TECHNICAL BULLETIN 103

MOOG

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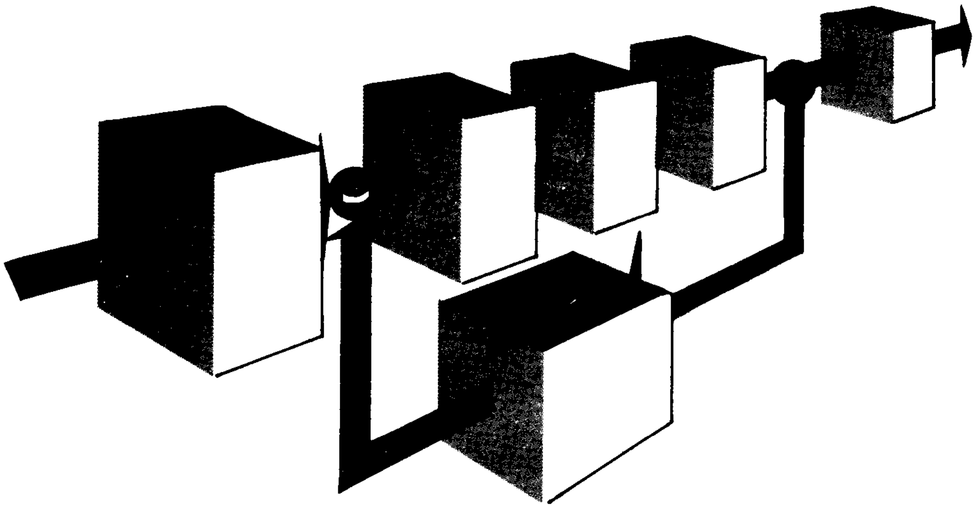

TRANSFER FUNCTIONS FOR MOOG SERVOVALVES

W. J. THAYER, DECEMBER 1958
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INTRODUCTION

It is often convenient in servoanalysis or in system synthesis work to represent an electrohydraulic servovalve by a simplified, equivalent transfer function. Such a representation is, at best, only an approximation of actual servovalve performance. However, the usefulness of linear transfer functions for approximating servovalve response in analytical work is well established.

The difficulty in assuming an explicit transfer function for electrohydraulic servovalves is that many design factors and many operational and environmental variables produce significant differences in the actual dynamic response. Consider the variables of the valve design. It is well known that internal valve paramaters (e.g., nozzle and orifice sizes, spring rates, spool diameter, spool displacement, etc.) may be adjusted to produce wide variations in dynamic response. An analytic approach for relating servovalve dynamic response to internal valve parameters is given in Appendix I of this technical bulletin.

Once a servovalve is built, the actual dynamic response will vary somewhat with operating conditions such as supply pressure, input signal level, hydraulic fluid temperature, ambient temperature, valve loading, and so forth. These effects are insignificant for small variations about design values, but should be considered where wide excursions are anticipated. It is important to appreciate and control these and other operational variables when performing measurements of servovalve dynamics. If such precautions are not taken, misleading and inaccurate results may be obtained. Appendix II to this Bulletin describes the production

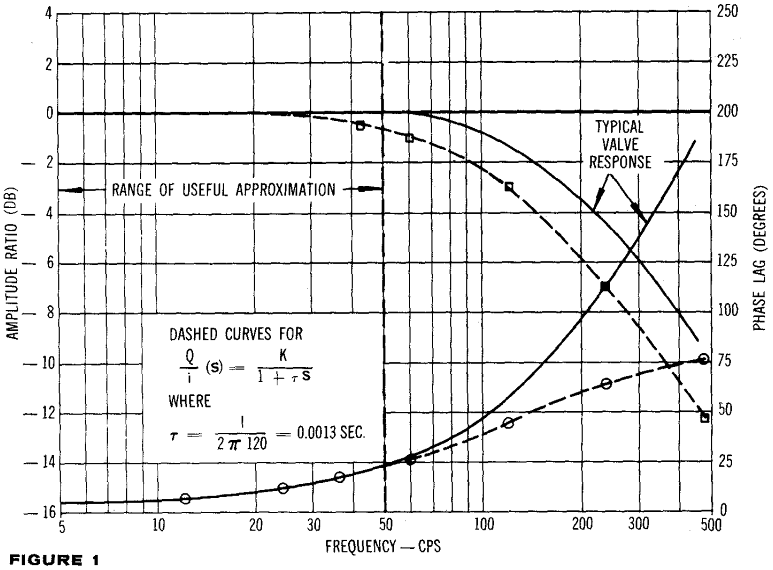


FIGURE 1

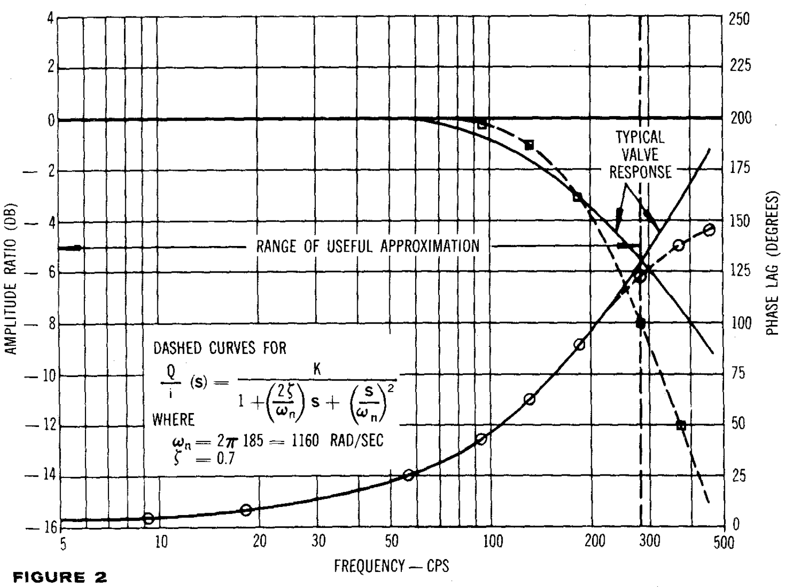


FIGURE 2

equipment presently used by Moog to measure servovalve dynamic response.

