TOYS ARE THE STORY — PLAYING AROUND WITH RP, PAGE 37

MACHINE DESIGN

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WELCOME BACK

NASA gets ready for a return trip, page 76

FEA makes composites predictable, page 82

Servohydraulics for the real world, page 90

BASICS OF DESIGN ENGINEERING: FASTENING AND JOINING

New developments in high-performance epoxy, page 99

Quick cure for acrylic adhesives, page 100

Good-looking seams with glue, page 104

Guidelines for sizing SERVOHYDRAULICS

The best controllers and software can't overcome a poorly designed hydraulic system.

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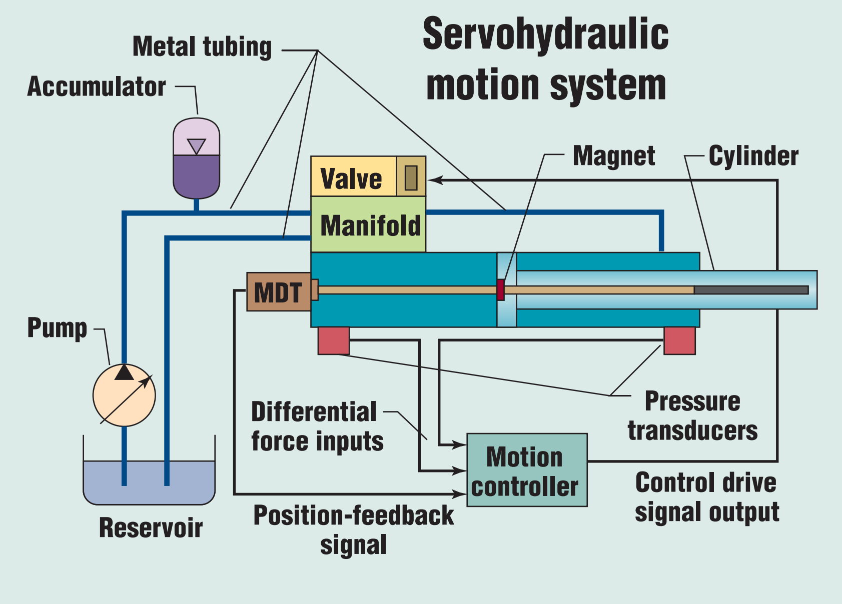
Global competition ensures that plant operators face the never-ending task of making equipment more productive. This often demands higher speed and precision from their motion-control systems. But in the case of closed-loop hydraulics, higher speeds and precision must start with good component design. No matter how much attention is given to electronic controllers and software, if the fluid-power cylinders and valves are sized or mounted incorrectly, system performance will suffer.



In typical servohydraulic motion systems, the valve mounts as close as possible to the cylinder. Experts also recommend motion controllers specifically designed for hydraulic applications.

CYLINDER SIZING

With linear actuators, application requirements tend to set stroke length and cycle times. Designers then determine cylinder diameter and oil pressure. A common problem is attempting to increase actuator speed by undersizing cylinders. Engineers often assume that for a given oil flow, smaller cylinders accelerate quicker and run faster. However, this only works for light loads. For actuators moving moderate to heavy masses, available force — not oil flow — limits acceleration, velocity, and deceleration. Because piston diameter determines output force, a cylinder that is too small will never attain the required speeds and cycle times.

A designer's first inclination may be to use the oversimplified equation $V = Q/A$. But it is only accurate if mass = 0. Only use the alternate form $Q = VA$ to calculate flow.

In fact, there is no simple tool for optimizing cylinder diameters. Two equations are shown here, each with advantages and limitations. We recommend calculating cylinder areas with both equations and using the larger of the two. The first equation is from *Design of Electrohydraulic Systems for Industrial Motion Control* by Jack Johnson.

$$A_p = \frac{1}{P_s}\left(\frac{V_2^2F_1 - V_1^2F_2}{V_2^2 - V_1^2}\right)$$

It accurately calculates the required area for the cylinder's powered end given the force and velocity at two operating points. For instance, one point can be when the cylinder deadheads; $F = A_pP_s$ and V

