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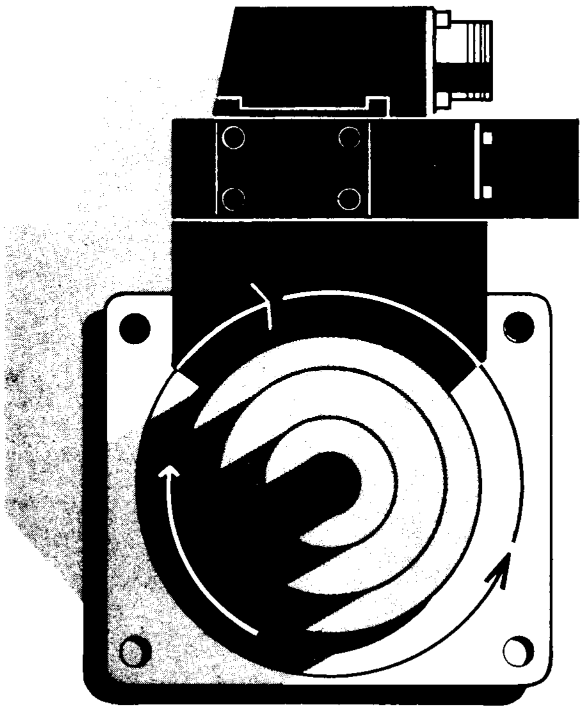

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SELECTION AND PERFORMANCE CRITERIA FOR ELECTROHYDRAULIC SERVODRIVES

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INTRODUCTION

An electrohydraulic servodrive is a marriage of a flow-control servovalve, a rotary hydraulic motor, and an instrumentation package that contains electrical transducers to measure performance, as illustrated in Figure 1. Servodrives are intended for service in closed-loop position or velocity control systems and are widely used in the machine tool industry as the "muscles" of numerical control. Typical electrical transducers are a tachometer for velocity feedback and a resolver or shaft encoder for position feedback.

There are several significant advantages of electrohydraulic servodrives over electric motor drives. Some of these are presented below:

1) Hydraulic drives have substantially higher power to weight ratios resulting in higher machine frame resonant frequencies for a given power level.

2) Hydraulic drives are much stiffer than electric drives, resulting in higher loop gain capability, greater accuracy, and better frequency response.

3) Hydraulic drives give smoother performance at low speeds and have a wide speed range without special control circuits. They can usually be direct-coupled to the load without the requirement for intermediate gearing.

4) Hydraulic drives are self-cooling and can be operated in a stall condition indefinitely without damage.

Both hydraulic and electric drives are very reliable provided that maintenance recommendations are followed. Hydraulic drives are usually less expensive for systems above several horsepower, especially if the hydraulic power supply is shared between several axes.

IDEAL SERVODRIVE CHARACTERISTICS

In this section, the power elements of the servodrive (i.e. the servovalve and hydraulic motor) are considered as ideal components and various torque-speed relations are presented. The ideal hydraulic motor provides shaft torque proportional to servovalve differential pressure and speed proportional to servovalve flow in accordance with Equations (1) and (2)*.

$$T_m = \frac{D_m}{2\pi} \Delta P_L \qquad (1)$$

$$\omega_m = \frac{2\pi}{D_m} Q_L \qquad (2)$$

The product of these two equations is the power delivered to the load. For the ideal motor, this power is transferred without loss from the hydraulic input power ($Q_L \, \Delta P_L$) to motor output shaft power ($\omega_m \, T_m$).

The flow and differential pressure to the servomotor are provided from a constant pressure hydraulic supply through an electrohydraulic servovalve (REF 1). Because the servovalve controls load pressure by throttling load flow across sharp-edged orifices in the valve second stage, the relation between load flow and load pressure is parabolic. This flow-pressure characteristic is expressed for an ideal servovalve by Equation (3).

$$\frac{Q_L}{Q_{nL}} = \frac{i_v}{i_r} \sqrt{1 - \frac{\Delta P_L}{P_s}} \qquad (3)$$

It is apparent that for a constant load differential pressure, load flow is linearly proportional to current in the servovalve torque motor coils. Also, for constant coil current, load flow is reduced in a square root relation by increased load pressure. When load pressure equals supply pressure, the load flow will be zero for any current (i.e. the load is stalled). The no-load flow by definition occurs with zero load differential pressure. In this case, all of the supply pressure is lost across the valve orifices and no torque is available at the motor shaft. This would roughly correspond to a rapid traverse condition in a machine tool, where high carriage speeds with no cutting torque are required.