Another difficulty in assigning simplified, linear transfer functions to represent servovalve response is that these valves are highly complex devices that exhibit high-order, nonlinear responses. If a first, second, or even third-order transfer function is selected to represent servovalve dynamics, still only an approximation to actual response is possible. Fortunately, for most physical systems, the servovalve is not the primary dynamic element, so it is only necessary to represent valve response throughout a relatively low frequency spectrum. For instance, if a servovalve-actuator is coupled to a load which exhibits a 50 cps resonant frequency, it is meaningful only to represent valve dynamic response in the frequency range to 50 cps. Similarly, for lower response physical systems, the contribution of valve dynamics throughout a correspondingly smaller frequency range need be considered. This simplification of actual servo response should be applied whenever practicable, for the reduced analytical task associated with the system analysis is obvious.

These approximations to servovalve response have resulted in such expressions as "the equivalent time constant of the servovalve is - seconds" or "the apparent natural frequency of the servovalve is - radians /second." If a representation of servovalve response throughout the frequency range to about 50 cps is sufficient, then a first-order expression is usually adequate. Figure I shows a typical valve dynamic response, together with the response of a first-order transfer function. The first-order approximation is seen to be quite good throughout the lower frequency region. The time constant for the first-order transfer function (i.e., the equivalent servovalve time constant) is best established by curve fitting techniques. If a quick approximation is desired, the equivalent time constant should correspond to the 45° phase point rather than the 0.7

amplitude point (-3 db). In general, these points will not coincide as the higher-order dynamic effects contribute low frequency phase lag in the servovalve response, while not detracting appreciably from the amplitude ratio.

If servovalve response to frequencies near the 90° phase lag point is of interest, then a second-order response should be used. In a positional servomechanism, a second-order representation of the servovalve response is usually sufficient, as the actuator contributes an additional 90° phase lag from the inherent integration. Figure 2 shows a second-order approximation to the servovalve dynamics of Figure 1. Here, the natural frequency is best associated with the 90° phase point, and the damping ratio with the amplitude characteristic. Other factors will often weigh more heavily in the choice of an approximate natural frequency and damping ratio. For example, it may be desirable to approximate the low frequency phase characteristic accurately and, to do so, a second-order transfer function which does not correlate with the 90° phase point may be used. A good deal of judgment must, therefore, be exercised to select the most appropriate transfer function approximation.

SERVOVALVE TRANSFER FUNCTIONS

Appropriate transfer functions for standard Moog servovalves are given below. These expressions are linear, empirical relationships which approximate the response of actual servovalves when operating without saturation. The time constants, natural frequencies, and damping ratios cited are representative; however, the response of individual servovalve designs may vary quite widely from those listed. Nevertheless, these representations are very useful for analytical studies and can reasonably form the basis for detailed system design.

FLOW CONTROL SERVOVALVES

This basic servovalve is one in which the control flow at constant load is proportional to the electrical input current. Flow from these servovalves will be influenced in varying degrees by changing load pressures, as indicated in Figure 4. For null stability considerations, only the region of this plot about the origin need be considered. Here, the influence of the load on flow gain of the servovalve can be considered negligible. In general, the assumption of zero load influence is conservative with respect to system stability analyses.

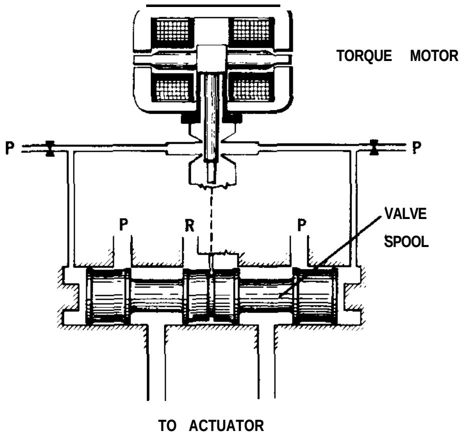


FIGURE 3

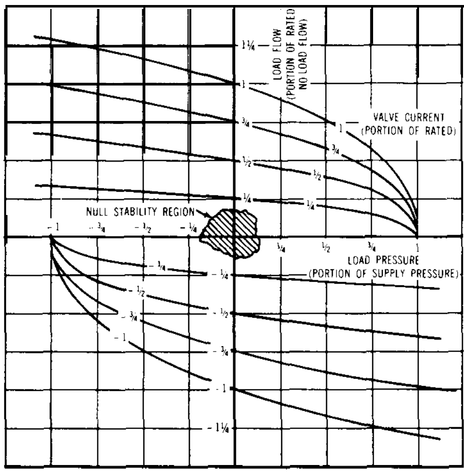


FIGURE 4

SYMBOLS FREQUENTLY USED

i	*differential current input to servovalve*	ma
Q	*servovalve flow to the load*	in^3/sec (cis)
p	*servovalve differential pressure output*	lbs/in^2 (psi)
K	*servovalve sensitivity, as defined*	
τ	*time constants*	sec.
ω_n	*natural frequencies*	rad/sec.
ζ	*damping ratios*	nondimensional
s	*Laplace operator*	

Another linearity assumption which is often made is that servovalve flow gain is constant through null. This is theoretically true for an ideal "zero lap" null cut of the valve spool; however, the actual lap condition will vary with production tolerances. If the spool becomes overlapped, the servovalve flow gain is reduced at null. Likewise, an underlap produces higher-than-normal servovalve gain. Normal production tolerances maintained at Moog hold the spool lap within ±0.0001 inch for all four null edges. This close control gives a very small range of possible nonlinear flow control through null (about ±3% for an "axis" null cut); but within this range, flow gain may be from 50% to 200% of nominal.

The change in servovalve flow gain at null may sometimes cause system instability; or, in other cases, poor positioning accuracy, or poor dynamic response of the actuator at low-amplitude input signals. This situation can be varied one way or the other by holding a nominal overlap or underlap, as appropriate.

The dynamic response of Moog flow control servovalves can be approximated in the frequency range to about 50 cps by the following first-order expression:

$$\frac{Q}{i}(s) = K\left(\frac{1}{1+\tau s}\right)$$

where

K = servovalve static flow gain at zero load pressure drop — cis/ma

τ = apparent servovalve time constant — sec

Standard flow control servovalves are available in several sizes and with many internal design configurations. The value of servovalve sensitivity K depends upon the rated flow and input current. Typically, for a 5 gpm valve at a rated 8 ma input current, K = 2.4 cis/ma.

