A long-stroke hydraulic cylinder does not necessarily have the optimum response characteristics for hydraulic automatic gage control (HAGC) because of oil compressibility leading to a substantial reduction in mill modulus. A double-acting cylinder with adjustable back-pressure provides fast roll gap opening and enhances system stability. A mathematical model has been constructed that simulates the dynamic characteristics of the HAGC system.

Evaluation of dynamic characteristics of HAGC system

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MORE than three decades ago, when high-pressure hydraulic servomechanisms started to become popular, the basic analytical work to simulate the servo system also began to be developed and results reported in the literature.[1-3] However, these studies only concentrated on relatively light duty systems consisting of a servovalve, with a small or even zero spring force, and double-acting hydraulic cylinders. Often these systems had a low natural frequency (5 to 20 Hz), low damping ratio and a low hydraulic pressure.

It was not until the 1960's and 1970's that servo systems were introduced into the steel industry for heavy duty gage control in rolling mills.[4] The first application was for the so-called, constant gap prestressed mill. The most important development since that application has been the introduction of closed-loop electro-hydraulic servocontrols.[5-7]

Then, over the past decade with the rapid development of electronics and modeling techniques, the applications of hydraulic automatic gage control (HAGC) became a proven technology. At the same time, the increased demand for high quality flat rolled products stimulated research efforts to further improve system efficiency and accuracy. Most of these studies concentrated on system design for gage control.[8-11]

However, because of the complexity of a control system for a tandem mill, simplified hydraulic systems were included in the overall control model that consisted of flow regulators that did not accurately simulate the behavior of the actual hydraulic components (ie, servovalves and cylinders). Although, the rationale for this approach can be justified on the basis of the system complexity, particularly in the case of large models, it suffers from the inability to evaluate the performance of the hydraulic system design.

The future enhancement of AGC systems is dependent on the hydraulic system. Gage control, even with an excellent control algorithm, cannot be perfected without a fast response, stable, hydraulic system. Currently, mathematical models for evaluating hydraulic systems are inadequate, particularly when compared with the sophisticated models on gage control system reported in the literature.

In recent developments of the HAGC system, the use of long-stroke hydraulic cylinders is promoted. However, it can be questioned whether the response of a long-stroke cylinder is as good as a short-stroke unit and, also, the role played by double and single-acting cylinders. The objectives of this article are to investigate the nonlinear effects of the hydraulic system and to compare the performance of various cylinder designs using both a position and pressure mode.

Theory

For a single-acting cylinder, Paul[12, 13] developed a mathematical model using ordinary differential equations to examine the stability effects for various pressure line lengths. In other articles, Laplace transform blocks were applied to quantitatively compare various hydraulic system designs.[14-16]

In this article, ordinary differential equations are first derived to explain the physical meaning of each hydraulic component followed by the generation of a state-space matrix using all equations.

A schematic drawing for an HAGC system that shows three different arrangements of a return line for a double-acting cylinder is illustrated in Fig. 1.

The six major dynamic components considered that form a complete hydraulic system are:

- Servovalve.
- Transmission line.
- Hydraulic cylinder.
- Mill (dynamics, natural frequency, modulus and damping effect).
- Return line.
- Sensor.

Control functions, such as lead-lag compensation, have been previously considered in the literature and are not included in this article.

Servovalve — The servovalve exhibits highly nonlinear characteristics. Generally, there are two stages of motion. A pilot stage follows the input signal to drive the torque motor

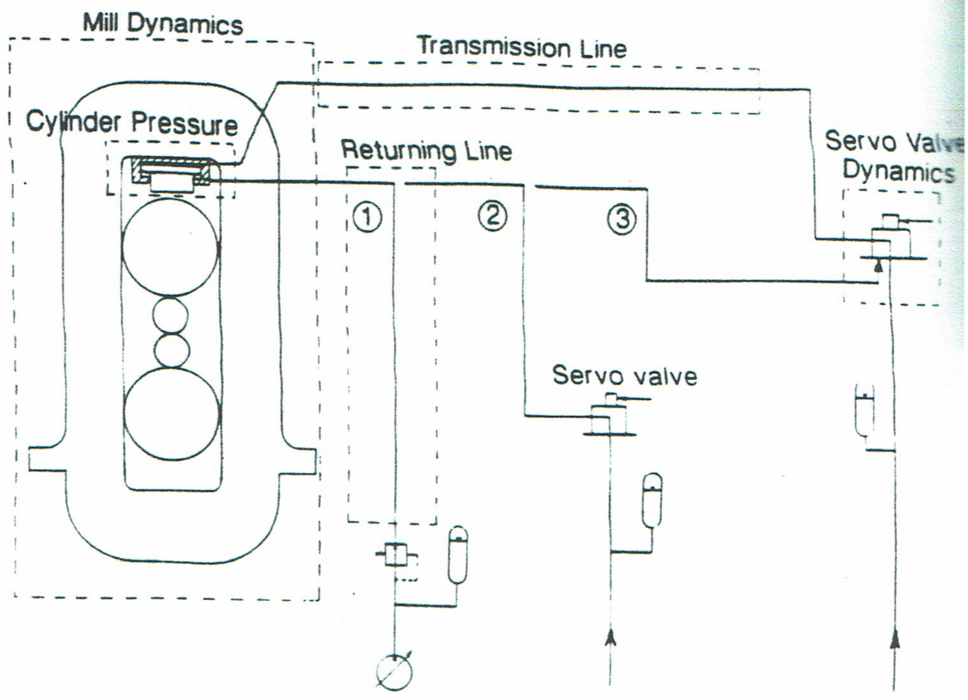


Fig. 1 — Hydraulic automatic gage control (HAGC) system.

which rotates the flapper to close or open the nozzles which, in turn, establishes the pressure to motivate the spool of the second stage. The displacement of the spool creates the opening that allows high-pressure fluid to pass. Spool position is then used as the feedback spring torque to counterbalance the motor torque due to input current.

Performance of a servovalve can be estimated from a frequency response test. Consequently, spool dynamics, including the torque motor, can be simulated by the following second-order differential equation which closely agrees with test results for frequencies less than 20 Hz:

$$\ddot{C} + 2\,\xi_s \omega_s^2\,\dot{C} = \omega_s^2\, I_c \tag{1}$$

where

C = servovalve opening due to spool motion
I_c = input torque motor signal
ω_s = servovalve natural frequency
ξ_s = servovalve damping coefficient

The natural frequency and damping coefficient can be obtained from the frequency response test provided by the servovalve manufacturer. Input current is taken from the feedback error signal amplified by a current gain factor:

$$I_c = K_c \left(\frac{u_y - u_x}{I_o} \right) \tag{2}$$

where

I_o = servovalve rated current
K_c = current gain amplifier
u_x = position transducer of mill displacement
u_y = control input signal

The equation for pressure mode operation is similar to the above equation.

