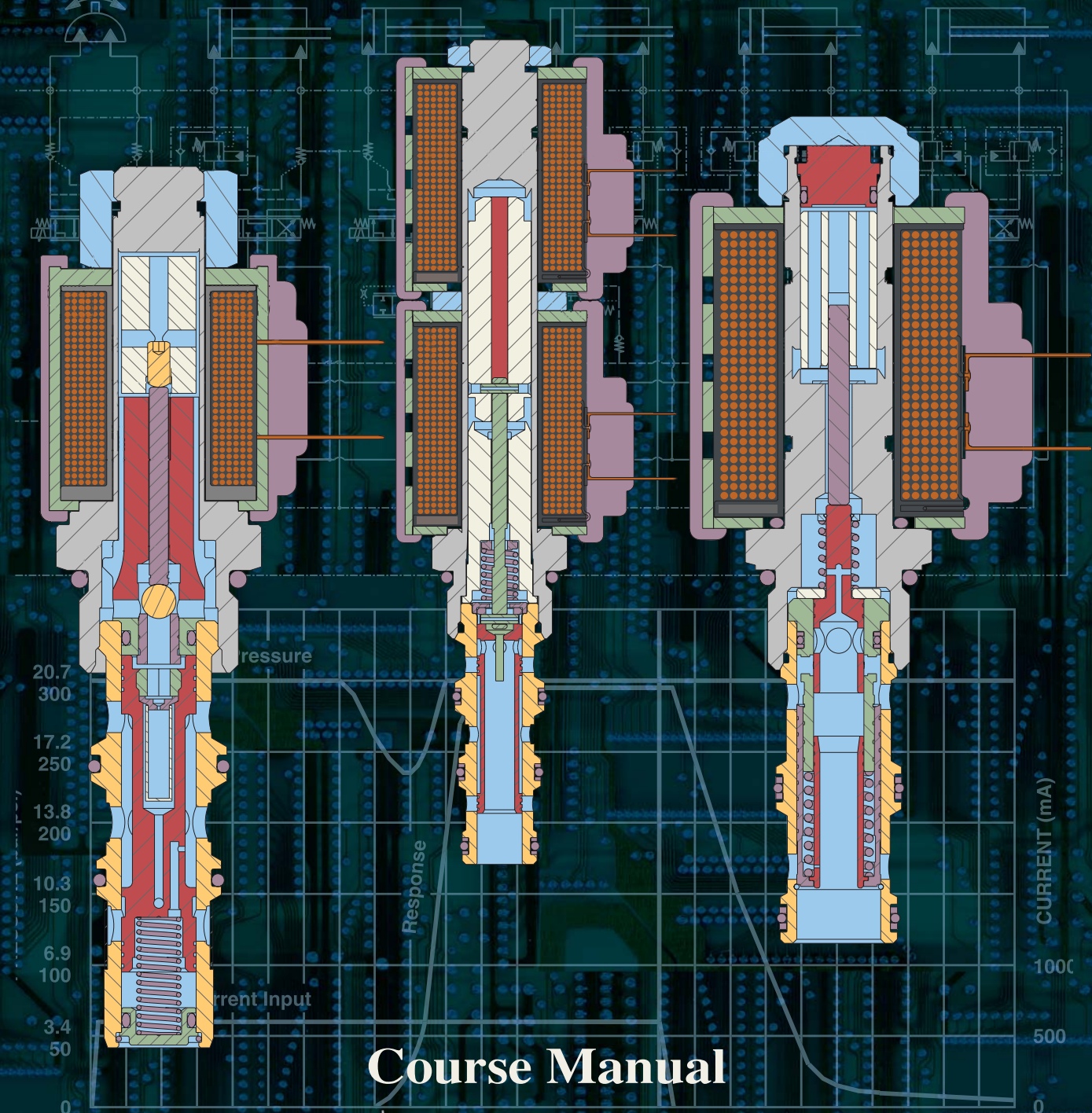
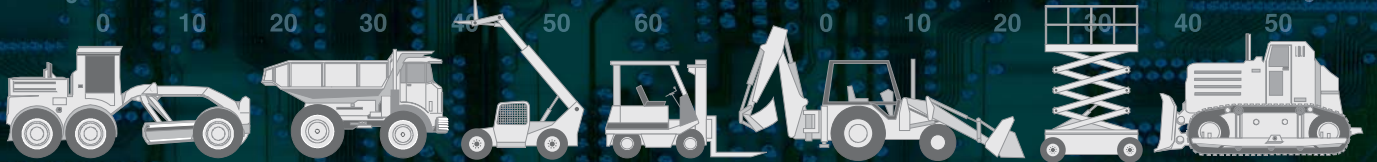


# Electro-Hydraulic Proportional Valves



## Course Manual



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# Electro-Hydraulic Proportional Valves Course Manual

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# Chapter 1: Overview

## Objectives

- Learn the definition of a proportional valve.
- Recognize the different types of proportional valves.
- Learn the definition of direct acting and pilot operated.

## Introduction

A proportional valve is one which can vary the output in response to the variation of electric input. The output of these valves depend on the magnetic force of the solenoid. There are three types of proportional valves, electrohydraulic proportional flow control, pressure control and directional valves.

The purpose of this manual is to explain how proportional valves work. To do this, the terminology associated with proportional valves, basic construction, operating parameters and how the valves are controlled, will be presented.

# HYDRAFORCE

## Summary of Chapters

- Chapter 1 – Overview
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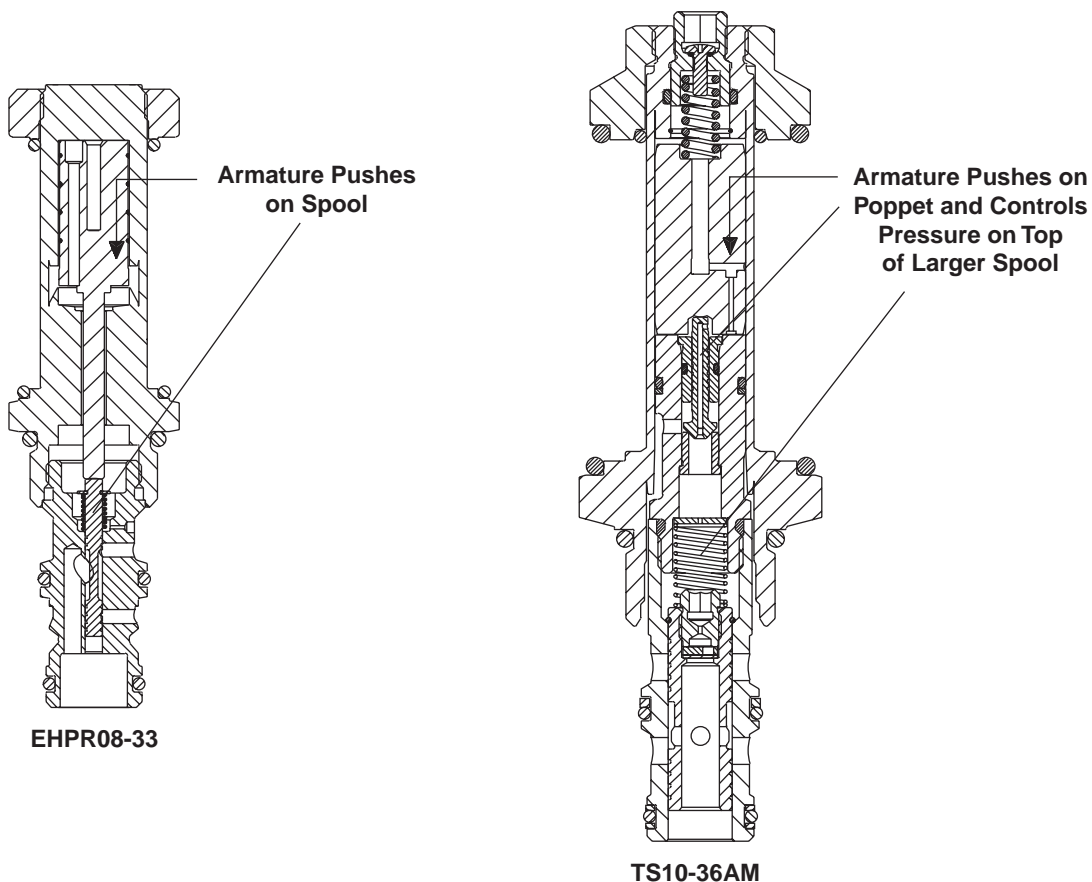
## Types of Proportional Valves