The relationship between servovalve flow, current, and load pressure is plotted in Figure 2. Because of the linear relations of Equations (1) and (2), motor shaft speed and torque could equally well be plotted on Figure 2, constituting the servodrive torque-speed characteristics for open-loop control.

* All symbols are defined in the nomenclature, (page 6).

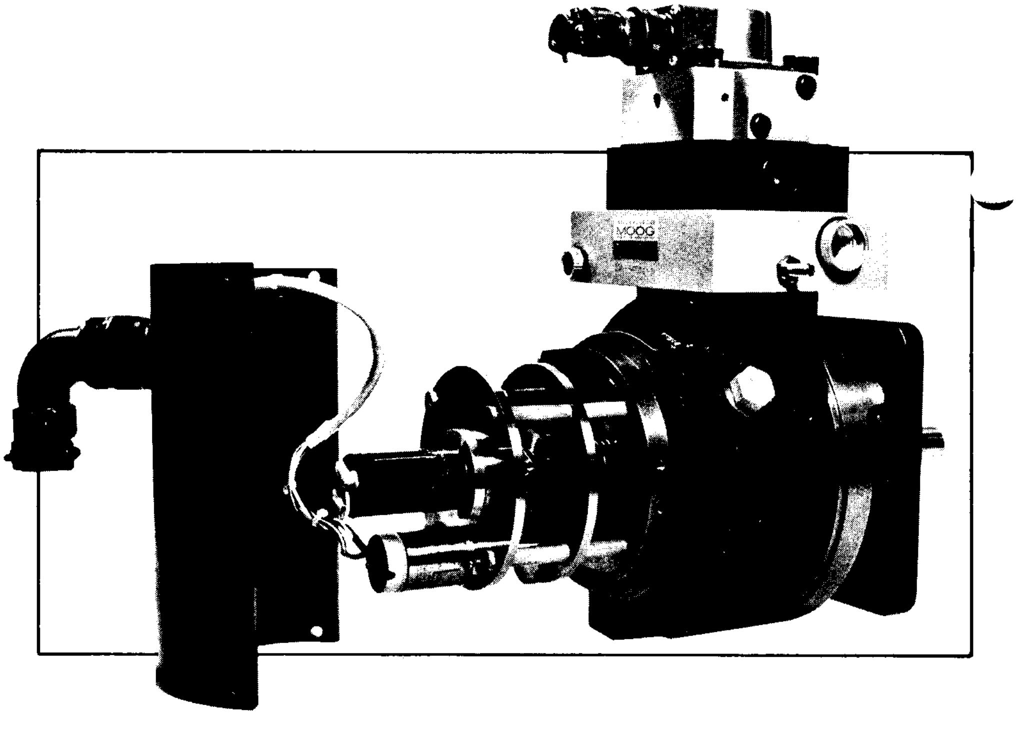


FIGURE 1
TYPICAL SERVODRIVE CONFIGURATION

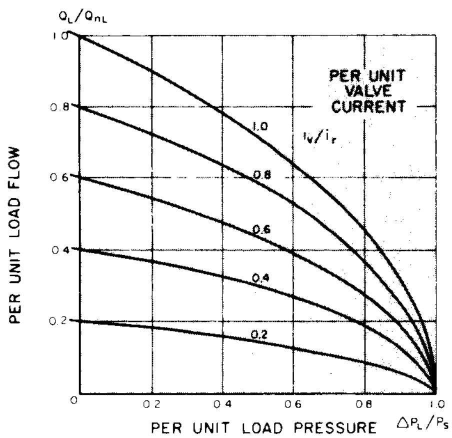


FIGURE 2 SERVOVALVE FLOW-PRESSURE CHARACTERISTICS

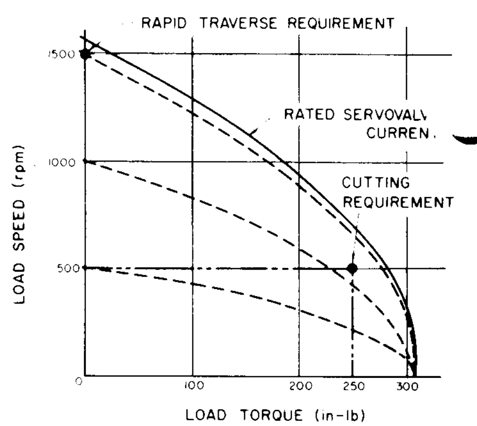


FIGURE 3 LOAD TORQUE-SPEED CHARACTERISTICS FOR EXAMPLE IN TEXT

Equations (1) through (3) are typically used to "size" a servodrive to the load requirements. For example, a machine tool axis may require the following performance (referred to the motor shaft):

Cutting Torque —
250 in-lb minimum from 0 to 500 rpm
Rapid Traverse —
1500 rpm with negligible torque

It is necessary to assume a value for either the hydraulic supply pressure or the servomotor displacement and then calculate other properties based on the assumption. Commercially available servomotors typically have displacements of 1.0, 2.5, 5 or 7.6 in³/rev. If we assume a 1.0 in³/rev motor, we can proceed as follows:

1) From Equation (2), calculate the no-load flow for a speed of 1500 rpm

$$Q_{nL} = \frac{1.0}{2\pi} \times \frac{2\pi}{60} \times 1500 = 25 \text{ cis}$$

2) From Equation (2), calculate the flow at 500 rpm (the maximum cutting speed)

$$Q_L = \frac{1.0}{2\pi} \times \frac{2\pi}{60} \times 500 = 8.33 \text{ cis}$$