The appropriate time constant for representing servovalve dynamics will depend largely upon the flow capacity of the valve. Typical time constant approximations for Moog Type 30 servovalves are given in the table below.

If it is necessary to represent servovalve dynamics through a wider frequency range, a second-order response can be used, as:

$$\frac{Q}{i}(s) \quad K\left[\frac{1}{1+\left(\frac{2\zeta}{\omega_n}\right)s+\left(\frac{s}{\omega_n}\right)^2}\right]$$

where

$\omega_n = 2\pi f_n$ apparent natural frequency — rad/sec

ζ = apparent damping ratio — nondimensional

Flow-Control Servovalve Series	Max. Flow Capacity at 3000 psi gpm	Approximate Dynamics 3000 psi 100°F P-P input = 50% rated 1st Order τ sec	2nd Order f_n cps	ζ
30	2	.0013	240	.5
31	6	.0015	200	.5
32	12	.0020	160	.55
34	18	.0023	140	6
35	30	.0029	110	.65

The first and second-order transfer function approximations for servovalve dynamic response listed in the above table give reasonably good correlation with actual measured response. It is possible to relate servovalve response to internal valve parameters, as discussed in Appendix I. However, the analytical approach to servovalve dynamics is most useful during preliminary servovalve design, or when attempting to change the response of a given design. It is better, and more accurate, for system design to use empirical approximations of the measured servovalve response.

PRESSURE CONTROL SERVOVALVES

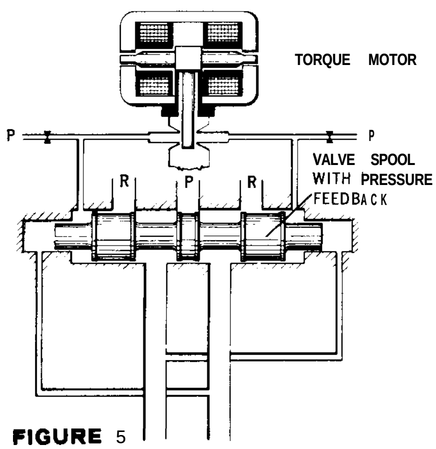


FIGURE 5

These servovalves provide a differential pressure output in response to the electrical input current. The static flow-pressure curves for a typical pressure control servovalve are shown in Figure 6. A small droop, or decrease in the controlled pressure with flow, does occur, even throughout the null region. This droop is usually small in pressure-control servovalves; however, in some applications even a small droop can significantly alter the system response. In pressure-flow servovalves, droop is purposely introduced. Transfer functions for these valves are discussed in the next section.

It is convenient to measure the dynamic response of a pressure control servovalve by capping the load lines and sensing the relationship of load pressure to input current. A second-order transfer function closely approximates the measured response in the frequency range to about 200 cps.

$$\frac{p}{i}(s) = K,\left[\frac{1}{1+\left(\frac{2\zeta}{\omega_n}\right)s+\left(\frac{s}{\omega_n}\right)^2}\right]$$

where

K, = pressure control servovalve static gain — psi/ma

$\omega_n = 2\pi f_n$ apparent natural frequency — rad/sec

ζ = apparent damping ratio — nondimensional

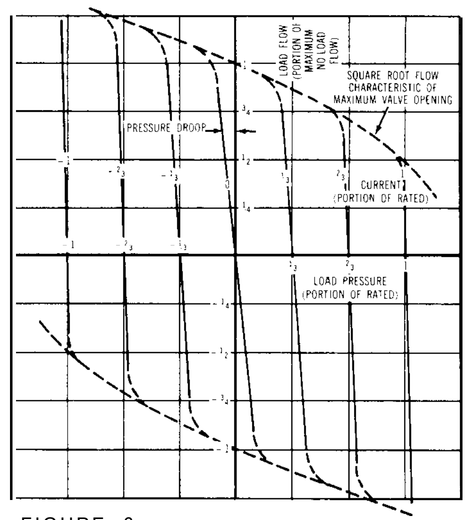


FIGURE 6

The controlled differential pressure may be any rated maximum up to the system pressure. For a 1000 psi rated control pressure at 8 ma electrical input, K, = 125 psi/ma.

With a blocked load, the apparent natural frequency for pressure control servovalves is approximately 250 cps, and the damping ratio is about 0.3 to 0.5. The actual blocked-load response for a pressure-control servovalve depends somewhat on the entrapped oil volume of the load, so the load volume should be noted with response data.

When a pressure control servovalve is required to supply flow to the load, the blocked-load transfer function no longer adequately describes servovalve response. Instead, the output pressure is determined by the concurrent values of input current and load flow. A linearized approximation to the overall servovalve dynamic relationship can be established by superposition. Thus, the pressure response to input current and the pressure response to load flow can be considered separable and non-interacting relationships. The transfer fuctions which result from this linearization are easily mechanized on analog computers, and system design based on these assumptions has proven to be valid for most cases.

The response of a pressure control servovalve to load flow at constant input current can be measured by techniques described in Appendix II. The characteristic response is approximated by the following transfer function:

$$\frac{p}{Q}(s) = -K_2\left[\frac{1+\tau s}{1+\left(\frac{2\zeta}{\omega_n}\right)s+\left(\frac{s}{\omega_n}\right)^2}\right]$$

where

K_2 = static droop characteristic psi/cis

τ = equivalent droop time constant sec

The droop constant K_2 represents the slope of the flow-load curves of Figure 6. Values for K_2 generally fall in the range 20 to 50 psi/cis. Specially designed spools can be utilized in pressure control valves to reduce the droop to almost zero. The equivalent droop time constant has a significant effect on the stability of many servo systems. Typical values correspond to a corner frequency near 10 cps, or 0.016 second. The second-order response of the droop transfer function is a high frequency effect, usually having a natural frequency near 200 cps.

