Chapter 15 Proportional valves

15.1 Introduction

As we mentioned in Chapter 14, the name proportional can be used to describe any action where one parameter varies in some proportion to another. In the case of a servovalve, the spool position moves in proportion to the input signal. If closed loop velocity control is used, the velocity changes proportionally to input current.

In the hydraulics industry, the term proportional valve refers to a specific type of valve which is quite distinctive to servovalves. Thus, when we talk about proportional valves, we are talking about a solenoid activated valve with very distinctive operating characteristics.

The function of proportional valves is to provide a smooth and continuous variation in flow or pressure in response to an electrical input. In these valves, it is important to link the electronics to the valve very carefully. This will become quite evident later on.

At this point, we shall try to define the differences between the more conventional "servovalves" and "proportional" valves. Essentially, it is the construction and flow-pressure-current characteristics which mark the differences.

Proportional Valve	**Servovalve**
- Open loop control	- Close loop control
- less costly than servovalves	- very expensive
- require more power (50W)	- low power input (.1 - .3W)
- moderate filtration (30 μm)	- high filtration (1-5 μm)
- spools are overlapped	- spools critically lapped
- flow-current characteristics very nonlinear	- flow-current characteristics very linear
- hysteresis large 0.5%	- very low hysteresis 0.1%
- can be used as flow, pressure and directional control valves (pressure compensation)	- used primarily in closed loop to create flow and pressure control
- can be used in closed loop control if expectations not high	- when used in closed loop, high performance is expected

In general proportional valves find most of their applications in open loops situations where pressure and flow are required to change continuously, where multiple fixed flow and pressure valves can be replaced by a single valve and where acceleration and deceleration under control are required.

In the following sections, we shall discuss the basic principles of operation of the proportional valve, its characteristics, its use in pressure, flow and directional control and some applications.

15.2 Basic Operating Principles and Characteristics

We know from our basic electronics that applying a current to a coil creates a magnetic field which when passed through an appropriate core material (called a pole piece) can result in a "magnetic force". The force when applied to a movable armature will result in a motion towards the core (the basic solenoid concept). In traditional solenoids, the force of attraction increases with decreasing gap between the armature and the pole piece. However, proportional solenoids are shaped (armature, coil and pole piece) such that the force of attraction is relatively constant over the stroke. This is illustrated schematically in Figure 15.1.

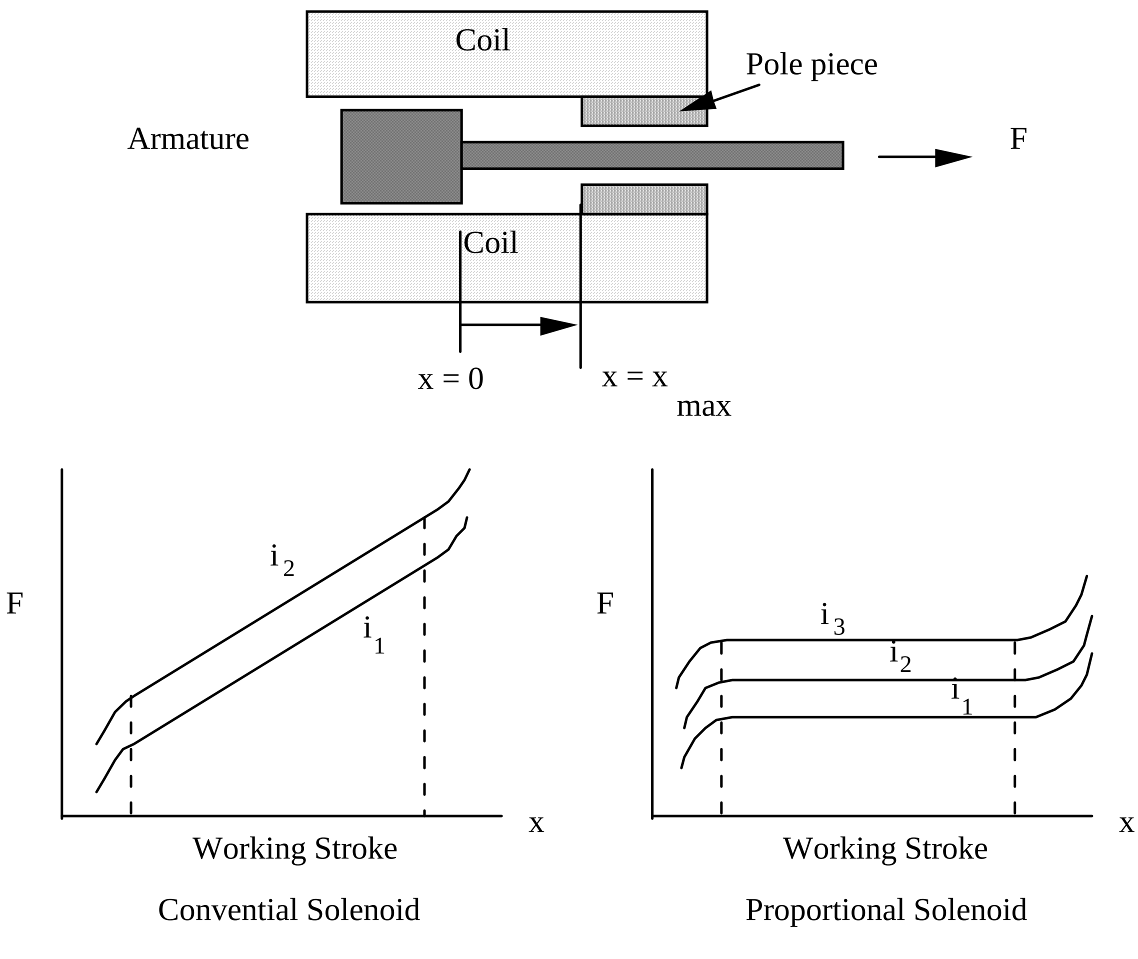

Figure 15.1 Proportional Solenoid

What this means is that if i is held constant, then the force of attraction is approximately constant over the working stroke. Thus, $F \alpha i$; $F = k_i i$. (Hence the name proportional solenoid).

As it is shown in Figure 15.1, the proportional solenoid is useless. The armature will accelerate uncontrollably until it hits the pole piece. In order for the armature to assume some defined and stable position, it must be "balanced" by some external force or the current must go to zero. Consider Figure 15.2 (a), we can balance the solenoid armature force with an external spring. The spring will compress until the force of the spring is equal to the armature force (a stable position). Increasing the current, increases the armature force resulting in an imbalance force on the spring, which compresses the spring until a balance is re-established.

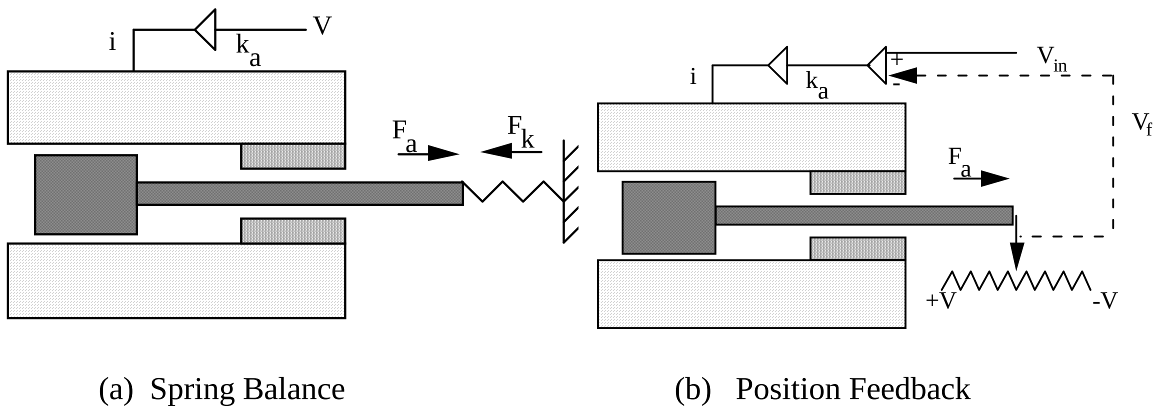

Figure 15.2 Controlling the armature position

We can analyze this from a transfer function (steady state only) point of view.

$$F_i = k_i\, i$$
$$F_k = k_x\, x$$

When balanced, $F_i = F_k$ or

$$k_x\, x = k_i\, i$$

or $$x = \frac{k_i}{k_x} i$$

Thus, for a specific value of k_i and k_x, we can say that the spring-armature position is proportional to the input current. If now we put a spool, poppet etc. between the armature and the spring, then all of a sudden, we have a spool (poppet) position that is proportional to the input current (potential applications should jump out at us).

This is the simplest application. Unfortunately, "things happen" at the spool such that the balancing forces are a combination of other factors (nonlinear) which means that $F_i \neq h_x$ x but $F_i = k_x$ x + other goodies. Thus, we could use the approach in Figure 15.2 (b) in which position feedback can be used. Here, the balancing force is due only to friction, flow forces and inertial effects. This approach relies on F_i going to a small value for any given input as the armature approaches its final position. Indeed, it is the traditional feedback controls principle in which the solenoid is driven by an error signal. At this point, it should be noted that without feedback, the hysteresis is quite severe, a problem in most proportional valves. The electronics are designed to minimize this effect but it still persists even after compensation.

