Lecture 27

SERVO VALVES

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

- Define servo valve.
- Compare servo valves with proportional valves.
- Appreciate the history of servo valves.
- Describe the working of a servo torque motor.
- Describe the working of single-stage spool-type servo valves.
- Describe the working of jet-type servo valves.
- Analyze the valve performance.
- Define dead band and hysteresis.
- Analyze mathematically the simple servo systems.

1.1 Introduction
Servo valves were developed to facilitate the adjustment of fluid flow based on changes in the load motion. Simply put, it is a programmable orifice. In machine motion control, servo systems involve continuous monitoring, feedback and correction. Also they are used to improve efficiency, accuracy and repeatability. The most common applications of servo valves are in aerospace vehicles, particularly in primary flight controls. In aircraft, control surfaces such as ailerons, elevators and rudders are positioned by servo units. In space vehicles, control during launch is provided by movable thrust nozzles that are positioned by servo units. Even the drill bits for angle drilling of oil wells are servo controlled.

A servomechanism is defined as an automatic device for controlling a large amount of power by means of a very small amount of power and automatically correcting the performance of a mechanism. The automatic and continuous correction requires return of information from the mechanism—feedback, in other words. Therefore, a servo valve is operated without feedback and it is not a true servomechanism. In Chapter 17 we have studied about proportional valves and Table 1.1 gives comparison between servo valve and electrohydraulic proportional control valves (EHPV).

Table 1.1 Comparison of servo valves and electrohydraulic proportional valves

<table>
<thead>
<tr>
<th>Feature</th>
<th>Servo Valve</th>
<th>EHPV</th>
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<tbody>
<tr>
<td>Electrical operator</td>
<td>Torque motor</td>
<td>Proportional solenoid</td>
</tr>
<tr>
<td>Manufacturing precision</td>
<td>Extremely high</td>
<td>Moderately high</td>
</tr>
<tr>
<td>Feedback circuitry</td>
<td>Main system as well as valve</td>
<td>Valve (depending on type), main system (seldom)</td>
</tr>
<tr>
<td>Cost (compared with a solenoid valve)</td>
<td>Very expensive</td>
<td>Moderately expensive</td>
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</tbody>
</table>

1.2 History of Electrohydraulic Servomechanisms
The earliest recognized servomechanism is the water clock invented around 250 BC (Fig. 1.1) by the Alexandrian inventor Ktesbios. In this device, time was recorded by the level of water in a graduated vessel. Water flows into the vessel at a controlled and constant rate from a water reservoir above it. The control of the flow rate from the reservoir involves a mechanism. Velocity flow rate from the outlet of a reservoir is determined by the equation

\[ v = \sqrt{2gh} \]
where \( V \) is the velocity, \( g \) is the acceleration due to gravity and \( h \) is the height of water above the outlet. The volume flow rate through the outlet depends upon the size of the outlet and the fluid velocity. Thus,

\[
Q = \text{Velocity} \times \text{Area}
\]

From these equations, we can see that as the water level in Ktesbios’s reservoir goes down, the flow rate from the reservoir decreases. Ktesbios’s solution to the problem was to use a second reservoir mounted above the first. He used a float to modulate an orifice through which water was fed into the primary reservoir. This kept the water level constant, resulting in hours of constant length.

Numerous servomechanisms were invented during Industrial Revolution in the mid-1700s and afterward. Many were associated with steam boiler technology where they were used to control water level, water and steam flow, steam pressure and the speed and position of steam-operated mechanisms.

**Figure 1.1** Water clock by Ktesbios.

**1.3 Electrohydraulic Servomechanism Concepts**

Figure 1.2 represents a typical fluid power system that uses a proportional valve to control the speed of hydraulic motor. The EHPV is set to provide the necessary flow to drive the motor at the required speed. As long as there are no disturbances, the speed remains constant. If there is any change in the operating parameters such as load, fluid temperature, viscosity and wear then the motor speed is likely to change. There is nothing designed in the system to detect the change and present the information to the valve controller that can automatically correct the change and return the speed to the required level. Speed correction is the responsibility of the operator who must make the required control adjustments.

**Figure 1.2** Proportional valve block diagram.

Although this type of circuit is perfectly satisfactory for a very large number of applications, some require automatic and continuous corrections. These circuits require servomechanisms. These mechanisms can be simply referred to as servo valves.