For system design, a simplified overall transfer function which includes both blocked load and flow droop effects can be used. This would be:

$$p(s) = \left[K_1 i - K_2(1+\tau s)Q\right] \cdots \left[\frac{1}{1+\left(\frac{2\zeta}{\omega_n}\right)s+\left(\frac{s}{\omega_n}\right)^2}\right]$$

where

K_1 = blocked-load pressure sensitivity psi/ma

K_2 = zero-load pressure droop psi/cis

τ = droop time constant about 0.016 sec.

ω_n = apparent servovalve natural frequency about 200 cps

ζ = apparent servovalve damping ratio about 0.5

For the analysis of most physical systems, the second-order bracketed term on the right can be replaced by a suitable first-order lag, thus simplifying the expression still further.

PRESSURE-FLOW CONTROL (PQ) SERVOVALVES

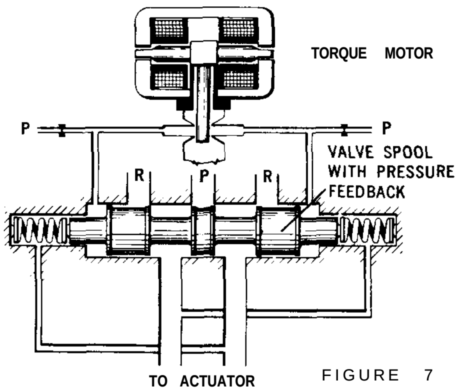


FIGURE 7

These servovalves combine the functions of pressure and flow control to provide characteristics which contribute effective damping in highly-resonant loaded servo systems. Flow from these servovalves is determined not only by the electrical input signal, but also by the differential load pressure. For a linear transfer function approximation to dynamic response, it may again be assumed that principles of superposition prevail. With this assumption, flow from the servovalve may be considered separately dependent upon input current and load pressure.

For most pressure-flow servovalves, the dynamic response of each flow relationship (i.e., flow to current, and flow to load pressure) can be approximated by a critically damped, second-order transfer function. In addition, it has been found experimentally that these dynamic responses are nearly equal. The assumption of identical dynamics further simplifies the overall transfer function, so that the dynamic performance expressed mathematically becomes:

$$Q(s) = (K_1 i - K_2 p)\left(\frac{1}{1+\frac{s}{\omega_n}}\right)^2$$

where

K_1 = servovalve sensitivity to input current cis/ma

K_2 = servovalve sensitivity to load pressure cis/psi

ω_n = equivalent servovalve natural frequency; critically damped rad/sec

The static flow-pressure characteristics for pressure-flow servovalves exhibit a nearly linear relationship between flow, current, and pressure as shown in Figure 8. A wide variation in the sensitivities K_1 and K_2 is possible by appropriate selection of internal valve parameters. Normally, these constants are selected to suit the individual requirements of specific servo systems. Typical values might be K_1 = 3.0 cis/ma and K_2 = 0.02 cis/psi. The equivalent natural frequency for these pressure-flow control servovalves is generally about 100 cps.

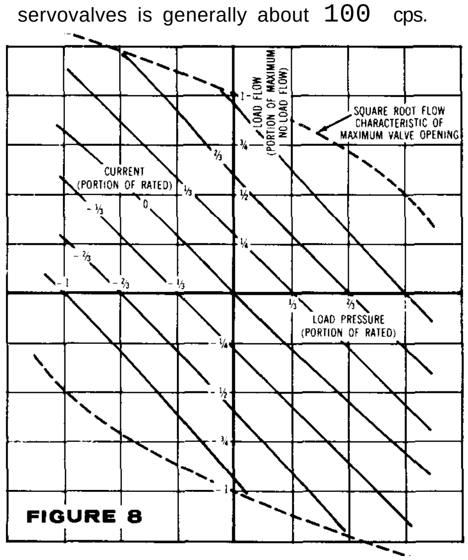


FIGURE 8

DYNAMIC PRESSURE FEEDBACK (DPF) SERVOVALVES

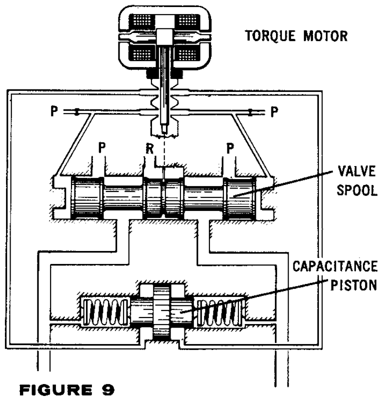


FIGURE 9

TO ACTUATOR

These servovalves function as pressureflow control valves under dynamic conditions, but act as pure flow control servovalves in the steady state, or statically. As such, the benefits of pressure feedback are realized as damping for a resonant load, but statically the system retains the high resolution and stiffness characteristics obtained with a flow control servovalve.

The dynamic performance of DPF servovalves is best represented for system analysis by the following expression, which presumes linearity by superposition:

$$Q(s) = \left[K_1\, i - K_2\left(\frac{\tau s}{1+\tau s}\right)p\right]\left(\frac{1}{1+\frac{s}{\omega_n}}\right)^2$$

where

K_1 = servovalve sensitivity to input current — cis/ma

K_2 = magnitude of the dynamic pressure feedback — cis/psi

τ = time constant of the dynamic pressure feedback filter — sec

ω_n = equivalent servovalve natural frequency; critically damped — rad/sec

The values of K_1, K_2, and τ can be set throughout wide ranges by choice of internal servovalve parameters. Most practically, these values are selected for each specific system application and will be influenced by various system requirements (e.g., system frequency or transient response, static and quasi-static system stiffness, system accuracies, etc.). Typical values of K_1, and K_2 would be similar to those mentioned for the pressure-flow control servovalves. Values for τ generally correspond to a corner frequency of about 1/3 the load resonant frequency. For instance, with a 10 cps resonant frequency load, τ would be set to approximately:

$$\tau = \frac{1}{2\pi\ (10/3)} = 0.05 \text{ sec.}$$

The equivalent natural frequency of DPF servovalves is generally about 100 cps.