In actual fact, a combination of springs and feedback are used together in most commercial proportional solenoids. This is shown in Figure 15.3 with a symbolic representation given in Figure 15.4.

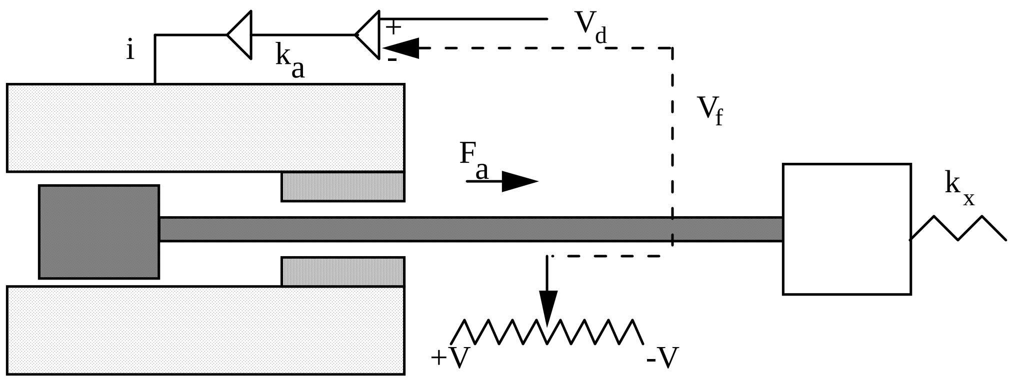

Figure 15.3 Basic proportional valve configuration with spring balancing and position feedback

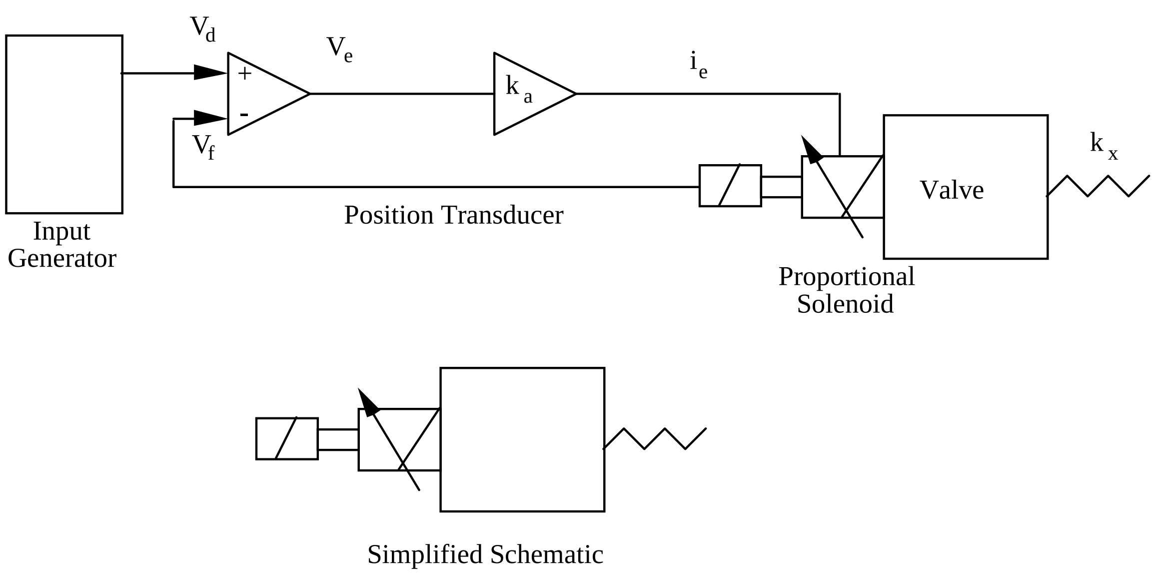

Figure 15.4 Symbolic schematic of proportional valves

Now, this does create some problems. If we think about what would happen in control terms, we can deduce that the proportional amplifier has to be a pretty large current amplifier to reduce steady state error. A spring when compressed must have a compression force on it. However, with feedback, if the final position has been reached, then the error is zero. Problem! If the error goes to zero, then the armature force goes to zero which is opposite to what we want. Thus, to balance the spring force, we must have a steady state error. From a controls points of view we can express this as follows: (Reference Figure 15.4).

$$F_i = k_i\, i_e \quad , \quad F_k = k_x\, x = F_i$$
$$V_f = k_f\, x \quad , \quad i_e = k_a\, V_e$$
$$V_e = V_d - V_f$$

∴ Our control "gain" diagram becomes (Figure 15.5)

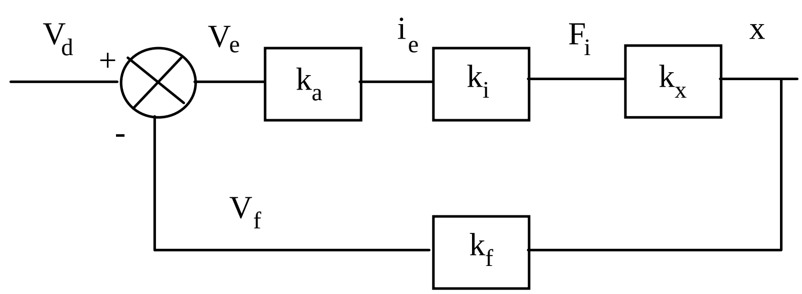

Figure 15.5 Control diagram of a proportional valve

Thus, $$\frac{x}{V_d} = \frac{k_a k_i k_x}{1 + k_a k_i k_x k_f}$$

Or $$\frac{V_e}{V_d} = \frac{1}{1 + k_f k_x k_i k_a}$$

If k_a is made large, the steady state error decreases. However, if all parameters are known, then the relationship between x and V_d is well established i.e. $x \alpha V_d$ which is what we really want.

At this point, we are going to insert a special section of the operation of solenoid valves written by My Brad Hicks who at the time of revisions of these notes is a full time engineer and a graduate student. He has graciously agreed to allow us to copy this description to our notes. Our thanks to him for this. Brad graduated from College in AgBio and took this class many years ago.

15.2 (A) Special supplemental notes on Solenoid Valves

By Brad Hicks, P.Eng

Solenoids are electro-mechanical interface devices that convert electrical input into an applied force (Hardwick 1984, Xu et al 1991, and Vaughan et al 1996 referenced throughout this discourse). Hydraulic valves utilize this principle to shift an internal component (spool) thereby altering flow supply or direction to a load. There are two variations of solenoids currently available in the marketplace; on/off and proportional. On/off type electro-hydraulic valves use a solenoid to move the valve spool from one position (off or no flow) to another position (on or maximum flow). A mechanical spring opposes spool movement and returns the spool to the opposite position once electrical power has been removed from the solenoid. This operation can be classified as *digital* since the flow paths can be in only one of two states. On the other hand, proportional type electro-hydraulic solenoid valves can position the spool at an infinite number of locations between the start and end positions. The mechanical spring employed in this application provides both the means for proportionality and again returns the spool to the start position when no electrical power is supplied. Consequently, proportional solenoid valves can be classified as *analog* devices which provide more functionality and opportunities within the field of electro-hydraulics.

15.2.1 Solenoid Operation

Both on/off and proportional solenoids have the same basic construction; a wire coil surrounds a ferromagnetic core with a paramagnetic guide tube (Figure 15.A1). A ferromagnetic armature is allowed to move linearly within the guide tube. When an electric current is supplied to the coil a magnetic field that intersects the core and armature is established. The coil creates two magnetic poles at the opposite ends of its length, where the flux paths circulate. These magnetic flux paths generate a linear force attempting to center the armature within the coil length (between the electromagnetic poles). An on/off solenoid's magnetic force increases as the gap between the core and armature decreases (classified as a *variable gap device*) because the reluctance of the magnetic flux circuit is decreased. The key difference between on/off and proportional solenoids is the shape and orientation of the core and armature to create a *constant gap device*. A proportional solenoid is constructed such that the gap is perpendicular to the direction of armature movement and therefore independent of armature position (Figure 15.A.2). Thus, for a given current through the coil a constant force over the working range of armature movement (spool stroke) is created. Examining the equation for force generated by either an on/off or proportional solenoid can show this principle:

$$F_M = \frac{\mu_o A}{2f}\left(\frac{I - I_o}{x + l_e}\right)^2 n^2 \qquad (1)$$

where:

	μ_o	= permeability of free-space [W/A m],
	A	= cross sectional area of the gap [m^2],
	f	= factor for flux leakage and area replacement [dimensionless],
	I	= input current [A],
	I_o	= initial current of the solenoid [A] (typically 4-7% of steady-state current),
	x	= the linear gap between armature and core [m],
	l_e	= equivalent reluctance length [m], and
	n	= number of coil turns.