Edited by Kenneth Korane

Bigger can be better

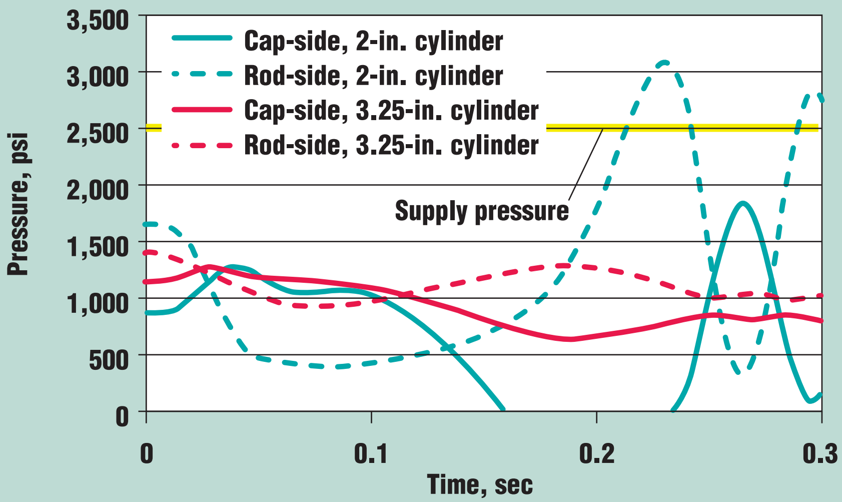
A metals-industry manufacturer recently faced problems with an underperforming servohydraulic system, and a hydraulic simulation suggested the machine's cylinders did not produce enough force to control rather large loads. Mathematical modeling recommended replacing the original 2-in. diameter cylinders with 3.25-in. ones, and installing appropriately larger hydraulic valves.

The graph shows how undersized cylinders may not provide enough force to quickly decelerate a large load. The 2-in. cylinder cavitates (pressure drops to zero) on the cap end while rod-end pressure exceeds system pressure. The system is out of control because oil on the rod side flows to the oil supply, and actually decreases deceleration.

A 3.25-in. actuator accelerates the load at the same rate, but its greater piston area means pressure doesn't change as much. Cylinder pressures throughout the cycle are in the middle of the pressure range and there is plenty of pressure drop across the piston to maintain positive control.

Increasing cylinder diameter also increases the system's natural frequency (stiffness), letting the motion controller handle faster accelerations and decelerations. This yields higher performance when properly tuned. However, keep in mind that big cylinders require large valves and more oil. Oversized cylinders cost more and, because larger valves tend to be slower, at some point increasing valve size will no longer boost system response.

Cylinder simulation



Cylinder-bore size simulations demonstrate relatively stable pressures for the larger cylinder.

= 0. The other point can be required force at the desired travel velocity. The equation is accurate when acceleration is zero but does not take into account oil compression and expansion when actuators accelerate and decelerate. It also leads one to believe that increasing the system pressure is a good way to reduce cylinder size, which is not the case in systems that accelerate and decelerate quickly.

For high-performance motion, Delta Computer Systems developed a second equation to ensure the system's natural frequency is three to four times higher than the motion frequency. For example, if acceleration frequency is 5 Hz, the actuator's natural frequency should be about 20 Hz. Determine this from:

$$A_c = (4f)^2\left(\frac{\pi^2 lW}{g\beta}\right)$$

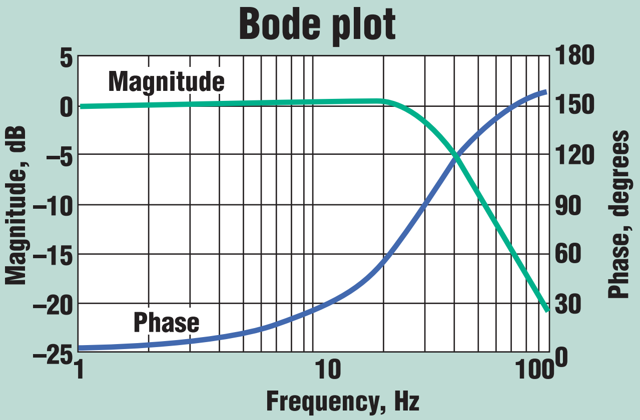
β is the incompressibility constant or bulk modulus of oil. It is ideally 200,000 psi, but usually falls below that due to entrained air.

This equation also tends to underestimate cylinder diameter because it makes some optimistic assumptions. The most significant one is that the valve sits directly on the cylinder. If this is not the case, increase cylinder length by about the length of the hose. The hose cross section is usually much smaller than the cylinder's, but hose is much more compliant. Hose or extra pipe between the valve and cylinder complicates calculations and reduces performance. For this reason, mount the valve as close as possible to the cylinder and connect them with metal tubing.

