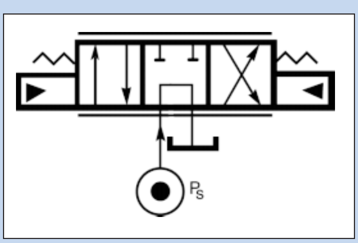
Electrohydraulic valves

Servo and servoproportional valves control pressure or flow and, ultimately, force or velocity. Unlike simple directional valves, they can maintain any position between fully open in one direction or the other.

High-performance valves are usually classified as either servo or proportional, a distinction that gives an indication of expected performance. Unfortunately, this classification tends to generalize and blur the true differences between various valve styles. Selection depends on the application, and each valve has merit when it comes to controlling pressure or flow.



To designate a proportional valve schematically, the familiar digital directional control valve symbol in the illustration above is augmented with two straight exterior lines outside the valve envelope that run parallel to the valve's long axis. These additional lines indicate that the spool has the ability to assume and maintain any intermediate position over the full range of spool travel.

Traditionally, the term servovalve describes valves that use closed-loop control. They monitor and feed back the main-stage spool position to a pilot stage or driver either mechanically or electronically. Proportional valves, on the other hand, move the main-stage spool in direct proportion to a command signal, but they usually do not have any means of automatic error correction (feedback) within the valve.

Confusion often arises when a valve's construction resembles a proportional valve, but the presence of a spool position feedback sensor (usually an LVDT) boosts its performance to that rivalling a servovalve. This reinforces the concept that designers and suppliers should use common terminology and focus on the performance reuirements of the particular application at hand.

Typically, proportional valves use one or two proportional solenoids to move the spool by driving it against a set of balanced springs. The resultant spool displacement is proportional to the current driving the solenoids. The springs also center the main stage spool. Repeatability of the main-stage spool position is a function of the springs' symmetry and ability of the design to minimize nonlinear effects of spring hysteresis, friction, and machining tolerance variations.

Servovalves

The term servovalve traditionally leads engineers to think of mechanical feedback valves, where a spring element (feedback wire) connects a torque motor to the main-stage spool. Spool displacement causes the wire to impart a torque onto the pilot-stage motor. The spool will hold position when torque from the feedback wire's deflection equals the torque from an electromagnetic field induced by the current through the motor coil. These two-stage valves contain a pilot stage or torque motor, and a main or second stage. Sometimes the main stage is referred to as the power stage. These valves can be separated primarily into two types, nozzle flapper and jet pipe, Figure 1.

The electromagnetic circuit of a nozzle flapper or jet-pipe torque motor is essentially the same. The differences between the two lie in the hydraulic bridge design. A hydraulic bridge controls the pilot flow which, in turn, controls the main-stage spool movement. In a nozzle flapper, the torque produced on the armature by the magnetic field moves the flapper toward either nozzle depending on command-signal polarity. Flapper displacement induces a pressure imbalance on the spool ends which moves the spool. In a jet pipe, the armature movement deflects the jet pipe and

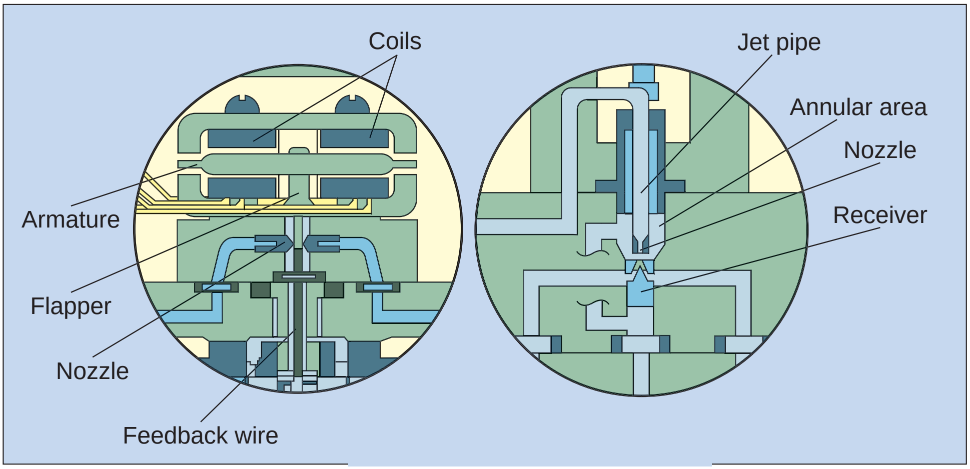


Fig. 1. First-stage configurations for nozzle flapper and jet-pipe valves.

asymmetrically imparts fluid between the spool ends through the jet receiver. This pressure imbalance remains until the feedback wire returns the jet pipe or flapper to neutral.

Historically, jet pipe and nozzle-flapper servovalves have competed for similar applications that require high dynamics. Typically, better first-stage dynamics gives the nozzle flapper better overall response, whereas improved pressure recovery of the jet/receiver bridge design gives the jet-pipe motors higher spool driving forces (chip-shearing capability). Both valves require low command currents and therefore offer a large mechanical advantage. Motor current for these style valves is typically less than 50.0 mA. Note that these servovalves are also proportional valves, because spool displacement and flow are directly proportional to the input command.

Direct-driven valves

Direct-driven valves, unlike hydraulically piloted two-stage valves, displace the spool by physically linking it to the motor armature. These valves usually come in two basic varieties, those driven by linear force motors (LFM) and those actuated by proportional solenoids. Within these two general classifications, the valves can be separated into proportional and servoproportional. The distinction is based on the use of a position transducer to provide spool position feedback. Servoproportional valves must incorporate closed-loop spool position feedback to increase repeatability and accuracy necessary for high-control applications. Typically, servoproportional, direct-driven valves have an overall lower dynamic response than hydraulically piloted two-stage valves with the same flow characteristics. This is usually due to the large armature mass of the LFM or solenoid and the large time constant associated with the coil, which is a function of the induction and resistance of the coil.

