TABLE 16.8 PERFORMANCE SPECIFICATIONS OF SERVOVALVES

Valve Type	Maximum Working Pressure (psi)	Maximum Flow at 1000 psi Pressure Drop (gpm)	Frequency at 90° Lag (Hz)	Hysteresis (%)	Resolution (%)
Spool					
One stage	5000	3500	200	0.1	0.01
Two stage	7000	1000	200	1	0.01
Three stage	4500	300	200	1	0.01
Flapper-nozzle/spool					
One and two stage	5000	1000	500	0.2	0.01
Three stage	5000	1000	500	1	0.01
Jet pipe/spool					
One and two stage	4500	300	500	2	0.1
Three stage	4500	300	200	2	0.1
Sliding plate					
One and two stage	300	40	150	3	0.1

Source: From Ref. 25.

16.8.2 Mathematical Model of a Spool-Type Valve

Spool-type valves are the most popular due to their ease of construction. They are also easier to analyze than other types of valves. Figure 16.33 shows a typical spool-valve configuration and defines the important variables and parameters. The valve has three "energy ports" where energy or power flows from or to the environment of the valve. Correspondingly, the valve can be modeled using the three mathematical equations given in functional form as follows:

$$Q_m = f(x, P_m, P_s) \tag{16.53}$$

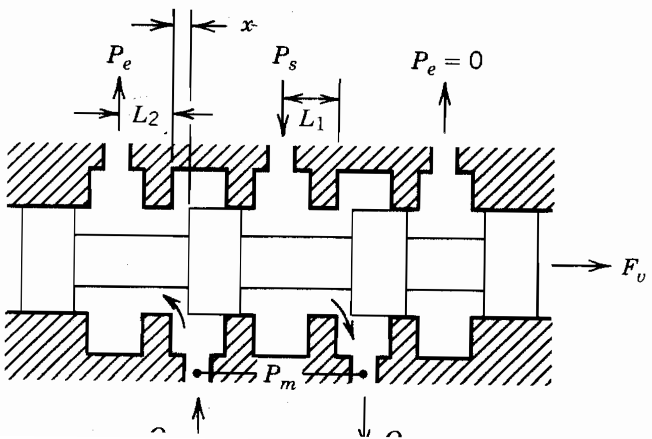
$$Q_s = f(x, P_m, P_s) \tag{16.54}$$

$$F_v = f(x, P_m, P_s) \tag{16.55}$$

Equation 16.53 gives the pressure-flow-displacement characteristics of the valve. These characteristics are needed in the dynamic analysis of a servoactuator which employs the valve. Equation 16.54 is used to compute the required flow rate from the source and will not be considered further here. Equation 16.55 is used to calculate the force required to move the spool (e.g., force output requirement of the torque motor in the case of an electrohydraulic servovalve).

The steady-state pressure-flow-displacement characteristics of the spool valve are characterized by the nonlinear orifice equation (Refs. 30 and 37):

$$Q_m = C_d\, w(x + U) \sqrt{\frac{P_s - P_m}{\rho}} \tag{16.56}$$



C_d = Effective coefficient of discharge
P_s = Supply pressure
P_m = Pressure drop across the servomotor (see Fig. 16.33)
Q_m = Flow rate to the servomotor (see Fig. 16.33)
U = Underlap of spool with respect to sleeve (see Ref. 30); $U = 0$ for an "idealized" valve
w = Circumferential width of metering ports in the valve
x = Displacement of the spool from its neutral position
ρ = Mass density of fluid

This model assumes that the flow rates through the metering orifices are steady, the fluid is incompressible, and the valve exhaust pressure $P_e = 0$. A linearized form of this model facilitates the dynamic analysis of a servoactuator containing the valve. The nonlinear model may be linearized by considering small changes of all variables about an initial steady-state operating point, with the result:

$$\Delta Q_m = K_1 \Delta x - C_1 \Delta P_m \tag{16.57}$$

where

$$K_1 = \left.\frac{\partial Q_m}{\partial x}\right|_{P_{m0},\, x_0} = C_d\, w \sqrt{\frac{P_s - P_{m0}}{\rho}} \tag{16.58}$$

$$C_1 = -\left.\frac{\partial Q_m}{\partial P_m}\right|_{x_0,\, P_{m0}} = \frac{C_d\, w x_0}{2\sqrt{\rho(P_s - P_{m0})}} \tag{16.59}$$

The terms ΔQ_m, Δx, and ΔP_m represent small changes of the corresponding variables about the steady-state operating point x_0, P_{m0}. The constants K_1 and C_1 are evaluated at the operating point. These expressions assume that the valve port shape does not vary with displacement.