Flow rate, q, is proportional to the valve opening and can also be determined by the pressure drop between two ports:

$$q = q_o\, C \sqrt{\frac{\Delta P}{500}} \tag{3}$$

where

P = servovalve pressure
P_s, P_t = tank pressure
ΔP = pressure difference
= $P_s - P$ for $C > 0$ (servovalve charging)
= $P - P_t$ for $C < 0$ (servovalve dumping)
q_o = servovalve rated flow rate

Using Taylor's expansion and neglecting the high order terms, the square root portion of equation (3) can be linearized. However, since the pressure drop changes every time the spool alternates its position, the system remains nonlinear.

Transmission line — The dynamic behavior of hydraulic transmission lines cannot be described by a simple, time dependent, differential equation. It is a distributed system and, therefore, depends on space and time. Although partial differential equations are the best approximation, they generate difficulties in matching with other dynamic components of the rolling system. A lumped model treats the hydraulic oil in the transmission line as a control volume.

Therefore, the transmission line can be characterized by ordinary differential equations rather than partial differential equations. Using the principles of mass and momentum conservation, the flow rate, q, is given by:

$$q = q_a + \frac{A_p L_p}{\beta_e}\, \dot{P} \tag{4}$$

where

A_p = area of pipe line
L_p = length of pipe line
q_a = flow rate into hydraulic cylinder
β_e = equivalent oil bulk modulus

The second term represents the flow rate due to oil compression. The pressure drop equation, which can be derived from the general form of Euler's equation of a potential compressible flow, provides the equation:

$$P - P_a = \frac{\rho L_p}{A_p}\, \dot{q}_a + R_p\, q_a \tag{5}$$

P_a = cylinder pressure
R_p = pressure drop coefficient
ρ = oil density

The first term comes from the inertia force of the oil in the transmission line while the second term represents the friction loss. Normally, fluid speed in the pipe line is small (order of −1) and the pressure drop coefficient, R_p, can be evaluated using the friction loss equation of the fully developed laminar pipe flow.

The equivalent bulk modulus, including oil bulk modulus, pipe elastic modulus and trapped gas effect can be calculated by:

$$\frac{1}{\beta_e} = \frac{1}{\beta_o} = \frac{Et_p}{d_p} + \frac{V_r}{\beta_g} \qquad (6)$$

where

d_p = pipe internal dia
E = Young's modulus of pipe material
t_p = pipe thickness
V_r = volume ratio of trapped gas to total fluid volume
β_g = gas bulk modulus
β_o = oil bulk modulus

Oil bulk modulus is not a constant but depends on oil pressure (high pressure has larger bulk modulus). Gas trapping occurs, inevitably, in operation and slows down the system response. Examinations are required from time to time to insure good working conditions.

Hydraulic cylinder — Cylinder pressure is determined by four major flow rates: cylinder flow, q_a; leaking flow, q_l; volume change rate due to compressed oil, q_c; and piston velocity, q_v. Combining these four terms, an equation for the cylinder pressure rate, $\dot{P}_a$ can be written:

$$\dot{P}_a = \frac{\beta_e}{V_0 + A_a x}\left[q_a - A_a \dot{x} - K_l (P_a - P_b)\right] \qquad (7)$$

where

A_a = cylinder area
K_l = leakage flow coefficient
P_b = cylinder back pressure
V_0 = initial cylinder volume
x = cylinder displacement
$\dot{x}$ = cylinder speed

Since cylinder displacement, x, is a term in the denominator, equation (7) is nonlinear. Also, for a single-acting cylinder, P_b does not exist and the leakage flow term, q_l, is dropped from the equation.

In modern designs of the HAGC system, the pressure line is kept as short as possible (less than 10 ft) and, in some cases, is eliminated by mounting the servovalve directly to the cylinder. Therefore, the pressure drop due to pipe line friction is negligible and the transmission line equations can be combined with equation (7) by summing the pipe volume with the initial cylinder volume. However, this will also ignore pressure wave propagation time and the natural frequency of the transmission line.

Mill dynamics — A mill dynamic equation is formulated on the basis of force equilibrium. There are six major components that contribute to the following equation: cylinder force; inertia force; damping force; spring force; Coulomb friction force; and mill load.

$$P_a A_a - P_b A_b = M\ddot{x} + B\dot{x} + Kx + f_c \,\mathrm{sgn}(\dot{x}) + f_x \qquad (8)$$

where

A_b = cylinder area, back-pressure side
B = mill damping coefficient
f_c = Coulomb friction force
f_x = rolling force
K = mill modulus
M = mill mass contribution to dynamic behavior

The rolling force term can be extended to connect with the strip deformation process, tension control loop and drive control system to establish a complete gage control model. Using the strip modulus concept, f_x can be rewritten as a linear function of x and merged with the spring force term. The rolling force, f_x, also plays a role of a balancing force in the initial condition to counteract roll weight and cylinder pressure force.

Mill dynamics includes the dynamics of the mill housing and rolls. They are continuous mediums and are too large to estimate as a lumped mass, which can represent the whole mill system. Since infinite numbers of dynamic modes exist, it is difficult to evaluate the mill mass and mill damping coefficient directly. Using mill modulus and mill natural frequency to derive the mill mass indirectly provides a better alternative.

Mill natural frequency — Although mill natural frequency can be measured by the hammer test, complex modes cannot be easily excited by the impact of the hammer and, consequently, detected. However, it is relatively easy to calculate these modes by the finite element method (FEM).

A simplified model using four degrees of freedom to simulate the entire mill indicates that the first resonant frequency is 125 Hz.[17] In a study that concentrates only on the top work and backup rolls, the first natural frequency is reported to be 79.7 and 75.6 Hz without and with strip in the mill.[18] The measured first natural frequency of a typical backup roll is 253 Hz and the calculated frequency by FEM is 226 Hz. In a model that considers only the roll stack (four rolls), the first natural frequency was found to be 84.5 Hz. In another model that considers one mill housing and uses beam elements to represent the rolls and chocks, the lowest first natural frequency was reported as 17.5 Hz.[19] Although the differences between models are large, the results suggest the following broad range of natural frequencies associated with a mill: backup roll, 200 to 250 Hz; roll stack, 70 to 120 Hz; and mill system, 20 to 150 Hz. The results also suggest that the frequency becomes lower as the model involves more machine elements. In addition, the natural frequency is affected by strip strength.

A complete FEM was generated by the author and associates to simulate mill dynamic behavior (Fig. 2). This model considers two mill housings and a roll stack by using spring elements, beam elements, 2-dimensional shell elements and pipe elements. The beam elements, shell elements and pipe elements carry their corresponding mass while the mass of the spring element is not taken into consideration. Beam elements are used to simulate the rolls and the shell elements for the housing. The pipe elements represent the screw, shims and other small or simple machine elements. Inclusion of spring elements vitalizes the dynamic activity in roll interfaces. There are eight generalized mass elements for the work and backup roll chocks. The model is based on the ANSYS finite element package. The first natural frequency was found to be 64 Hz without strip in the mill. A discussion of additional information obtained from the model, such as mode shape, is beyond the scope of this article.