There are three main categories of proportional valves. These include flow, pressure and directional control valves. Flow control valves control the flow into or out of a hydraulic cylinder or motor. The flow is controlled to regulate the speed of these actuators. In other words, the rate the cylinder extends and retracts or how fast the motor turns. The pressure control valves regulate the pressure applied to both of these devices. By regulating the pressure, the force the cylinder applies, or the torque of the motor is controlled. Finally, the directional control determines if the cylinder extends or retracts, or if the motor turns clockwise or counterclockwise.



## Direct Acting vs. Pilot Operated

The movement of the hydraulic components (such as the spool), inside these valves can be controlled directly by the solenoid actuator. Valves which use this type of actuator are known as direct acting valves. Another method of controlling this movement is to use a small direct acting valve which controls the pressure applied to a larger spool. This is known as a pilot operated valve. In this valve, a small element pilots (controls) a large element. The diagrams below are examples of how these two functions and methods of actuation are accomplished. Both examples are pressure reducing valves. The EHPR08-33 is direct acting (controlled by the solenoid) and the TS10-36 is pilot operated (small valve controlling the larger spool). The operation of each is discussed in later chapters.



**Notes:**

# Chapter 2: Basics of Controllers

## Objectives

The objectives for this chapter are as follows:

- Discover the different types of voltage signals used in proportional valves.
- Understand what threshold and maximum current are.
- Learn about voltage and current control.
- Recognize a potentiometer and what it is used for.
- Discover what electronic controllers are.
- Learn about features available in controllers, such as ramp,  $I_{MIN}$ ,  $I_{MAX}$  and dither.

## Introduction

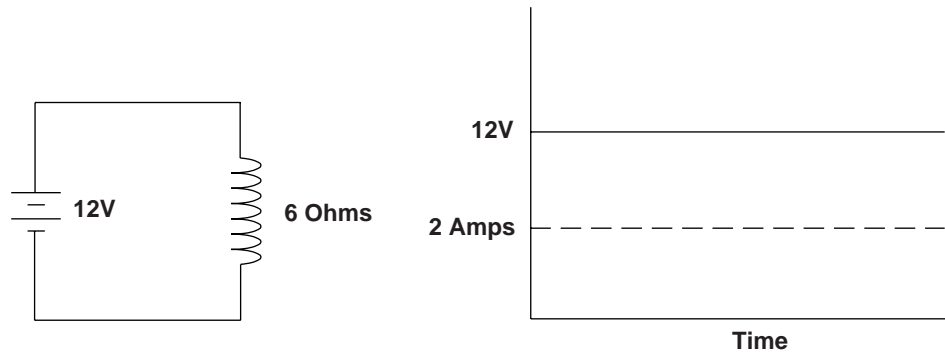
This chapter will outline some of the basic electrical terminology associated with the control of proportional valves. PWM and Dither, which are the voltage signals applied to the coil, will be discussed first. Following, the different current levels associated with the control of the valve will be discussed. Finally, the differences behind voltage and current control as well as devices designed for each, will be presented.

## Voltage Signals

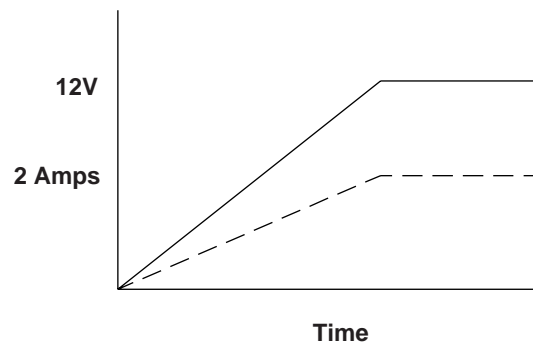
Proportional valves are made up of mechanical and electromechanical components. The mechanical components include the spool and cage, which control the oil pressure and flow. The electromechanical components include the coil, plunger and pole piece, which control the spool. In order to create the magnetic force which the plunger applies to the spool, voltage must be applied to the coil. There are several variations of voltage that can be applied to the coil, including smooth DC, PWM and dither. These will be discussed in the following section. In each of these cases, the voltage is DC or non-alternating current.

### Smooth DC

Smooth DC is available from a battery and is the first of the three variations of voltage we will discuss. Graphed against time, the steady state or constant level of voltage and current appear as a straight line. For this reason it is called smooth DC. The graph below shows the current and voltage recorded when 12 V from a battery is applied to a 6 ohm coil.

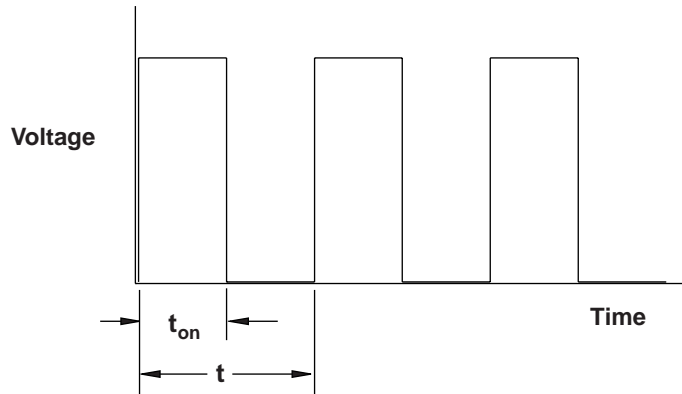


The following graph shows that the voltage and current remain smooth even as the voltage is varied or ramped from 0 to 12V.



**PWM**

The second type of voltage signal we will consider is PWM, which is an acronym for Pulse Width Modulation. It can be defined as the rapid turning on and off (pulsing) of voltage within a fixed amount of time (width of modulation). The graph below shows several voltage pulses recorded vs. time.



t: The time period of one full cycle is the frequency in which the voltage is turned on and off. This time is typically measured in cycles per second or Hz (Hertz). The frequency used with HydraForce valves is typically between 75 to 400 Hz.

