

1.4.5 Two-Stage Proportional Directional Control Valves

The pressure output from a proportional pressure-reducing valve is directed to move the spool of the main valve against a control spring. Energizing solenoid 1 causes pressure to be applied to pilot port X and hence to current in solenoid 1. As the main spool lands are notched, a movement to the right progressively opens the flow paths from P to B and A to T. De-energizing solenoid 1 de-pressurizes spring chamber C and the control spring centralizes the spool.
Similarly, solenoid 2 controls the flow paths P to A and B to T. The symbol for such a valve is shown in Fig. 1.1. The operating time of valve from mid-position to one extreme is the minimum of 40–60 ms. The operating time can be dependent on the rate of increase or decrease of the control current. The output flow from the valve depends on the pressure drop across the valve and the control current in the solenoid.

1.4.6 Two-Stage Proportional Relief Valve

This is similar to a conventional two-stage relief valve but with a proportional pilot relief valve controlling the main spool. The system pressure is applied via the control orifice to the pilot stage. When the pressure exceeds the force generated by the proportional solenoid, the pilot stage opens. This causes a flow across the control orifice with a resultant pressure drop. The pressures on the main spool are out of balance and so the spool lifts thereby relieving the fluid. A small conventional direct-acting relief valve can be incorporated into the design as an overload pilot to protect the system from any possible malfunction of the proportional valve or electrical control circuit.

1.4.7 Proportional Flow Control

Small flows may be controlled by using one pair of ports of a four-port, two-position proportional directional control valve and higher flows may be controlled if two ports are coupled together. These methods of connections are illustrated in Fig. 1.18. The flow through the valve is proportional to the current flowing in the solenoid and to the pressure drop across the valve. The flow characteristic is not precisely linear because the flow opening is not exactly proportional to the applied current. By carefully designing notches in the spool, it is possible to obtain a variable sharp-edged orifice; this reduces the effect on the flow of variations in the fluid viscosity.

**Figure 1.17** Two-stage directional control.

**Figure 1.18** Connection of a four-port proportional direction control valve single-path flow and double-path flow capacity.
1.4.7.1 Pressure-Compensated Proportional Flow Control

If a constant pressure drop is maintained across the flow control valve orifice, the flow through the valve is independent of any upstream or downstream pressure variations. This is achieved by using a pressure-compensating cartridge, as employed in conventional flow control valves. The compensator can be considered as a remotely operated pressure control valve, which continuously varies its orifice opening to maintain a fixed pressure drop over the flow control orifice. The pressure difference between the two pilots on the compensator valve, that is the pressure drop across the flow control orifice, is equivalent to the fixed force set by the control spring of the compensator valve.

1.4.7.2 Electrical Control of Proportional Valves

A block schematic for a proportional amplifier together with a current–time graph showing ramping is shown in Fig. 1.19. The ramp-up control determines the rate at which the control signal increases and hence the acceleration of the actuator. The ramp-down control corresponds to the deceleration [Fig. 1.19(b)]. The input level control determines the maximum value of the control signal. A low-level dither signal is superimposed onto the control signal. It is an AC level at 100 Hz. Its function is to keep the spool oscillating to overcome the effects of static friction [Fig. 1.19(a)].

The feedback signal is either from the output current or from the spool position in the valve. The feedback does not indicate the actuator conditions and it is still an open-loop system. In order to close the loop, the transducer has to measure of the actuator (either position or speed), feed the signal back and compare it with the output of the actuator, feed the signal back and compare with the desired values. The difference between these values is converted into a new input signal. A block diagram of a closed loop control system is shown in Fig. 1.20. Although closed-loop control can be achieved using proportional valves, it would not be as accurate or have as fast response as an electrohydraulic servo-valve-based system.
Figure 1.19 Proportional amplifier: (a) Block schematic diagram; (b) current–time characteristics.
1.5.1 Response Speed and Dynamic Characteristics

A short travel speed of minimum mass and consequently low inertia is used in servo valves, giving high response speeds and making servo valves suitable for dynamic applications. On the other hand, a proportional valve spool has a longer travel and is biased to one position by a control spring – the spool and spring combination have a much higher inertia than an equivalent servo valve. The application of a dither signal reduces the effects of spool “stiction” and inertia in both proportional and servo valves. Its value is usually adjustable and is set to give maximum response speed without any flow or pressure fluctuations being set up by the dither current.

1.5.2 Hysteresis Effect

Spool position in a servo valve is controlled by a nozzle and flapper or jet pipe system with a feedback link correction for the spool position. A proportional valve relies on the force exerted by a DC coil acting against a spring to position the spool. There is a considerable difference in the valve output depending on whether the current is increasing or decreasing. Proportional valves have higher hysteresis than servo valves.

1.5.3 Null Position

Because of the underlap spool, a very slight change in the control current varies the output of a servo valve about the zero flow position. In a proportional valve, there is no output until the control current exceeds 200 mA that is required to overcome the spring preload and spool overlap. Wherever fast response and accurate control are required, servo valves are best suited, whereas proportional valves are economic. The differences between proportional and hydraulic servo valves are given in Table 1.1. However, proportional valves are much more dirt-tolerant and provide economical and satisfactory alternative for many applications.
### Table 1.1 Comparison of proportional and servo valves

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Proportional Hydraulic Valve</th>
<th>Electrohydraulic Servo Valve</th>
</tr>
</thead>
<tbody>
<tr>
<td>Valve lap</td>
<td>Overlap spool, causing a “dead zone” on either side of the null position.</td>
<td>Zero or underlap valve spool. No dead zone.</td>
</tr>
<tr>
<td>Response time for the valve spool to move fully over</td>
<td>40–60 ms</td>
<td>5–10 ms</td>
</tr>
<tr>
<td>Maximum operating frequency</td>
<td>Approx. 10 Hz</td>
<td>Approx. 100 Hz</td>
</tr>
<tr>
<td>Hysteresis</td>
<td>Without armature feedback approx. 5%</td>
<td>Approx. 0.1%</td>
</tr>
<tr>
<td></td>
<td>With armature position feedback approx. 1%</td>
<td>Standalone as a control unit for position control.</td>
</tr>
</tbody>
</table>