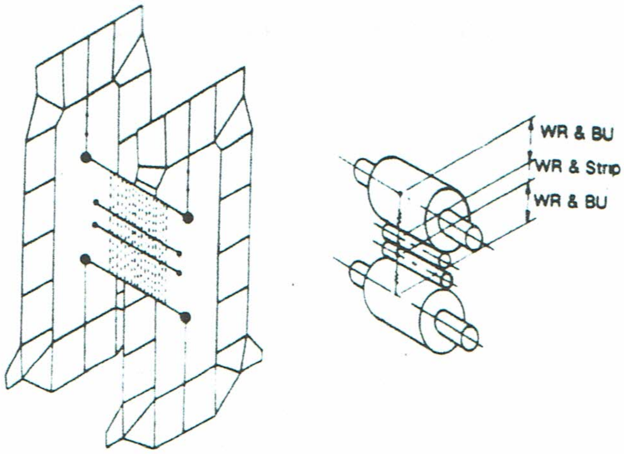


Fig. 2 — Finite element model for analysis of mill dynamics.

Mill modulus — Mill modulus is approximately 25 to 35 million lb/in. depending on mill design and the working stroke of the hydraulic cylinder. It decreases as the working stroke of the cylinder increases (Fig. 3). For example, a 5-in. stroke weakens the mill modulus by nearly 30%. A simple spring constant calculation that combines two springs in series provides a good approximation of this effect:

$$\frac{1}{M_m} = \frac{1}{M_o} = \beta_e \frac{L}{A} \tag{9}$$

where

A = cylinder area, high-pressure side
L = cylinder working stroke
M_m = mill modulus
M_o = mill modulus with zero working stroke

Since the oil working stroke is considered in the dynamic equation in this study, a rigid mill modulus with zero working stroke is adopted. The mill mass was then calculated using the mill natural frequency (64 Hz) and mill modulus (assumed 30 million lb/in.). It yielded an equivalent mill mass of 35.8 tons which is subsequently used in the study.

Mill damping factor — The mill damping force is generated from force interaction between the strip and work rolls. The length of arc contact is reported to play a key role in determining the damping force.[20] When the roll moves downward, the length increases and the separation becomes larger, which impedes the roll, and vice versa. Based on this mechanism, the fluctuation of arc contact length was formulated in terms of work roll angular velocity and delivery thickness. The damping force, D, was derived from the contact length:

$$D = \frac{SW}{\omega_p} \tag{10}$$

where

S = yield stress at delivery side
W = strip width
ω_p = work roll angular velocity, radians/s

In a cold mill, a higher angular velocity tends to decrease the damping force with a higher yield stress, conversely, tending to increase the damping force.

Pawelski[21] also found that the roll gap of a mill stand acts as a nonlinear damping element and that the damping force is a function of strip speed and strength (Fig. 4). Rolling force changes for mill stands No. 1 and 5 and rolling speeds of 1200 and 1800 metres/min, plotted against the roll displacement, demonstrated continuous hysteresis loops. Stand No. 5 had lower hysteresis and smaller force disturbances. In both cases, the area enclosed by the hysteresis loop, which is the energy dissipation per cycle, increased approximately 70% by reducing the rolling speed from 1800 to 1200 metres/min. Stand No. 1 provided a larger damping force and so did the lower mill speed. However, since stand No. 5 had a higher yield stress due to work hardening, according to equation (10), it should have a higher damping factor. Thus, there is a contradiction between equation (10) and the measured results. Strip thickness, as an important factor determining contact length, should be considered. Thicker strip, based on these results will help to stabilize the system.

Damping behavior of the rolling process is nonlinear, speed dependent, and material property and geometry sensitive. The calculated damping factor using equation (10) is only 0.15 for 50-in. wide strip at 3000 fpm and 50 ksi yield stress. Since the value varies accordingly, this study employed zero damping factor when evaluating system nonlinearity and assumed 0, 0.5 and 1.0 damping factors for performance comparison between single and double-acting cylinders.

Fig. 3 — Effect of working stroke of 32-in. dia hydraulic cylinder on mill modulus.

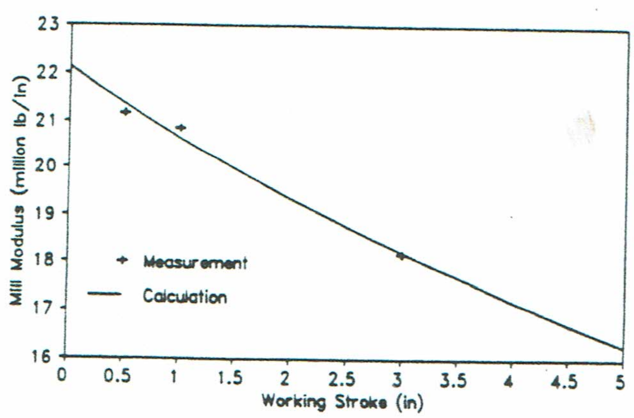
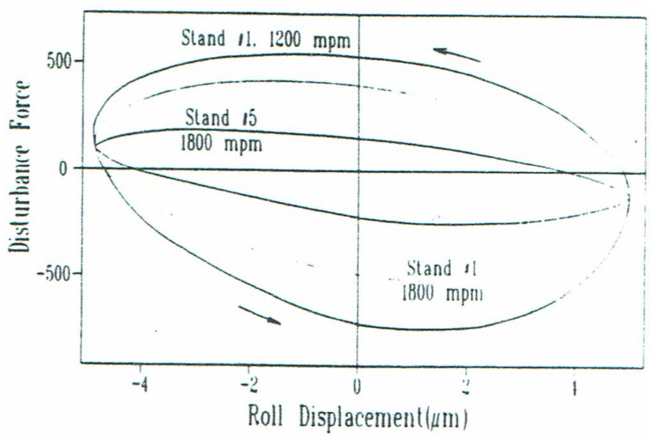


Fig. 4 — Mill damping force as a function of mill speed and material strength (as shown in mill stand number).[21]



Return line — As shown in Fig. 1, there are three different types of circuit design involving a double-acting cylinder. The first type is to connect the low-pressure side of the cylinder with the hydraulic oil tank with pressure regulating valve and a low-pressure pump usually installed to maintain back pressure. In the second type of design, the return line is connected with the servovalve which is also used for the high-pressure line. The four ports of the servovalve are all used in this design. The third type is to connect the return line with a separate servovalve.

Since two servovalves are used in the third type, the hydraulic cylinder can be prestressed by adjusting both servovalves. Normally, one valve operates in a position mode to set up the roll gap with the other valve running in a pressure mode to determine the pressure of the cylinder. Therefore, the oil is in compressed state and oil bulk modulus reaches the maximum value. Once the mill is subject to the rolling load, the return side servovalve instantly shifts to a full dump position and the whole system becomes the first type design. This is particularly useful in a plate mill with a large reduction using a long-stroke cylinder.