Unlike hydraulically piloted servos, direct-driven valve performance does not vary with changes in supply pressure. This makes them ideal for applications where pilot flow for first-stage operation is not available. Direct-driven valves also tend to be viscosity insensitive devices, whereas nozzle-flapper and jet-pipe valves work best with oil viscosity below 6,000 SUS. However, most direct-driven valves cannot generate the high spool driving forces of their hydraulically piloted counterparts.

Like the torque motor used in the nozzle flapper/jet pipe servos, the LFM allows for bidirectional movement by adding permanent magnets to the design and therefore making the armature motion sensitive to command polarity. In the outstroke, the LFM must overcome spring force plus external flow and friction forces. During the backstroke to center position, however, the spring provides additional spool-driving force which makes the valve less contamination sensitive. Magnetic-field forces are balanced by a bidirectional spring that lets the spool remain centered without expending any power.

Unlike the LFM, the proportional solenoid is a unidirectional device. Two solenoids oppose each other to achieve a centered, no power, fail-safe position. When a single solenoid is used, holding the spool at midstroke requires a continuous current to balance the load generated by the return spring. This makes the design less energy efficient than its LFM or a dual-solenoids counterpart. During a power loss, the LFM and dual proportional solenoid designs fail to a neutral position and block flow to the load, that is the piston. When a single solenoid de-

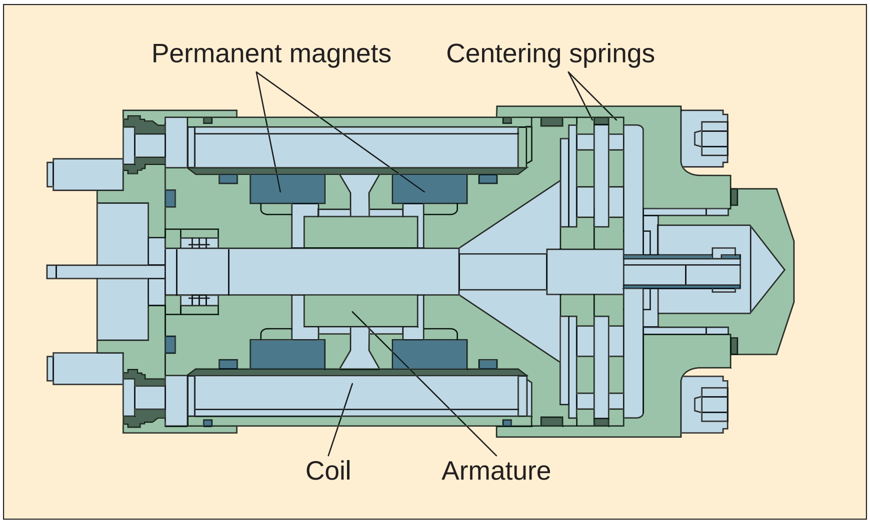


Fig. 2. The linear force motor often is used to drive the spool of high-performance valves directly. An alternative is to use one or two proportional solenoids to drive the spool.

sign loses power, the spool must move through an open position that tends to cause uncontrolled load movements.

Multistage valves

All of the aforementioned designs can be used to create a multistage hydraulic valve. The approach for each design is specific to the application requirements. Usually, most designs do not exceed three stages. Mounting a nozzle flapper, jet pipe, or direct-driven valve onto a larger main stage satisfies most requirements for dynamics and flow. Sometimes, the jet-pipe valve is used in a multistage configuration where the mechanical feedback of a traditional jet pipe is replaced with electronic feedback. This servojet style has pilot characteristics of a typical jet pipe. Depending on the required control, many multistage valves close a position loop about the main stage using a linear variable differential transducer. This device monitors the spool position. In case of hydraulic power loss, springs on opposite sides of the main stage spool return it to a neutral position.

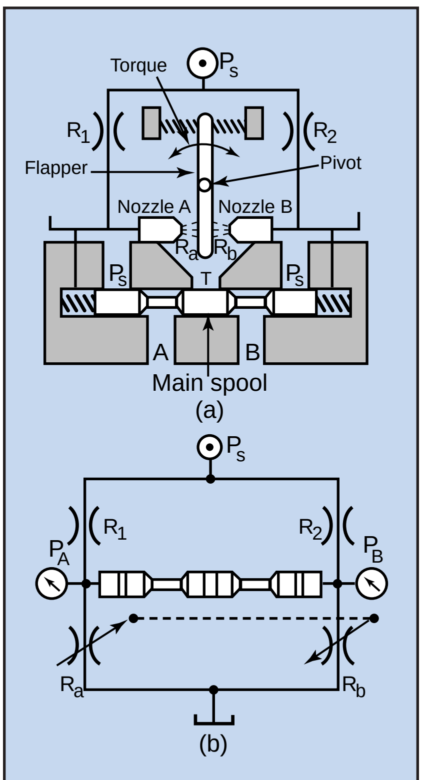


Fig. 3. When the flapper nozzle pilot section (a) is drawn in schematic form, (b), it is obvious that a bridge circuit exists. By moving the flapper, restrictions Ra and Rb change in opposite directions. This unbalances the bridge and causes the spool to move against its centering springs.

Hydraulic system design

To choose the proper hydraulic valve for a specific application, designers must consider specific application and system configurations. Supply pressure, fluid type, system force requirements, valve dynamic response, and load resonant frequency are examples of the various factors affecting system operation.

Hydraulically piloted valves are sensitive to supply pressure disturbances, whereas direct-driven valves are unaffected by supply pressure variation. Fluid type is important when considering seal compatibility and viscosity effects on performance over the system's operating temperature range.

Total force requirements must include all static and dynamic forces acting on the system. Load forces can aid or resist, depending on load orientation and direction. Forces required to overcome inertia can be large in high-speed applications and are critical to valve sizing.