The frequency sensitive filter $\left(\frac{\tau s}{1+\tau s}\right)$ which operates on the feedback pressure is created by hydraulic orifices and capacitive elements. As such, the actual filter response is nonlinear, reflecting the square root relationship of hydraulic orifices. The result is an apparent change in the filter time constant, τ with the amplitude of the load pressure. This nonlinearity gives a longer time constant with increasing amplitude of the load pressure variation. A longer time constant allows more effective pressure feedback; hence, more system damping. As the amplitude of the load pressure variations approach zero, the pressure feedback time constant becomes shorter. Theoretically this could lead to system limit cycle oscillations; however, in practice this seldom occurs due to finite frictions in the actuator and load. A conservative design approach is to size the DPF filter for adequate pressure feedback at a reasonably low amplitude of load pressure variations; for instance, equal to 1/10 supply pressure. This allows a linear analysis of system performance using the relationships given previously, and gives good agreement with actual system dynamic response.

STATIC LOAD ERROR WASHOUT (SLEW) SERVOVALVES

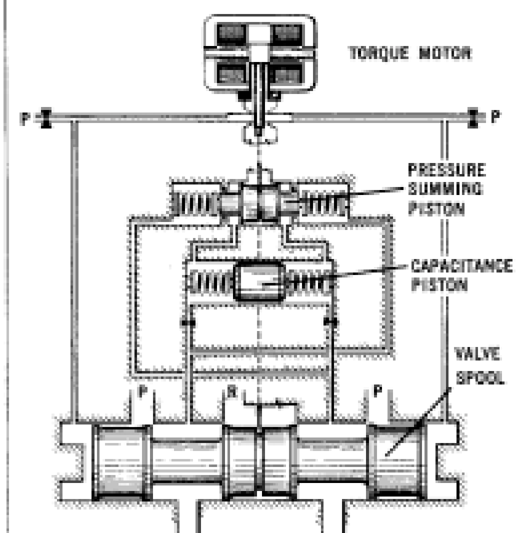


FIGURE 10 TO ACTUATOR

These servovalves represent a further extension of the pressure-feedback technique for damping resonant-loaded servosystems. In PQ servovalves the pressure feedback is proportional, so acts for static, as well as dynamic, load forces. Under static loading, the pressure feedback produces a servoloop error which must be offset by a corresponding position signal. The effect is that of apparent compliance of the servoactuator; that is, the actuator position will change proportional to load force. The DPF servovalve washes-out the static pressure feedback, and essentially removes the position error due to actuator compliance.

In most resonant-loaded servosystems, the position feedback signal is derived from the actuator. This means that the physical compliance of the structure, which is contributing to the basic resonance problem, is outside the position servoloop. This compliance causes errors in position of the load under static load-holding conditions, the same as caused by actuator compliance. SLEW servovalves include an additional static pressure feedback effect to correct for load position errors caused by structural position errors.

This is accomplished by combining two pressure feedbacks within the SLEW servovalve; one, a conventional proportional pressure feedback which has negative polarity for damping of the load resonance; and the other, a positive pressure feedback which causes the actuator to extend under static loading to correct for position errors from both actuator and load compliant effects. The positive pressure feedback path includes a low-pass filter, so that it is effective only under quasi-static loading conditions.

The following transfer function can be used to represent the SLEW servovalve:

$$Q(s) = \left[K_1 i - \left(K_2 - \frac{K_3}{1 + \tau s}\right) p(s)\right] \left(\frac{1}{1 + \frac{s}{\omega_n}}\right)^2$$

where

K_1 = flow sensitivity to input current — cis/ma

K_2 = proportional pressure feedback sensitivity — cis/psi

K_3 = positive pressure feedback sensitivity — cis/psi

τ = time constant of low-pass filter — sec

ω_n = equivalent natural frequency of servovalve; critically damped — rad/sec

Note that if $K_2 = K_3$, then the SLEW servovalve becomes equivalent to the DPF servovalve. However, if $K_3 > K_2$, then SLEW correction becomes effective. Normally K_2 is set to give the desired load damping, and will have a value corresponding to that used in either the PQ or DPF servovalves. The value of K_3 is generally about 1.5 K_2 but should be set for the actual compliance of the servosystem. The corner frequency, f_c, of the low-pass filter (where $\tau = \frac{1}{2\pi f_c}$) is generally about one-tenth the load resonant frequency. Actual values for the pressure-feedback parameters are best established through analog computer study of the complete actuation system.

ACCELERATION SWITCHING (AS) SERVOVALVES

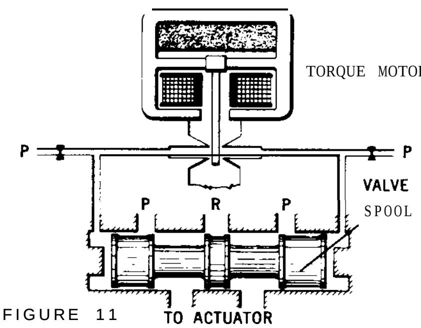


FIGURE 11

In these servovalves, the conventional proportional input current to the first stage is replaced by an alternating switching action. Control of the relative on and off time durations of the positive and negative input currents produces a corresponding rate of change of flow from the servovalve. In transfer function notation, this response is:

$$\frac{Q}{i_t}(s) = \frac{K}{s}$$

where

K = acceleration switching servovalve gain — $\frac{in^3/sec^2}{\text{time unbalance}}$

i_t = time unbalance of the current input — nondimensional

This relationship is valid for system analysis throughout the frequency range to near the switching frequency, which is generally 150 cps or higher. It must be pointed out, however, that servo performance for signal information which approaches the switching frequency is limited by the sampled-data nature of the system. Typical values for K would be from 100 to 1000 in^3/sec^2.

The s in the denominator of the transfer function indicates that the servovalve acts as an integrator. This is true under most conditions of operation. A nonlinearity which occurs is apparent when the servovalve is required to pass flow to the load. Due to flow-reaction forces acting on the valve spool, the true integral effect created by the spool is upset. This means that in the steady-state a finite electrical signal is required to sustain valve flow. A constant input velocity to an acceleration switching servo will, therefore, create a small, but finite, steady error.

With constant-amplitude sinusoidal inputs to a switching amplifier and AS servovalve supplying flow output, this nonlinearity appears as a break in the slope of the amplitude ratio of the servovalve response. Below the break, the response exhibits a zero-slope or proportional amplitude ratio. The frequency of this break point is dependent upon internal valve parameters and the amplitude of the sinusoidal input. In well-made servovalves, the break frequency is generally below 1 cps; so, for system stability and dynamic response considerations, this effect is unimportant and can be ignored.