The second design type requires high-pressure seals in both sides of the cylinder where the friction due to the seal is expected to be higher than with a single-acting cylinder. Also, since four ports are used, with the same opening, the instantaneous flow rate would be $1/\sqrt{2}$ of the single-acting cylinder which utilizes only three ports.

The first type of design is the simplest involving double-acting cylinders. Since it connects to the tank and operates in a low pressure range, the fluid in the pipe line behaves as an incompressible flow. Using continuity equations together with the Bernoulli equation, the back pressure of the cylinder can be calculated by:

$$P_b = P_{bo} + \rho\, L_r \frac{A_b}{A_r}\ddot{x} + R_r\, \dot{x} \tag{11}$$

where

A_r = cross-sectional area of return line
L_r = total length of return line
P_{bo} = initial back pressure
R_r = pressure drop coefficient

For laminar flow, R_r is a constant and can be computed by the following equation:

$$R_r = 8\pi\mu\, L_{rp}\frac{A_b}{A_r^2} \tag{12}$$

where

L_{rp} = equivalent pipe length for pressure drop
μ = fluid dynamic viscosity

In case of turbulent flow, R_r becomes a function of $\dot{x}$ and the equation becomes nonlinear. Calculations show that most of the step and frequency response test cases and, even the real-time operation, experience laminar flow in the return line. This type operates in a similar manner to the single-acting cylinder except that a controllable damping factor is available due to the return line.

Sensor — The sensor is usually simulated by a first-order differential equation. The equations for both position and pressure transducer are similar:

$$\text{For position mode, } T_x\, \dot{u}_x + u_x = x \tag{13}$$
$$\text{For pressure mode, } T_p\, \dot{u}_p + u_p = P \tag{14}$$

where

T_p = time constant of pressure transducer
T_x = time constant of position transducer
u_p = pressure signal from pressure transducer
u_x = position signal from position transducer

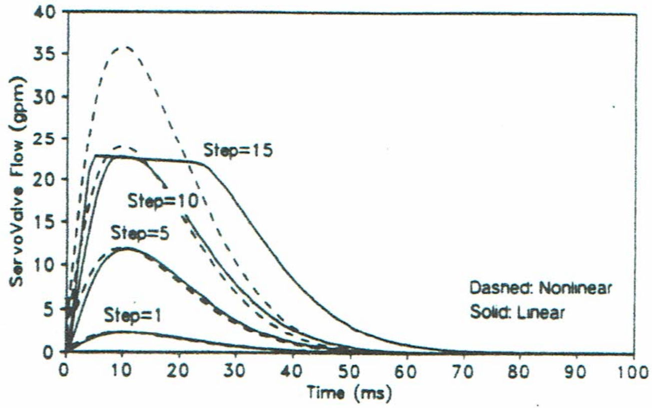
State-space matrix and numerical solution — The complete equation set for the hydraulic system is formed by combining equations (1) to (14). For a complete evaluation of gage performance of the control system, it may be necessary to consider not ony the hydraulic system as discussed in this article but also the drive system response, tension loop, rolling dynamics, control regulators and the interrelation between mill stands. However, this is outside the scope of this article. Instead, only the hydraulic system of an individual mill stand will be discussed.

There are six states for the system with direct mounted servovalves and eight states with the pressure line. Nondimensionizing each variable helps numerical stability. The state-space matrix can be built easily from these equations and will not be reiterated here. A computer program for both linear and nonlinear systems was generated using the state-space matrix and the fourth-order Runge-Kutta method. Time interval was set to 0.5 ms in the study. Some of the linear solutions, particularly the frequency response, were calculated using Matrix_x software package created by Integrated Systems, Inc.

Component characteristics

Servovalve saturation — Servovalve manufacturers indicate that an oversize servovalve will attenuate the system response because it has a larger time constant. On the other hand, an undersize servovalve will saturate the flow rate and also handicap the response. Solutions with and without servovalve saturation are shown in Fig. 5. The dashed and solid lines represent the linear and nonlinear solutions respectively. Although the linear and nonlinear results are similar for small step inputs less than 10 mils because the servovalve is not saturated, there are distinct differences caused by other nonlinear factors. A step input of 10 mils will just saturate the servovalve. Larger inputs, eg, 15 mils, will produce a large difference in the flow rate. The effect of servovalve saturation on response time is illustrated in Fig. 6. Time constants for step inputs of 1, 5, 10 and 15 mils are 18.7, 18.8, 19.2 and 25.1 ms respectively.

Fig. 5 — Servovalve flow rate for 1, 5, 10 and 15 mil step input (L_o = 0.05 in., damping = 0, gain = 2, P_s = 4500 psi, single-acting and indirect mounted servovalve).



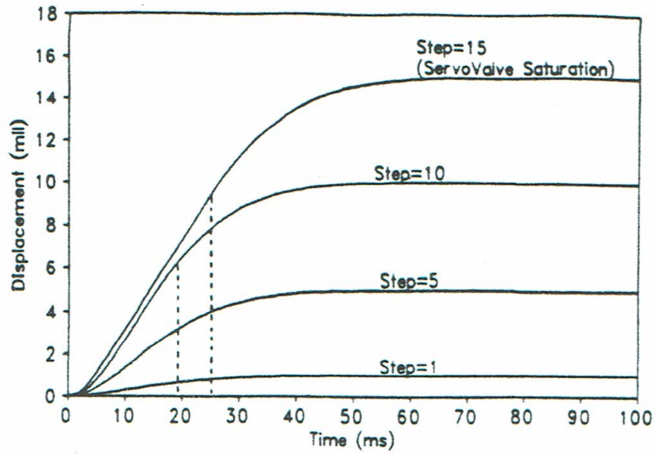


Fig. 6 — Nonlinear effects of servovalve saturation on system response time for 1, 5, 10 and 15 mil step input (L_o = 0.5 in., damping 0, gain = 2, P_s = 4500 psi, single-acting and indirect mounted servovalve).

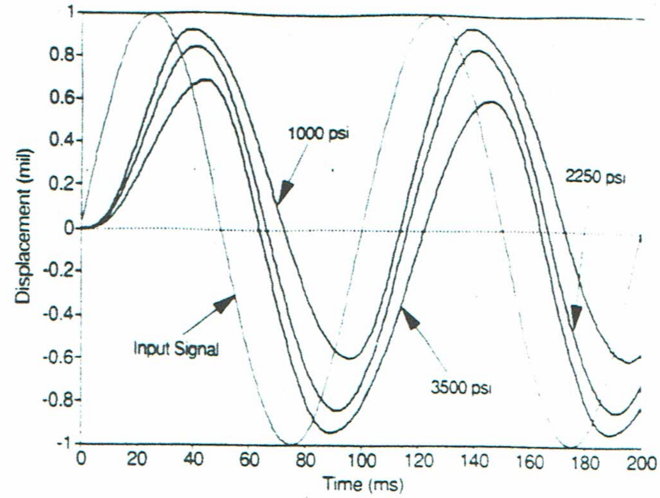


Fig. 7 — Nonlinear effects of cylinder pressure on step response time (L_o = 0.5 in., damping 0, gain = 2, P_s = 4500 psi, single-acting and indirect mounted servovalve).