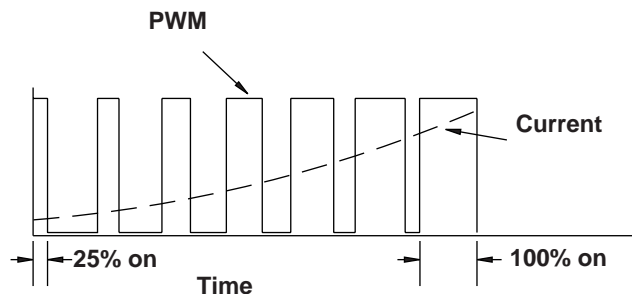
ton: This is the amount of time that power is applied to the coil during the above time periods.

The two time periods above are typically combined to define what is known as the **duty cycle**. The duty cycle is found using the following equation:

$$(t_{on} \div t) \times 100 = \text{duty cycle \%}$$

Simply stated, duty cycle is the percentage of time that power is applied to the coil.

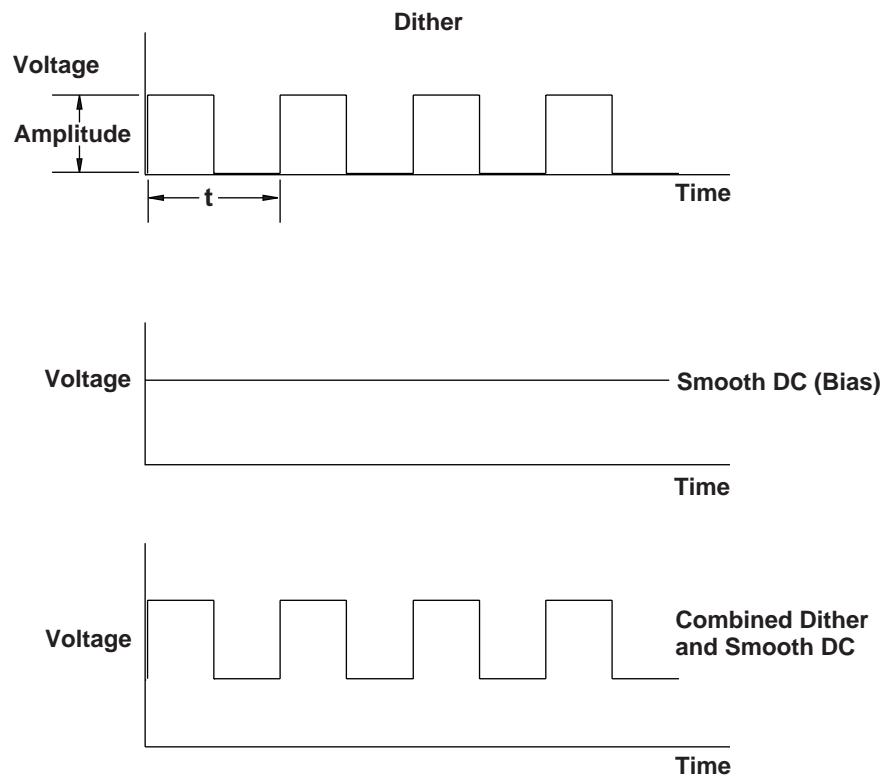
The current is varied by changing the duty cycle or the percentage of on time. If the duty cycle is low (voltage on for a short time) then the current is low. When the duty cycle is high, the current is high. The graph below shows how the current in the coil increases as the duty cycle increases.



PWM is used to keep the plunger constantly moving. Basically, by using PWM, the magnetic solenoid force continuously builds and falls. This means that for a brief moment, the armature is attracted to the pole piece and then it is no longer attracted. The amount of magnetic force generated during each pulse is high because the voltage level (amplitude) at the top of each pulse, is typically the system voltage. This constant movement of the armature reduces the viscous and mechanical friction forces between the moving and fixed parts of the valve. This in turn reduces hysteresis (hysteresis will be described in the following chapter). PWM is used in the majority of proportional valve applications, rather than straight DC because it reduces hysteresis.

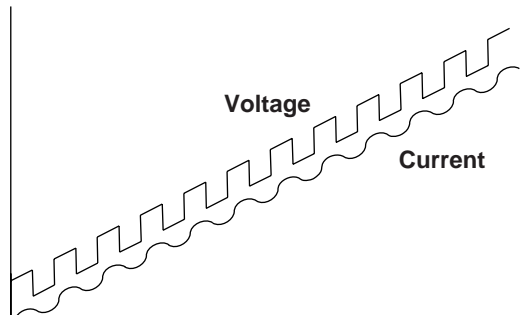
### Dither

The third type of voltage variation we will consider is dither. It can be defined as a state of indecisive action. The indecisive action in our discussion is the voltage level. Dither is similar to PWM in the fact that the voltage is dithered, or turned on and off rapidly. Like PWM, dither keeps the actuator moving to reduce hysteresis. There are two factors which distinguish it from PWM. The first factor is that the amplitude or level of the pulse is smaller in dither. The second factor is that the duty cycle is fixed at 50%. This smaller, fixed pulse or dither is added to smooth DC. Smooth DC is also known as bias. It is called this because the dither is shifted by increasing or decreasing the smooth DC. In other words, it is biased by the level of the smooth DC. The combined output is then applied to the proportional valve. The graphs below illustrate the concept of voltage vs. time.





In order to vary the current, the smooth DC level must be varied, not the dither. This is shown in the following graph. Notice that the dither, amplitude and frequency remain the same, but the DC bias is increased.



**$I_{MIN}$  &  $I_{MAX}$**

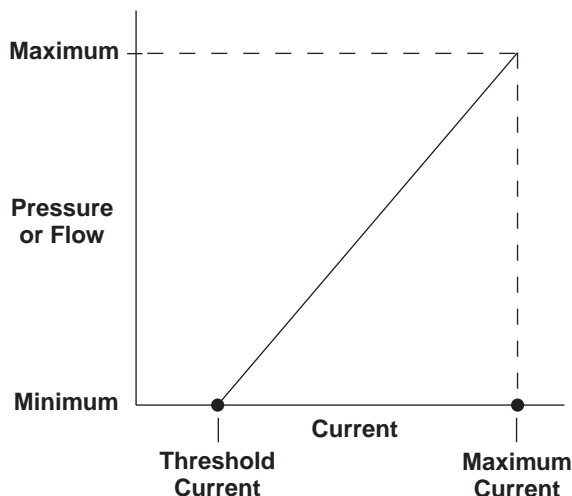
The terms threshold current and maximum current are sometimes referred to as  $I_{MIN}$  and  $I_{MAX}$ . These points typically define the operating limits of a proportional valve.

**Threshold Current**

Threshold current ( $I_{MIN}$ ), is the current applied to the proportional solenoid coil, which results in the initial change in pressure or flow. (Note:  $I_{MIN}$  is related to a setting available on electronic controllers. This will be discussed later in this chapter).