3) From Equation (3), calculate the fraction of system pressure required t supply the cutting torque at maximum cutting speed (500 rpm), assuming full rated current into the servovalve.

$$\frac{8.33}{25} = 1.0\sqrt{1 - \frac{\Delta P_L}{P_s}}$$

$$\frac{\Delta P_L}{P_s} = 0.89$$

4) From Equation (1), calculate the supply pressure required to provide the desired load torque.

$$250 = \frac{1.0}{2\pi}(0.89\ P_s)$$

$$P_s = 1765 \text{ psi}$$

Conservative design practice allows 10 to 20 percent margin on the selection of system pressure to allow for hydraulic line losses as well as leakage and viscous friction losses in the hydraulic motor. In this example, using a 10 percent supply pressure margin,

$$P_s = 1.1 \times 1765 = 1940 \text{ psi}$$

It remains only to relate no-load valve flow at the calculated supply pressure to rated servovalve flow. Rated flow is almost always expressed as the valve flow at a valve pressure drop of 1000 psi. This relationship is expressed by Equation (4).

$$\frac{Q_{nL}}{Q_r} = \sqrt{\frac{P_s}{1000}} \qquad (4)$$

Applying this equation to the example, we get

$$\frac{25}{Q_r} = \sqrt{\frac{1940}{1000}}$$

$$Q_r = 17.9 \text{ cis} = 4.66 \text{ gpm}$$

The next largest rated flow servovalve (5.0 gpm at 1000 psi valve drop for most manufacturers) would be selected. This selection will then provide margin for cross-port leakage and leakage from the control ports to case within the servo-motor.

The torque-speed characteristics corresponding to this example are shown in Figure 3.