For a completely linear system, the time constant should be the same, ie, 18.7 ms. For a small step input, the system operates almost linearly, However, a large input saturates the servovalve and the response deteriorates.

For a mill modulus of 30 million lb/in., a 10-mil step input is equivalent to 0.3 million lb which is 6.7% of a separation force of 4.5 million lb. Hence, if the set-up calculation error exceeds 6.7%, the AGC system response would be slower because of servovalve saturation.

Using two smaller size servovalves, one for the steady state and one for the transient state may take advantage of a fast response without servovalve saturation.

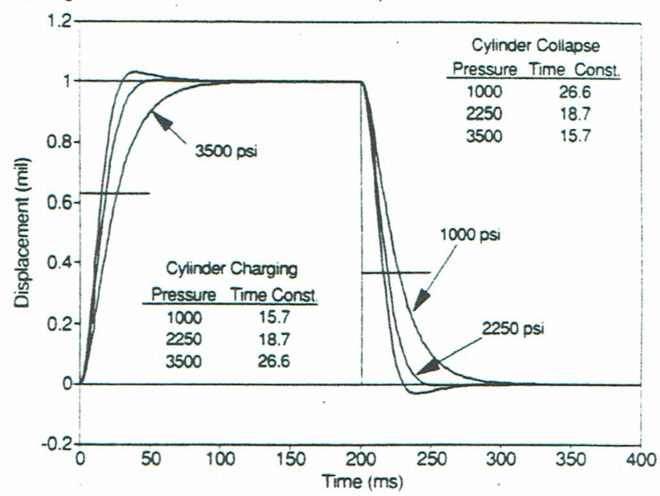
Servovalve flow rate and pressure drop — Flow rate is also proportional to the square root of the pressure drop. Depending on cylinder pressure, supply pressure and tank pressure, the pressure drop is different for charging and dumping conditions. Neglecting tank pressure, if the cylinder pressure is half of the supply pressure, then the pressure drop for both conditions is the same. However, if the cylinder pressure is larger than half of the supply pressure, the pressure drop is smaller for the charging condition than the dumping condition and vice versa. This phenomenon is illustrated in Fig. 7 and 8. A square wave response is obtain with cylinder pressures of 1000, 2250 and 3500 psi and a 4500 psi supply pressure (Fig. 7). This performance shows that the higher the cylinder pressure the slower is the response to expand the cylinder (or stretch the mill) and the faster the cylinder collapse. The response curve for 3500 psi during the cylinder expansion process is nearly the same as that of 1000 psi during the cylinder collapse process. The time constants for both expansion and collapse are the same for a pressure of 2250 psi, which is half of supply pressure.

A similar phenomenon occurs with the frequency response (Fig. 8). The highest cylinder pressure, 3500 psi, has worst response during expansion but the best response during collapse. The peak-to-peak amplitude ratios of the response curve to the input signal for 1000, 2250 and 3500 psi are 0.768, 0.841 and 0.768 respectively. A pressure of 2250 psi, which produces equal response with both conditions, gives the best amplitude ratio. Both low and high pressure cases display the same lower amplitude ratio. This implies that equalizing the response guarantees a better response. The easiest way to achieve this objective is to add an additional gain factor, which is the square root of the pressure difference ratio, $(P_s - P)/(P - P_t)$, for the cylinder collapse process.

Cylinder displacement — Cylinder displacement is another nonlinear term in equation (7). It expands the volume of the high pressure side to accommodate more oil and to relax cylinder pressure. Linear and nonlinear performance (without and with consideration of the cylinder displacement respectively), for step signals of 1, 5 and 8 mils are illustrated in Fig. 9 and 10. To insure no valve saturation, an 80-gpm servovalve with 1.5 current gain was used. The effect of cylinder displacement is almost zero for small signals of 1 and 5 mils and small for a larger signal of 8 mils (Fig. 9). The pressure rate, which is affected directly by the cylinder displacement according to equation (7) is shown in Fig. 10. All linear results have a high frequency oscillation (1000 Hz) and a larger amplitude than nonlinear. Cylinder displacement smooths the pressure increment but does not generate differences in displacement response. Since the responses are close, the displacement term is dropped from equation (7).

Nonlinear terms have been examined in the previous discussion, eliminated if inconsequential and linearized if

Fig. 8 — Nonlinear effects of cylinder pressure on frequency response (L_o = 0.5 in., damping 0, gain = 2, P_s = 4500 psi, single-acting and direct mounted servovalve).



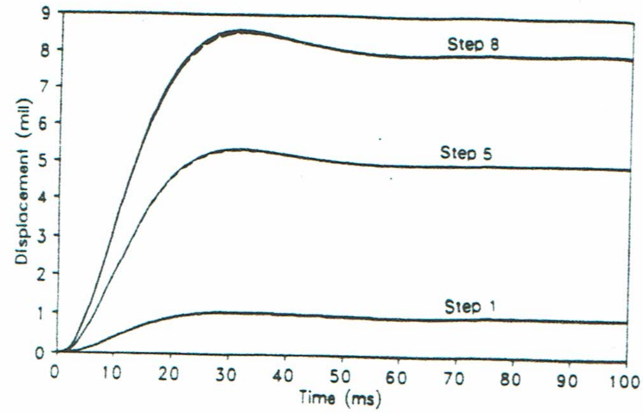


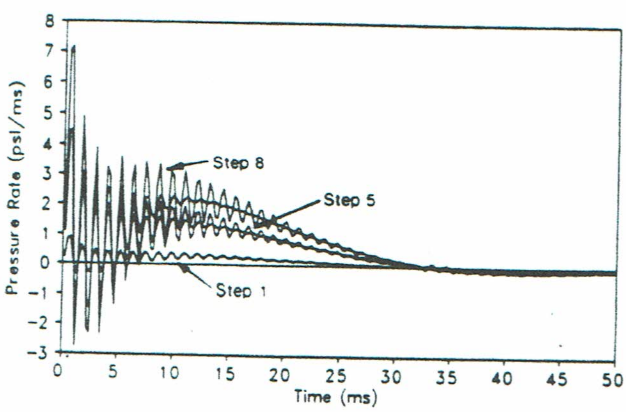
Fig. 9 — Linear and nonlinear effects of cylinder displacement on step response for 1. 5 and 8 mil step inputs. (L_o = 0.5 in., damping 0. gain = 1.5. P_s = 4500 psi. single-acting and direct mounted servovalve).

possible, by adding a control block. The effect of these nonlinear elements is small as long as the input signal is small. Uneven response of the servovalve can be corrected by an additional gain amplifier. For convenience, subsequent discussion is based on a small signal input and only linear results will be considered.