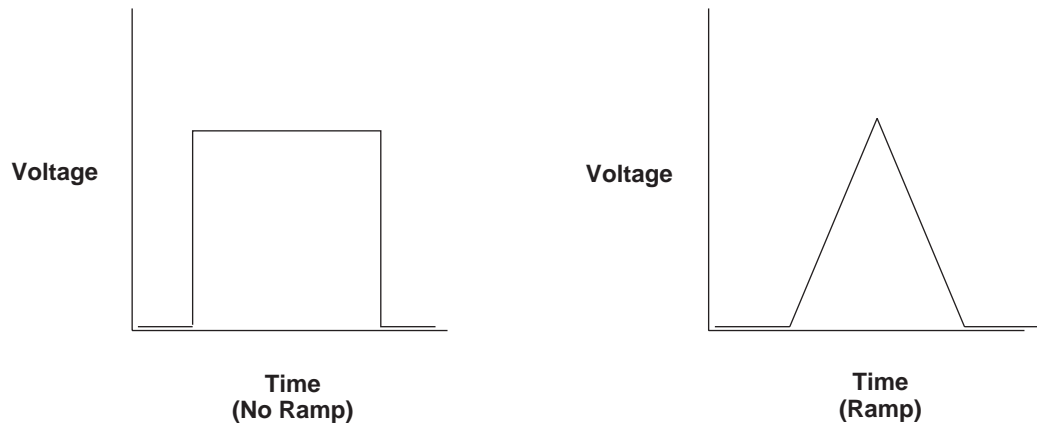
**Maximum Current**

Maximum current ( $I_{MAX}$ ), is the level of current applied to the coil that results in the maximum rated output from the proportional valve. The following pressure vs. current graph, gives an example using these two operating points.



## Ramp

Ramp is a term associated with controlling the voltage applied to the coil, over a given amount of time. The ramping function allows the voltage to build or fall gradually and evenly. Without this function, the voltage could build or fall instantly when the switch is closed. The following graphs illustrate how the voltage increases and decreases quickly, as with a switch, as well as changing gradually. A ramp function can either be created manually by varying the input voltage slowly, or through electronics available on an electronic controller.

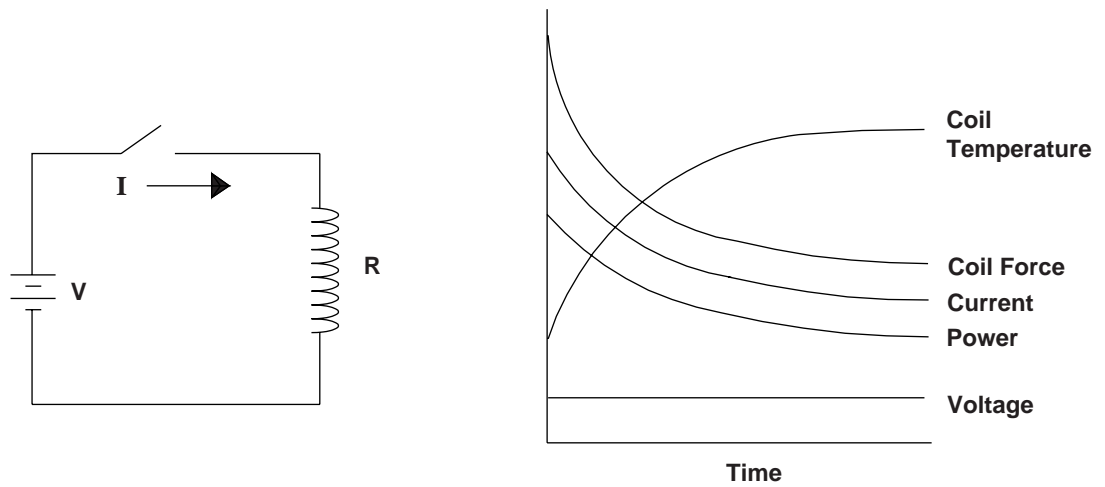


## Voltage vs. Current

The power applied to a solenoid coil can be controlled by varying the voltage or the current. These methods are known as voltage driven and current driven systems. Each system is described in the following section as well as the devices used to vary or regulate the power applied to the coil. The first method, voltage driven, is less costly because of the simple device used to regulate the input to the coil. The second method, current driven, regulates the force the solenoid creates, by regulating the current. While the current driven method regulates the output from the valve better, it is more complex than the voltage driven method because it involves electronics.

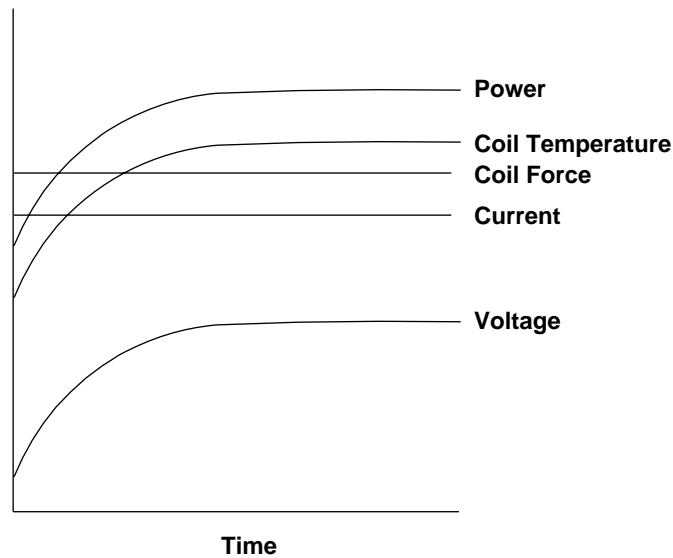
### Voltage Driven

The term voltage driven is associated with powering the coil from a source which has no control on current, such as a battery. When the battery (shown in the following circuit) is connected to the coil, the current draw in the circuit is limited by the resistance in the coil ( $V = IR$ ). As the coil heats up over time, from the power applied, the resistance goes up and the current goes down. Eventually, the temperature and resistance stabilize and the voltage, current and power stay the same. In addition, the force developed by the solenoid decreases in proportion to the current, as shown in the following graph.



### Current Driven

Current driven refers to applying a source to the coil which regulates current rather than voltage. This type of source is better suited for proportional valves than voltage driven because although the coil heats up, the current remains the same, as does the force from the solenoid. This is shown in the diagram below. However, the power available must be sufficient to increase the voltage to maintain the current level. If it is not, the current will begin to decrease as in the voltage driven coil.

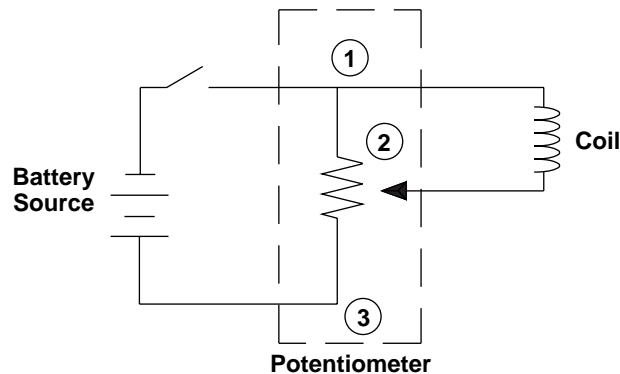


## Potentiometers vs. Electronic Controls

Proportional valves work with variable current or voltage applied to the solenoid coil. In order to take a fixed voltage level and vary it, for example from a 12V battery, a device is required between the battery and the coil. This device can be as simple as a potentiometer or as complex as an electronic closed loop current controller.

### Potentiometer

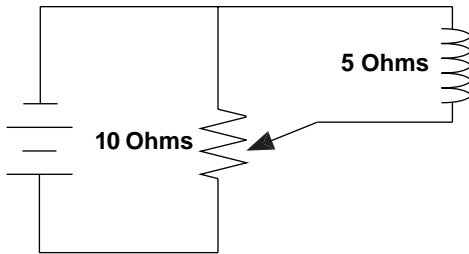
The full voltage from the battery is not directly applied to the proportional valve as shown in the Voltage Driven section of this chapter. A potentiometer can be connected between the battery and coil to vary the voltage. The potentiometer is simply a variable resistor. The symbol and how it is connected in a circuit is shown below.