The static and dynamic behavior of the valve spool (Eq. 16.55) can be modeled by considering the forces which act on the spool (Ref. 30). These forces include the externally imposed force (input) as well as steady and unsteady flow forces resulting from flow through the orifices. Additional forces that may be present include the viscous damping between the spool and the sleeve and any mechanical spring forces acting on the spool (not shown in Fig. 16.33). The force balance equation for the spool shown in Fig. 16.33 is

$$F_v = m_s \ddot{x} + \frac{\mu A_s}{h}\dot{x} + \left[C_d\, w \sqrt{2\rho\left(\frac{P_s - P_m}{2}\right)}\,(L_1 - L_2)\right]\dot{x} + \left[2C_d\, C_v\, w\left(\frac{P_s - P_m}{2}\right)\cos\theta_j\right]x \tag{16.60}$$

where

A_s = Net shear area of spool lands
C_d = Metering orifice discharge coefficient
C_v = Metering orifice velocity coefficient (see Ref. 30)
F_v = External force on spool (e.g., imposed by torque motor)
h = Radial clearance between spool and sleeve (valve body)
L_1 = Length of fluid column to be accelerated at inlet (see Fig. 16.33)
L_2 = Length of fluid column to be accelerated at outlet (see Fig. 16.33)
m_s = Mass of spool
P_m = Pressure drop across servomotor
P_s = Supply pressure
w = Circumferential width of metering ports in valve
x = Displacement of spool
$\dot{x}$ = Velocity of spool
$\ddot{x}$ = Acceleration of spool
ρ = Mass density of fluid
μ = Absolute viscosity of fluid
θ_j = Effective angle of fluid jet (see Ref. 30)

The fourth term on the right-hand side of Eq. 16.60 is the steady flow-induced force and the third term on the right-hand side is the unsteady flow-induced force. The steady flow force is a "springlike" force that always opposes the motion of the spool, and hence is a stabilizing force. The unsteady flow force is a "damping-like" force that changes its direction of action depending on the flow direction, and hence it can be a stabilizing or destabilizing force. The valve is dynamically stable if $(L_1 - L_2) > 0$. A more complete discussion of the dynamic modeling of the valve spool is given in Ref. 30.

16.8.3 Mathematical Models for an Electrohydraulic Servovalve

The steady-state pressure-flow characteristics for an electrohydraulic servovalve of the type shown in Fig. 16.30 are identical to those of the spool-type valve in the previous section except the input x is replaced by the current I. That is, in the steady state, the motion of the spool in the electrohydraulic servovalve is directly proportional to the current input to the valve. The steady-state pressure-flow-current characteristics for the "idealized" electrohydraulic servovalve (e.g., Fig. 16.30; spool matched perfectly with sleeve such that effective underlap $U = 0$) are given by the equation

$$Q_m = K_v I \sqrt{\frac{P_s - P_m}{\rho}} \tag{16.61}$$

where

K_v = A size factor
I = Current input to servovalve
P_s = Supply pressure
P_m = Pressure drop across the servomotor
Q_m = Control flow rate to the servomotor
ρ = Mass density of fluid

This model assumes that the exhaust pressure $P_e = 0$. Equation 16.61 may be linearized for operation about an initial steady-state operating point with the result:

$$\Delta Q_m = K_1 \Delta I - C_1 \Delta P_m \tag{16.62}$$

where

$$K_1 = \left.\frac{\partial Q_m}{\partial I}\right|_{P_{m0}, I_0} \quad \text{(Flow sensitivity)} \tag{16.63}$$

$$C_1 = -\left.\frac{\partial Q_m}{\partial P_m}\right|_{I_0, P_{m0}} \quad \text{(Flow-pressure sensitivity)} \tag{16.64}$$

Equation 16.62 is valid for cases when $U \neq 0$ as well. The terms ΔQ_m, ΔI, and ΔP_m represent small changes of the corresponding variables about the initial steady-state operating point I_0, P_{m0}. The constants K_1 and C_1 are evaluated at the operating point.