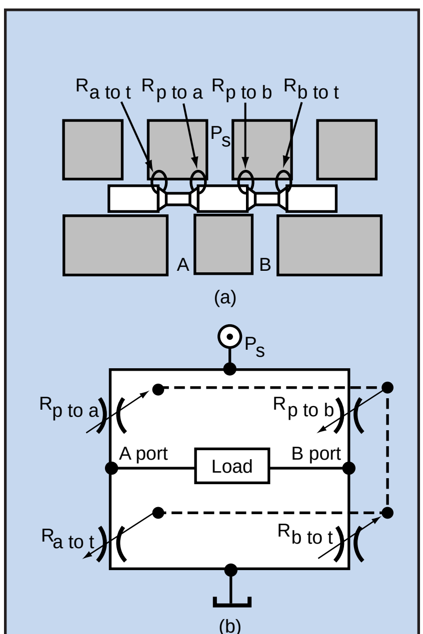


Fig. 4. The 4-way spool valve has four individual lands that vary in unison as the spool shifts — two lands open while the other two close. When drawn in schematic form, it is clear that the four lands constitute a bridge circuit, and spool movement unbalances the bridge one way or the other to cause a reversal in load flow.

The load resonance frequency is a function of the overall travel stiffness, which is the combination of the hydraulic and structural stiffness. For optimum dynamic performance, a valve's 90° phase point should exceed the load resonant frequency by a factor of three or more.

The valve's dynamic response is defined as the frequency where phase lag between input current and output flow is 90°. This 90° phase lag point varies with input signal amplitude, supply pressure, and fluid temperature so comparisons must use consistent conditions.

A closer look

Viewing the principal internal parts of a flapper-nozzle servovalve, Figure 3, it should be clear that a torque applied from a torque motor to the flapper arm, say in the clockwise direction, moves the flapper closer to nozzle *A* and tends to close it.

Concurrently, the flapper moves away from nozzle *B* to allow more flow through it, so the net result is a rise in pressure P_a and a drop in pressure P_b. The pressure difference, P_a — P_b is felt across the two ends of the main valve spool, driving it to the right and creating communication from port *P* to port *B* ,and from port *A* to port *T*.

A 4-way directional control valve is represented in Figure 4. When the valve spool moves to the right, $R_{p\ to\ a}$ and $R_{b\ to\ t}$ open while $R_{p\ to\ b}$ and $R_{a\ to\ t}$ close. Fluid flows from the valve's *A* port to the load and returns via *B* port to tank. Left spool movement opens $R_{p\ to\ b}$ and $R_{a\ to\ t}$ so that fluid flows from the valve's *B* port to the load, returning to tank via *A* port.

Electromechanical actuators

It is possible to construct proportional electrohydraulic interface devices (I_Hs) only because of the invention of certain proportional electromechanical interface devices (I_Ms). The I_Ms commonly used in the fluid power industry include:

- torque motors,

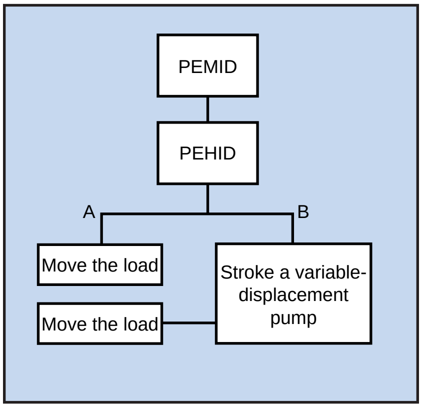


Fig. 5. Proportional electromechanical interface device (PEMID) causes an electrohydraulic valve (PEHID) to shift with two results: valve-control method (A), where valve output moves the load, and (B), where the valve changes the displacement of a pump with resulting pump output then moving the load, an example of the pump control method.

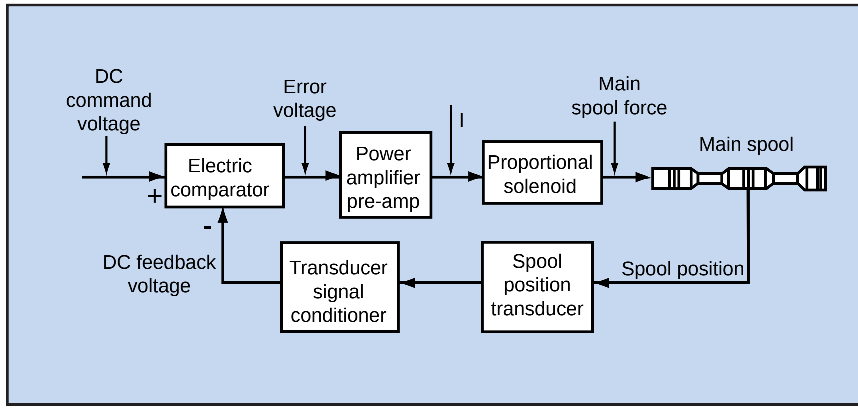


Fig. 6. A small current must cause a small spool shift and a large current must cause a large spool shift in continuously variable electrohydraulic valves. To assure such proportional, stepless spool positioning, some valves use a spool-position transducer to measure actual spool position. The spool is made to stop in a position commensurate with the command voltage through feedback- loop closure. This closed feedback loop often is called the inner loop.

- linear force motors, and
- proportional solenoids.

The I_Ms receive an electrical current input, convert it to mechanical force and motion, and then transform the energy into some sort of hydromechanical action. The direct mechanical action is always within a valve, although that valve may stroke a pump or directly power a load. A circuit designer would select one path or the other of the family tree, Figure 5, for a given application. Valve control is called the energy-loss control method in path A, because the valve, being a restrictive device, consumes excess power as a necessary part of its control function.

Path *B*, on the other hand, is called the *volume-control* or *load-demand* method that supplies only as much power as the load can use. The only losses encountered using this method are those caused by the modest inefficiencies of the pump and actuator. Altogether, these are nearly always less than those for the energy-loss method, all other things being equal. This leads to the following fundamental truths regarding proportional hydraulic systems:

- path *B* is always more efficient than path *A*, and
- path *A* always has lower initial cost because valves are less expensive than variable-displacement pumps, and one fixed-displacement pump can supply pressurized fluid to more than one valve and functioning circuit branch.