Figure 1.3 shows the circuit that has the same purpose as that of Fig. 1.2, but in this circuit the operator has been relieved from the responsibility of speed corrections. Instead, a tachometer generator has been installed that senses the load speed. This information is automatically and continuously fed back to the control electronics (usually a printed circuit board) where it is compared with the operator command signal input.
Servo control provides automatic and continuous corrections for any changes in motor rpm.

If any difference is found between these signals, the electronic circuitry automatically generates a correction signal proportional to the difference. This signal repositions the valve to correct the flow rate as required. This “sense and correct” function is continuous, so any and every change in load speed is automatically corrected. The system required to perform this function includes three major segments: the servo valve, the command electronics and the feedback transducer.

1.4 Servo Valves
Servo valves can be used in virtually any aspect of fluid power system operations, including the following:

1. Positioning of cylinders and rotary actuators.
2. Speed of cylinders and motors.
4. Acceleration and deceleration.
5. System pressure.
6. Flow rate.

The most common applications are for cylinder positioning and motor speed control. The valves for these functions incorporate both direction and flow control in sliding spool arrangement that is positioned by a torque motor.

1.4.1 Torque Motor
A torque motor is illustrated in Fig. 1.4. It is a simple electromagnetic device consisting of one or two permanent magnets, two pole pieces, a ferromagnetic armature and two coils. The permanent magnet polarizes the upper and lower pole pieces, so that they present equal and opposite magnetic fields. Torque motors are very low-power devices operated on low-voltage DC power.

The armature is mounted at its mid-point so that it is free to rotate through a very limited are either clockwise or counter-clockwise. The ends of the armature are extended into the gaps between the pole pieces. The magnetic field holds the armature in a neutral position. The two coils surround the arms of the armature to form two small electromagnets. When a current is passed through the coils, a
magnetic field is generated. The polarity of the field depends on the direction of the current flow. In Fig. 1.5, the current flow causes the left end to become the South Pole and right end to become the North Pole, resulting in counter-clockwise rotation of the armature.

The two coils of a torque motor may be connected in three different configurations: parallel, series and the so-called push–pull arrangement. These options are illustrated in Fig. 1.6. The push–pull arrangement is the most common. In this arrangement, both the leads B and D are connected to ground through the control circuit amplifier. Leads A and C are connected to separate output terminals on the command amplifier. When the voltage input to both coils equals, the armature is centered. Increasing the voltage input to one coil while simultaneously reducing the input to the other coil by the same amount causes the armature to rotate. The voltage can be varied from zero to its maximum value for each coil, but the polarity is never reversed. This means that the position of the armature is determined by the differential torque. When the voltage is the same for both coils, the torque is equal and the armature is centered. Any change in the voltage to either coil results in the rotation of the armature.

This push–pull method of connecting the coil is preferred for at least three reasons:

1. Any change in the current as a result of voltage fluctuations, temperature changes or other causes is canceled by the equal and opposite effects on the coil.
2. There is more stability in armature positioning because of the opposing torque.
3. The power consumption is lower than in the other two configurations for the other two circuits.

The input to this arrangement is expressed as a differential current \( \Delta I \), where \( \Delta I \) is the difference between the coil currents. The control power is calculated from the equation

\[
P = (\Delta I)^2 R
\]

where \( P \) represents the control power, \( \Delta I \) represents the differential current \( I_{AD} - I_{BC} \) and \( R \) represents the resistance of one coil. In parallel connection, the direction of rotation depends on the polarity of the input signal. The coils in the parallel circuit assist each other rather than oppose (as in a push–pull
circuit). That is, both of them attempt to move either clockwise or counter-clockwise. Reversing the polarity of coils reverses the polarity of the rotation. The control power is then found from

\[ P = (I_p)^2 \left( \frac{R}{2} \right) \]

where \( I_p \) is the total current through the circuit and \( R \) is the resistance of the coil. In a series circuit, the coils assist, rather than oppose, the armature rotation. As with the parallel circuit, a polarity change is required to change the direction of rotation. The control power of a series circuit is given by

\[ P = (I_s)^2 \frac{2R}{2} \]

where \( I_s \) is the current in the series circuit and \( R \) is the resistance of each coil. Notice that the series and parallel circuits have the same maximum power requirement, which is exactly twice that of the push–pull circuit.

