SERVOCONTROL with pneumatic actuators

Advances in flow control let pneumatics handle closed-loop tasks that were once practical only with expensive motors.

Pneumatic cylinders now incorporate direct-drive voice-coil powered servovalves (inset) as a means of realizing servocontrolled motion. Cylinder position sensing is through an LVDT (not shown).

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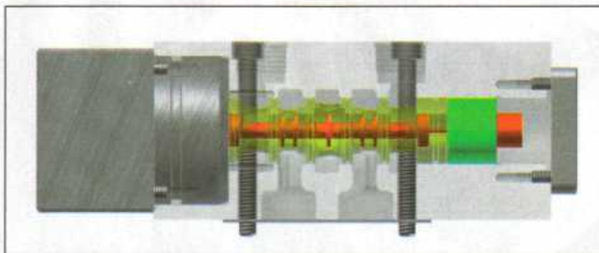
Any aviation buff is well aware of the Wright Brothers' role in the history of flight. But it's likely that many can't identify the Wright's seminal contribution: aerodynamic control. When their plane made its public debut — thrilling a crowd with well-controlled banks, turns, and figure eights — the balance of the aviation community was resigned to wobbly, white-knuckle flights the distance of a Tom Dempsey field goal.

After millennia of da Vincianesque dabbling, and more than a century of engineering effort, Wilbur and Orville Wright had solved the elusive piece of the puzzle. Who would have guessed that dynamic control of an air machine would prove to be so troublesome? Perhaps any engineer who's attempted to realize servocontrol with a pneumatic actuator.

In the arena of industrial automation, air cylinders have long given machine designers a fast and convenient means of discrete state actuation. Pneumatics combined with flow controls and position switches can give acceptable acceleration smoothing on a Spartan budget. But industry increasingly needs actuators with continuously variable positioning. These actuators have typically been stepper or brushless-dc motors coupled to a means of mechanical transmission. Unfortunately, the expense of this approach often sends a parsimonious purchasing manager into sticker shock.



Consequently, fluid-power manufacturers have pursued the development of pneumatic cylinders capable of continuously variable output — in other words, pneumatic servos. But these actuators often have lacked the speed and control to be functional alternatives to stepper motors or state-of-the-art electric servos. The price has been right, but the performance has been wrong.

And while research, development, and marketing of the technology proceeds, many erstwhile advocates of pneumatic servos now consider such efforts a festival of folly. They'll likely suggest you'd have better luck getting Michael Moore to vote Republican than getting a pneumatic cylinder to abruptly stop at midstroke. The bugbear ubiquitously identified as the cause for this deficiency in control is compressibility. Unlike the incompressible fluid powering hydraulic servos, the compressible gases powering pneumatic systems render the actuators soft and compliant. Or so the argument goes.

Let's evaluate that contention by assessing the effect of fluid compressibility on actuator stiffness. (Stiffness is here defined as a measure of force required to displace a cylinder piston from midstroke with both head and rod-end chambers sealed to prevent fluid leakage.) Consider a cylinder with a 2-in. bore and 12-in. stroke coupled to a 100-lb inertial load. When this cylinder is filled with hydraulic fluid, a force exceeding 210,000 lb

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is required to displace the actuator 1 in. from midstroke. (For a full analysis, see the sidebar "Computing displacement forces.")

In contrast, when the identical cylinder is filled with air at 40 psi, a disturbance force of 50 lb will yield the same 1-in. shift. This equates to an equivalent spring-mass natural frequency of 140 Hz for the hydraulic system; and for the pneumatic system, the natural frequency is only 2 Hz. When disposed in a positional servo arrangement, a well-tuned PID controller can deliver a closed-loop bandwidth between 15 and 30 Hz with the hydraulic system. But the same controller will merely manage a 0.2 to 0.4-Hz bandwidth with the air system. (Assuming an estimated damping ratio of 0.1 and a high-speed servovalve common to the art.) Ergo the prejudice, out goes the pneumatics.

Well — not so fast. Magnetic fields are also compressible and the force generated by a linear motor is independent of position. Based on the definition of stiffness, that makes for an actuator that's almost infinitely compressible. The same holds true for slotless motors. Yet both are renowned for their speed, accuracy, and controllability. Yes, air is compressible. But only when this physical reality is ignored does the performance of a pneumatic servoactuator so dismally suffer.