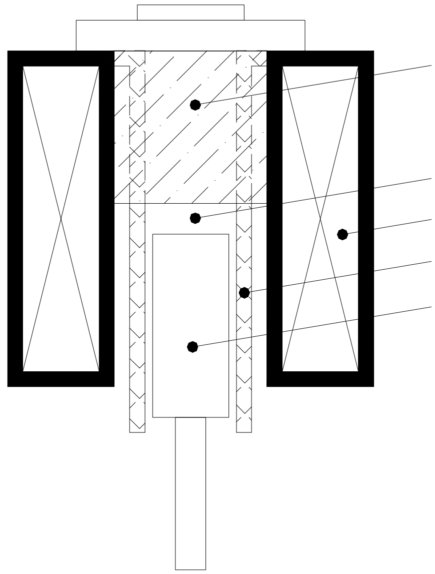

Figure 15.A.1: Solenoid Construction

Neglecting the influence of flow reaction forces, it can be shown using Newton's first Law of Motion that the forces acting on the armature at steady state are (Figure 15.A.3):

$$\sum F = 0$$
$$F_M = F_S \qquad (2)$$

where: F_S = Force applied by mechanical spring [lb].

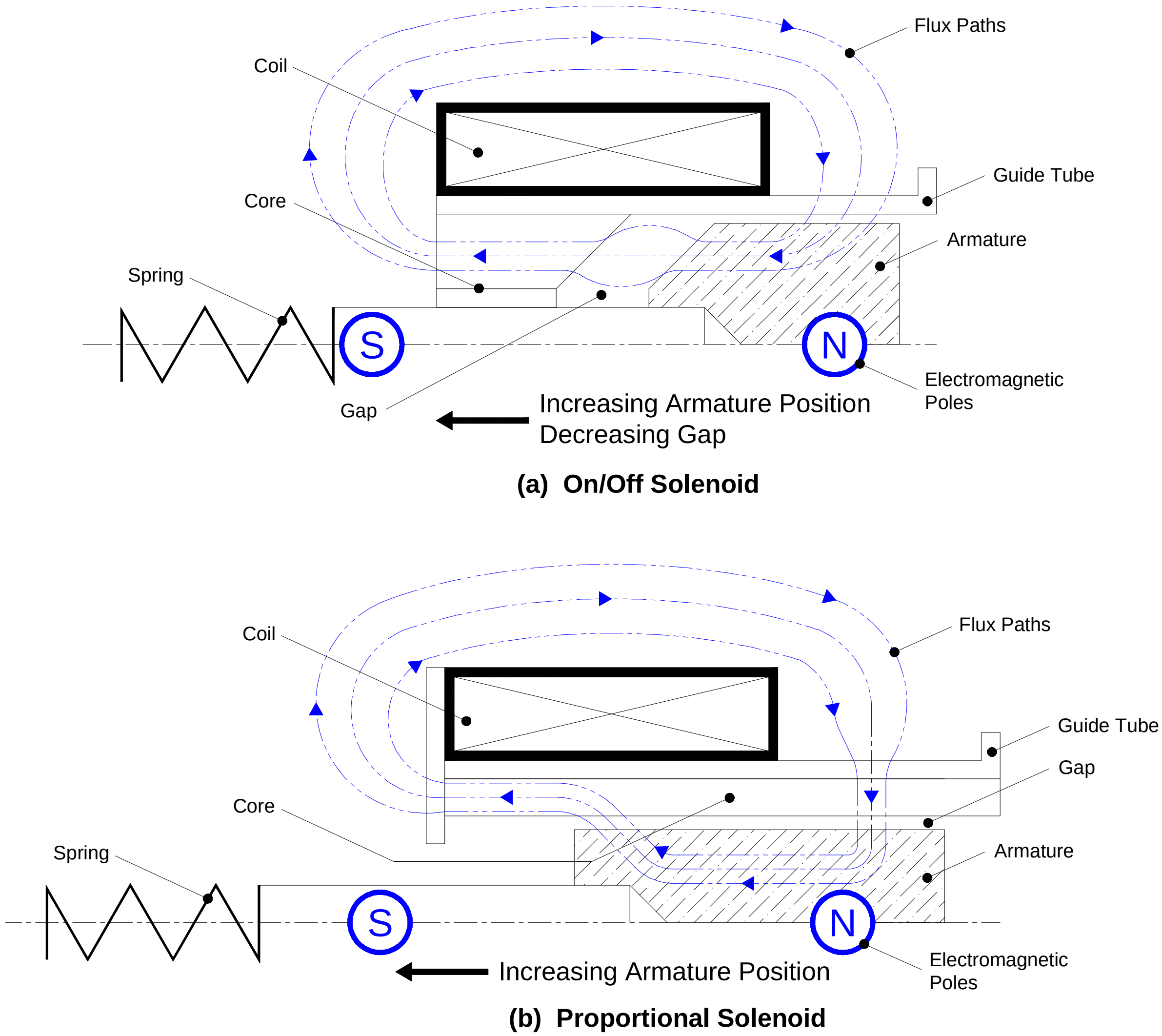

Figure 15.A.2: On/Off versus Proportional Solenoid Gap Construction

Using equation (1) families of force versus armature position (spool stroke) curves for varying input currents are generated for both on/off and proportional solenoids (Figure 15.A.4). For an on/off solenoid the force generated increases as x (linear gap) decreases. By superimposing the return spring curve on this family of solenoid curves it is shown that there are a limited number of intersection points where the spring force and solenoid force would be in equilibrium. On/off solenoids are designed for single current supply systems, where the input current creates a large enough force to exceed the spring force at all positions, moving the spool to its maximum displacement.

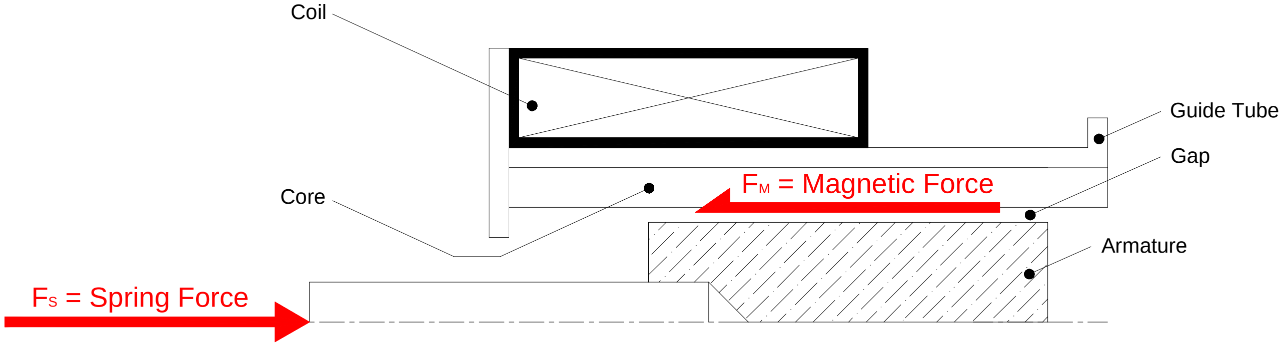

Figure 15.A.3: Forces Acting on Armature

With a proportional solenoid the gap is constant throughout the spool stroke. Thus, when the return spring curve is superimposed on the proportional solenoid family of input current curves many intersection (equilibrium) points are found. By utilizing a variable current supply to change the input current many spool positions can be created; thus spool position is proportional to input current.

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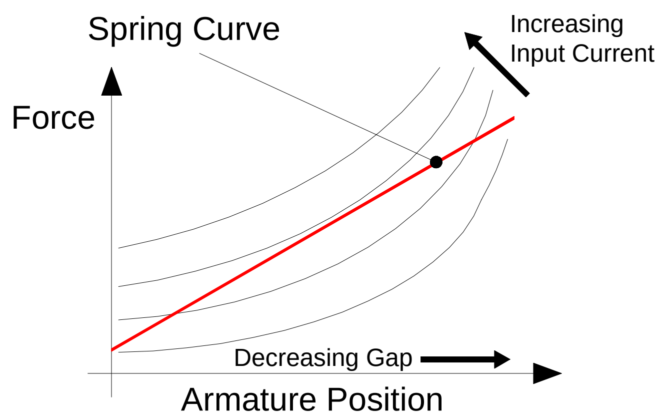
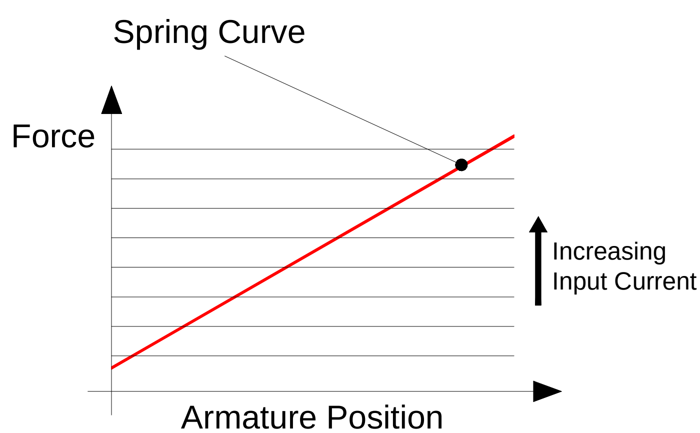
Vaughan, N.D. and Gamble, J.B. "The Modeling and Simulation of a Proportional Solenoid Valve." Journal of Dynamic Systems, Measurement, and Control v.118 (1996) p. 120-125.