It is interesting to note that these two low-power torque motors can control a two- or three-stage valve that may be flowing 300–400 LPM or more at 140–300 bar. Taking in-between values, we see that the power output of the valve approaches 90000 W. If we define the power gain of the valve as the output power divided by the control power, we have

\[ \text{Power gain} = \frac{90000 \text{ W}}{1.6 \text{ W}} = 5.625 \times 10^4 \]

1.4.2 Valve Spools

At first glance, the hardware of a servo valve looks very similar to that of any spool-type directional control valve, that is, a sliding spool that operates in the bore of the valve body to open and close the flow path between ports. The actual differences are found more in the manufacturing process and clearance specifications than in the basic design.

The servo valve spool and the bore (in which spool moves) are very high precision components. Typically, spool and bore straightness and diametrical tolerances are held to ±1 \( \mu \)m. The radial clearance between the spool and the bore is typically 3–5 \( \mu \)m. In most valves, the radial expansion that results from holding the spool in hand for a few minutes would prevent its insertion into the bore. To achieve this precision, a great deal of hand finishing is involved in the manufacturing process. The spool and the body are quite often a matched set, and the parts are not interchangeable.

Special spool surface finishes are often employed. Nitriding is often used to provide extra surface hardness and a glass-like smooth finish. This reduces the friction and improves the wear characteristics. Servo valves may be either three- or four-way. The spool may have two, three or four lands, depending on the function and on the manufacturer’s preference. It has been shown that four-land spools can have slightly large clearances without incurring unacceptable leakage. This means that they have improved wear characteristics and are somewhat more tolerant of contaminants in the fluid. The two outer lands also assist in keeping the spool precisely centered.

As with most spool-type valves, circumferential grooves are machined into the spool lands. The purpose of the grooves is to reduce the side loading by equalizing the pressure around the spool and holding it centered into the bore. A spool with three grooves can have as little as 6% of the side force as found in the ungrooved spools.

Spool “lap” defines the width of the lands relative to the width of the ports in the valve bore. There are three possible lap configurations: Overlap, underlap and line-to-line. These are shown in Fig. 1.7.
Figure 1.7 (a) Zero overlap. (b) Underlapped. (c) Overlapped

By far, the most common condition is the line-to-line (or zero-overlap) spool. Here, the bandwidth exactly matches the port width. Thus, when the spool is centered, there is no flow. Any movement of the spool, regardless of how little, results in flow through the valve. This valve is suitable for closed-loop position, speed, and force control applications because of its precise metering characteristics about the null (neutral) position. Unfortunately, even a small amount of wear on either the land or port edge results in leakage in the null position.

Overlapped spools have lands that are 0.5–5% wider than the ports. These spools have the advantage of providing lower leakage flow in the null position than the line-to-line configuration. However, the overlap means that the precision achievable about the null position is compromised because of the relatively large dead band. For instance, when used as a position controller, a cylinder that is being extended stops at a different position when being retracted even with the same command input. An overlapped valve can be satisfactorily employed as a speed controller as long as it is operated well away from its null position.

In many servo valve control circuits, dither is used to reduce the effects of static friction (termed stiction). Dither is a very low amplitude command signal superimposed over the normal command signal that results in a continuous, very short stroke, lateral oscillation of the spool. In such systems, a slight overlap may be used to prevent unexpected leakage in the null position.

An underlapped spool has lands that are 0.5–1.5% narrower than the ports. This design is often referred to as “open center” although there really are no open-center servo valves. The underlap is far too small to be a true open center. This type of valve provides very rapid response to commands about the null position, but it has the disadvantage of having non-linear flow characteristics near null. This compromises control to some extent.

1.4.3 Valve Configurations
Servo valves may be single-stage (also called direct-acting), two-stage or three-stage, depending primarily on the flow requirements of the system. Single-stage valves may be used when the flow requirements are low (usually less than 20 LPM, depending on the valve design). These valves commonly utilize a sliding spool mechanically connected to the torque motor armature. The flow capacity is dictated by the low force available from the torque motor and the limited stroke of the spool.

1.4.4 Single-Stage Spool-Type Servo Valve
Figure 1.8 shows a single-stage servo valve. The mechanical connection between the torque motor armature and the spool is a stiff wire. When there is no command input to the torque motor, the armature is in the neutral (nullled) position, which, in turn, causes the spool to be in the nullled position, and there is no flow through the valve. A clockwise deflection of the armature pushes the spool to the left, opening up flow path from P to B and A to T. A counter-clockwise deflection opens P to A and B to T.
For higher flow rates, two- or even three-stage valves must be used. In these valves, second and third stages are always sliding spools that are pilot operated from the previous stages. The first stage may use the sliding spool, but there are other designs also.