ACTUAL SERVODRIVE PERFORMANCE

Although the servodrive properties described in the previous section are sufficient for valve and motor selection, a number of non-ideal properties of the servovalve and hydraulic motor exist which influence performance. The following summarizes some of the anomalies that must be considered in designing a closed-loop control system using a servo-drive where speed and torque ranges exceeding 1000 to 1 must be available with maximum smoothness:

1. Variation of motor displacement with shaft angle
2. Motor leakage, both constant and shaft angle dependent
3. Breakout and running friction of the motor and load
4. Servovalve null shift, threshold, and hysteresis
5. Oil compressibility effects (load resonance)

Each of these non-ideal properties will be examined below in terms of its influence on servodrive performance:

Variation of Motor Displacement

Motor displacement variation with shaft angle results from changes of effective radius and center-of-pressure location of the pressurized power elements of the motor as the shaft rotates. These displacement variations, as well as changes in the lengths of internal motor leakage paths, are usually associated with the motor valving used to create unidirectional shaft rotation for constant input flow. The change in motor displacement and leakage with shaft angle creates a variation in motor speed and torque for a constant flow and load pressure, called "ripple". By measuring and feeding back the motor speed, in a closed-loop control system, the servovalve spool will respond as necessary to minimize motor ripple as indicated by Figure 4. Here the same motor/valve combination is run first open-loop and then closed-loop at various values of input current. The ripple is reduced in inverse proportion to static loop gain within the bandwidth of the closed-loop servo system. As shaft speed increases, the servovalve spool ultimately reaches a point where it cannot follow the ripple frequency. Beyond this speed, the percentage ripple remains relatively constant so the absolute ripple increases linearly with shaft speed, as indicated in Figure 5. In machine tool service, low-speed ripple shows up directly as imperfections in the work piece. At high-speeds, ripple can excite machine frame resonances and these may affect surface finish.

Motor Leakage and Friction

There are two primary leakage paths within a hydraulic motor; (1) leakage from each control port to return, and (2) leakage from one control port to the other due to load differential pressure. Port-to-return leakage tends to reduce the quiescent pressures at the control ports because the leakage must be supplied by flow across the pressure lands of the servovalve spool. It is desirable to maintain quiescent control port pressures at or somewhat above one-half of supply pressure. This improves the linearity of load pressure to valve current and also reduces cavitation on the low pressure side of the motor when braking over-hauling loads. For these reasons, it is common to slightly undercut the valve spool pressure lands to supply the leakage flow and maintain adequate quiescent control port pressures.

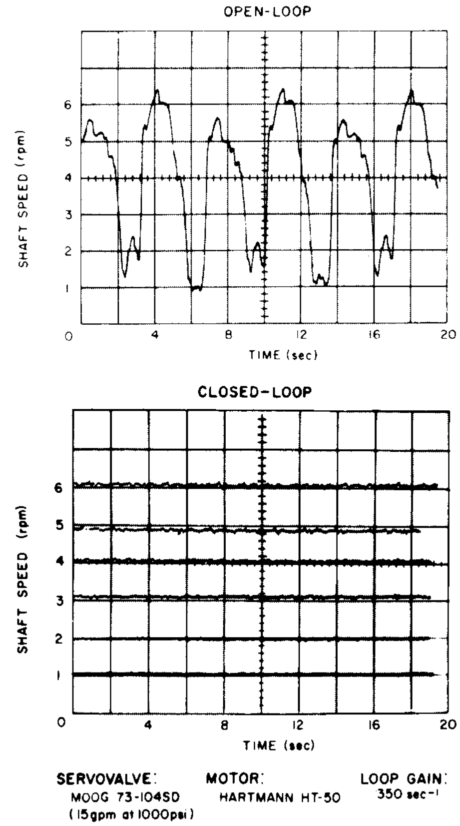


FIGURE 4 OPEN AND CLOSED-LOOP SERVODRIVE LOW-SPEED PERFORMANCE

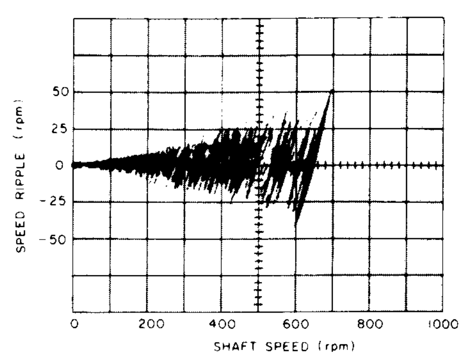


FIGURE 5 RIPPLE VARIATION WITH SHAFT SPEED

The cross-port leakage path has a more direct influence on servodrive performance. It is usually both beneficial in the sense of supplying damping to a resonant load, and harmful by reducing the effective pressure gain of the servovalve. With reduced pressure gain, a larger valve current (error signal) is required to create the same load differential pressure. For example, in a servo system with combined motor and load breakout friction of $\triangle P_f$, the valve error current required to just overcome the friction is

$$i_{vf} = \frac{\triangle P_f}{K_p} \qquad (5)$$

where K_p is the blocked load pressure gain of the servodrive given by Equation (6).