(a) On/Off Solenoid



(b) Proportional Solenoid

Figure 15.A.4: Solenoid Force versus Armature Position – On/Off and Proportional Solenoids

15.3 Valve Applications

15.3.1 Throttle (Choke) Value

The simplest application of using a proportional solenoid is that of a throttle valve. A throttle valves creates a variable orifice in a line. If the upstream pressure is at the dead head/R.V. setting, then flow through the valve is proportional to the product of the orifice area and the pressure drop across the valve. i.e. Q = $kA_o(x)\sqrt{\Delta P}$. For any ΔP, we can change the flow (and hence the velocity of the actuator) by changing the orifice area. Flow is in one direction only and is not controlled (because ΔP can change even though A_o (x) is controlled). A schematic is shown in Figure 15.6.

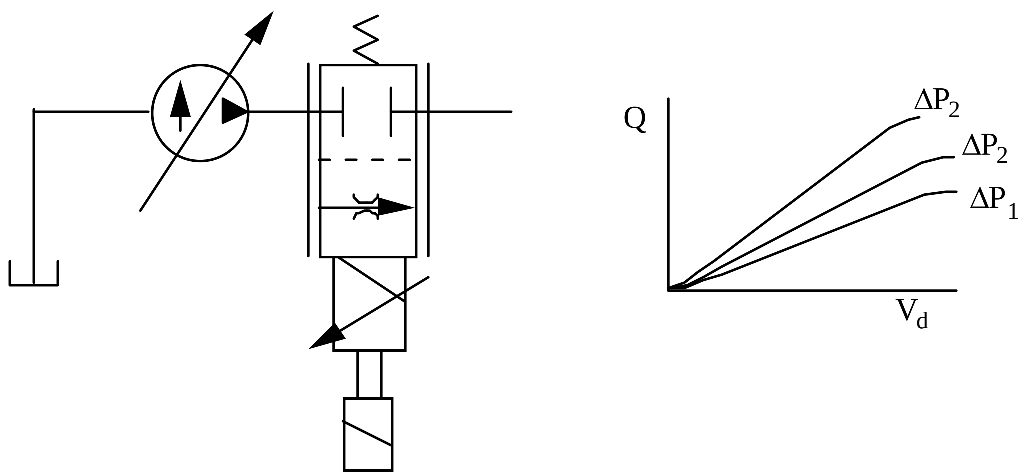


Figure 15.6 Throttle valve application

Theoretically, Q should be proportional to input voltage for a constant ΔP. However, due to dead zone (discussed presently) and losses within the valve, the flow characteristics appear nonlinear as illustrated in Figure 15.6.

One of the advantages of using proportional valves lie in their cost. However, if a zero lapped spool is used in the valve part, the costs escalate significantly. Thus, most spool valves are slightly over lapped to reduce machining costs. This creates a dead zone where the spool moves but no flow occurs. However, most of the commercial proportional amplifiers available have compensation loops built into them which provide a “bias” signal at x = 0 to move the spool quickly over the dead zone point. Linearity, however, is never restored and thus the flow characteristics will always be slightly non ideal.

We can get cute and make this valve a pressure compensated flow control (two way) valve by placing a “hydrostat” upstream to the proportional valve as illustrated in Figure 15.7.

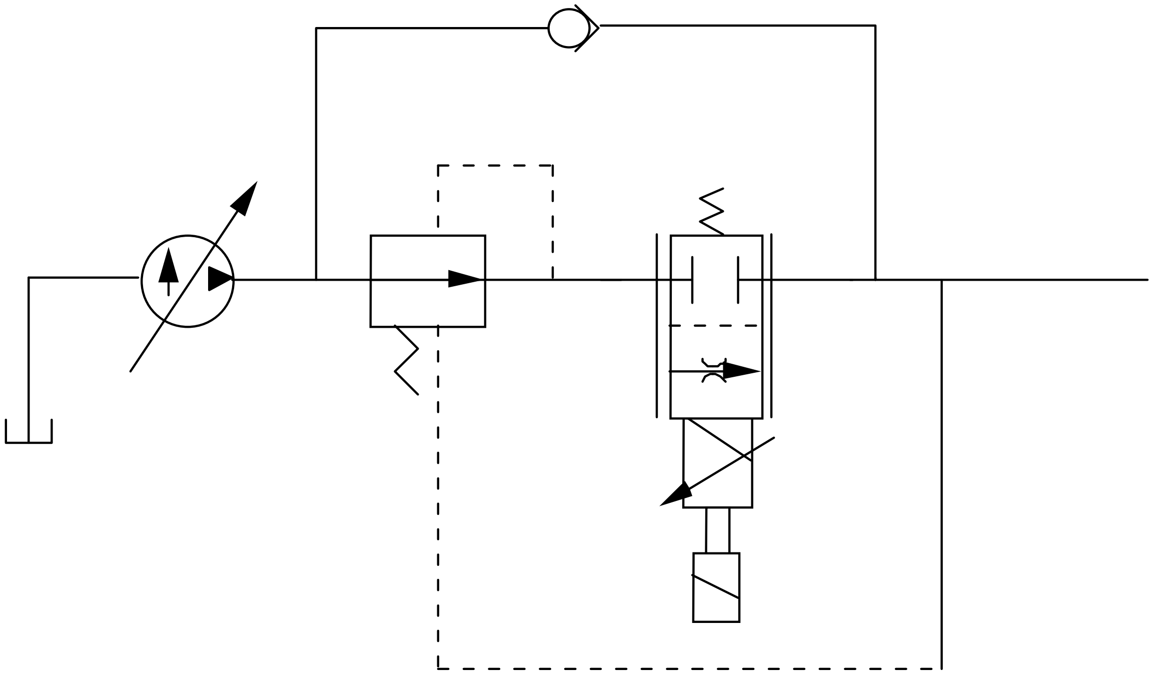


Figure 15.7 Pressure compensated flow control (two way) valve

It should be noted that two stage pilot proportional flow control valves are available to handle large flows. In these valves, the feedback of position comes from the main stage rather than from the pilot.

15.3.2 Proportional Directional - Control Valves

A logical extension to the proportional throttle valve is the directional control valve. A schematic is shown in Figure 15.8.

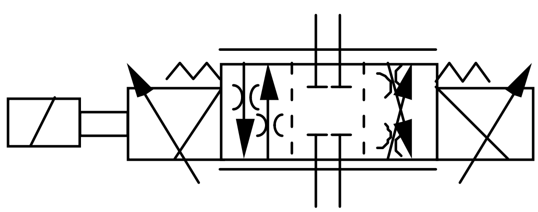


Figure 15.8 Proportional directional control valves

The spool has symmetric metering orifices and two proportional solenoids (one for each direction). The amplifiers know which solenoid to activate via the position transducers.

The amplifiers are designed to reduce the dead zone to about 5% of full stroke (compared to 15.25% with no compensation). By changing the shape of the metering orifices, it is possible to have different resistances in the various parts (Figure 15.9).

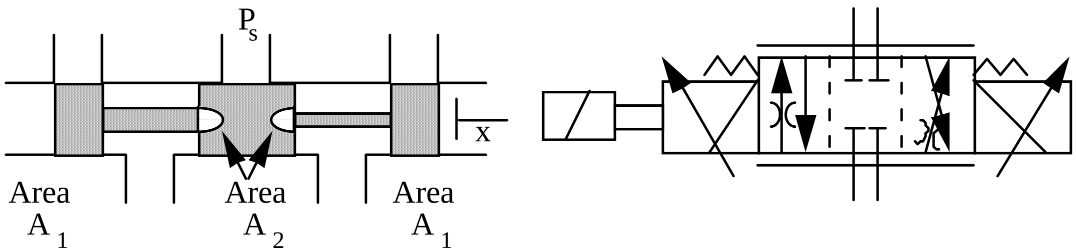


Figure 15.9 Different metering orifices

In this case, as the spool is moved to the right (x_v +) the orifice A_1 is much larger than at A_2 for any x_v. Although metering does occur over A_1 it is small compared to the area A_2. Thus we have "metering in" but limited metering out.

The proportional directional control valve can be made into a pressure compensated flow control valve by employing a hydrostat and a shuttle valve as illustrated in Figure 15.10.