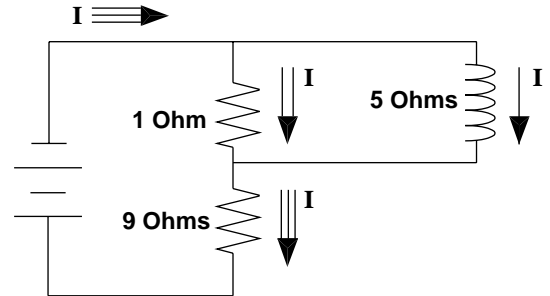
As shown in the diagram, the source is connected across terminals 1 and 3. Between terminals 1 and 3 there is a resistor. The size of the resistor or the amount of voltage drop across this resistor varies with the position of the wiper connected at terminal 2. The wiper is a part of the potentiometer that varies the resistance between terminals 1 and 2 from a small amount to a large amount. When the resistance is small, or the wiper is close to terminal 1, all the current flows through the potentiometer because the current wants to flow through the path of least resistance. When the wiper is closest to terminal 3, the resistance between terminal 1 and 2 is larger than the coil, so the majority of the current will flow through the coil.

This is further explained in the following example. The diagrams represent two extremes of the resistance, regulated by the wiper. Assume that the total resistance of the potentiometer is 10 ohms and the resistance of the coil is 5 ohms. Also assume that the wiper shown in the following diagram is moved to a position in which it cuts the resistor between terminals 1 and 3. With this configuration, the resistance between terminal 1 and 2 is 1 ohm and the resistance between terminal 2 and 3 is 9 ohms. In this case, most of the current from the battery will flow through the 1 ohm resistor rather than through the 5 ohm coil because the current takes the path of least resistance. A small portion of the current also flows through the coil.

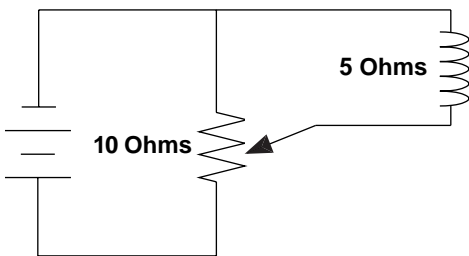
The 10 Ohm resistor is split by the wiper and the circuit can be redrawn as: →



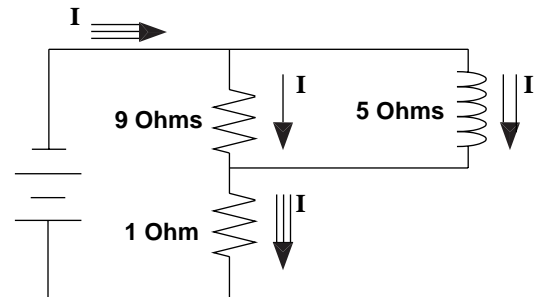
(The number of tails attached to the arrow head is an indication of the relative amount of current flowing through the circuit)



The next circuit, shows the potentiometer with most of the current flowing through the coil.



→



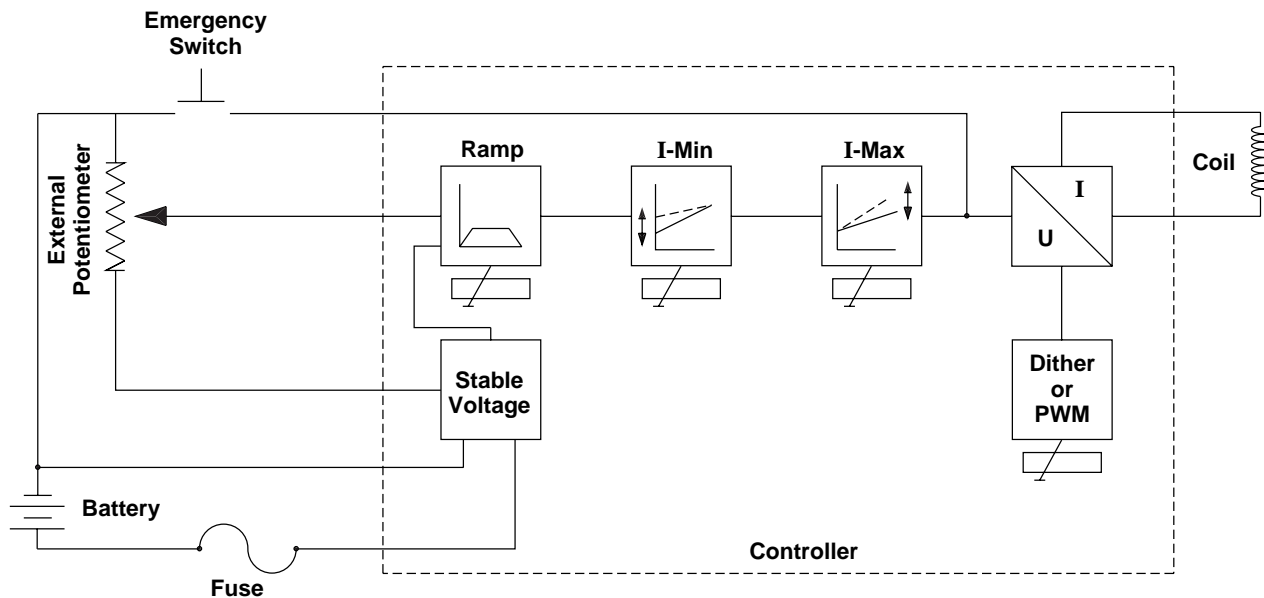
Again, the current takes the path of least resistance. In this case, the 5 ohm coil has a lower resistance than the 9 ohm portion of the potentiometer in parallel with the coil.

An example of a potentiometer in use is a dimmer switch. When the dimmer switch is first turned on the light is very low (little current is flowing through the light bulb). At this point, the current is flowing through the potentiometer which is rationing current to the light bulb. When the switch is turned, the brightness of the bulb increases until the knob cannot be turned farther.

### Controller

The electronic controller or amplifier is used to regulate the current applied to the coil. By regulating the current, the solenoid actuator force is regulated. Therefore the output from the valve is better regulated than by just controlling the voltage through a potentiometer. In addition to regulating the current, controllers are equipped with features which improve the performance of the system.

The performance of the valve is improved because the controller has dither or PWM built-in. The performance of the system is improved because of enhanced features in the controller, such as ramping. For example, if a flow control is controlling a motor, ramping allows for a slow increase in current, which translates into a slow increase in speed. Additional features are described in the following sections. The following block diagram shows an electronic controller with several features built in.



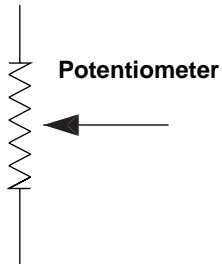


The function of each of the components in the block diagram is described below.

**Emergency Switch**

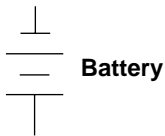


The emergency or disable switch disconnects the coil from the output of the controller. This switch will connect the output from the ramp,  $I_{MIN}$ ,  $I_{MAX}$  portion of the circuit to ground so there is no output to the coil. The controller works or is enabled when the emergency switch is open.