Electrohydraulic valves

Continuously variable electrohydraulic valves illustrate that a continuously varying control current always results in a continuously varying, controlled output variable. That output variable could be flow, pressure, or simply the position of a spool that affects final flow and/or pressure. Broad categories of these continuously variable valves are:

- direct-driven valves where the force of the proportional solenoid acts directly upon the main spool to provide the desired degree of hydraulic control, and
- pilot-operated valves in which the I_M acts first on a primary hydromechanical device whose output acts on a main spool. These are also sometimes called multi-stage valves and are either 2-or 3-stage but never more.

Direct-driven valves

Further subdivisions include those valves that use some means of spool-position feedback and those that do not. The non-feedback types simply take the force of the proportional solenoid and put it against a restoring spring. Thus, the main spool would take a position commensurate with the force generated by the solenoid if those were the only two forces acting on the main spool. Unfortunately, there are two other significant forces that act on the spool: flow forces and stiction forces.

Flow forces are a natural phenomenon in all control valves that result from the momentum change that takes place as the result of the valve's throttling effect. This occurs when potential (pressure) energy is converted into kinetic (velocity) energy in the constricting region of the valve. In spool valves, the flow force always acts to **close** the valve regardless of the direction of flow. The consequence is that when using the valve, say at a low pressure drop (flow), the spool is in a position where the solenoid force is balanced by the restoring spring. As the valve's pressure drop increases, either because of a reduction in load restriction or an increase in supply pressure, the flow force increases so as to close the valve.

As a result, the spool takes a position not totally controlled by the control current (solenoid force). This does not mean the valve does not function; it does mean that the spool's exact position at any moment is harder to predict. It is true, however, that in some direct-acting valves, flow forces can be so high that they cause an automatic near-closure at high pressures and flows. Load-dependent spool position can be detected by mathematical analysis of valve pressure drop at a steady control current but varying flows. The curve, if there is no load-induced spool shift, will relate pressure drop to the square of the passing flow. If the data does not fit the square relationship, flow forces are probably causing a spool shift.

Stiction forces also act upon the spool and the solenoid's armature, so that spool position does not smoothly vary as control current continuously and smoothly changes. Instead, the flow (spool position) has a staircase effect. Additionally, the curve trace for increasing control current is not the same as the curve trace for decreasing control current, producing *stiction-induced hysteresis*. If the valve is to be used in manually operated control systems, this hysteresis is not a major problem because a human operator can compensate easily for such performance aberrations. But when automatic controls using feedback are contemplated, hysteresis can cause a continual hunting or oscillation rather than smooth and stable operation.

Incorporating electronic dithering — that is, causing the spool to be in a continuous but acceptably small state of agitation — can help significantly to reduce the detrimental effects of stiction-induced hysteresis. When properly implemented, a closed-loop feedback control system *around the spool*, called the *inner loop* in the hydraulics industry, Figure 6, can all but eliminate stiction and flow-force effects. This loop is closed by measuring actual spool position, usually with an LVDT position transducer, and comparing it to the commanded position. If the position is incorrect, the electric current to the valve's proportional solenoid is adjusted until spool position becomes correct. Thus, the spool is always in the exact spot commanded, dynamically induced lags notwithstanding.

While not exactly true, the foregoing statement is acceptable for all practical purposes. The spool position feedback transducer of choice is nearly always an LVDT. Because LVDTs must be operated with AC voltage, they must always be accompanied by a special electronic signal conditioner that has:

- an oscillator section that generates AC voltage to excite the transformer. This AC voltage is not derived from the 60 Hz power- company line; it is generated by a solid-state electronic oscillator usually outputting a few volts at a fixed frequency, generally between 3 kHz and 10 kHz, and
- a phase-sensitive demodulator section that converts a transduced AC signal into an equivalent DC signal with the full sense of the algebraic sign of the measured position.

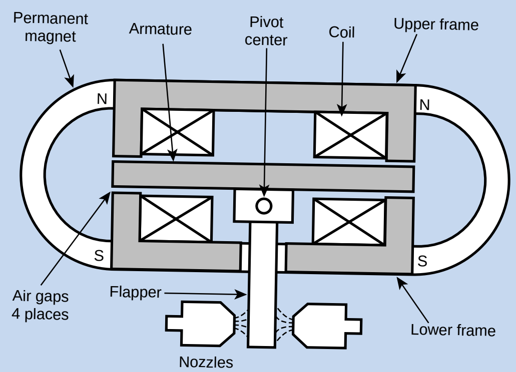


Fig. 7. The four nominally equal air gaps of an electromagnetic torque motor each carry equal magnetic flux from the permanent magnets, producing zero net torque on the armature. When current enters the coil, coil-induced magnetic flux adds to or subtracts from the four air-gap fluxes, creating a torque on the armature. Armature movement typically causes a flapper to move, changing resistivity of the two nozzles.