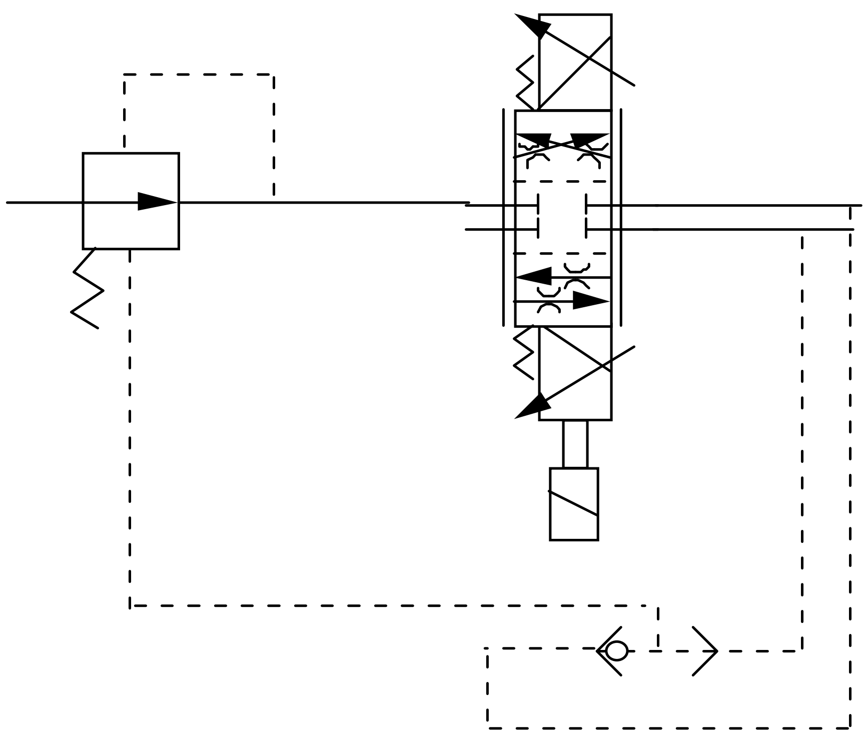


Figure 15.10 Pressure compensated flow control valve (proportional)

To handle large flows, a pilot operated proportional flow control valve is recommended. The pilot spool is somewhat different than normal in that it has four positions as shown in Figure 15.11 (because pilot flows are very small, only one solenoid is used). The fourth position (#4 in Figure 15.11) is used in a fail position if loss of power occurs. The spring pushes the spool to this position. The two "ends" of the main stage spool are connected together and allowed to bleed off to tank. When this happens, the main stage spool is centered via the two centering springs. The load ports are either locked or "shorted" as dictated by the center position. The three right hand positions are the normal operating regions when the solenoid is actuated. The normal null position is #2 (closed or open center). We must note that the current is not zero at this point. (If it was, the spool would shoot over to position #4). To limit the pressure to the pilot valve and make its operation less sensitive to pressure variations in the downstream load, a pressure reducing valve is often incorporated in the supply pressure line (pilot).

A hydrostat is included before the main stage valve for pressure compensation. Special internal drilling in the main spool allows the upstream pressure from the actuator to be sensed in either direction. This kind of two stage valve can be used to control large flows with reasonable accuracy.

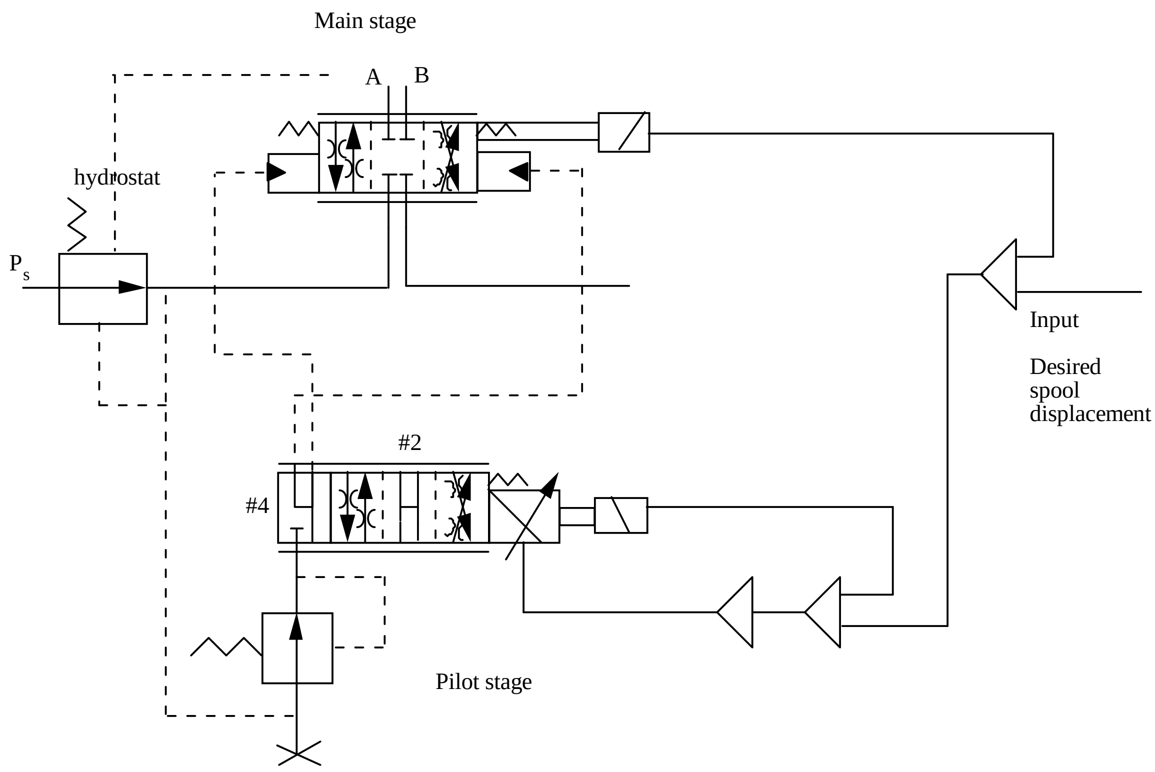
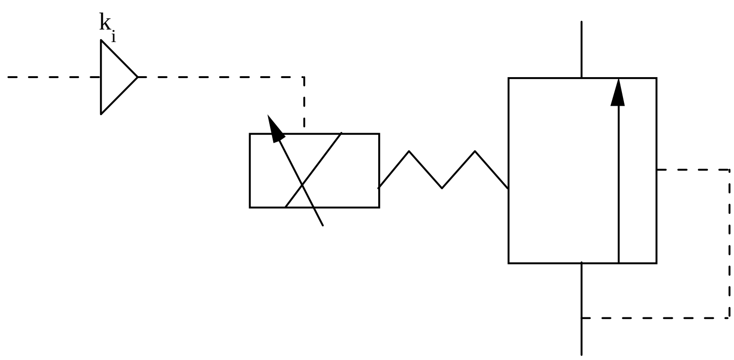


Figure 15.11 Pilot operated pressure compensated proportional flow control

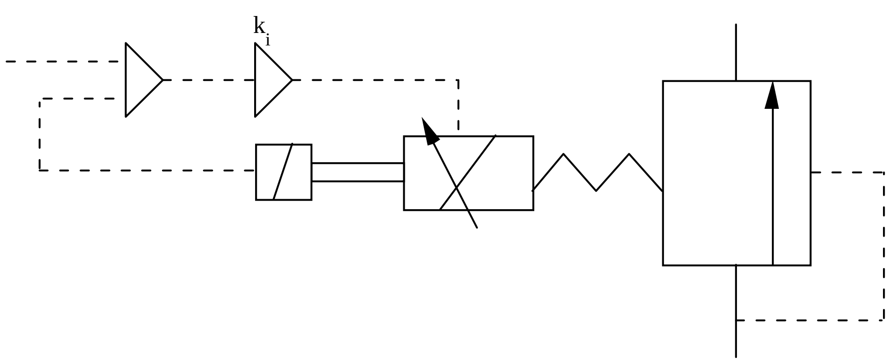
We should note that two feedbacks are normally used in these two stage valves. The first feedback is used to ensure that the pilot spool goes to the null position (#2) when the error input signal commands it to do so. The second transducer feedback is used on the main stage spool. If one examines the control diagram of Figure 15.11, it becomes evident that the error signal (the difference between the actual main stage position and the desired position) is used to drive the pilot stage. It must also be that the amplifier driving the pilot stage must have a bias to ensure that position #2 (null) is achieved when the error signal is zero (otherwise, as stated above, the error spring would push the spool back to position #4). The main reason we use feedback on the pilot stage in this case is to ensure that friction forces etc. can be overcome when needed.

15.3.3 Proportional Pressure Control Valve

If we recall that to set the pressure of a relief valve, reducing valve, counterbalance valve, sequence valve, pressure compensator etc., all we had to do is to adjust the spring pretension by compressing the spring via some "knob". Well if we put a proportional solenoid to compress the spring, we can pre-compress the spring remotely. Indeed, this is the exact scenario we talked about in Section 16.2. Such a scheme is illustrated in Figure 15.12. This valve is a single stage direct operating relief valve where the solenoid is used to compress the R.V's poppet spring to give a required cracking pressure.