Typical steady-state pressure-flow-current characteristics for an "idealized" electrohydraulic servovalve that employs a spool-valve second stage (governed by Eq. 16.61) are shown in Fig. 16.34. Characteristics for other types of electrohydraulic servovalves are given in Refs. 34 and 35.

Another important characteristic of an electrohydraulic servovalve is its hysteresis due to the characteristics of the permanent magnets in the torque motor. The hysteresis characteristic is determined from a measurement of the output flow rate as a function of the input current for a constant (usually zero) pressure drop across the valve (load pressure drop). A typical hysteresis characteristic is shown in Fig. 16.35. The slope of the flow-current curve is the "flow sensitivity" of the valve (i.e., K_1 in Eq. 16.63).

It is often convenient in the dynamic analysis of servoactuators to have an approximate dynamic model for the servovalve. Experience has shown that linearized transfer functions based on empirical approximations from measured servovalve responses are adequate for most system designs. Reference 39 outlines considerations underlying the determination of approximate transfer function models for electrohydraulic servovalves. Figure 16.36 shows typical frequency-response plots for an electrohydraulic flow-control servovalve, along with approximate transfer functions. For a frequency range of 0–50 Hz, the following first-order expression has been found to be adequate for two-stage electrohydraulic servovalves:

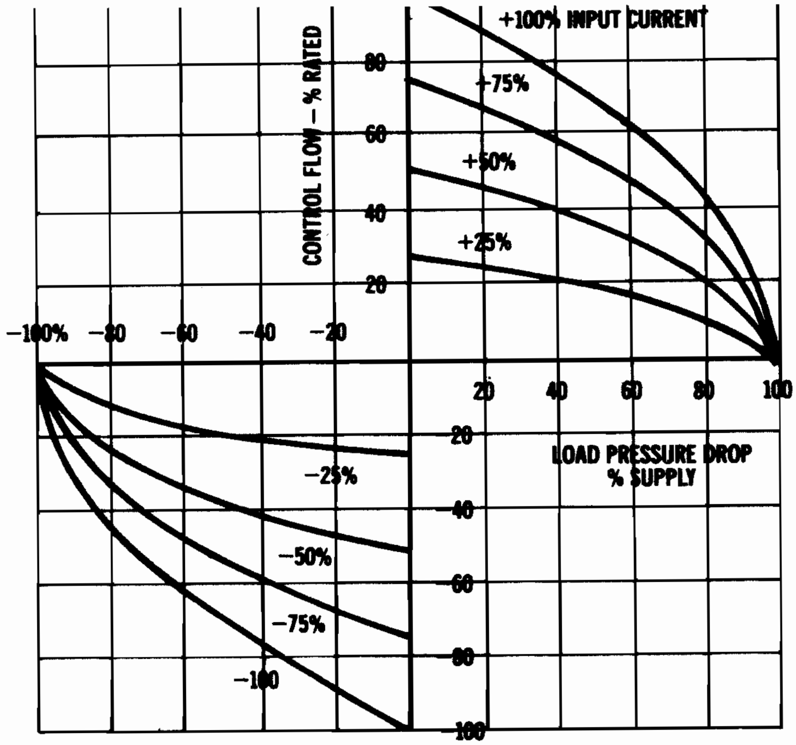


Fig. 16.34 Typical steady-state pressure-flow characteristics of an electrohydraulic servovalve. (Courtesy of Moog Inc., East Aurora, NY)