$$K_p = \frac{K_v}{K_L} \frac{1}{1 + \dfrac{K_v}{K_L K_{pv}}} \qquad (6)$$

Typical values of pressure gain for a servovalve alone without motor leakage can be calculated from Equation (7).

$$K_{pv} = 30 \frac{P_s}{i_r} \qquad (7)$$

The uncertainty or error of servomotor output required to generate the valve current necessary to cause the load to move is determined by dividing Equation (5) by the product of the gain factors between the motor output and the valve input. These factors normally are the feedback transducer and servoamplifier gain as illustrated in Figure 6. For example, in a motor shaft position servo, they are expressed by Equation (8).

$$\theta_\epsilon = \frac{i_{vf}}{K_\theta \ K_a} \qquad (8)$$

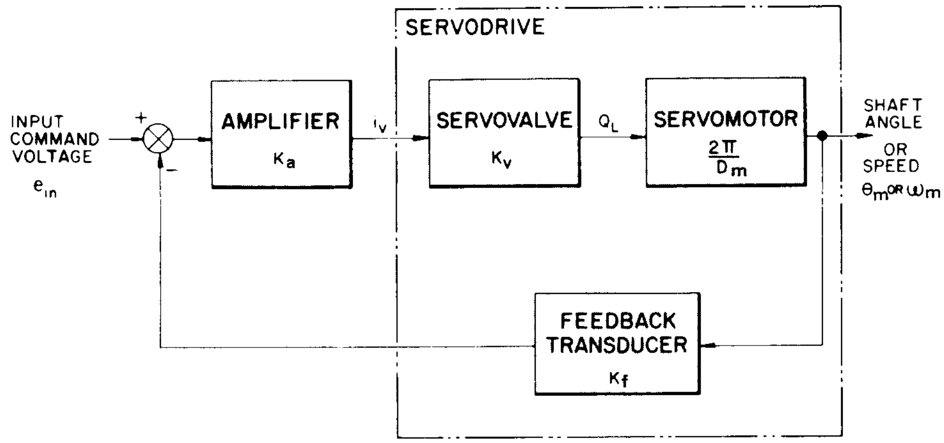
Combining Equations (5) and (8),

$$\theta_\epsilon = \frac{\triangle P_f}{K_p \ K_\theta \ K_a} \qquad (9)$$

Three facts are evident from Equations (6) and (9):

1) Pressure gain K_p must be high for greatest accuracy. This requires minimum values of K_L, the cross-port leakage coefficient.

2) Feedback transducer and amplifier gain (i.e. electronic gain) must be high for greatest accuracy. For a given level of loop gain, it is desirable to distribute as much gain as possible into the electronic elements between the motor output and the valve input.



$$\text{LOOP GAIN} = K_{VL} = \frac{2\pi}{D_m} K_a K_v K_f$$

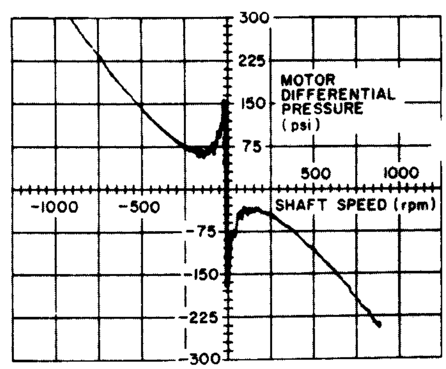
FIGURE 6 SIMPLIFIED HYDRAULIC SERVO BLOCK DIAGRAM

3) Every effort should be made to minimize friction, both in the load and by selection of a hydraulic motor with low breakout pressure. A typical plot of motor differential pressure versus shaft speed for a high quality servomotor is given in Figure 7.