(a) without feedback



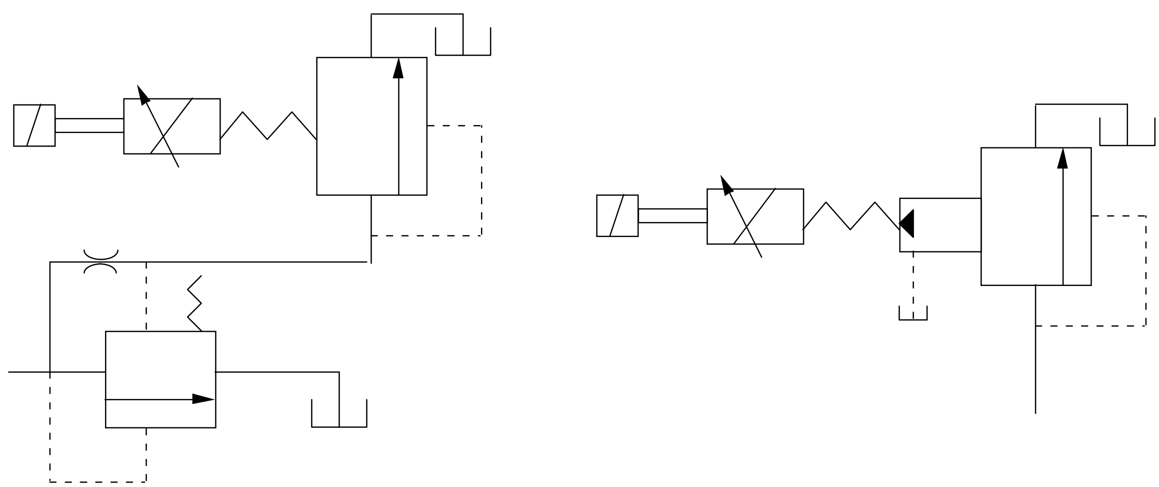
(b) with feedback

Figure 15.12 Single stage proportional relief valve

To get better control on the cracking pre-set setting a position feedback is used as in Figure 15.12 (b). This kind of valve really is only good as a pilot stage (low flows < 3 ℓpm) and should be incorporated with a main stage to form a two stage proportional pressure valve.

A schematic of a two stage relief valve is shown in Figure 15.13 (a) with a simplified version illustrated in Figure 15.13 (b).

By changing where the pressure is sensed, or by changing the normally open or closed position of the main stage, all types of pressure controls can be handled including pressure reducing, pump pressure compensator, counterbalance etc.



(a) Pilot operated pressure RV (b) Simplified symbol

Figure 15.13 Proportional pilot operated RV

15.4 Electronics (Proportional amplifiers)

As we have implied in the preceding sections, the proportional solenoids and associated valves can be used in applications where pressure or flow needs to be varied in a continuous fashion on demand. The main problem with these devices is the fact that high performance is not one of their strong points. Hysteresis in spring applications, and dead zone in valve applications are quite severe. Using position feedback can reduce the magnitude of these problems considerably. The amplifiers for these devices are quite sophisticated but add to the overall cost of the product.

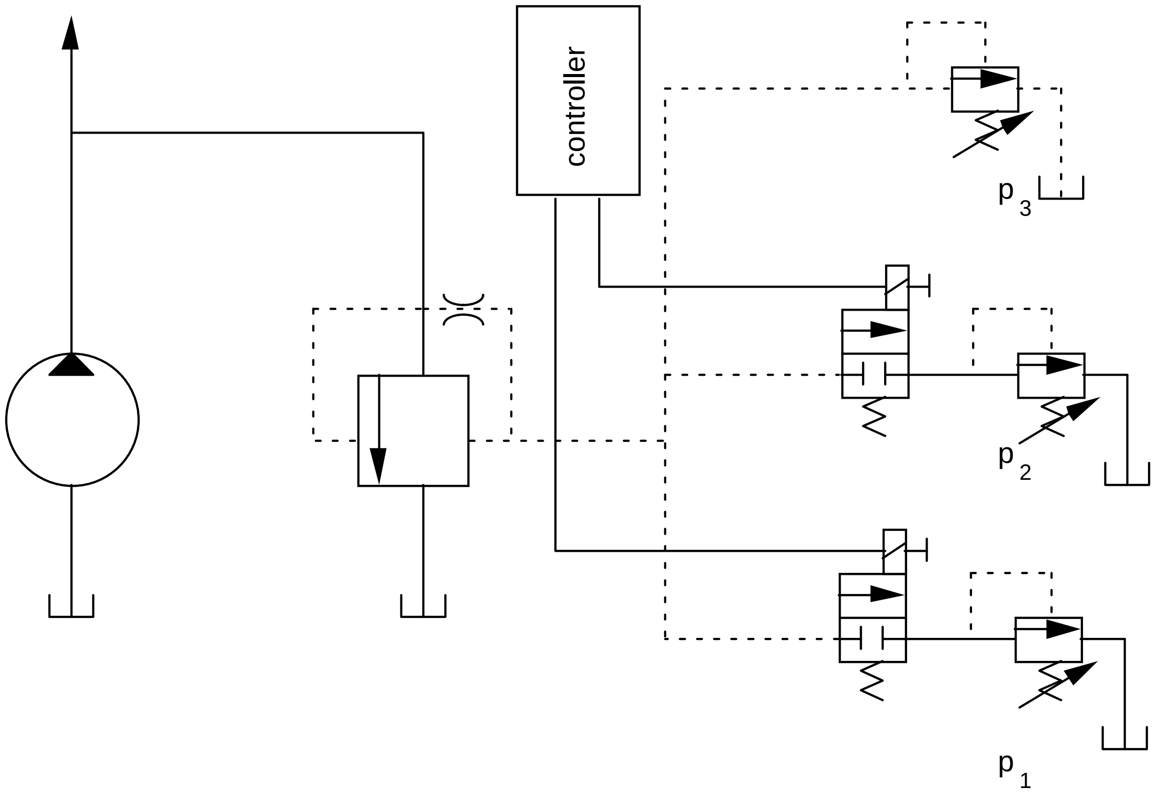
The amplifier can output 2-5A (required to drive the solenoids) with an approximate power output of 40-50 W. We shall not go into detail about the operation of the amplifier but shall summarize the various options/performance that can be found in commercial amplifiers.

1. Settings to compensate for the dead zone.
2. Settings to adjust the maximum output to input ratio (gains). This is important in applications where the gain is required to be different in different directions of the actuator (e.g., to compensate for the area differential of an actuator in a meter-in application)
3. Feedback comparators
4. AC signals to drive the LVDT's (position transducers)
5. Power (current) amplifiers to drive the solenoids. It should be mentioned here that often, the solenoids are driven by a pulse width modulated signal. P.W.M. signals are constant amplitude and frequency pulses where the widths are varied in accordance to an input signal. If the input signal is zero, the width of the pulse is zero. If the input signal is maximum, the width of the pulse is maximum (always on).
    i. Intermediate input signals vary the width accordingly. The basic concept is that the system the PMW signal is driving, reacts to the average of the pulse signal integrated over the full period - that is the physical system filters the pulses and
    ii. reacts to the average. This has been shown to create a small high frequency dither on the armature which can help overcome stiction but not create ripples on the pressure or flow downstream to the valve.
6. Ramp generators. For acceleration and deceleration control, these generators are extremely valuable and can be programmed for rate, magnitude and time when activated. These units can also detect actuator direction reversal and ensure that acceleration ramps are maintained properly with direction reversal.
7. They can often be integrated with process controllers so that a programmed sequence of events can be detected by a central unit.
8. Other features can be included but they do cost big bucks.

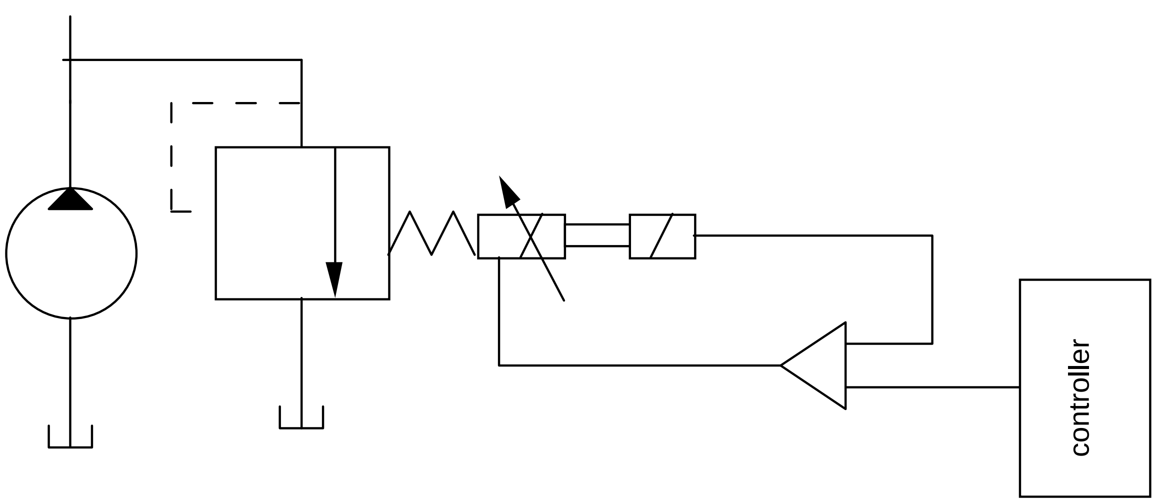
15.5 Applications

15.5.1 Multiple R.V settings

If we used relief valves and solenoid on/off valves, we would need N pilot relief valves and N-1 solenoid on/off valves. If we used proportional relief valves, our circuit becomes far easier to implement. Not only this, we have an infinite number of pressure settings which can be changed continuously if we want. These circuits are illustrated in Figures 16.14 (a) and (b).