Pressure line length — Pressure line length is one of the most important factors affecting HAGC response; the longer the pressure line the slower the response. Mounting the servovalve directly to the AGC cylinder can eliminate the pressure line completely and, hence, improve the system response. The effectiveness of the pressure line can be examined from three aspects: oil volume of the pipe line; natural frequency of the transmission line; and travel time of the pressure wave front.

As mentioned previously, if the pipe line length is small enough to neglect the pressure drop due to friction, the oil volume of the pipe line can be considered as part of the cylinder volume. Studies based on this simplified assumption are illustrated in Fig. 11 to compare the step response times of two cases: zero pipe length (direct mounted); and a 20-ft pipe length. The top portion of the illustration shows almost no differences in the cylinder displacements between the two cases for gain factors of 2 and 8. Cylinder pressure, shown at the bottom of the illustration, exhibits

Fig. 10 — Linear and nonlinear effects of cylinder displacement on step response for 1. 5 and 8 mil step inputs. (L_o = 0.5 in., damping 0. gain = 1.5. P_s = 4500 psi. single-acting and direct mounted servovalve).



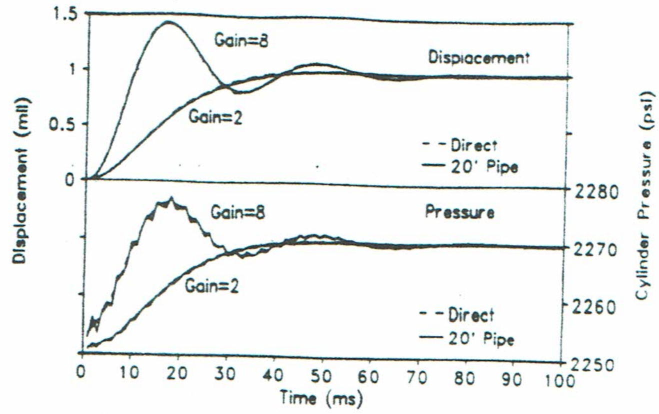


Fig. 11 — Effect of cylinder mounting types of cylinder displacement and pressure for 2 and 8 gain factors. (L_o = 0.1 in., damping 0, P_s = 4500 psi, single-acting).

a bumpy response for the 20-ft pipe length in the transient state.

The natural resonant frequency and the damping coefficient of the transmission line can be derived by combining equation (4) and (5):

$$\omega_l = \frac{1}{L_p}\sqrt{\frac{\beta_e}{\rho}} \tag{15}$$

$$\xi_l = A_p \frac{R_p}{\sqrt{2\,\beta_e\,\rho}} \tag{16}$$

where

ξ_l = damping coefficient of transmission line
ω_l = natural resonant frequency of transmission line

The natural frequency is inversely proportional to the pipe length. Since the pipe modulus is normally much larger than oil bulk modulus, the bulk modulus and density of the hydraulic fluid are the two key factors determining the equivalent bulk modulus. For an oil density of 0.0325 lb/cu in. and an equivalent bulk modulus of 260,000 psi, the natural frequency is equal to 737 Hz divided by the pipe length in feet. The first natural frequency of the mill is in the region of 60 to 100 Hz. With a pipe length in the range of 7 to 12 ft, the hydraulic resonant frequency approaches that of the mill frequency and the mill could be excited by the hydraulic system. Although the damping coefficients may help to stabilize the system, care must be exercised to keep the natural frequency of the hydraulic system away from the mill frequency.

Since the hydraulic system was modeled by a lumped model, the pressure changes (as the function of the pipe length) are not taken into account. In fact, for compressible flow, the cylinder pressure should not change until the pressure wave front arrives. The velocity of pressure wave, V_p, in fluid is calculated by:

$$V_p = \sqrt{\frac{\rho}{\beta_e}} \tag{17}$$

Travel time of the pressure wave is the inverse of the natural frequency of the hydraulic line and the delay time is approximately 0.22 ms/unit pipe length (in ft). According to the performance test described in reference 14, there is an 8-ms dead time difference between the direct mounting type and conventional type (using the pressure line). Although the pipe length is not mentioned in reference 14, the length based on equation (17) could be 36 ft at the most which is reasonable in conventional design.

Cylinder working stroke — The cylinder working stroke is determined by the cylinder type (long or short-stroke) and the roll size. For a pancake type cylinder, the total stroke is usually less than 0.5 in. and, for a long-stroke cylinder that can have a total stroke length up to 10-in., the working stroke depends on the roll size selected. Generally, the longer the working stroke the larger is the time constant. The step response for working strokes of 0.1, 1, 5, and 10 in. for a single-acting cylinder with direct mounted servovalve is illustrated in Fig. 12. There are 104 and 80-Hz frequency oscillations with working strokes of 5 and 10 in. respectively and the oscillation is sustained after reaching the target. The time constants are substantially different for a 1 and 10-in. stroke but are considerably closer for 0.1 and 1-in. strokes. As suggested by Huzyak,[16] a cylinder with a stroke less than 1 in. is classified as a short-stroke actuator and greater than 1 in. as a long-stroke actuator.

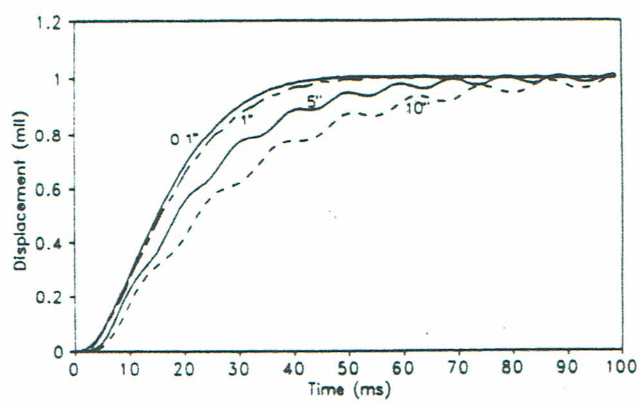
Many factors contribute to the dead time such as mill inertia and friction force, sensor and servovalve response time, cylinder oil compression process, and pressure wave propagation time. Dead times due to oil compression with 0.1 and 1-in. strokes (Fig. 12) are nearly the same but are considerably larger in the long-stroke cylinder with larger than 1-in. working strokes.

The frequency response, as shown in Fig. 13, is 80 and 104 Hz for working strokes of 5 and 10-in. respectively. (Mill natural frequency and servovalve frequency is also in this range.)

Using a long-stroke cylinder could excite the mill. The cutoff frequencies of −3 db gain are 14.09, 13.01, 9.59 and 7.26 Hz for working strokes of 0.1 to 10 in. This performance again demonstrates the disadvantages of using the long-stroke cylinder. However, these response curves meet at a frequency of 2 Hz, which implies that if a mill runs at a slow speed, such as an in-line temper mill in a coating line, a long-stroke cylinder (used in place of an electromechanical screwdown system) can provide a satisfactory frequency response even though the step response lags behind that obtained with a short-stroke cylinder.