The pioneers of aviation failed if they attempted to design airplanes to look like birds. Similarly, modern fluid-power engineers fare poorly if they attempt to control a pneumatic servo as if it were hydraulic. A pneumatic cylinder does look like its oleic counterpart — but it quacks more like a motor.

In 2002, the founders of Sunstream Scientific set out to address the deficiencies of early pneumatic servos with a comprehensive, clean-sheet design. After

Computing displacement forces

It can be helpful to understand how to calculate displacement forces in hydraulic and pneumatic systems. These calculations yield the resonant frequencies of these systems which can be valuable in determining the necessary control schemes for a pneumatic or a hydraulic system.

Consider a typical piston moving in a cylinder. The piston consists of a head and a rod. Assume for the sake of an example that the head area of the piston, $A_H = \pi$ in.^{2} and area of the piston minus that of the rod, $A_R = 2.835$ in.^{2} First consider the case for a conventional hydraulic system. From hydraulic-control systems theory,

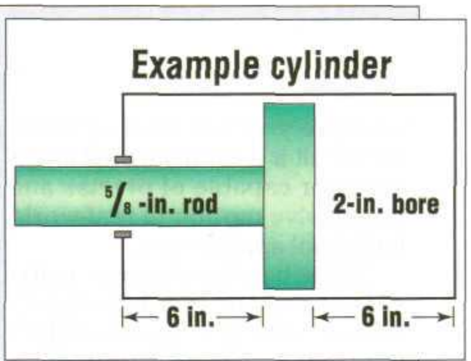
$$\frac{1}{\beta_e} = -\left(\frac{\Delta V_t}{V_t}\right)\frac{1}{\Delta P}$$

where β_e = the effective bulk modulus of the fluid, about 210,000 psi; V_t = tube volume, and P = tube pressure. Then for $\Delta x = 0.1$ in.,

$$\Delta P_r = -\left(\frac{\Delta V_r}{V_r}\right)\beta_e$$
$$= -\left[\left(\frac{A_r \Delta x}{A_r \bullet 6 \text{ in.}}\right)\right]\beta_e$$
$$= -\left(\frac{0.1}{6}\right)\beta_e$$

$$\Delta P_H = -\left(\frac{\Delta V_H}{V_H}\right)\beta_e$$
$$= -\left[\frac{(A_H(-\Delta X))}{(A_H \bullet 6 \text{ in.})}\right]\beta_e$$
$$= \left(\frac{0.1}{6}\right)\beta_e$$

$$\Delta\text{Force} = A_H \Delta P_H - A_R \Delta P_R$$
$$= \left(\frac{0.1}{6}\right)\beta_e(\pi + 2.835)$$
$$= 20{,}917 \text{ lbf (for 0.1-in. displacement)}$$



Effective spring rate K_e is the change in force divided by the corresponding displacement, so

K_e = 20,917 lbf/0.1 in.
= 209,170 lbf/in.

For a 100-lbf inertial load,
Mass = 100 lbf/386 in./sec^2
= 0.259 slugs

The undamped resonant frequency, $\omega_n = \sqrt{(209{,}170/0.259)}$
= 899 rad/sec

So for a hydraulic cylinder, f_n = 143 Hz

In the case of a pneumatic system,

$$\Delta P_R = -\left(\frac{\Delta V_R}{V_R}\right)nP$$

where n = 1.2 for mixed adiabatic isothermal compression and P = nominal cylinder pressure. Thus the term nP is equivalent to β_e for hydraulic fluid. Then

$$\Delta P_R = -\left(\frac{0.1}{6}\right)nP$$

$$\Delta P_H = \left(\frac{0.1}{6}\right)nP$$

$$\Delta\text{Force} = \left(\frac{0.1}{6}\right)nP(\pi + 2.835)$$
$$\approx 0.1nP$$

The effective spring rate = $K_e = nP$ and from the previous calculation, mass = 0.259 slugs. Then if n = 1.2 and P_{nom} = 40 psi,
$\omega_n = \sqrt{(nP/0.259)}$ = 13.6 rad/sec

So for a pneumatic cylinder, f_n = 2.2 Hz.

multiple years of development, the result is a sophisticated linear actuator capable of precise and responsive motion in real-world industrial applications.