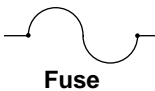
**Potentiometer**

The function of the potentiometer is to interface between the operator and the controller. A more detailed explanation can be found in the previous section. The potentiometer can take several different forms such as in a joystick or a rotating knob.



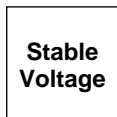
**Battery**

This symbol represents a battery or DC voltage source.



**Fuse**

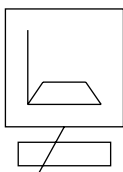
The fuse is located in the circuit to protect the electronics in case the coil draws too much current. This could occur if the coil shorts. The fuse is typically made from a thin wire. As more current flows through the fuse, this wire heats up. If too much current flows, the wire burns and opens, or disconnects the battery from the circuit.



**Stable Voltage**

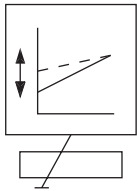
This device provides constant voltage to the electronics inside the controller. It is needed because electronic devices perform better with a set voltage level. Stable voltage is typically not available with a battery because voltage varies as the battery begins to drain. High frequency PWM is used to generate stabilized voltage. Unlike the low frequency PWM described previously, the frequency used to generate stabilized voltage ranges between 5,000 and 20,000 Hz.

**Ramp**



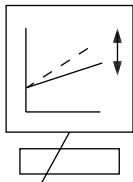
The ramp function controls the rate at which the current applied to the coil, will increase or decrease (known as ramp time). Some controllers have two adjustments for the ramp. These are known as independent ramps. One controls the time it takes for current to increase and the other controls the time it takes the current to decrease. If one potentiometer is used, it controls one or both of the ramps. If it controls both, then the increasing and decreasing ramp are known as dependent ramps. A small potentiometer is built into the controller to vary the ramp. Depending on the direction in which it is turned, it either increases or decreases the ramp time.

**I-Min**



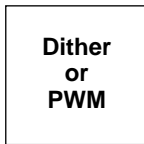
The  $I_{MIN}$  adjustment is used to modify the minimum current applied to the solenoid, in relation to the minimum setting of the external potentiometer or joystick. For example, the external potentiometer could be turned to the minimum setting (to give 0.0 Amps), but the  $I_{MIN}$  could be set to 200 mAmps. The adjustment of  $I_{MIN}$ , as in the ramp function, is accomplished by turning a potentiometer, which is built into the controller.

**I-Max**

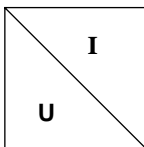


This adjustment is used to limit the output from the controller when the setting of the external potentiometer is at its maximum. For example, assume that when the potentiometer is set to maximum, the maximum output from the controller could be 2.0 Amps (or whatever current gives the maximum desired output from the valve). With the  $I_{MAX}$  adjustment, the output could be limited to 1.0 Amp.  $I_{MAX}$  can also be varied by a potentiometer on the controller.

**Dither  
or  
PWM**



The box marked Dither represents either Dither or PWM (both are explained in detail in the previous section). In each, this box controls the frequency of the signal which is varied by a potentiometer located on the controller.



The voltage to current converter takes all the preceding signals or inputs along with the system voltage, adds them together and outputs a regulated current. Regulated current is used because the force of the solenoid is determined by the current multiplied by the number of turns on the coil. If the output was in volts, the external potentiometer would constantly need to be adjusted, to maintain a constant current at the coil. This is discussed in more detail in the previous section of voltage vs. current.

## Summary

In this chapter the following concepts were presented:

- Smooth DC, PWM and Dither are different voltage signals that could be applied to the coil.
- How the coil force would decrease if constant voltage was used to control a proportional valve.
- How a potentiometer works.
- The features built into an electronic control such as  $I_{MIN}$ ,  $I_{MAX}$  and Ramp.
- How PWM and Dither are used to decrease hysteresis in a valve.

### Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. Why would a potentiometer be used instead of an electronic controller? \_\_\_\_\_
2. What source does smooth DC come from? \_\_\_\_\_
3. What does the acronym PWM designate? \_\_\_\_\_
4. Define PWM. \_\_\_\_\_
5. At what percentage is the duty cycle of dither fixed. \_\_\_\_\_
6. Threshold and maximum current can also be referred to as? \_\_\_\_\_ / \_\_\_\_\_
7. True or False. Maximum current is the level of current applied to the coil that results in the maximum rated output for proportional valves. \_\_\_\_\_
8. What does the ramp function do? \_\_\_\_\_
9. What are dependent ramps? Independent ramps? \_\_\_\_\_ / \_\_\_\_\_
10. True or False. Current driven is associated with powering the coil from a source which has no control on current, such as a battery. \_\_\_\_\_
11. What is the advantage of using current control over voltage control? \_\_\_\_\_
12. What is the function of a potentiometer when connected to an electronic controller? \_\_\_\_\_
13. Give an example of a device in which potentiometer can be found. \_\_\_\_\_

# Chapter 3: Pressure Controls - Introductory

## Objectives

The objectives for this chapter are as follows:

- Become familiar with the terms pressure rise and pressure droop.
- Learn what the pressure control range is.
- Understand what hysteresis is, how it is determined and what causes it.
- Discuss how hysteresis is minimized through the effects of PWM.

## Introduction

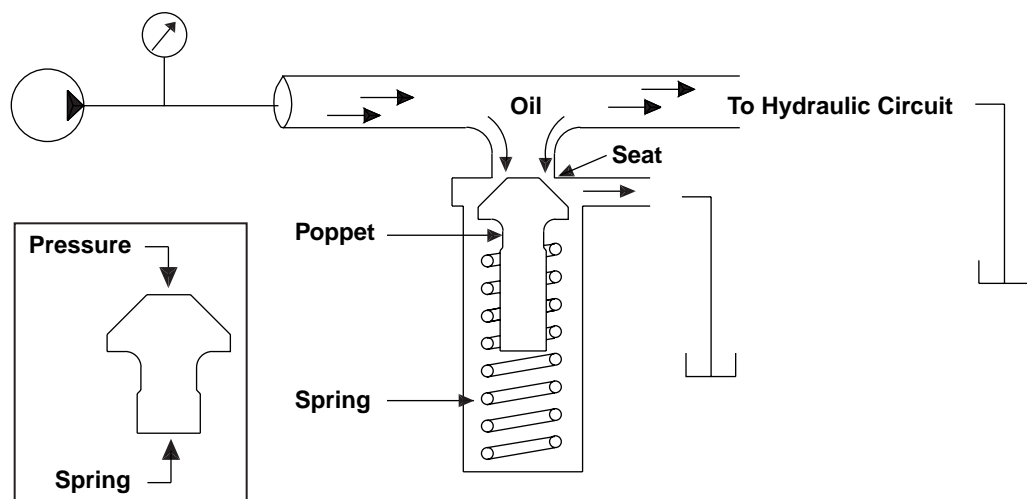
This chapter begins by explaining simple terminology associated with the performance characteristics of proportional valves. These include control range, hysteresis, and response time. After defining hysteresis, the factors that contribute to it are discussed.