APPENDIX I

ANALYTIC ANALYSIS OF SERVOVALVE DYNAMICS

It is possible to derive meaningful transfer functions for electrohydraulic servovalves, and several papers have reported such work (ref). Unfortunately, servovalves are complex devices and have many nonlinear characteristics which are significant in their operation. These nonlinearities include: electrical hysteresis of the torque motor, change in torque-motor output with displacement, change in orifice fluid-impedance with flow and with fluid characteristics, change in orifice discharge coefficient with pressure ratio, sliding friction of the spool, and others.

Many servovalve parts are small so have a shape which is analytically non-ideal. For example, fixed inlet orifices are often 0.006 to 0.008 inch in diameter. Ideally, the length of the orifice would be small with respect to its diameter to avoid both laminar and sharp-edge orifice effects; however, this becomes physically impractical with small orifices due to lack of strength for differential pressure loading, and lack of material for adequate life with fluid erosion. Therefore, the practical design from the performance standpoint is not necessarily the ideal design from the analytical standpoint.

Experience has shown that these non-linear and non-ideal characteristics limit the usefulness of theoretical analysis of servovalve dynamics in systems design. Instead, the more meaningful approach is to approximate measured servovalve response with suitable transfer functions, as discussed in the body of this technical bulletin.

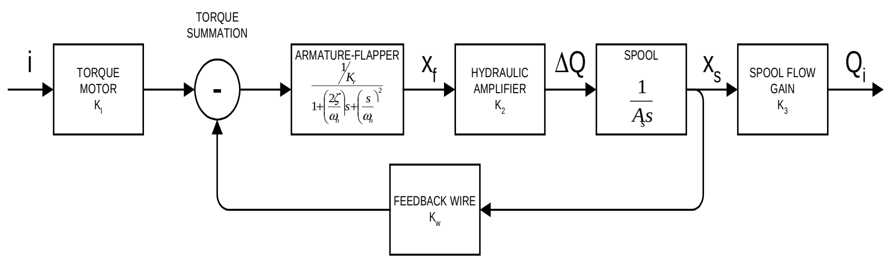
The analytic representation of servo-valve dynamics is useful during preliminary design of a new valve configuration, or when attempting to alter response of a given design by parameter variation. Analysis also contributes to a clearer understanding of servovalve operation.

Rather elaborate analyses of servovalve dynamic response have been performed at Moog, including computer studies which involve several nonlinear effects, and up to eight dynamic orders (excluding any load dynamics). Unfortunately, these complex analyses have not contributed significantly to servovalve design due to uncertainties and inaccuracies associated with the higher-order effects.

These analyses have been extremely useful when reduced to their simpler form. A very adequate transfer function representation for the basic Type 30 mechanical feedback servovalve is given in Figure 12. This simplified representation results from the following assumptions:

1. An ideal current source (infinite impedance) is used.
2. Negligible load pressure exists.
3. All nonlinearities can either be approximated by linear dynamic effects, or can be neglected.
4. The armature/flapper can be represented as a simple lumped-parameter system.
5. Perturbation conditions can be applied to the hydraulic amplifier orifice characteristics.
6. Fluid compressibility and viscosity effects are negligible.
7. Motions of the flapper are small with respect to spool motion.
8. The forces necessary to move the spool are small with respect to the driving force available.

The last assumption implies that the differential pressure across the spool is



SIMPLIFIED SERVOVALVE BLOCK DIAGRAM

FIGURE 12

negligible during dynamic conditions. If so, then spool mass, friction, flow forces, and other spool force effects can be neglected. At first this assumption may seem unreasonable; but it can be shown to be quite valid, and the simplification which results more than justifies its use.

The simplified block diagram is a third order system consisting of the armature/ flapper mass, damping and stiffness, together with the flow-integration effect of the spool. The spool, in this case, is analogous to the piston of a simple position servoloop.

The rotational mass of the armature/ flapper is quite easy to calculate. The effective stiffness of the armature/flapper is a composite of several effects, the most important of which are the centering effect of the flexure tube, and the decentering effect of the permanent magnet flux. The latter is set by charge level of the torque motor, and is individually adjusted in each servovalve to meet prescribed dynamic response limits. The damping force on the armature/flapper is likewise a composite effect. Here, it is known from experience that the equivalent ζ is about 0.4.

The hydraulic-amplifier orifice bridge reduces to a simple gain term with the assumptions listed earlier. This gain is the differential flow unbalance between opposite arms of the bridge, per increment of flapper motion.

Internal loop gain of the servovalve is determined by the following parameters.

$$K_v = \frac{K_2 \; k_w}{k_f \; A_s} \qquad \text{sec}^{-1}$$

The hydraulic amplifier flow gain, K2, can be related to nozzle parameters by the following:

$$K_2 = c_o \;\; \pi d_n \sqrt{\Delta P_n}$$

where c_o = nozzle orifice coefficient
d_n = nozzle diameter
ΔP_n = nozzle pressure drop

Any of the loop gain parameters can be altered to change servovalve response. For example, the following changes would increase internal servovalve loop gain: (1) smaller spool diameter, (2) larger nozzle diameter, (3) higher nozzle pressure drop, (4) higher torque motor charge level. The higher torque motor charge gives a lower k_f which increases loop gain, but this also lowers the natural frequency of the first stage. Unfortunately, the directions of these two effects are not compatible in that higher loop gain cannot be used with a lower natural frequency first stage. Therefore, an optimum charge level exists which produces maximum loop gain for the stability margin desired.