(a) Digital approach to multiple pressure limits

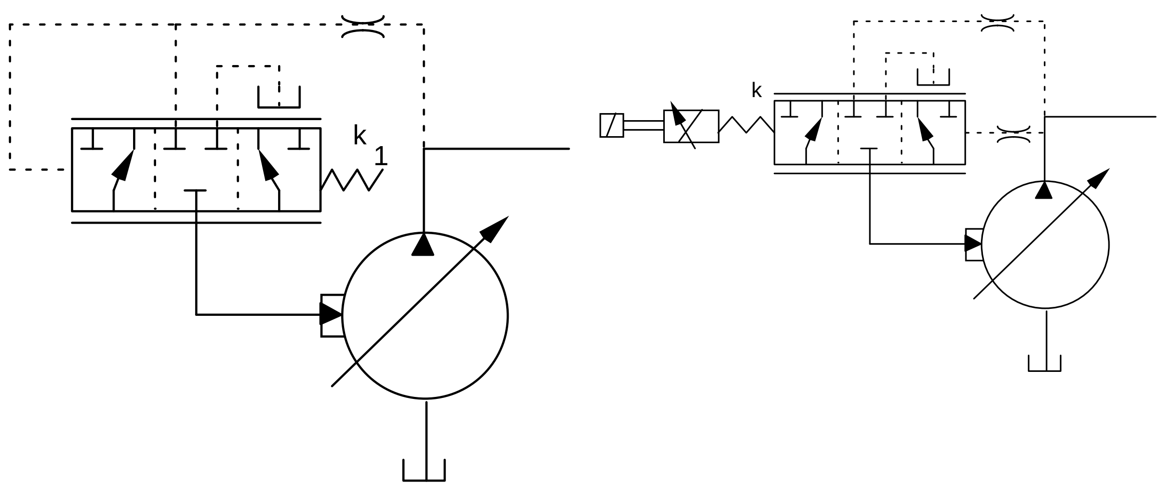


(b) Proportional relief valve

Figure 15.14 Comparison of digital vs. continuous pressure limit settings

15.5.2 Flow and Pressure Control of Pumps

We have discussed pressure compensated pumps. To change the cut-off pressure, we must change the spring pre-compression. (Same scenario as with R.V's). The use of a proportional solenoid appears to have an application here. Consider Figure 15.15.



(a) Conventional (b) Proportional with pressure valve

Figure 15.15 Pressure compensation

This is the classical pressure compensated pump. The dead head setting is adjusted by the spring k_1. Using a proportional pressure valve, we get a remote control of the dead head setting from an electrical amplifier process unit.

If we want flow control, we can use load sensing concepts in which a pressure drop across an orifice is maintained. This is illustrated in Figure 15.16. Replacing the manual flow control valve by a proportional solenoid valve allows us to change the flow rate remotely from a processor / amplifier unit. The pressure drop across the flow control valve is dictated by the spring constant k. We can add a proportional pressure controller in parallel to the flow compensator to limit the maximum system pressure in the circuit. This is indicated by the pilot line A in Figure 15.16.

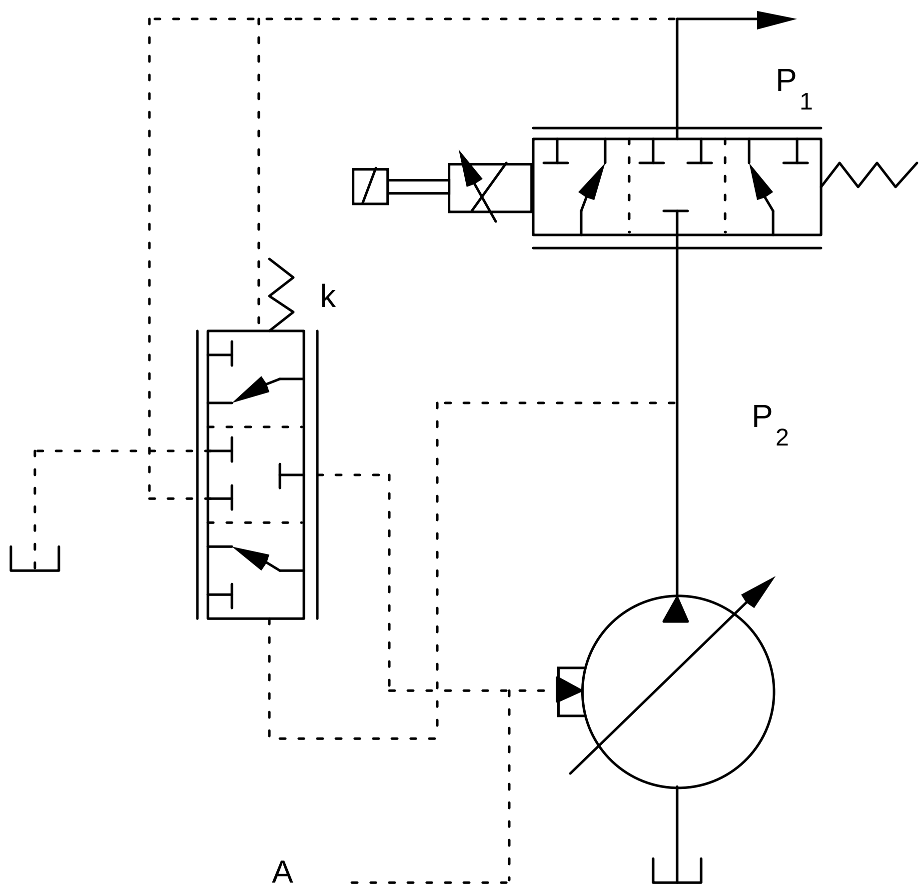


Figure 15.16 Load sensing variable displacement pump

15.5.3 Acceleration/deceleration control of an actuator and load

To limit acceleration we know we can use pressure limiting valves, variable flow through valves, cushions, cam activated deceleration valves, variable capacity pump etc. In many cases, simplicity in design is gained using these approaches. However, if the load profile is complex, then the circuit can become very complex and indeed, rather costly and inflexible. The use of proportional valves for open loop control has definite potential but it must be emphasized that all alternatives must be considered. Proportional valves are great but they do require expensive controllers (amplifiers in addition to the valves).

An example of a proportional directional control valve being used to maintain a constant (and the same) speed in both directions for a single rod cylinder is shown in Figure 15.17. If the same flow was delivered in both direction, the velocity would change. Similarly the rate of increase for acceleration/deceleration is not necessarily the same. Thus we need the following.

1. bi-directional proportional flow control

2. a power supply to send out the appropriate rate of change signal which reflects the change in direction of the actuator.

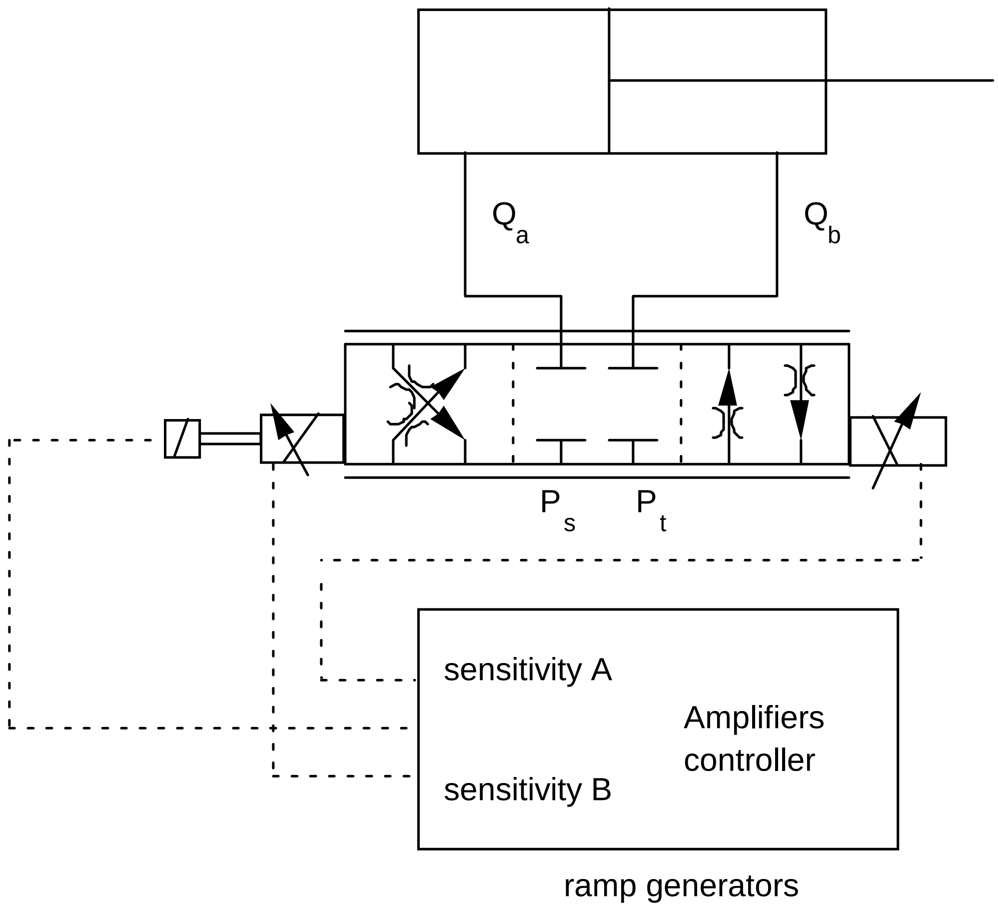


Figure 15.17 Proportional flow to a single rod actuator

A circuit which can accomplish this, is illustrated in Figure 15.17. Note, there is no pressure compensation across the valve so we do not have flow control. If the burden is constant and well known over the stroke, then this becomes less of concern.