VALVE CONSIDERATIONS

Focusing first on cylinder size ensures the system will have the dynamic response needed to meet acceleration and deceleration requirements. This usually means calculating the necessary system pressure, too.

The next step is typically to determine valve size (flow rating), which is relatively straightforward after calculating cylinder diameter. A note of caution: Servovalves and servo-quality proportional valves are generally rated at pressure drop ΔP = 70 bar (1,015 psi) whereas other proportional valves are often rated at ΔP = 10 bar (145 psi). The difference



Typical Bode plots show amplitude response and phase lag of the spool as a function of control-signal frequency.

is significant. Flow at 1,000 psi pressure drop is typically about 2.65 times the flow at 150 psi. Selecting the right valve involves more than sizing, however, because most valves have many functional options.

Valve choices. The first decision is whether to use servo or proportional valves. The main difference lies in how spools shift. Proportional valves move the spool with an electric coil and magnet, like the voice coil of an audio speaker. Servovalves, on the other hand, use a small torque motor to control hydraulic pilot pressure which, in turn, moves the spool.

NOMENCLATURE

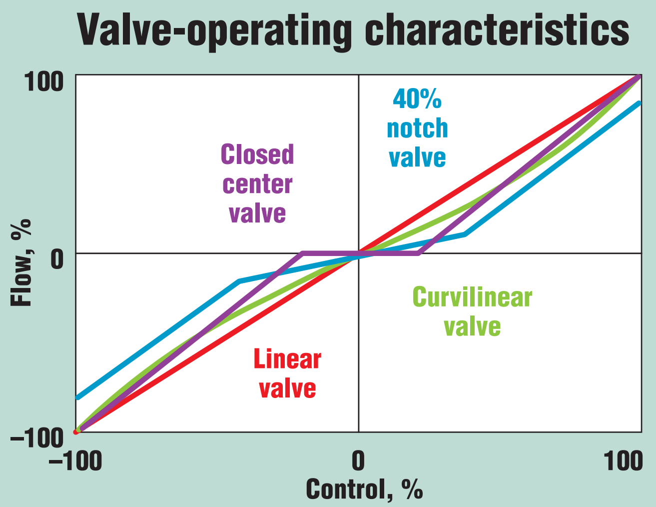
- A = Cylinder piston area
- A_c = Average cylinder area
- A_p = Powered-end piston area
- F = Cylinder force
- F_1 = Force at point 1
- F_2 = Force at point 2
- f = Frequency of motion
- g = Acceleration due to gravity, 32 fps^2
- l = Length
- P_s = Supply pressure
- Q = Flow
- V = Velocity
- V_1 = Velocity at point 1
- V_2 = Velocity at point 2
- W = Load weight
- β = Oil bulk modulus

The force generated to shift the spool — and valve response — differs with each method. Servovalves generally shift faster because of a higher ratio of hydraulic force to spool mass, although some proportional valves have nearly the response of servovalves. Proportional valves must generate enough force to move the spool, in-line LVDT, and solenoid core, as well as overcome spring-centering forces.

Precise tolerances and small orifices associated with pilot-operated servovalves drive up costs and make them susceptible to contamination. For many applications, this steers people to proportional valves. Nonetheless, servovalves still have their place. For example, they often work better in high-flow applications.

In some cases, proportional valves do not have the power to overcome Bernoulli forces caused by high flows. The valves momentarily lose control until flow forces drop. And when troubleshooting such problems, there is often a tendency to fault the controller instead of the valve. An oscilloscope or other diagnostic tool that records control signals, along with spool and actuator positions, helps resolve such issues. In these applications, servovalves tend to be a safer design choice. They perform better because of a faster, more-linear response and, thus, are easier to control.

Proportional-valve amplifiers. Proportional valves need an amplifier to convert the motion controller's output voltage to a high-current signal that drives the spool. With a servo-proportional valve, the amplifier compares the error between the control or reference signal and spool-position feedback from the valve's LVDT. Some amplifiers use simple proportional control whereas others use PI or PID control. If the amplifiers are not "tuned" for the valve, performance suffers. It is best to select proportional



The graph shows operating characteristics of various valves. Linear valves with servo-quality spools provide flow proportional to the control signal as long as pressure drop across the valve is constant.

valves with onboard electronics to help ensure the amplifier is properly tuned. Purchasing separate amplifier cards requires additional effort and knowledge to adjust amplifier gains so the spool responds quickly to control signals.