TYPICAL PARAMETERS FOR 31 SERIES SERVOVALVE

i	=	torque motor current	± 10 ma
x_f	=	flapper displacement at nozzles	± 0.0012 in.
x_s	=	spool displacement	± 0.015 in.
ΔQ	=	hydraulic amplifier differential flow	± 0.18 in^3/sec
Q_L	=	servovalve control flow	± 4 gpm
K_1	=	torque motor gain	0.025 in-lbs/ma
K_2	=	hydraulic amplifier flow gain	$150 \frac{\text{in}^3/\text{sec}}{\text{in}}$
K_3	=	flow gain of spool/bushing	$1030 \frac{\text{in}^3/\text{sec}}{\text{in}}$
A_s	=	spool end area	0.026 in^2
k_f	=	net stiffness on armature/flapper	93 in-lbs/in
k_w	=	feedback wire stiffness	13.5 in-lbs/in
b_f	=	net damping on armature/flapper	$0.016 \frac{\text{in-lbs}}{\text{in/sec}}$
I_f	=	rotational mass of armature/flapper	$4.4 \times 10^{-6} \frac{\text{in-lbs.}}{\text{in/sec}^2}$
ω_n	=	$\sqrt{\frac{k_f}{I_f}}$ natural frequency of first stage	730 cps
ζ	=	$\frac{1}{2} \frac{b_f}{k_f} \omega_n$ damping ratio of first stage	0.4
K_v	=	$\frac{K_2 \; k_w}{k_f \; A_s}$ servovalve loop gain	840 sec^{-1}

REFERENCES

"Describing Functions for Electrohydraulic Valves"
J. Zaborszky and H. J. Harrington
AIEE Transactions, Volume 76, Part I, May 1957 and Volume 76, Part II, January 1958

"Analysis of Electrohydraulic Valves and Systems"
R. S. Cataldo
ISA-11-60, Presented at JACC, Boston, Massachusetts, September 1960

"Mathematical Models for Time-Domain Design of Electro-Hydraulic Servomechanisms"
P. K. C. Wang
AIEE CP#61-82, Presented at New York, New York, January 29, 1961

"Damping the First Stage of a Two-Stage Electro-Hydraulic Servo Valve"
T. W. Thompson and P. F. Hayner
AIEE CP#61-154, Presented at New York, New York, January 29, 1961

APPENDIX II

MEASUREMENT OF SERVOVALVE DYNAMIC RESPONSE

Moog has developed highly specialized test equipment for production measurement of servovalve dynamic response. This appendix describes this equipment and expiains the design philosophy which insures accurate and consistent dynamic response measurements.

Servovalve dynamic response is defined as the relationship of output to input with all other operational variables held constant. This relationship is conveniently described in terms of the amplitude ratio and phase angle of the output in response to a sinusoidal input of varying frequency. The input to Moog servovalves is usually considered to be the differential current between the two motor coils. The two coils may be connected in series aiding; or the valve may be supplied with a single two-lead coil, in which case the input becomes the absolute current in the coil.

Servovalve output is normally considered the primary controlled variable; e.g., flow to the load or pressure in the load lines. Measured servovalve output is dependent upon the nature of the load, so it is important for consistent and useful response information to maintain precisely known loading conditions. The philosophy of dynamic testing at Moog is to attempt to maintain an ideal load. In this way, the servovalve response is most completely isolated with the valve operating as a single component. Practically, this philosophy is well suited to servovalve dynamics, for the measured information is applicable to all systems and is consistent if measured at different times and with different pieces of test apparatus.

The ideal load for a flow control servovalve would be massless and frictionless, presenting absolutely no obstruction to flow from the servovalve. In practice, this "no load" operation can be approached with sufficient purity to assure no detectable influence due to loading. The mechanization of this loading condition is described in detail later.

For a pressure control servovalve, the ideal load would have zero flow and zero compliance. This so called "blocked load" is easily obtained with low compliance, capped load lines.

In servovalves which utilize load pressure feedback (i.e., pressure control, pressure flow, dynamic pressure feedback and SLEW servovalves), the output is determined not only by the coil current, but also by the action of the pressure feedback. The principle of superposition may be employed to individually measure the servovalve response to current with no load, and the response to load variations with zero or constant current. Techniques for the latter test represent an extension of those described here in that the appropriate response is measured while a forced hydraulic load is applied to the servovalve under test.

DYNAMIC TEST EQUIPMENT

A simplified schematic for production dynamic test equipment is presented in Figure 13. Photographs of a control console and test actuator appear in Figures 14 and 15, respectively.

The servovalve under test is driven with sinusoidal input current from the electronic oscillator. The amplifier cir-