At the heart of the pneumatic servoactuator is a high-bandwidth servovalve capable of shifting from center to a fully open aperture in less than 3 msec. (The metering element is positioned to an aperture resolution of 0.03 mm^2.) And at the head is a digital signal processor capable of the multitasking needed to integrate multiple sensors. The DSP also provides the high-speed computation necessary to implement complex, nonlinear-control algorithms. Without the ability to correct for the intrinsic nonlinearity of the pneumatic system, compressibility is indeed the bugbear of motion-control mythology.

It's difficult to place absolute values on the performance of the pneumatic servo, as the operation of any closed-loop system depends on numerous factors outside the actuator such as friction, inertia, and mechanical compliance. With these limits to quantitative assurances in mind, here are the general performance qualities of the pneumatic servo:

Accuracy: The final accuracy of the pneumatic servo depends particularly upon friction and inertia. A properly designed pneumatic servosystem, with static friction no more than 10 to 20% of load inertia, should produce an accuracy of 0.020 in. Repeatability, moreover, may be as high as 0.001 in. If integral action is recruited in the PID control loop, steady-state error can dramatically drop to well below 0.001 in., but at the expense of a lower cycle rate.

Response: Consider typical systems with minimal mechanical compliance and with load inertias which are well known and relatively time invariant. Here, the functional bandwidth of the pneumatic servo will approach 10 Hz. Let's examine actual test results with a 10-in. stroke, 2-in. bore cylinder coupled to a 100-lb mass on a rigid sled, and supplied with 80-psi factory air. The pneumatic servo will track a small-signal, 5-Hz sinusoidal position input with a phase lag of 15°, and do so without overshoot. A point-to-point, 9-in. traversal can take place in 400 msec without overshoot.

Maximum velocity is limited by seal material (and distance available for acceleration) rather than dynamic controllability. It does no good to go fast if you blow a seal in the process. The recommendation is that velocity remain below 40 ips in the interest of rod-seal life. But high velocity can be delivered if necessary. In the latter setup with the inertial load reduced to 40 lb, the pneumatic servo can traverse the 9 in. in 250 msec, while reaching peak accelerations of $\pm 3\,g$ and a peak velocity over 70 ips.

Dynamic tracking: The bandwidth of the pneumatic servo is 10 to 100 times higher than that of conventional closed-loop pneumatic actuators. Thus it can smoothly track real-world positional trajectories and maintain acceleration and velocity limits. Acceleration and velocity limits are factory set and can be easily reprogrammed through a teaching pendant, PC, or real time during operation. For each point-to-point positional movement, the servo electronics will compute a positional trajectory based on a trapezoidal acceleration profile.

Following error is determined by the system bandwidth and actuator velocity. As with any servosystem, it can be mitigated with the introduction of feed-forward gain.

Inertia ratio: The pneumatic servo does not have a "gearbox" through which the load can be isolated. Consequently, changes in load will alter the system dynamics and have a subsequent effect on performance. Nevertheless, a properly sized servo will perform well over a broad range of inertial loads. For example, a servo tuned for 50 lbfm will have minimal overshoot if the load drops to 25 lbfm, or rises to 75 lbfm. As a general rule, keep the lightest inertial load within 40% of the heaviest for best performance.

Maximum acceleration will of course diminish with heavier loads. But a properly sized pneumatic servo will be responsive and controllable even in systems with high inertia.

Disturbance rejection: The pneumatic servo can reject stochastic load disturbances (such as the addition of a random mass in a vertical axis) to a degree compatible with that of a 5 to 10-Hz linear system. It will be able to correct for the disturbance, but may experience a temporary offset. It can handle deterministic load disturbances (such as the addition of a known mass in a vertical application) with proper programming. Random pressure supply fluctuations will not affect the pneumatic servo output.

Pneumatic servocontrol is still very much a nascent technology. The digital processors needed to handle nonlinear controls have become affordable just in the past few years. New sensors have become available that deflate the price point of position sensing. And the pioneers who will compete in this field are still emerging. **MD**

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