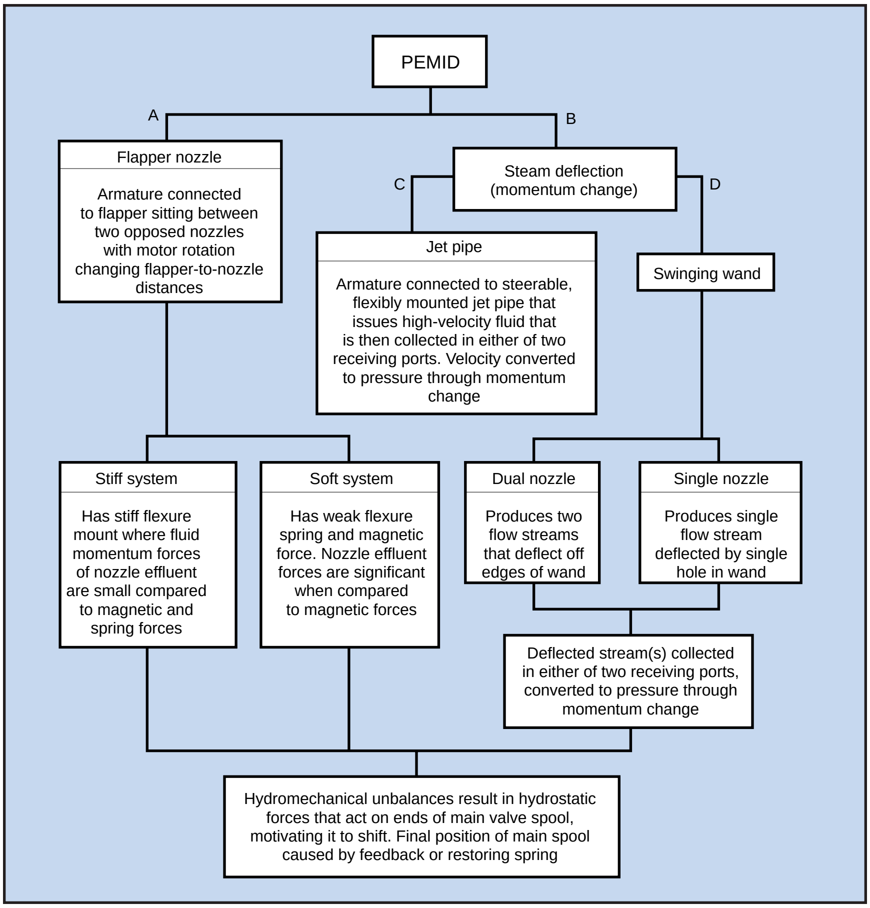


Fig. 8. A family tree of torque-motor electromechanical interfaces indicates all of the common piloting methods presently used in industry.

Pilot-operated valves

Valves with larger flow capacities need pilot stages to boost the power necessary to shift the larger spools. The electromechanical methods used for this staging are torque motors, force motors, and proportional solenoids.

Torque motors, Figure 7, are electromechanical rotary machines whose rotational travel is restricte — often to less than one or two revolutions — and are nearly always used for a piloting function. They are fitted with permanent magnets as the major flux source with the flux paths arranged to form a force bridge. Their limited rotation allows the armature to be mounted on a stiff flexure spring rather than bearings, although there is one known proprietary exception that uses a soft spring. The stiff spring and lack of bearings virtually eliminate hysteresis caused by bearing restriction.

Incoming current creates a second set of magnetic fluxes that unbalance the force bridge and results in net torque. The torque causes angular rotation until the flux-induced torque equals the counter-torque of the flexing spring plus any external load. An important characteristic of the torque motor is that the direction of rotation is affected by the direction of current through the coil. The electromagnetic field caused by the current is compared to the field of the permanent magnet in the magnetic bridge circuit and rotation ensues in a commensurate direction.

In the final valve assembly, the torque-motor armature is connected to a flapper sitting between two opposed nozzles, a jet pipe, or a swinging wand or blade. These last two steer a fluid stream, Figure 8, branch *B*. Basic operating principles and conceptual construction of flapper-nozzle and jet pipe servovalves are indicated in Figures 9 and 10, respectively. Torque motors almost exclusively pilot servovalves, and usually requires less than one watt of power to fully operate although that is not a hard-and-fast rule.

Torque from the torque motor of a jet pipe servovalve steers the jet to one receiver or the other, unbalancing spool end pressures. Movement of the main spool continues until the feedback spring between the main spool and jet forces the jet pipe back to near null. Main spool position then is commensurate with coil current.