The power supply/amplifier has ramp input selects that dictate the acceleration and deceleration rates. To accommodate the different flows, the output sensitivity of each solenoid can be adjusted separately (Note, if only one solenoid is used, this gets much more complicated; hence, the use of two solenoid in Figure 15.17). The amplifiers also are set to compensate for dead-zone and hysteresis.

A problem that we have encountered arises again. If the orifices are matched and symmetrical, then for a single rod actuator, the controlled flow into the blank end is not equal to the flow out (area ratio). If the burden suddenly changes direction, we do provide some protection from run-away condition because of the down stream orifice. This means flow control does not exist unless the resistance is high enough to create a substantial back pressure on the rod side of the actuator. In fact, we can compensate somewhat by having asymmetric orifices in the directional proportional flow control valve. By having a higher resistance in the downstream orifice, a substantial burden change would have to occur to be in a run-away mode.

Let us consider an example. In Figure 15.18, the hydraulic force profile has been established. Let us examine what the use of asymmetric orifices would have on the performance.

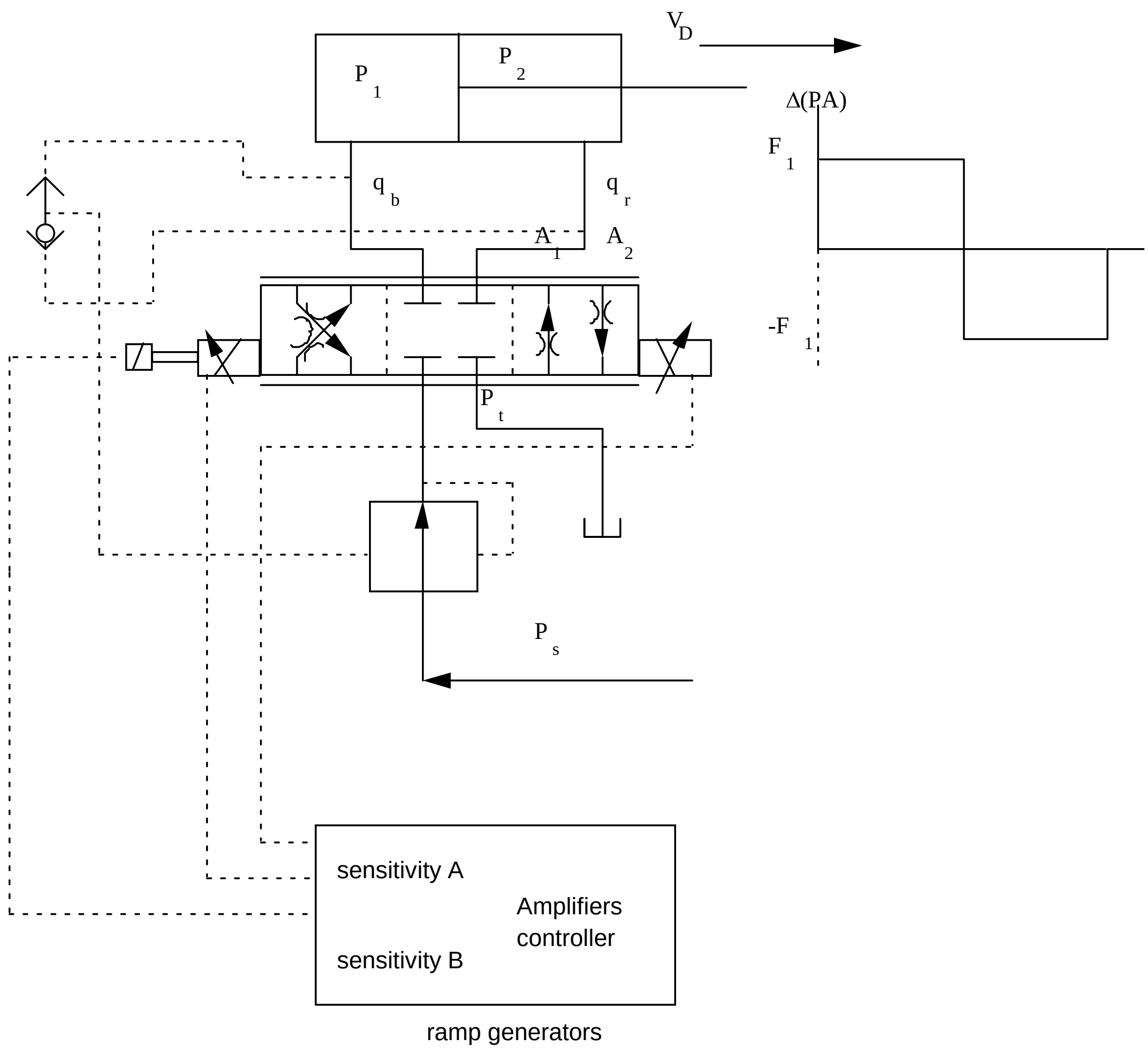


Figure 15.18 Proportional directional control with asymmetric orifice

For control, we want V_D = constant (q_b = constant). If our burden reverses direction, we want the resistance of the orifice downstream to be sufficiently high enough to ensure that P_1 is just slightly greater than zero (to prevent cavitation) but to maintain the desired velocity. To analyze this situation, we can set $P_1 \approx 0$. In the following analysis, F_1 is negative (run-away conditions) and is assumed to be the maximum run-away force.

$$q_b = k_1 A_1 \sqrt{\Delta P} \qquad (16.1)$$

where $k_1 = Cd\sqrt{\frac{2}{\rho}}$

where Δp is controlled by hydrostat = constant and
A_1 = area of the upstream orifice of the valve

$\therefore$ $q_b = C_1 A_1 = A_B V_D$ (V_D = desired velocity for control and Ab = area of the blank side of the actutaor)

where $C_1 = Cd\sqrt{\frac{2}{\rho}\Delta P}$

$$\therefore \quad V_D = \frac{q_b}{A_B} = \frac{C_1 A_1}{A_B} \qquad (16.2)$$

Now, $V_D = \frac{q_r}{A_r} = k_1 A_2 \sqrt{P_2}$ (assuming P_t is at tank)

We also know that $P_2 = \frac{F_1}{A_r}$ (because $P_1 \approx 0$)

$$\therefore \quad V_D = k_1 A_2 \sqrt{\frac{F_1}{A_r}} \qquad (16.3)$$

Equating Equation (16.2) to (16.3)

$$\frac{C_1 A_1}{A_B} = k_1 A_2 \sqrt{\frac{F_1}{A_r}}$$

or

$$\frac{A_1}{A_2} = \frac{k_1 A_B}{C_1}\sqrt{\frac{F_1}{Ar}} = A_B \sqrt{\frac{F_1}{\Delta P A_r}} \qquad (16.4)$$

(where we substituting in for C_1).

$\therefore$ For a given k_1, A_B, A_R, C_1 and F_1 (max), we can calculate a critical area ratio which will ensure that V will not change from the desired value.

We must notice that if F_1 is less than F_{1max} , then P_1 is not zero. This is not a problem because our flow control valve sees a resistive load and the hydrostat compensates for any changes in P_1.

This kind of circuit is not a good idea. This is because the presence of the downstream orifice is always present and hence for resistive hydraulic loads, we have substantial pressure drops and hence power losses. We do know that we could use pilot operated counterbalance valves which are far more effective during conditions of resistive loading.

It should also be noted that this analysis is only valid for pressure compensated flow control. You cannot control flow by just putting a resistance in the downstream line.

In summary then, proportional valves have many potential applications. However, in design, it is a very good idea to design for a simple worst case scenario and then examine the circuit to see if proportional or servovalves can be used and, indeed, are cost effective. Versatility is important but if the cost is too high, no one will buy (implement) the product.