It should be evident from these considerations that the purposeful introduction of cross-port leakage to augment damping of the load resonance should be discouraged in systems where maximum accuracy is desired. Viscous motor losses (shown in Figure 7 for speeds above 250 rpm) are useful for load damping but extract an obvious penalty in lost efficiency. Electronic damping techniques such as dynamic pressure feedback (REF 2) or load acceleration feedback should be used where practical.

Servovalve Anomalies

The flow-control servovalve has several non-ideal properties that influence servodrive performance. The most critical valve parameters are null shift, threshold, and hysteresis. These parameters are expressed as a percentage of rated valve current and they are more or less independent of valve flow rating. Null shifts may be caused by temperature, supply pressure, and return line pressure variations. The percent null shift represents the level of steady valve current required to restore valve null. Threshold is a measure of internal friction within the servovalve and represents the amount of valve current change necessary to cause a corresponding change of servovalve output. Hysteresis is caused by friction within the servovalve as well as the electromagnetic properties of the first-stage torque motor. It appears as an uncertainty of valve output as related to the previous levels of current input.



SERVOVALVE: MOOG 73-104SD **SYSTEM PRESSURE:** 2000 psi **MOTOR:** HARTMANN HT-50

FIGURE 7 DIFFERENTIAL PRESSURE VERSUS SHAFT SPEED — UNLOADED SERVOMOTOR

Each of these undesirable properties can be expressed as a valve current that must be furnished by an error in the closed-loop control system. In each case, the corresponding error can be calculated by dividing the spurious valve current by the servoamplifier and feedback gains, as illustrated for position feedback in Equation (8). This amplifies the conclusion that high levels of electronic gain are desirable for best repeatability of the closed-loop system.

Fluid Compressibility Effects

Hydraulic oil compliance is expressed in terms of the fluid bulk modulus which is the change in pressure required to cause a unit change in volume of the oil. The oil compliance results in a loss of stiffness in the servomotor caused by compression of oil both within the motor and between the servovalve and the motor. The oil stiffness can be calculated from Equation (10).

$$k_o = \frac{B \sigma D_m}{\pi^2} \tag{10}$$

where $\sigma = \dfrac{D_m}{V_t}$ = Ratio of motor displacement to total fluid volume under compression between the servovalve and both sides of the motor.

B = Oil bulk modulus (for DTE-24 hydraulic fluid at room temperature, B = 150,000 psi is a conservative value that allows for some air entrainment in the oil).

Note that σ is always less than 1.0 as V_t includes D_m as well as the fluid volume of the motor ports, manifold, and servovalve. Thus, in order to maximize oil stiffness, it is necessary to mount the servovalve as close to the motor as possible. The servovalve is manifold mounted directly to the motor body in all servodrives marketed today. Table I (page 6) lists typical values of motor shaft oil stiffness for several popular servomotor sizes, as well as other important physical properties to be discussed later.

Introduction of the oil spring into the servoloop allows the dynamics of the load to reflect in the loop gain and phase characteristics, thus affecting servo stability. It can be shown that for a resonant load, the maximum loop gain that can be achieved before servo instability occurs is that given by Equation (11).

$$K_{v_L\,max} = 2\zeta\omega_n \tag{11}$$

The load most typical for a servodrive in machine tool service is almost pure inertia when reflected back through lead screws and gearing to the motor shaft. The total load inertia (including motor and instrumentation) interacts with the oil spring to form the load resonance of frequency given by Equation (12).

$$\omega_n = \sqrt{\frac{k_o}{I_t}} \tag{12}$$

Because of the excellent bearings typically used, the load absorbs little energy due to friction and viscous losses, so the damping ratio is small (typically 0.1 or less). The damping ratio, together with the load resonant frequency, establishes a definite upper limit on allowable loop gain (Equation 11). For this reason, it is generally accepted that for machine tool service special servo stability compensation techniques will be necessary unless the load resonant frequency exceeds 300 rad/sec.