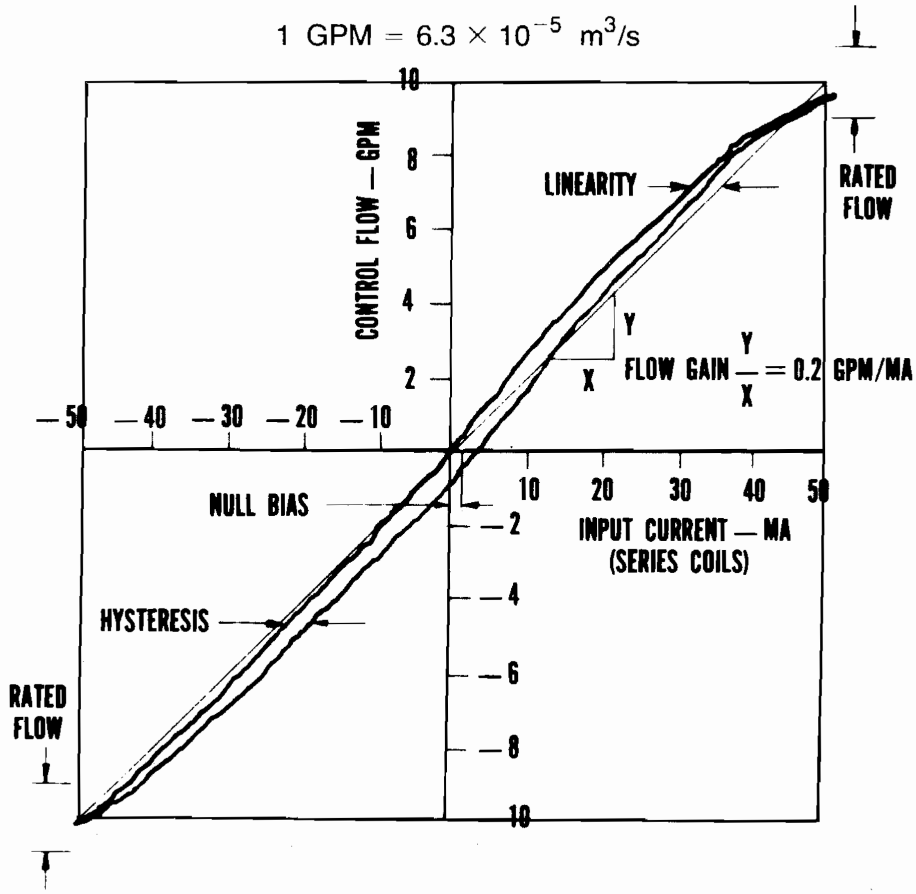


Fig. 16.35 Typical steady-state flow-current characteristics of an electrohydraulic servovalve. (Courtesy of Moog, Inc., East Aurora, NY)