Single and double-acting cylinder — In the previous discussion, performance has been considered based on single-acting cylinders without mill damping. For example, Fig. 13 shows that the cutoff frequencies are all less than 20 Hz with a 2-ma/mil gain amplifier. Higher cutoff frequencies might be expected by increasing the gain factor to 3 ma/mil.

Fig. 12 — Step response for cylinder working stroke of 0.1, 1, 5 and 10 in. (damping 0, gain = 2, P_s = 4500 psi, single-acting and direct mounted servovalve).



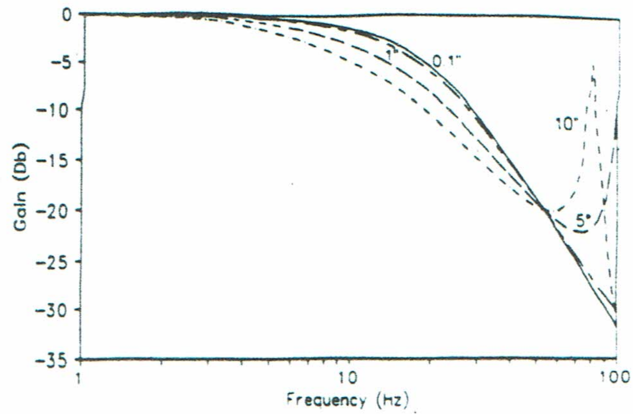
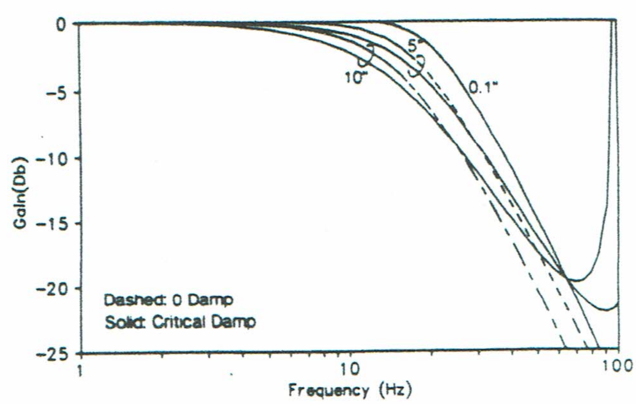


Fig. 13 — Frequency response for cylinder working stroke of 0.1, 1, 5 and 10 in. (damping 0, gain = 2, P_s = 4500 psi, single-acting and direct mounted servovalve).

The effect of mill damping with working strokes of 0.1, 5 and 10 in. is illustrated in Fig. 14. These results suggest that a short-stroke cylinder, in contrast with long-stroke cylinders, would be insensitive to mill damping because the two cutoff frequencies at a −3 db gain are identical. For a zero damping coefficient, the long- stroke cylinders have a resonant frequency of approximately 100 Hz which disappears if the damping coefficient is increased to 1.0. The cutoff frequencies with a 5-in. stroke are 16.38 and 18.15 Hz for zero and critical damping respectively, and 12.16 and 13.95 Hz for a 10-in. stroke respectively. In addition, the results also show that critical damping of long-stroke cylinders is more beneficial in the lower frequency range below 10 Hz. It is concluded that mill damping, while improving the performance of a long- stroke cylinder, has little effect on short- stroke units.

As mentioned previously, there are three types of double-acting cylinder designs. To model the first type, an incompressible flow was assumed in the return line. If the average fluid speed in the return line is less than 0.3 in./s (most cases have considerably lower speeds), laminar flow can be assumed and friction loss becomes a function of the equivalent pipe length and fluid speed. Since fluid speed is proportional to cylinder speed, it implies that the return line contributes a damping force to the mill system. The equivalent damping coefficient and natural frequency can be derived by combining equations (8), (9) and (10). The natural frequency decreases due to the inertia force of the oil in the returning line. However, the damping coefficient becomes adjustable by changing the cylinder area in the

Fig. 14 — Effect of mill damping facotr on frequency response with working stroke of 0.1, 5 and 10 in. (gain = 3, P_s = 4500 psi, single-acting and direct mounted servovalve).



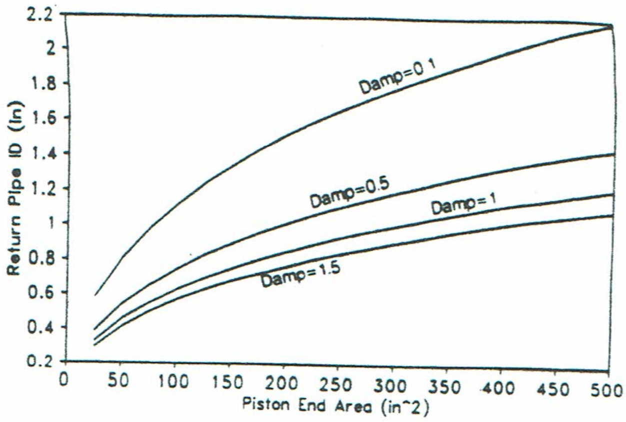
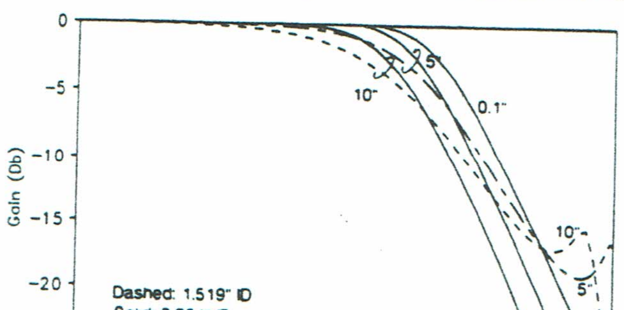


Fig. 15 — Equivalent damping coefficients of various return pipe diameters for double-acting hydraulic cylinder.

back-pressure side and the pipe diameter and length. The equivalent damping coefficients for various pipe diameters are illustrated in Fig. 15 which shows, for example, that a 0.854-in. dia pipe with a 200-sq in. cylinder area results in a damping coefficient of 1.0. The frequency response for stroke lengths of 0.1, 5 and 10 in. with return pipe diameters of 0.854 and 1.519 in. are shown in Fig. 16. As expected, the short-stroke cylinder shows no difference. However, the long-stroke cylinders with larger pipe provide a small damping effect. Compared with a single-acting cylinder, the 1.519-in. dia return pipe reduces the resonance peak and resonant frequency. As the return pipe diameter is reduced, the damping factor becomes larger and the cutoff frequency and phase angle increase.

The performance of single and double-acting cylinders can be evaluated using Fig. 17. A critical mill damping factor is used for the single-acting cylinder. Assuming zero mill damping factor, the selection of a 0.854-in. dia return pipe for the double-acting cylinder generates an equivalent critical damping factor. The purpose of this arrangement is to compare the results with the same damping factors. The cutoff frequencies of the double-acting cylinder are slightly larger than those of the single-acting cylinder. Two corresponding curves meet at the frequency of 20 Hz. Therefore, for a frequency less than 20 Hz, the operation of double-acting cylinder is superior. And, at the high frequency range, the double-acting cylinder can further attenuate the high frequency noise.