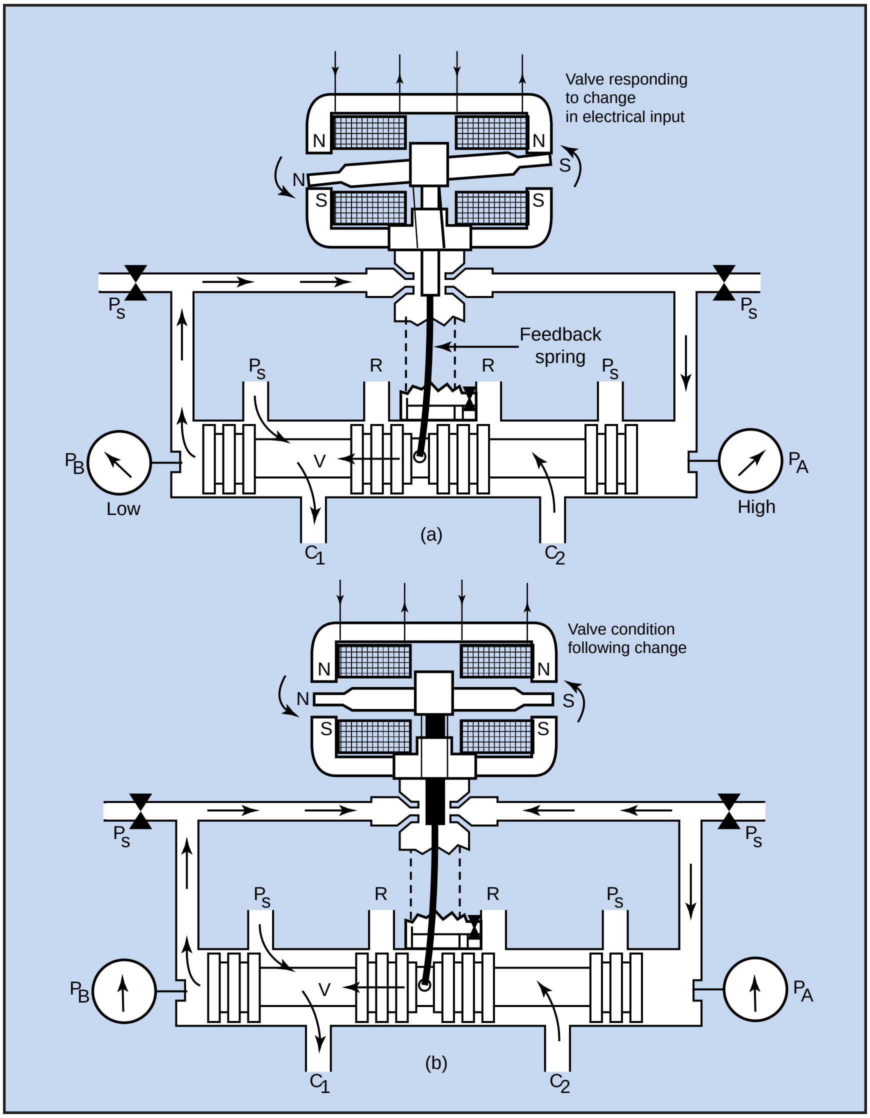


Fig. 9. Current entering the torque-motor coil, (a), causes the armature to rotate against a stiff feedback spring. The flapper, attached to the armature, blocks nozzle A and relieves nozzle B, causing pressure PA to rise and PB to fall. This unbalance moves the spool to the left. As the spool moves, (b), the feedback spring, anchored to the spool and the flapper, forces the flapper toward center. Eventually, the flapper and spool reach a position where the flapper is nearly centered, the pressures are nearly equal, and the spool comes to rest at a position commensurate with the amount of torque (coil current).

The flapper-nozzle has two different implementations: the one already mentioned is the stiff design, wherein the force due to the impinging nozzle flow is small in comparison to spring and torque-motor forces. In soft designs, the torque motor and nozzles are deliberately sized so that nozzle effluent causes a significant force on the flapper. One argument concludes that this design is more tolerant of certain contamination problems. The argument goes like this: when the two fixed orifices are open fully and unclogged, the unpowered flapper will center due to a combination of fluid-momentum force acting on the flapper, restoring force on the light spring, and magnetic force in the torque motor.

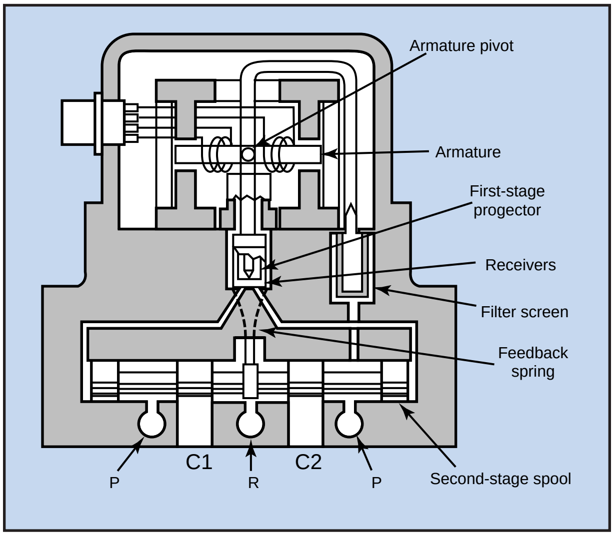


Fig. 10. Current in the torque motor of a jet-pipe servovalve steers a jet nozzle, causing a pressure difference between two collector ports. If A-port pressure is high, for example, the main spool moves to the right. Concurrently, the feedback spring drags the jet nozzle toward center and approximately equalizes collector pressures. Thus, the main spool has been positioned as directed by the coil current.

Should one of the nozzles or fixed orifices become partially blocked, reduced effluent flow produces less force on the blocked side of the flapper. The flapper then moves toward the clogged bridge leg until reduced force from the opposite receding nozzle equals the diminished flow force from the partially blocked nozzle. Current input to the torque motor then causes the flapper to move about a shifted neutral, but the pressure does not go to a hard-over level. The main spool might not shift fully in one direction, however.

The swinging wand, Figure 8 path *D*, has a mechanical-to-hydraulic interface that is proprietary. The versions of this interface include:

- dual nozzle where the two fluid streams are deflected off the outside edges of the wand, and
- single nozzle, where a single fluid stream passes through a central hole in the wand.

Consider the dual-nozzle version, Figure 11. The two fluid streams issuing from the source side of the pilot head are collected in opposing receiving ports. When a current into the torque motor causes the wand to swing, one receiving port experiences a rise in pressure while the other experiences a pressure reduction. As in the case of the jet pipe and flapper-nozzle pilots, the resulting difference in pressure shifts the valve's main spool.

The single-nozzle version has a hole laterally bored and centrally located in the wand such that the single fluid stream issuing from the single nozzle must pass through the hole. When the wand is centered, equal pressures are collected in the two receiving ports. A current into the coil causes the wand to shift and the fluid stream is deflected off the inside edge of the central hole resulting in different pressures being collected in the two receivers. The resulting differential pressure between the two receiving ports shifts the main spool.

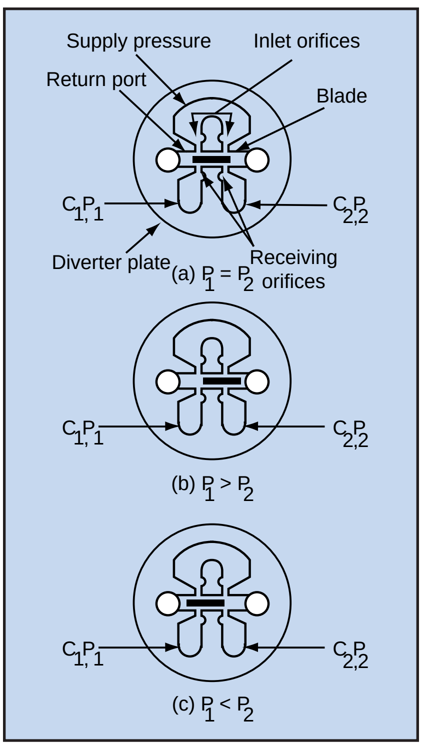


Fig. 11. The swinging-wand pilot stage generates a differential pressure in receiver ports C1 and C2 by deflecting two fluid streams off each edge of the wand. An unseen torque motor moves the wand in proportion to the amount of current. Thus the pressure difference between C1 and C2 is a reflection of coil current. Port pressures are equal, (a), C1 pressure is higher, (b), and lower, (c).