# HYDRAFORCE

## Pressure Control Introduction

### Relief Valve

Before explaining the terminology associated with a pressure control, we will briefly discuss what a pressure control is. A pressure control valve is one that regulates or maintains the pressure of a hydraulic circuit. This is typically accomplished by holding a poppet on a seat. However, other methods also exist. Shown below is a simple schematic of a relief or pressure control valve.



In the schematic above, the system pressure measured by the pressure gage is maintained by the poppet opening and closing. The opening of the poppet allows oil to escape when the pressure force exerted by the oil, exceeds the spring force (see the figure to the right). In other words, the valve relieves the pressure.

The following two sections define two operating parameters or performance characteristics of a pressure relief valve. These are crack pressure and pressure rise. Both will be further described in the following chapter under Performance of TS38-20 and Operation of TS38-20.

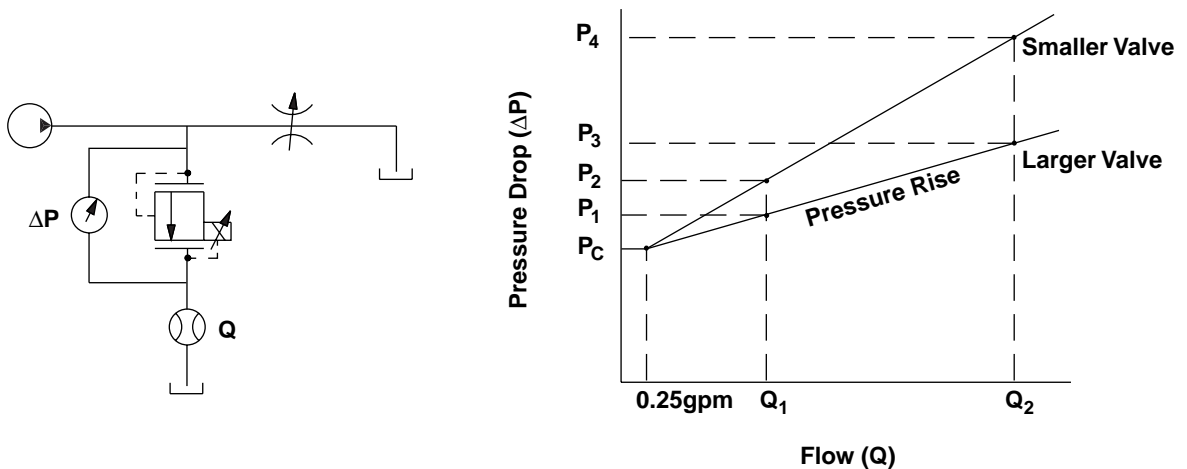
### Crack Pressure

Crack pressure is the pressure which exerts a force that is just greater than the spring force. HydraForce typically defines the crack pressure (also known as the pressure setting) as the pressure which allows 0.25 gpm to flow through the pressure control. (The crack pressure is also indicated on the following graph as  $P_c$ ).



### Pressure Rise

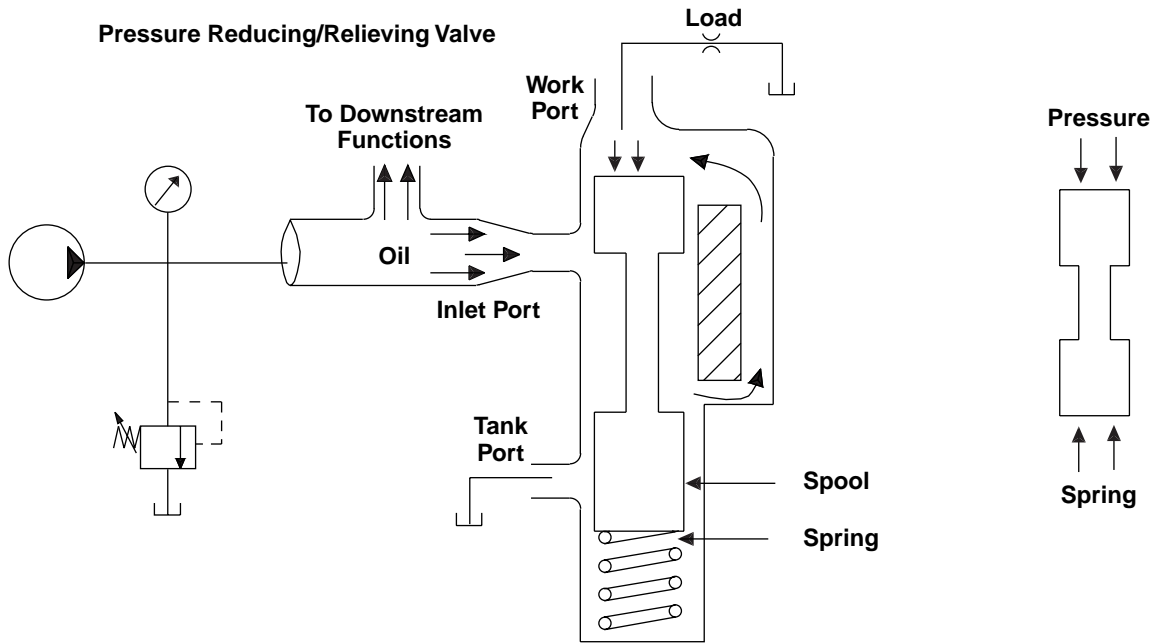
Pressure rise is an increase in pressure, above the pressure setting of the valve. This characteristic is typically associated with relief valves. An example circuit which has a pressure control such as the TS10-26, is shown below. Assume that the current applied to the TS10-26 is such that the regulated pressure (crack pressure) is 2000 psi. The graph next to the circuit shows how the pressure rises as increased flow passes through the TS10-26 when the needle valve closes.



In order to separate the pressure drop of the system from the performance of the valve, the pressure is typically measured using a differential pressure transducer. This is noted above as  $\Delta P$ . The differential pressure is simply a measurement made by subtracting the pressure at the outlet of the valve from the pressure at the inlet of the valve. The pressure rise is described as a ratio or the change in pressure divided by the corresponding change in flow. This can be found using the following relation:

$$\begin{aligned} \text{larger valve: } & \frac{\text{change in pressure}}{\text{change in flow}} = \frac{P_3 - P_1}{Q_2 - Q_1} = \text{pressure rise per flow} \\ \text{smaller valve: } & \frac{\text{change in pressure}}{\text{change in flow}} = \frac{P_4 - P_2}{Q_2 - Q_1} = \text{pressure rise per flow} \end{aligned}$$

Pressure rise is a parameter or performance characteristic used to select the size of the relief valve. As the graph above shows, the larger valve has a lower pressure rise than the smaller valve. A lower pressure rise characteristic is desirable to better protect a hydraulic system. A relief valve is used to protect the system from being over pressurized. The setting of the relief valve is typically set at or just below the maximum continuous operating pressure that is allowed by the other components in the hydraulic system. These components include the pump, hoses, cylinders and other valves. It is set at this maximum level to take full advantage of the other components which maximize the efficiency of the hydraulic circuit. If the pressure rise is too large or steep, the pressure applied to the other hydraulic components may be so great that it causes them to fail.