1.5.5 Two-Stage Servo Valve
Figure 1.9 shows a two-stage servo valve, pilot operated. This valve can be used to control the direction and speed of a hydraulic motor. The pilot spool is positioned by the torque motor. The pilot stage is sleeve. This sleeve is associated with the internal feedback mechanism of the valve. There are two pressure sources: $p_c$ is the control pressure for piloting the main spool and $p_s$ is the supply pressure for operating the system.

In neutral position, as shown in Fig. 1.9, the middle spool land blocks off the pilot part to the left (large) end of the main (second-stage) spool. The pressure on the large end of the spool is half the control pressure. The pressure at the small end of the spool is always equal to the control pressure. In order to be balanced, the large end must have twice the area of the small end. The result is that the main spool is static. Because no input has been made to the pilot spool, the main spool is static. Because no input has been made to the pilot spool, the main spool is in neutral position, and so there is no flow to the hydraulic motor.
In neutral, large pilot end is blocked at pilot valve in the static condition. This pressure is $\frac{1}{2}p_c$.

Control pressure is present here at small end of main spool.

1. Large spool end area is twice the area of opposite end which is subject to control pressure at all times.

2. A counter-clockwise movement of torque motor armature pushes the pilot spool to the left. This opens up the port (through the pilot-stage sleeve) to the large end of the main spool. The pressure in that end increases, causing a force imbalance that shifts the main spool to the right and opens flow paths P to A and B to T.

This movement of the main spool also initiates the internal feedback action. As the main spool moves to the right, it pushes the feedback linkage. This linkage transmits the main spool movement to the pilot-stage sleeve, causing the sleeve to slide to the left. When the sleeve port moves to the point where it is blocked by the pilot spool land, flow to the large end of the main spool stops. Consequently, the main spool stops. The exact stopping point is predetermined and is controlled by the deflection of the torque motor. Because the opening created by the movement of the main spool constitutes a control orifice, both the direction and the speed of rotation of the hydraulic motor are set.

De-energizing the torque motor returns the pilot spool to its neutral position, which opens up a flow path through the pilot-stage sleeve and ports the large end of the main spool to the tank. The resulting force imbalance shifts the spool to the left. This allows the pilot-stage spring to push the spool back to the right. When the sleeve port realigns with the pilot spool land, all pilot flow stops, the main spool stops (in its neutral position), and the hydraulic motor stops. A similar but opposite sequence takes place when the torque motor armature is deflected clockwise.

1.5.6 Double-Flapper Nozzle Pilot Stage

Figure 1.10 shows a double-flapper nozzle pilot stage. The nozzle is a physical part of the torque motor armature and extends into a fluid cavity inside the valve. Control pressure for piloting the main
The spool enters the pilot section through the ports on opposite sides of the flapper. The pressure is then channeled to nozzles located in the fluid cavity into which the flapper extends. These channels are also connected to the pilot chambers at the ends of the main spool.

![Two-stage spool-type servo valve](image)

**Figure 1.10** Two-stage spool-type servo valve.

When the torque motor is not energized, the armature is centered, which positions the flapper exactly in the middle of the cavity. Because the distances between the nozzle lands and the flapper are very small, this, in effect, forms small orifices that restrict the flow and generate a pilot pressure. With the flapper exactly centered, both orifices are of the same size, resulting in the same pressure in both pilot chambers. This causes a force balance on the main spool and holds it in its neutral position.

If the armature is deflected counter-clockwise, the flapper moves toward the right-hand nozzle. This movement reduces the orifice associated with that nozzle while increasing the orifice for the opposite nozzle. This increases the pressure in the right-hand pilot chamber and decreases the opposite pressure. The result is a force imbalance that shifts the main spool to the left and opens the related flow paths.