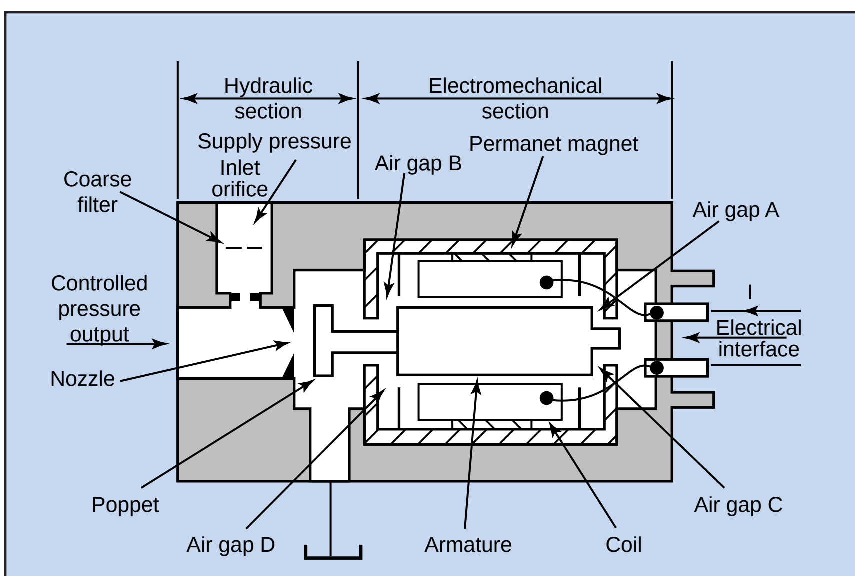


Fig. 12. A permanent magnet creates equal fluxes in the four air gaps of electromagnetic force motor that results in net zero force on the armature. Current into the coil in the direction shown, for example, strengthens flux in gaps B and D and weakens flux in gaps A and C. Now there is a net force to the left, pushing the poppet against the nozzle. Through control of force, the current controls output pressure.

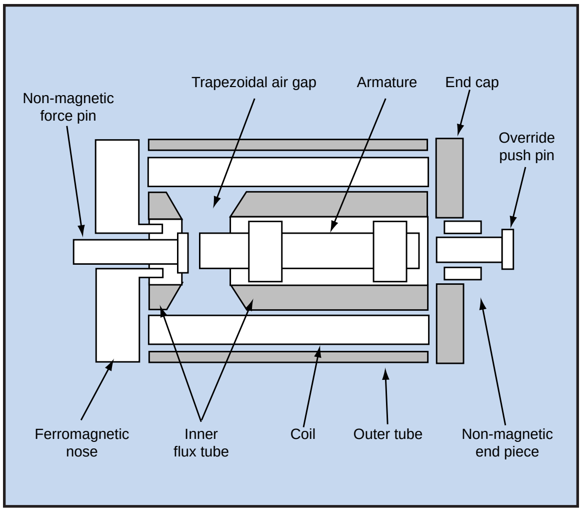


Fig. 13. The trapezoidal air gap of a proportional solenoid is shaped to create a relatively constant force regardless of armature position when the current is constant. Because there are no permanent magnets, the force is always in one direction (to the left here), regardless of current direction. Thus, bidirectional valves always require two proportional solenoids.

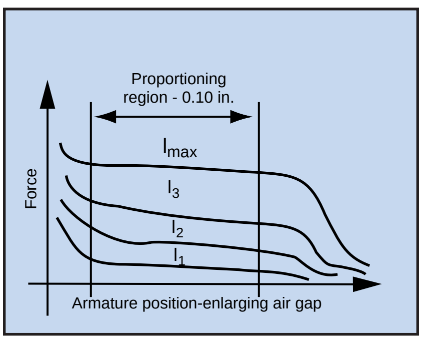


Fig. 14. Typical force vs. armature position curves show region of proportional solenoid armature travel where there is relatively constant force at constant current. Valve designers must use the solenoid so the armature operates in this proportional region. With current technology, the region is about 0.10-in. wide.

This swinging-wand design has a supply pressure limitation in that the pilot head must be sized for a particular supply-pressure range. If flow issuing from the nozzle(s) is excessive, fluid momentum force acting on the wand can pin it against the receiver side, locking it there. Installing an orifice — matched with the supply pressure and the needs of the pilot stage — in series with the nozzle side, remedies this problem. An approximately 2:1 change in supply pressure is possible with a single orifice.

Force motors are the linear equivalent of torque motors in that they also have permanent magnets inside. Therefore the direction of motion depends upon the direction of input current, Figure 12. There is only one manufacturer of force motors in the U. S.: Fema Corp., Portage, Mich. The two permanent magnets each create attractive forces, each urging the armature toward it, but nominally offsetting one another when centered. Additionally, a stiff centering spring prevents either of the natural regenerative attractive forces from pulling the armature either way.

When a current is applied in the direction shown in Figure 12, the resulting electromagnetic fields act to strengthen the magnetic fields by increased flux density in air-gaps *B* and *D* while at the same time weakening the fields in air-gaps *A* and *C*. The resulting force moves the armature and poppet to the left. State-of-the-art force-motor design produces a maximum of about five pounds of stall force, about 0.02-in. of travel (no load) at about 5 watts of power.

Proportional solenoids

Proportional solenoids are about 30 years old and are manufactured by a number of companies world-wide. Some market their solenoids to U. S. industry, while others supply themselves. All competing products have similar performance specifications. State-of-the-art proportional solenoid design yield these approximate *typical* specifications:

- maximum force, 20 lb
- proportioning travel, 0.10 in. current, 12-V coil, 1.5 to 2.5 A, and
- power, 15 to 25 watts. Figure 13 indicates approximate construction detail of a proportional solenoid. The secrets to success lie in forming the proper trapezoidal air gap dimensions and in keeping stiction low.