A block diagram for which the oil compliance can be combined with any arbitrary load transfer function to mathematically model servomotor performance is given in Figure 7. Note that the undefined transfer function $\dfrac{\omega_m}{\Delta T}(s)$ can include other degrees of freedom such as machine frame resonances. In Figure 7, the leakage flow is proportional to motor differential pressure, and the shaft torque reflects the loss of effective motor pressure due to friction and viscous losses. If we assume that the load is pure inertia with no inherent damping, the load transfer function is

$$\frac{\omega_m}{\Delta T}(s) = \frac{I}{I_t S} \tag{13}$$

If friction is neglected, the servomotor transfer function for this example is given by Equation (14).

$$\frac{\omega_m}{Q_L}(s) = \frac{2\pi}{D_m} \times \frac{1}{\left[1 + \dfrac{60 K_\omega K_L}{D_m}\right] + \left[\left(\dfrac{2\pi}{D_m}\right)^2 K_L I_t + \dfrac{15 K_\omega D_m}{\pi^2 k_o}\right] S + \left[\dfrac{I_t}{k_o}\right] S^2} \tag{14}$$

Equation (14) is representative of servodrive applications where the primary load is inertia. Several important conclusions are evident from this transfer function:

1) Steady-state motor speed is reduced by the combined effect of viscous losses and cross-port leakage. This is because viscous losses require steady load pressure for constant speed which results in leakage flow that subtracts from servovalve flow causing a speed reduction.

2) Both motor viscous losses and cross-port leakage augment load damping and therefore permit higher loop gain with stability.

3) It is desirable to maximize stiffness and minimize inertia to achieve a high load resonant frequency and, therefore, higher loop gain with an adequate stability margin.

Cross-Port Relief Valves

In many cases, it is necessary to provide cross-port relief valves to limit maximum motor differential pressure to a safe value consistent with the motor rating or load acceleration limits. This is particularly true when the load can store substantial kinetic energy, as in the case of an inertia rotating at a high speed. A conservative estimate for the peak pressure levels that would be reached without relief valves if the servovalve spool were suddenly centered while the load inertia is in motion can be calculated by assuming that the kinetic energy is converted without loss into the potential energy of oil compressibility. Equation (15) gives the desired relation.

$$\Delta P_{max} = 0.658 \frac{k_o}{\omega_n D_m} \bar{\omega}_m \tag{15}$$

If ΔP_{max} calculated from this equation exceeds the prescribed safe value, then cross-port relief valves are required.