The internal feedback mechanism in the flapper-nozzle valve is flexible metal rod (usually termed a spring) that is attached to the end of the flapper and inserted into a ball joint in the main spool. As the main spool shifts to the left, the feedback spring exerts a force on the flapper (and on the armature) that tends to return it to its neutral position. The force is proportional to the distance moved by the spool. So as the spool shifts to the left, the restoring force in the flapper increases. When the force is sufficient to center the flapper, the control orifices are again equal, as are the pressures in the pilot chambers, and so the main spool stops. It remains in that position until a subsequent command signal causes the armature to move again.

### 1.5.7 Jet Pipe Servo Valve

Figure 1.11 shows another pilot stage – jet pipe stage. In this device, the jet nozzle is attached to and moves with the torque motor armature. The pilot fluid is directed by the jet nozzle into two receiver ports that connect to the pilot chambers at the ends of the main spool. In neutral position, the fluid is evenly divided, which results in equal pressures in the pilot chambers. As a result, the main spool remains stationary. A deflection of the torque motor directs the jet more directly into one of the receiver ports, causing the force imbalance and main spool movement. Internal feedback is provided by a feedback spring in a manner similar to that in the flapper nozzle system.
A variation of the jet pipe concept is the deflector jet valves shown in Fig. 1.12. In this valve, the jet pipe concept does not change. Rather, deflectors are moved into the fluid jet, which direct it into the appropriate receiver ports.

The standard symbols for servo valves are shown in Fig. 1.13. The symbols do not hint the number of stages or type of pilot stage used. They show only function. The directional flow control valves are always three-position, infinitely positionable units and usually have closed centers. The actuator symbols represent the summing junction of the electronic amplifier, which receives two input signals (reference and feedback) and produces a single output signal.
1.5.8 Pressure Flow Characteristics

Servo valves are generally considered to be “high-pressure” devices. Usually the pressure rated is 200 bar, although most are capable of operating higher than 300 bar. Servo valves are typically flow rated at 60–70 bar differential; that is, the flow rate stated for the valve is the flow that occurs at 60–70 bar pressure drop across a fully opened valve. There is a very deliberate logic in the choice of 60–70 bar. The majority of servo systems use a working pressure of 200 bar. It can be mathematically shown that the maximum transmission of power from the pump to the cylinder or motor occurs when the pressure drop through the valve is one-third of the system working pressure. This parameter is related to the basic orifice equation, which describes the flow rate through orifice. Thus, the valves are flow rated at the maximum power transfer pressure drop based on the popular 200 bar. It has also been determined that valve control is optimized at the 65 bar pressure drop.

If the pressure drop across the valve is different from the optimum, the flow rate changes proportionately. The non-optimum flow rate can be calculated using the equation

\[ q = q_r \sqrt{\Delta p / \Delta p_r} \]  

where \( q \) is the adjusted flow rate, \( q_r \) is the rated flow, \( \Delta p \) is the actual pressure drop and \( \Delta p_r \) is the rated pressure drop (usually 65 bar).

1.5.9 Valve Performance

The performance of a valve can be described by numerous parameters. They are generally divided into two categories: dynamic response and static response. The dynamic response characteristics are the same as those for proportional valves, frequency response and amplitude ratio. Figure 1.14 is a Bode diagram showing these characteristics (frequency response and amplitude ratio) for a specific valve.

During the normal operation of a valve, the valve is likely to experience the same current input (set point) frequently. Sometimes, this specific current input is approached from a lower current setting. At other times, it is approached from higher settings. All valves have the characteristic that this current set point results in different spool positions, depending on whether it is approached from a lower or higher current.
This characteristic is termed hysteresis. Typical hysteresis curves for servo valves and proportional control valves are shown in Fig. 1.15. Hysteresis is expressed as the percent difference in the rated current required to give the same output when approached from higher and lower inputs. For servo valves, it is typically 1–2%. To overcome the problem of hysteresis, some controllers are designed so that the set point is always approached from the lower side. This requires a deliberate undershoot when approaching from the high side.

A second important valve characteristic is the valve dead band. Dead band occurs only at the null position, as shown in Fig. 1.16. It is defined as the current required to move the spool from the exact centered position to the position where the first flow output is seen. It is usually expressed in milliamps or percent-rated current. Dead band is the result of the spool inertia, overlap, static friction and any other forces that might impede the initial motion.
A similar phenomenon is threshold. Threshold current is the smallest input current required to overcome spool inertia and other impending forces to cause the spool to move. The primary difference between threshold and dead band is that threshold occurs throughout the spool stroke, whereas dead band occurs only at the null position. Threshold contributes to the dead band rating.