Fig. 16 — Frequency response for cylinder working stroke of 0.2, 2, 5 and 10 in. with return line pipe dia of 0.854 and 1.519 in. (gain = 3, P_S = 4500 psi, double-acting and direct mounted servovalve).



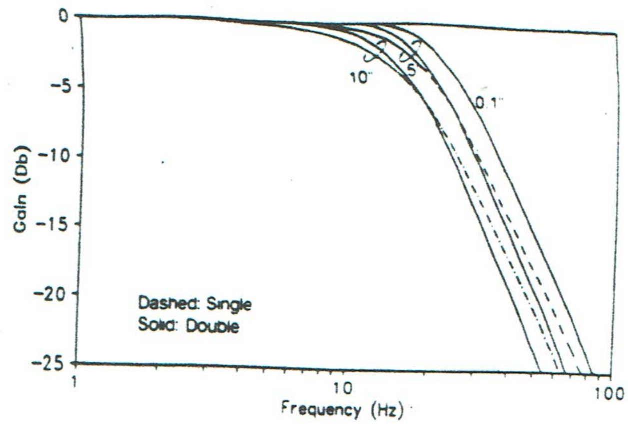


Fig. 17 — Frequency response for single and double-acting cylinders with working stroke of 0.1, 1, 5 and 10 in. (gain = 3, P_S = 4500 psi, direct mounted servovalve, 0.854-in. dia return pipe for double-acting cylinder.

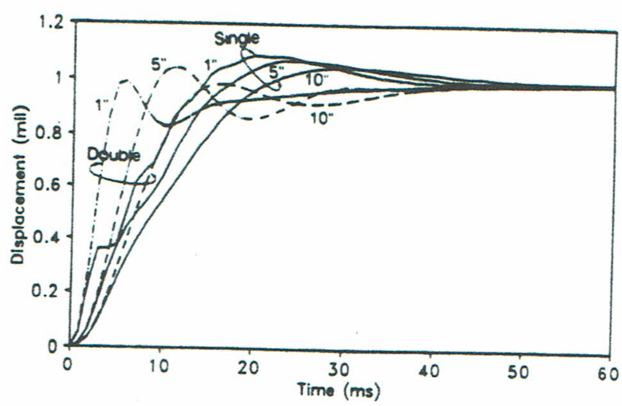
Pressure mode operation — Using the BISRA AGC, which is associated with the position mode, enhances the roll eccentricity effect. A constant force mode (or pressure mode) together with the mass flow principle and feed-forward control concept can minimize the roll eccentricity effect. Pressure mode operation has gained importance in the recent years.

Step response using the pressure mode for single and double-acting cylinders with a 0.5 damping factor and 0.854-in. dia return pipe (for double-acting cylinder) is illustrated in Fig. 18. The double-acting cylinders respond considerably faster in the first 10 ms and then undershoot, oscillate and then arrive at the target. The single-acting cylinders have larger time constants, overshoot and decay down to the target. The frequency response is shown in Fig. 19. The cutoff frequencies of the double-acting cylinders are larger than the single-acting cylinders. For the frequency range less than 20 Hz, the single-acting cylinder performed slightly better than the double-acting cylinder.

Conclusions and summary

The objective of this article is, primarily, to demonstrate the nonlinear effects of the hydraulic system and the damping effect due to the return line.

Fig. 18 — Step response using pressure mode for single and double-acting cylinders with 1, 5 and 10-in. working stroke (damping = 0.5, gain = 3, P_S = 4500 psi, direct mounted servovalve, 0.854 in. dia return pipe for double-acting cylinder).



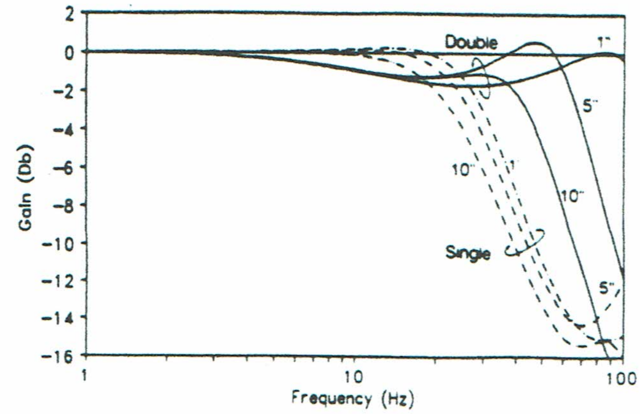


Fig. 19 — Frequency response using pressure mode for single and double-acting cylinders with 1, 5 and 10-in. working stroke (damping = 0.5, gain = 3, P_S = 4500 psi, direct mounted servovalve, 0.854-in. dia return pipe for double-acting cylinder).

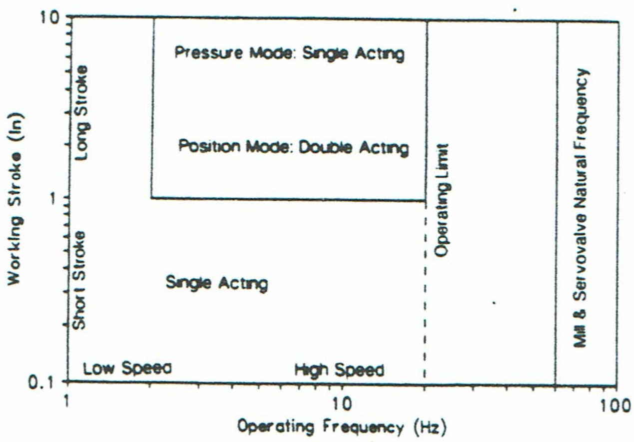
The nonlinear effects of valve saturation and uneven pressure drop can be negated by using two servovalves and an additional gain amplifier for the dumping process. The return line does provide the necessary damping effect particularly for a long-stroke cylinder.

A comprehensive chart that facilitates the selection of the hydraulic cylinder is illustrated in Fig. 20. Since the natural frequencies for mill application servovalves are between 60 and 100 Hz, the operating frequency should be less than 20 Hz (as recommended by servovalve manufacturers).

The first two natural frequencies of the mill occur at 64 and 120 Hz from this study. Although mill natural frequencies depend on the mill design and strip characteristics, the operating limit frequency should be less than 20 Hz as recommended by Collison[4] which is less than one third of the mill first natural frequency.

For a low-speed mill (less than 2 Hz), there are no significant differences between long or short-stroke, single or double-acting cylinders. The long-stroke single-acting cylinder should be an appropriate selection from the view point of initial and maintenance costs. A single-acting type cylinder should be selected if the working stroke is less than 1 in. because it has the same response as the double-acting type. For frequencies higher than 2 Hz, the double-acting type is superior in the position mode. However, for the pressure mode, although the cutoff frequencies are larger than 20 Hz for both, the single-acting type is slightly better in the low frequency range (less than 20 Hz).

Fig. 20 — Selection criteria for hydraulic cylinders.



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