Bode plots. Bode plots are essential valve-selection tools. They use logarithmic scales and show the spool's amplitude response and phase lag as a function of control-signal frequency. Normally, amplitude response is relatively flat for the first few Hertz and then it falls off. Phase lag is also relatively small for the first few Hertz and remains fairly flat, but rises rapidly at higher frequencies.

The point where phase lag reaches 90° determines a valve's frequency rating. At this point, amplitude is often approximately +6 dB, or about half the original value. For instance, a sinusoidal application that works well at 10 Hz — where a Bode plot shows little drop in gain or increase in phase lag — may not work well at 40 Hz where valve response may be only half the amplitude. Motion controllers can compensate for the loss in valve gain and phase lag with feedforward gains. At higher frequencies, the controller increases the control signal to compensate for the drop in valve gains. But the motion controller cannot increase the control signal beyond 100%. Therefore, designers must either leave plenty of "head room" in the system for the extra control signal or use higher-frequency valves that have little drop in gain or increase in phase lag over the application's frequency range.

Valve ratings. Another reality to contend with is that not all valve manufacturers generate Bode plots under the same test conditions. Different manufacturers may rate their valves at different spool-travel amplitudes. Spool-travel response is usually much worse for 100% control signals than for 5% signals. This means two valves rated at 30 Hz may actually be quite different if one is rated using a 5% control signal and the other with a 50% control signal. Many Bode plots show response for a 5 or 25% sine wave. The 5% ratings are good for applications that dither the valve around 0%, such as pressure and force-control applications. However, they are not useful for high-speed applications where valve-spool travel approaches 100%. A good rule of thumb is to take the rise time from 0 to 100%, multiply it by four, then divide into one to get the frequency for full travel.

Linear valves with fast responses are necessary for high-performance position/pressure-control systems. Of course, valve performance is not perfect and a good motion controller is still required to compensate for valve response and the spring-mass effect of the actuator and load. To maximize performance of well-engineered hydraulic systems, select motion controllers designed for hydraulics. They have features such as separate extend and retract gains, position-pressure/force control, and can connect directly to magnetostrictive linear-displacement transducers.

SPOOL CHOICES

Proportional valves are so named because the valve spool shifts in proportion to the control signal driving the valve. However, flow is not necessarily proportional. Proportional valves often have many different types of spools and selecting the correct one is critical for best system performance.

Servo-quality spools. For position and pressure/force control, select a proportional servo-quality or axis-cut spool. The spools provide flow proportional to the control signal as long as pressure drop across the valve stays constant. The valves have constant gain because response is linear.

Closed-center spools. Nonlinear spools come in many forms with many names. The most common is the overlapped or closed-center spool that appears to have a "dead band" or zero-gain region because the valve does not permit flow when the control signal is small. This may reduce leakage and make it easier to keep a system stopped while in manual control, but it also makes the valves poor choices for position or pressure control because the spool must shift quickly across center to provide fine control.

The larger the dead band, the longer it takes to shift the spool. During these few milliseconds there is essentially no flow response from the valve, resulting in no change in position and pressure input to the motion controller. This feedback discontinuity limits the ability for a motion controller to maintain precise positions and pressures. Only use closed-center valves when the spool does not need to shift quickly across the dead band, for instance where motion does not change direction rapidly or often. Speed control in conveyors is a good example of such an application.

Dual or variable-gain spools. Other spools have flow gains that vary with the control signal. The valves usually have low flow gains when the control signal is close to zero and higher flow gains as the signal approaches ±100%. Notch or dual-gain valves have distinct low and high-gain regions and curvilinear valves have continuously varying gains. For manual systems, both types offer fine control at slow speeds along with substantial flow for high speeds.

Although not a problem with manual or open-loop control, a nonlinear valve makes the entire system nonlinear and closed-loop control difficult. A controller must change gains on the fly as the valve shifts between high and low-gain regions. In theory, valve linearization (compensation for varying gains as a function of the control signal) can be done within the motion controller using look-up tables or specific calculations. However, the need to match specific valve characteristics limits the usefulness of this approach. And the spool must travel further for a given flow change in the low gain region, slowing response in this region and reducing systems performance.

Notch spools work well when closed-loop control is only required at slow speeds. Open-loop or manual control can be used at higher speeds where the valve is in the high-gain region. This way, the closed-loop controller does not need to change its gains on the fly as the valve shifts between high and low-gain regions. However, for most position and pressure-control applications, it is better to avoid notch or curvilinear spools. **MD**

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