Information concerning hysteresis, dead band and other valve performance characteristics is available from the valve manufacturers. These characteristics can be significant in evaluating the suitability of a valve for a specific application. Threshold is the current that must be applied before a response is detected. Good-quality two-stage valves have a threshold less than 0.5% of rated current.

1.4.10 Gain and Feedback

One of the most important concepts of servo system is its ability to constantly monitor its output and automatically make corrections to ensure that the output remains at the commanded level. This is accomplished through the use of some type of feedback from a transducer that monitors the output parameter. Gain is defined as the ratio of output by input. Two gains are defined for servo valves: flow gain and pressure gain. Flow gain is the ratio of flow to input current. Flow gain is determined by measuring control flow versus input current. Flow gain is the slope of the graph of control flow versus input current. Pressure gain is defined as the ratio of pressure and input current. Pressure transducers are mounted on the output ports and pressure difference is measured as a function of input current. The range of input current to produce a load pressure change from −40% to +40% of supply pressure is determined. Pressure gain is difficult to measure.

1.4.10.1 The Control Ratio Equation

![Figure 1.17 A basic feedback – closed loop.](image-url)
One of the most important aspects of system analysis is an understanding of system gain. Figure 1.17 is a block diagram of a generic system with feedback. The command input signal is given a designator R for reference. The gain of the loop leg is designated G and termed the forward loop gain. The gain of the feedback loop is denoted by H and the controlled output is shown as C.

As the system operates, a feedback signal F is continuously generated. Its value is based on the value of C and the feedback gain. Thus,

\[ F = CH \]

This feedback signal is fed to the summing junction (in the op-amp), where it is compared with the reference signal. The result is an error signal E whose value is

\[ E = R - F = R - CH \]  

(1.2)

This results in a change in the controlled variable C, which then becomes

\[ C = EG \]  

(1.3)

and the process repeats itself. Equating Eqs. (1.2) and (1.3), we get

\[ C = (R - CH)G = RG - CGH \]

so that

\[ RG = C + CGH = C(1 + GH) \]

and, eventually,

\[ \frac{C}{R} = \frac{G}{1 + GH} \]  

(1.4)

This equation is known as the control ratio, and closed-loop gain or the closed-loop transfer function of the system. The right-hand side of Eq. (1.4) defines the system gain and is commonly referred to as the closed-loop gain of the system.

**Example 1.1**
A torque motor is connected in a push–pull circuit. Each coil has a resistance of 20 Ω and is rated at 200 mA. Find

(a) The voltage of each coil when the armature is centered.

(b) The maximum value of ΔI.

(c) The maximum control power for the torque motor.

**Solution**

(a) The maximum voltage for the coil is

\[ E = IR = 200 \text{ mA} \times 20 \Omega = 4 \text{ V} \]

The armature is centered when

\[ E = \frac{E_{\text{max}}}{2} = 2 \text{ Volts} \]

(b) The differential current is

\[ I = I_{AD} - I_{BC} \]

The maximum value will occur when the maximum voltage is applied to one coil, so that zero voltage is applied to the other. In this case,

\[ \Delta I = I_{AD} - I_{BC} = \frac{E_{\text{max}}}{R} - 0 = \frac{4 \text{ V}}{20 \Omega} = 200 \text{ mA} \]

(c) The maximum control power is then

\[ P = (\Delta I^2)R = (200 \text{ mA})^2(20 \Omega) = 0.8 \text{ W} \]

**Example 1.2**
A torque motor is connected in a parallel circuit. Each coil has a resistance of 20 Ω and is rated at 200 mA. Find

(a) The voltage of each coil when the armature is centered.
(b) The maximum value of $\Delta L$.
(c) The maximum control power for the torque motor

**Solution:**

(a) The voltage to each coil will remain the same (4 V).

(b) The current through the circuit will increase because of lower resistance. For a parallel circuit made up of two equal resistors, the equivalence resistance is $R/2$; in this case, 10 $\Omega$. The value of $I$ is the total current, which we find from

$$I_e = \frac{E}{R} = \frac{4 V}{10 \Omega} = 0.4 A = 400 mA$$

(c) Control power is given by

$$P = (I_p)^2 \left( \frac{R}{2} \right) = (400 mA)^2 (20 \Omega/2) = 1.6 W$$

**Example 1.3**

A torque motor is connected in a series circuit. Each coil has a resistance of 20 $\Omega$ and is rated at 200 mA. Find

(a) The voltage of each coil when the armature is centered.