Figure 14 shows a representative force-displacement curve for a proportional solenoid. Its unique characteristics are a region of relatively constant force as the armature changes position, plus relatively linear changes in force for changes in solenoid current, both performance goals sought by the solenoid's designers. It is probably true that neither the constancy of force nor the linearity with current are as important as the manufacturers would claim. Things incorporated external to the solenoid substantially affect performance of the total valve: use of pressure feedback or armature-position feedback, for example. Furthermore, the enormous versatility of the hardware and software of modern computers makes control and linearization a fairly straightforward task.

Device comparisons

Some differences between proportional solenoids and force/ torque motors are apparent in the specifications while others are not. Torque/force motors require lower current levels. Proportional solenoids:

- require much higher electrical input power than their motor counterparts
- produce substantially greater mechanical travel than motors
- produce higher levels of force
- produce higher levels of stiction
- operate with greater hysteresis, and
- generate force in a direction independent of current direction. Therefore, to make a 4-way directional valve operate requires two proportional solenoids but only one force/torque motor.

All of these factors make proportional-solenoid drive electronics more complex than that necessary for force/torque motors. Proportional-solenoid power requirements have caused manufacturers of proportional-valve drive electronics to adopt pulse width modulation (PWM) as the power-output method of choice. The major reason for the use of PWM is to handle the high-output power required of the solenoid without overburdening the power-output transistors.

There is a second benefit of using PWM: if the PWM frequency is sufficiently low, it automatically provides a mechanical dither that helps minimize stiction-induced hysteresis. In some valves the effects of dither can only be described as dramatic when looking at the reduction in hysteresis. The correct dither frequency must be determined after the valve is designed. Furthermore, the frequency selected must be a compromise between propagating the dither pulsations imperceptibly into the hydraulic circuit and yet achieving sufficient reduction in stiction. A low frequency helps the stiction problem but if too low, the user of the hydraulic system can feel the pulsations.

U.S. industry uses PWM frequencies from about 33 Hz to about 400 Hz. At least one European manufacturer uses 40 kHz and receives no dither effects whatsoever. Their amplifier supplies dither with a separate on-board dither generator. There is an advantage to this method: dither power remains constant throughout the modulation range, whereas when relying on the PWM frequency for dithering, the dither power varies with the amount of modulation. There is none at the 0% and 100% modulation points, but maximum at 50% modulation.

Summary of pilot-operated valves

Figure 15 shows a family tree of all electrically modulated, continuously variable pilot-operated valves. It is a peculiarity of the U.S. hydraulic-valve manufacturing industry that each terminus of the tree also tends to define a specific manufacturer's product. For example, the *A/C/E/I/N/V* path rather accurately describes the servovalve built by Sauer-Danfoss's Controls Div. in Minneapolis. (Note that its mate, *U*, has no supplier.) In contrast, though, the *A/C/E/J/O* path is well populated. That is where the products of Moog, Eaton/Vickers, Bosch-Rexroth, Parker Hannifin's Dynamic Valve, and others have congregated.

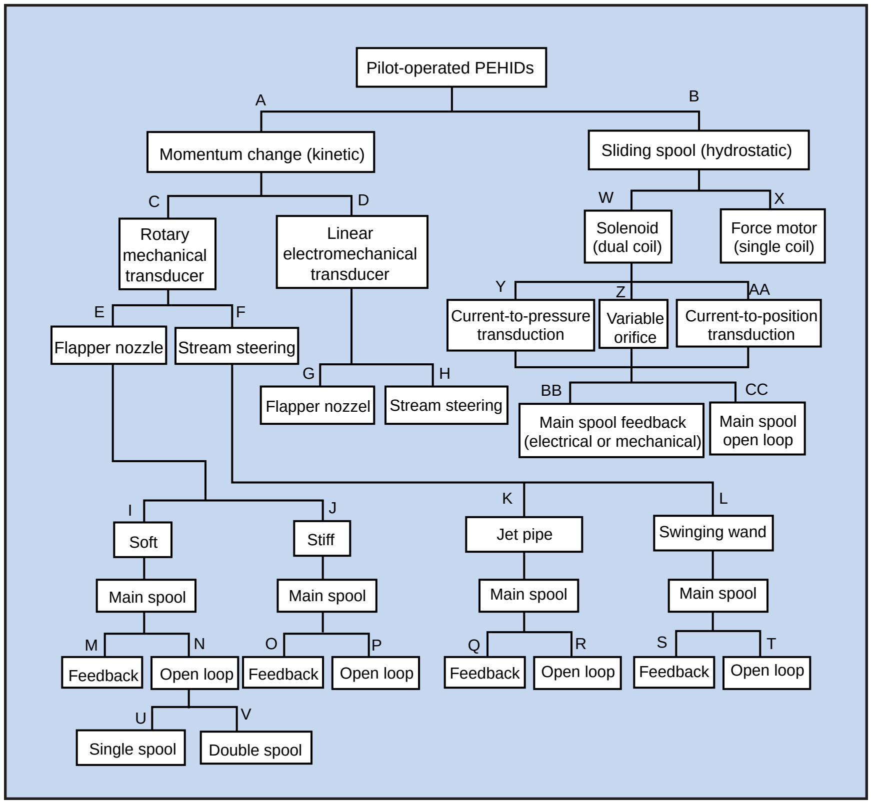


Fig. 15. Family tree of pilot-operated PHEIDs.

For complete listings of manufacturers of electrohydraulic proportional and servovalves (including addresses, phone and fax numbers, and e-mail and Internet addresses), consult the **Section B** of this directory. Detailed product information for participating companies can be found in **Section C**. Local sources of products, including distributors and manufacturer's representatives, can be found in **Section D**.

For specifications on products, such as flow and pressure ratings, frequency response, etc, consult the Annual Designer's Guide to Fluid Power Products, which is published in every issue of Hydraulics & Pneumatics magazine. This Designer's Guide can also be accessed on the Internet at our website, Fluid Power Web. Go top **www.fpweb.com**, then clisk on the Deisgner's Guide button.