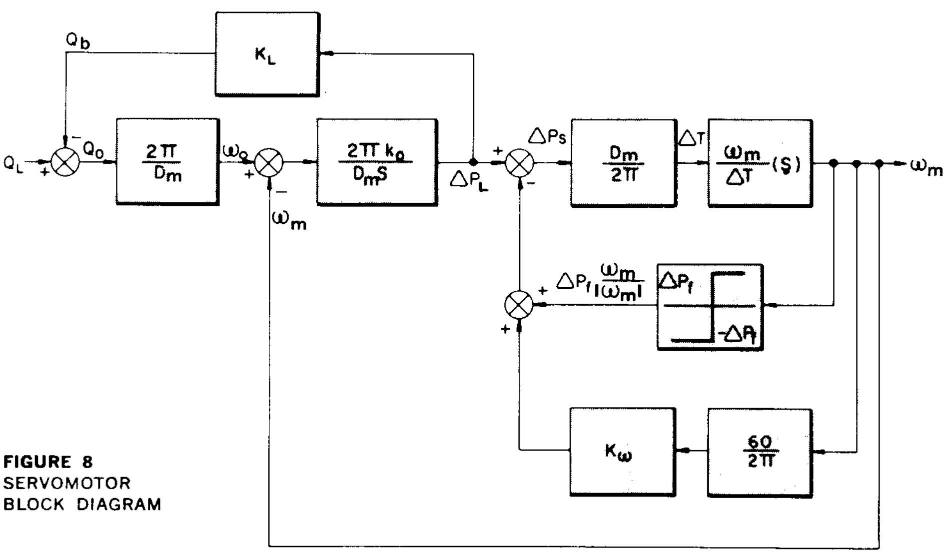


FIGURE 8
SERVOMOTOR
BLOCK DIAGRAM

TABLE 1 SOME TYPICAL SERVOMOTOR PROPERTIES

	MOTOR DISPLACEMENT	MOTOR OIL STIFFNESS	MOTOR INERTIA	UNLOADED MOTOR RESONANT FREQ.	CROSS-PORT LEAKAGE COEFFICIENT	VISCOUS LOSS COEFFICIENT
HARTMANN MOTOR	D_m in³/rev.	k_o in.-lb./rad.	I_m in.-lb.-sec.²	f_m Hz.	K_L cis/psi	K_ω psi/rpm
HT-10	1.0	7000	0.002	298	0.0004	0.13
HT-22	2.2	22,000	0.006	305	0.0003	0.12
HT-50	5.0	59,000	0.015	316	0.0010	0.27
HT-76	7.6	86,000	0.027	284	0.0012	0.21

TABLE 2 NOMENCLATURE

Symbol	Description	Typical Units
T_m	Ideal servomotor shaft torque	in-lb
$\triangle T$	Change in motor shaft torque around an operating point	in-lb
$\triangle P_L$	Servovalve load differential pressure	psi
$\triangle P_f$	Differential pressure to break-out servomotor friction	psi
$\triangle P_s$	Effective servomotor differential pressure after subtracting losses	psi
$\triangle P_{max}$	Peak servomotor differential pressure when braking a load with kinetic energy	psi
P_s	Hydraulic power supply (or system) pressure	psi
θ_m	Servomotor shaft position	rad
θ_ϵ	Servomotor shaft position error	rad
ω_m	Servomotor shaft speed	rad/sec
$\overline{\omega}_m$	Servomotor shaft speed $\overline{\omega}_m = \frac{60}{2\pi}\omega_m$	rpm
ω_o	Ideal servomotor shaft speed neglecting oil compressibility	rad/sec
Q_L	Servovalve load flow	cis
Q_{nL}	No-load servovalve flow	cis
Q_r	Rated servovalve flow at 1000 psi valve pressure drop	gpm
Q_o	Net servomotor flow after substracting leakage flow	cis
Q_b	Cross-port leakage flow	cis
i_v	Servovalve torque motor coil current	ma
i_{vf}	Servovalve current required to breakout servomotor friction	ma
i_r	Rated servovalve current	ma
k_o	Servodrive oil stiffness	in-lb/rad
B	Hydraulic fluid bulk modulus	psi
D_m	Servomotor volumetric displacement	in³/rev

NOMENCLATURE

Symbol	Description	Typical Units
V_t	Total fluid volume under compression between the servovalve and both sides of the motor, including the motor volumetric displacement	in^3
σ	Ratio of servomotor volumetric displacement to total fluid volume under compression between the servovalve and both sides of the motor	. .
I_m	Servomotor rotary inertia	$in\text{-}lb\text{-}sec^2$
I_t	Total drive inertia at the motor shaft, including the load, instrumentation, and servomotor	$in\text{-}lb\text{-}sec^2$
S	Laplace transform variable	rad/sec
ω_n	Load resonant frequency (oil spring and total drive inertia)	rad/sec
ζ	Load resonance damping ratio	. . .
f_m	Resonance frequency of unloaded servodrive (oil spring and servomotor inertia)	Hz
$\frac{\omega_m}{\Delta T}(s)$	Generalized load transfer function	rad/sec-in-lb
$\frac{\omega_m}{Q_L}(s)$	Servomotor transfer function	rad/in^3
K_a	Servoamplifier gain	Proportional — ma/volt Integral — ma/sec-volt
K_v	Servovalve no-load flow gain	cis/ma
K_f	Feedback transducer gain	volts/unit servo output
K_θ	Position servo feedback gain	volts/rad
K_{pv}	Servovalve blocked-port pressure gain	psi/ma
K_p	Servodrive blocked shaft pressure gain	psi/ma
K_L	Servomotor cross-port leakage coefficient	cis/psi
K_ω	Servomotor viscous loss coefficient	psi/rpm
$K_{vL_{max}}$	Maximum servoloop gain for stability	rad/sec

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