(b) The maximum value of $\Delta L$.

(c) The maximum control power for the torque motor

**Solution:**

(a) A torque motor is connected in a series circuit. Therefore, total resistance is $2R$ or 40 $\Omega$. The maximum current will be 200 mA.

(b) A torque motor is connected in a series circuit therefore the maximum voltage is

$$Voltage (E) = IR = (200 mA)(40 \Omega) = 8 V$$

(c) The control power is

$$P = (I)^2 (2R) = (200 mA)^2 (2)(20 \Omega) = 1.6 W$$

**Example 1.4**

A servo valve is flow rated at 56 LPM at 65 bar differential (Fig. 1.5). It is to be operated in a 130 bar system. What will its adjusted flow rate be at the optimum power transfer $\Delta p$?

**Solution:**

For optimum power transfer, the pressure drop across the valve should be

$$\frac{130}{3} = 43.33 \text{ bar}$$

From Eq. (1.1), we have

$$q = q_r \sqrt{\frac{\Delta p}{\Delta p_r}} = 56 \sqrt{\frac{43.3}{65}} = 45 \text{ LPM}$$

**Example 1.5**

Figure 1.18 shows a servo valve which is rated at 65 bar pressure drop across the valve in a 195 bar system.

(a) Find the pressure drop when motor is fully loaded.

(b) Find the pressure drop when motor is not fully loaded.
(c) Find the pressure drop when motor is overloaded. Will there be a flow?
(d) Derive an expression for fluid temperature rise across the valve.

Note that the term “6 bar pressure drop” imply that the drop across one flow path through the valve.

![Diagram of fluid system with pressure drops](image)

**Figure 1.18**

**Solution:**
(a) When motor is at full load: From Fig. 1.18, with the pump compensator set at 195 bar, the pressure is reduced to $195 - 32.5 = 162.5$ bar across the P to A path through the valve. If we assume the motor to be operating at full load, 130 bar is dropped across the motor. The final 32.5 bar is then dropped through the B to T side of the valve.

(b) When motor is not at full load: If the motor is not fully loaded, then the system pressure drops below 195 bar. In this case, the 65 bar drop across the valve still occurs, but the power transfer diminishes.

(c) When motor is overloaded: If the motor is overloaded so that more than 130 bar is required to rotate it, then the 65 bar drop across the valve is not possible. The result is a lower flow rate [Eq.(1.1) applies], so the motor will turn more slowly. The extreme of this condition is a stalled motor. In this case, the full 195 bar will be dropped across the motor. Obviously, there is no flow in this case.

(d) Heat generation and fluid temperature rise: Because the servo valve represents a major pressure drop, it will also generate a considerable amount of heat. Let

$$HGR = \text{Heat generation rate} = W \times C_p \times \Delta T$$

$$P = \text{Power loss across the valve} = \Delta p \times Q$$

The fluid temperature rise across the valve can be found using the equation

$$\Delta T = \frac{HGR}{C_p W} \quad (1.5)$$

where $\Delta T = \text{temperature rise (°C)}$, $C_p = \text{specific heat of the oil} = 1.8$ kJ/kg K, $W$ (kg/s) = weight flow rate = $\gamma Q$, and $\gamma$ = fluid-specific weight.
Example 1.5
For the servo valve of Example 1.5, determine the power loss, heat generation rate and the temperature rise across the valve. The fluid has a specific weight of 8620 N/mm$^3$.

**Solution:** The power loss is
\[ P = \Delta p \times q = 43.3 \times 10^5 \times (45/1000) \times (1/60) \]
\[ = 3.25 \text{ kW} \]
The resultant heat generation rate is found from Eq.(1.5). The increase in the fluid temperature is
\[ \Delta T = \frac{\text{HGR}}{C_p W} \]
The weight flow rate is
\[ W = \gamma Q \]
Oil flow rate in kg/s = \( \gamma Q \) (oil flow rate in m$^3$/s)
\[ = 895 \times 45 \times \frac{10^{-3}}{60} \]
\[ = 0.67125 \text{ kg/s} \]
Therefore,
\[ \Delta T = \frac{\text{HGR}}{C_p W} \]
\[ = \frac{4.97}{1.8 \times 0.889} = 2.689^\circ C \]
This is the temperature rise experienced by every drop of oil that flows through the valve. The rise occurs in the time required for the fluid to flow through the portion of the valve where the pressure drop occurs.
Although the weight flow rate appears in the equation, the temperature rise is actually independent of flow rate. So the temperature rise for any fluid is a function of pressure drop only.