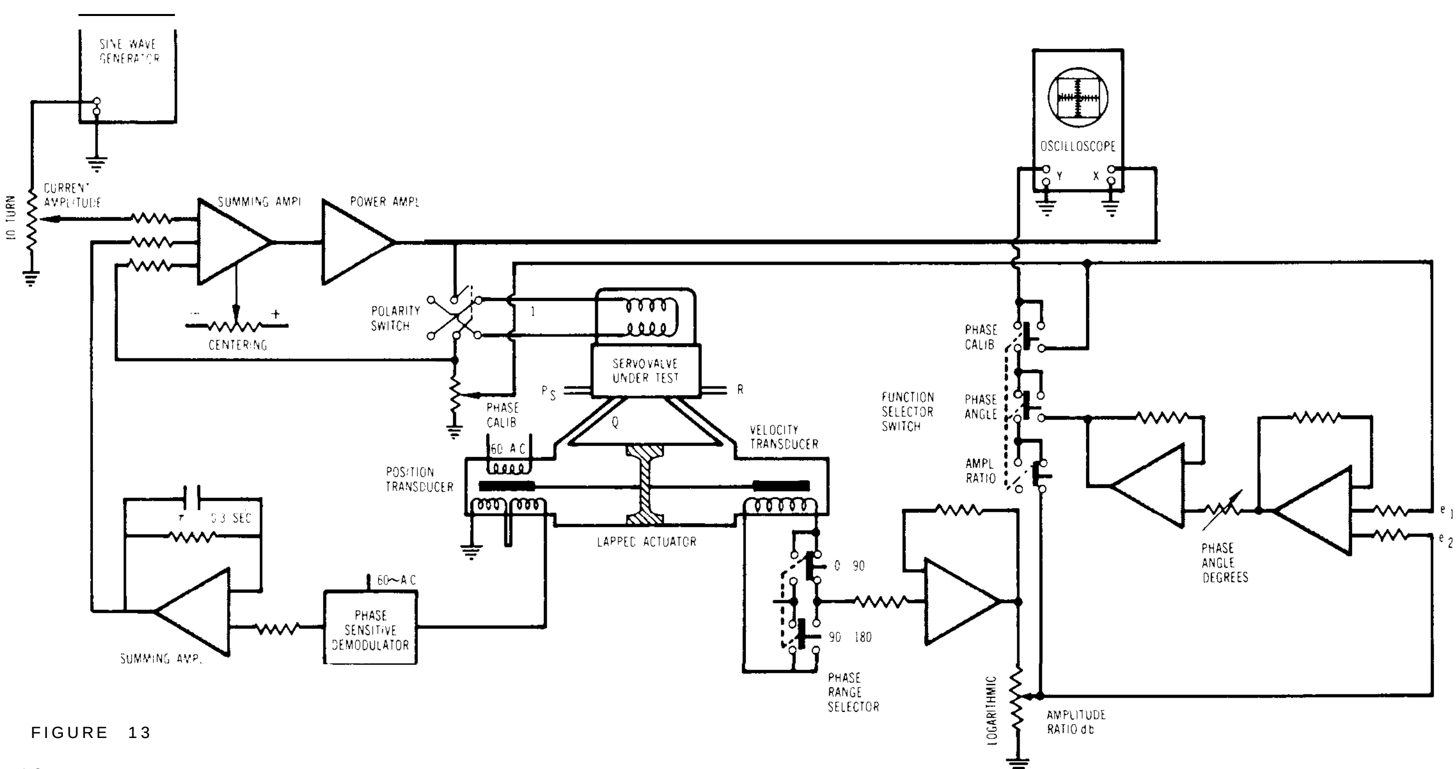


FIGURE 13

cuitry which supplies the input current to the servovalve operates with a high degree of current feedback. Operation with current feedback is essential for precise testing for several reasons, including:

1. The dynamic effect of servovalve coil inductance, which would otherwise upset the proportional relationship of coil current to command input voltage, is virtually eliminated.
2. Purity of the sinusoidal input current is obtained throughout the response range (to several hundred cps).
3. Calibrations of input current per unit input voltage are unaffected by changing from valve-to-valve, regardless of torque motor coil resistance.
4. The long term amplifier stability and accuracy associated with high feedback are attained.

As indicated in Figure 13, the torque motor coils are connected in series aiding in this test equipment. Servovalve operation in this manner is identical to operation with push-pull differential cur. rent, with or without quiescent coil current. The series coil connection is used only for simplicity of the electrical circuitry.

For dynamic flow tests, servovalve output is plumbed to a test actuator. The actuator has a lap-fit, extremely low mass piston which has no end seals or rod-end bearing surfaces. Instead, small rods extend from each face of the piston to carry, respectively, the core for a variable reluctance position transducer and the core for a translational motion velocity generator. Both transducers are immersed in oil so that no external piston seals are required.

The velocity transducer generates a voltage which is proportional at each instant to the flow from the servovalve. This indication of output flow is not limited by resolution of the electrical sensor nor is the signal reduced in amplitude for small excursions of the test actuator. The position transducer signal is utilized for continuous centering of the piston. This position signal is supplied to the servovalve by a low pass, low gain, negative feedback loop so the average actuator position is maintained near center.

Electrical signals from the piston velocity transducer, which indicate output flow, are passed through an isolating amplifier, then to a logarithmic potentiometer. The potentiometer dial is calibrated directly in decibels (db). Using the oscilloscope as an error detector, the operator is able to compare the electrical output signal at any test frequency with a reference signal amplitude which is proportional to the input current. The db potentiometer can then be adjusted for zero error. The amplitude ratio of servovalve output to input current is directly indicated by the calibration of the db dial.

The phase angle of servovalve response is determined by appropriate summation of the output flow and the input current signals. With sinusoidal current and flow signals, the amplitude of the summation signal is related to phase angle. This is seen by the following:

Input current signal $e_1 = A \sin \omega t$

Output flow signal $e_2 = B \sin(\omega t + \theta)$

where A and B are constants

θ is the relative phase angle

If these signals are subtracted;

$$e_1 - e_2 = \left[2A \sin\left(\frac{\theta}{2}\right)\right]\left[\cos\left(\omega t - \frac{\theta}{2}\right)\right]$$

(utilized for phase angles from 0 to 90°)

and by adding the signals:

$$e_1 + e_2 = \left[2A \cos\left(\frac{\theta}{2}\right)\right]\left[\sin\left(\omega t - \frac{\theta}{2}\right)\right]$$

(utilized for phase angles from 90 to 270°)

In each case, the amplitude of the summed signal is related to the phase angle between the signals, regardless of the signal frequency.

Circuitry for performing the phase measuring function utilizing these relationships is included in the test equipment. Following the amplitude ratio measurement, the operator depresses the appropriate phase selector button. The amplitude of the oscilloscope display is then adjusted to the reference amplitude by rotating the phase dial. Rotation of this dial adjusts the gain of the sum-difference amplifier. The dial is calibrated directly in degrees of phase angle which can be recorded by the operator.

[redacted]

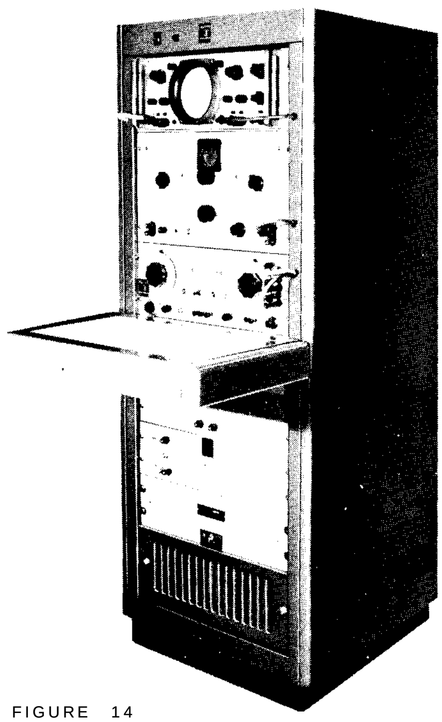


FIGURE 14

[redacted]

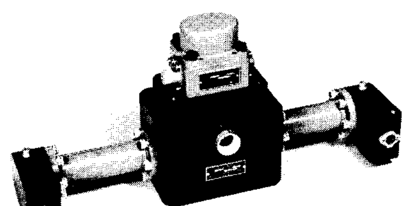


FIGURE 15