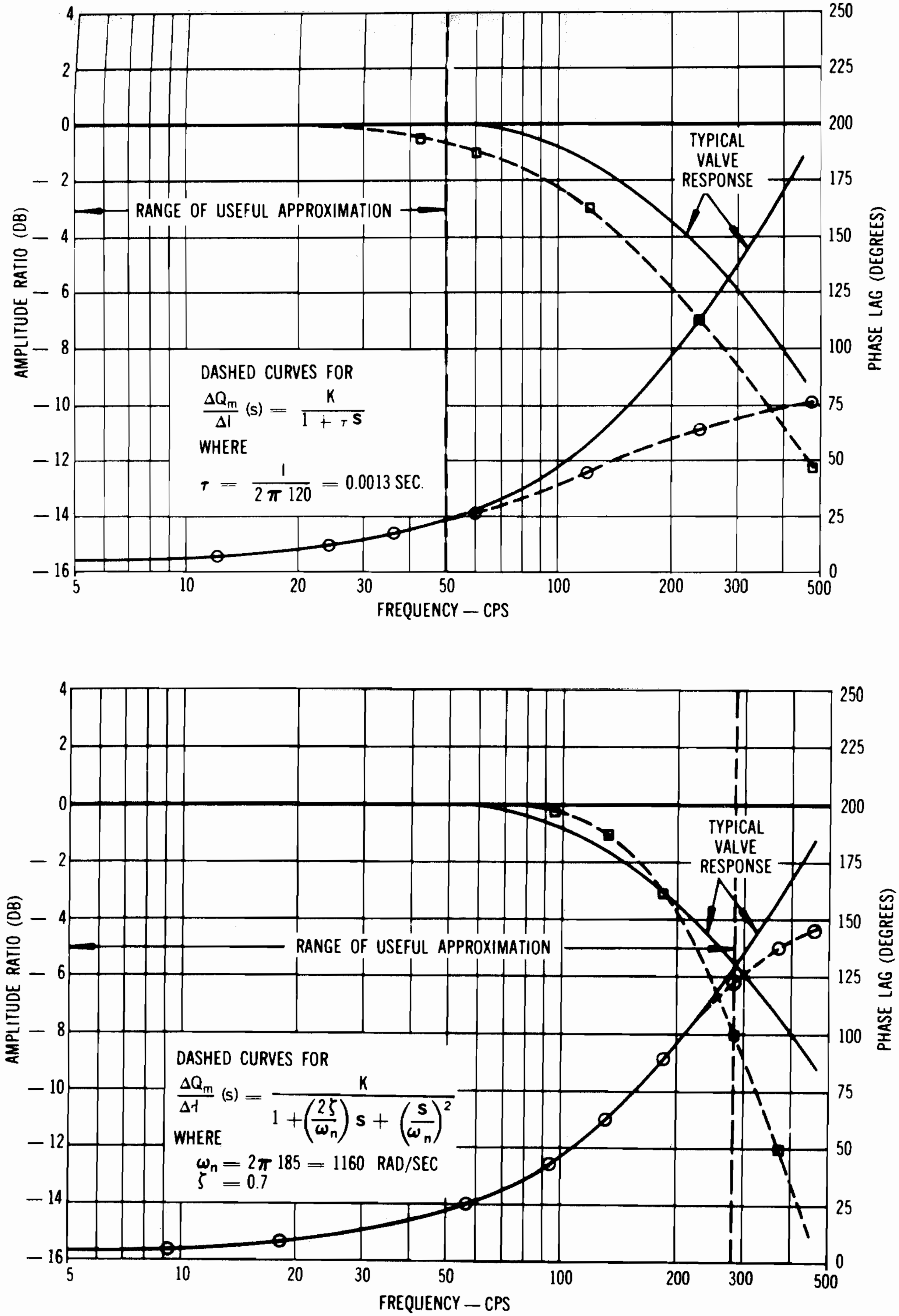


Fig. 16.36 Typical dynamic behavior of electrohydraulic servovalves (taken from Reference 39).

TABLE 16.9 TYPICAL DYNAMIC CHARACTERISTICS OF TWO-STAGE ELECTROHYDRAULIC FLOW-CONTROL SERVOVALVES

Flow-control servovalve	Max. flow capacity at 3000 psi gpm	Approximate dynamics 3000 psi 100°F Peak-to-Peak Input at 50% rated current		
		1st order	2nd order	
		τ sec	f_n cps	ζ
A	2	0.0013	240	0.5
B	6	0.0015	200	0.5
C	12	0.0020	160	0.55
D	18	0.0023	140	0.6
E	30	0.0029	110	0.65

Source: From Reference 39.

$$\frac{\Delta Q_m(s)}{\Delta I(s)} = \frac{K_1}{\tau s + 1} \tag{16.65}$$

where

$Q_m(s)$ = Laplace transform of control flow rate to the servomotor
$I(s)$ = Laplace transform of the current input to the servovalve
K_1 = Servovalve flow sensitivity at $P_m = 0$, $I = 0$
τ = Apparent servovalve time constant (s)

Typical time constants for electrohydraulic flow-control servovalves are given in Table 16.9.

If a good approximation is desired over a wider frequency range, the following second-order model may be preferred:

$$\frac{\Delta Q_m(s)}{\Delta I(s)} = \frac{K_1}{(s/\omega_n)^2 + (2\zeta/\omega_n)s + 1} \tag{16.66}$$

where

$\omega_n = 2\pi f_n$ = Apparent natural frequency (rad/s)
ζ = Apparent damping ratio (dimensionless)

Typical values of f_n and ζ for two-stage electrohydraulic flow control servovalves are given in Table 16.9.

16.9 ELECTROMECHANICAL AND ELECTROHYDRAULIC SERVOSYSTEMS

16.9.1 Typical Configurations of Electromechanical Servosystems

An electrical servomotor may be combined with an electrical or electronic modulator to form an electromechanical servoactuator. The addition of a feedback transducer forms a servosystem. Figure 16.2 shows an electromechanical linear-motion servosystem which incorporates a rotary brushless DC servomotor and tachometer feedback; the electronic modulator is not shown.

16.9.2 Typical Configurations of Electrohydraulic Servosystems

A servoactuator comprising an electrohydraulic servovalve and a servomotor may be combined with an electronic servoamplifier (or modulator) and an appropriate feedback transducer to form a high-performance servosystem. Schematic diagrams of three typical electrohydraulic servosystems are shown in Fig. 16.37.

Figure 16.38 shows a photograph and a cross section of a high-performance servosystem which incorporates an electrohydraulic servovalve, an axial-piston servomotor, and a tachometer. The servo-