Example 1.7
In the given servo system (Fig. 1.19), suppose the forward loop contains a hydraulic motor. The forward loop gain is 300 rpm/V and the feedback loop consists of a tachogenerator that has a gain of 0.2 V/rpm. What will be the speed of the hydraulic motor for an input of 3.5 V?

![Figure 1.19](image_url)

**Solution:** Given
Reference input, \( R = 3.5 \text{V} \)
Forward loop gain, \( G = 300 \text{ rpm/V} \)
Feedback loop gain, \( H = 0.2 \text{ V/rpm} \)
The equation of closed-loop transfer is given by
\[
\frac{C}{R} = \frac{G}{1 + GH} \quad (1.6)
\]

This gives
\[
C = \left( \frac{G}{1 + GH} \right) R
\]
\[
= \left( \frac{300}{1 + 300 \times 0.2} \right) \quad (3.5)
\]
\[
= 17.2 \text{ rpm}
\]

Normally, if the feedback system is not included, the 3.5 V input should give 1050 rpm. Because there is feedback, the output is 17.6 rpm.

**Example 1.8**

Figure 1.20 shows a closed-loop feedback system with multiple elements in the forward loop. Derive an expression for closed-loop gain.

Solution: Let us have a system that has more elements in the forward loop. The gain can still be found by using the Eq. (1.6). Let us first resolve all the forward loop gains into a single value \( G \). The forward loop gain \( G \) is the product of all the individual element gains between the summing junction and the output. Thus,
\[
G = G_1 G_2 G_3 \quad (1.7)
\]

So
\[
\frac{C}{R} = \frac{G}{1 + GH} = \frac{G_1 G_2 G_3}{1 + G_1 G_2 G_3 H} \quad (1.8)
\]

**Example 1.9**

Determine the speed of the hydraulic motor in the circuit shown in Fig. 1.21.
Solution: Using Eq.(1.7) we can write

\[ G = G_A G_{SV} G_M \]

\[ = (50 \text{ mA/V})(0.1 \text{ gpm/mA})(30 \text{ rpm/gpm}) \]

\[ = 150 \text{ rpm/V} \]

Using the closed-loop transfer equation, we have

\[ \frac{C}{R} = \frac{G}{1 + GH} \]

\[ \Rightarrow C = \left( \frac{G}{1 + GH} \right) R \]

\[ \Rightarrow C = \left( \frac{150}{1 + 150 \times 0.2} \right) (5) = 24.2 \text{ rpm} \]
**Objective-Type Questions**

**Fill in the Blanks**

1. A servomechanism is defined as an automatic device for controlling a large amount of power by means of a very small amount of _______.
2. Servo valves are operated by _______ motors.
3. Typically, spool and bore straightness and diametrical tolerances are held to _______.
4. Overlapped spools have lands that are 0.5% to _______% wider than the ports.
5. In many servo valve control circuits, _______ is used to reduce the effects of static friction (termed stiction).
6. _______ valves may be used where the flow requirements usually less than 20 LPM_______.

**State True or False**

1. Servo valves are less expensive than proportional valves.
2. There is no dead zone for a servo valve.
3. A servo valve uses always zero or underlap spool.
4. The maximum operating frequency of a servo valve is 10 Hz.
5. Servo that uses feedback electronics is more accurate.

**Review Questions**

1. Define a servo valve.
2. How do servo valves differ from proportional control valves?
3. Explain the operation of a torque motor.
4. Define underlap, overlap and line to line in the context of servo valve spools.
5. Define dead band.
6. Define threshold.
7. Define hysteresis.
8. List and define the types of hydraulic amplifiers.
9. At what pressure drop are servo valves usually rated?
10. Define gain.
11. Define flow gain.
12. Define pressure gain.
13. What are the uses of servo valves?
14. What is a torque motor?
15. What is a spool lap?
16. What are the three types of servo valves?
Answers
Fill in the Blanks

1. Power
2. Torque
3. ±1 μm
4. 5%
5. Dither
6. single stage

State True or False

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