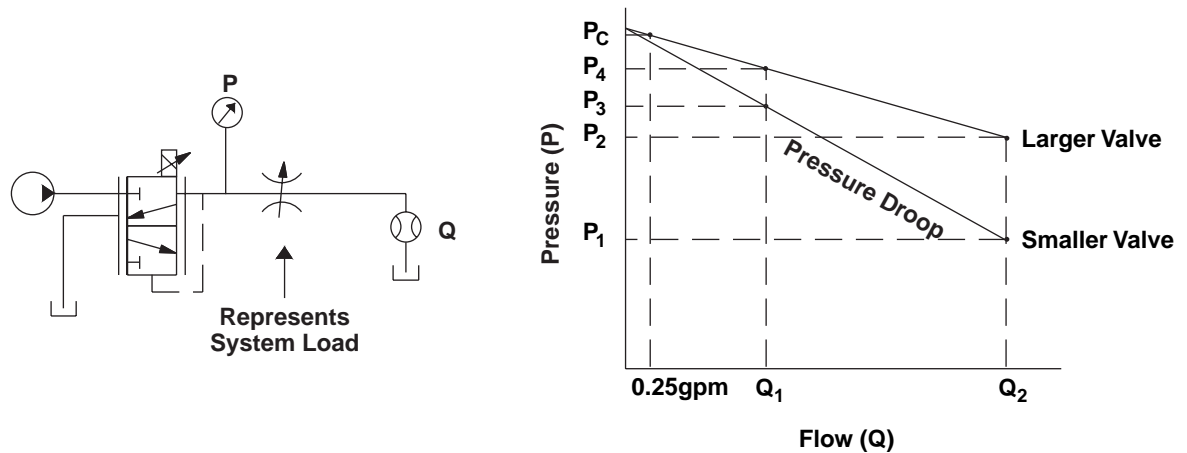
### Pressure Reducing/Relieving Valve

The schematic above shows a simple pressure reducing/relieving valve. The spool controls the pressure at the work port, to which a load is connected. Notice that there is also a relief valve in this circuit. This is required to control the pressure in the entire circuit. The pressure reducing/relieving spool is only used to decrease the pressure below the setting of the relief valve. As in the relief valve, the pressure pushes against a spool which in turn pushes against the spring. When the pressure exceeds the spring force, the spool moves toward the spring. The connection between the inlet and the work port is then decreased. If the pressure at the work port continues to increase, it will push against the spool and compress the spring more. Eventually, some of the oil will be relieved to the tank.

The following section defines a performance characteristic of the reducing/relieving valve known as pressure droop. The cause of pressure droop will be discussed in the following chapter under operation of the EHPR08-33. Also noted above is that this type of valve can act as a pressure relieving valve. The pressure rise and pressure droop characteristic will be combined into a single graph in the following chapter under Performance of EHPR08-33.

### Pressure Droop

Pressure droop is another term associated with pressure control valves. This occurs when the pressure, which is maintained by the pressure reducing valve, falls below the set pressure as flow increases. In order to separate the effects of system pressure losses, a differential pressure measurement can be made.



The equation to describe pressure rise is also the same as used to describe pressure droop. The equations are shown below.

$$\text{larger valve: } \frac{P_4 - P_2}{Q_2 - Q_1} = \text{pressure droop per flow}$$

$$\text{smaller valve: } \frac{P_3 - P_1}{Q_2 - Q_1} = \text{pressure droop per flow}$$

Pressure droop, like pressure rise is a performance characteristic used to select the correct size valve. When reviewing pressure droop, a smaller amount is more desirable. However, the amount of droop must be weighed against the size, cost and maximum flow rating of the valve.

## Repeatability and Reproducibility

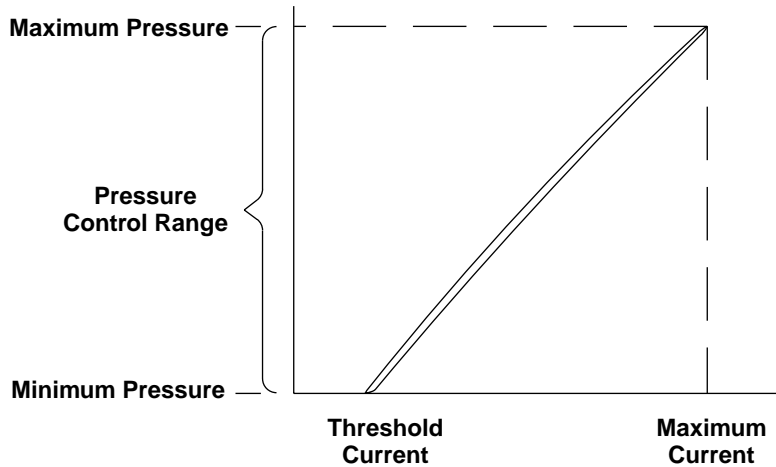
Repeatability is a term that refers to the performance of one valve. It is found by testing one valve 30 times. While testing the valve, a pressure vs. current graph is recorded. This graph looks similar to the one shown under Pressure Range. The maximum and minimum pressure or flow at a given current level are used to determine repeatability. This is found by dividing the minimum pressure by the maximum pressure and multiplying by 100. This percentage defines the repeatability of the valve.

Reproducibility is a term that refers to the performance of several valves. It is defined by testing 30 valves one time each. As with repeatability, the maximum and minimum pressure or flow at a given current level can then be used to define reproducibility. Again, this is found by dividing the minimum pressure by the maximum pressure and multiplying by 100. This percentage defines the reproducibility of the valve.

Both repeatability and reproducibility are typically defined between 20 to 80% of the operating range. PWM voltage is applied to the coil such that the hysteresis is at a minimum. The current is ramped at a smooth and consistent rate over a period of 60 seconds or more.

### Control Range

The control range is associated with the terms threshold and maximum current which were introduced in Chapter 2. It is the pressure which can be controlled between the threshold and maximum current. This is illustrated in the graph below.



Any pressure between the upper and lower pressure defined by this range can be selected by applying the corresponding current between the threshold and maximum current.

## Hysteresis

Hysteresis is the difference between increasing and decreasing input at a given output. This concept is described in the following example.

Where:

$I_{th}$  = threshold current

$I_h$  = current where hysteresis is measured

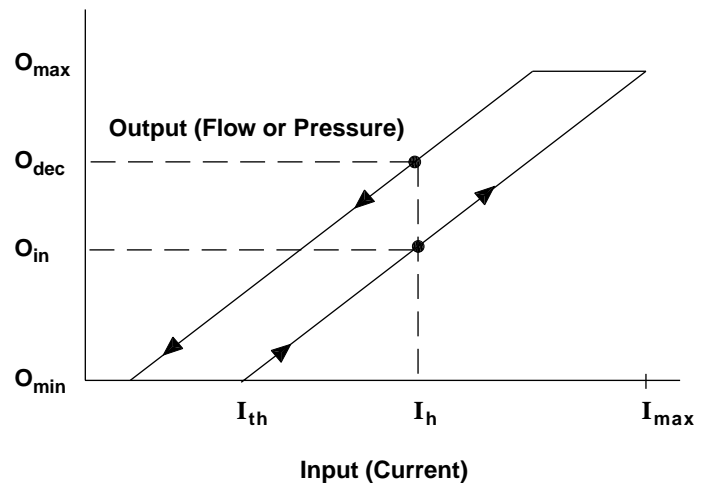
$I_{max}$  = maximum current

$O_{min}$  = minimum output

$O_{in}$  = output when current is increased

$O_{dec}$  = output when current is decreasing

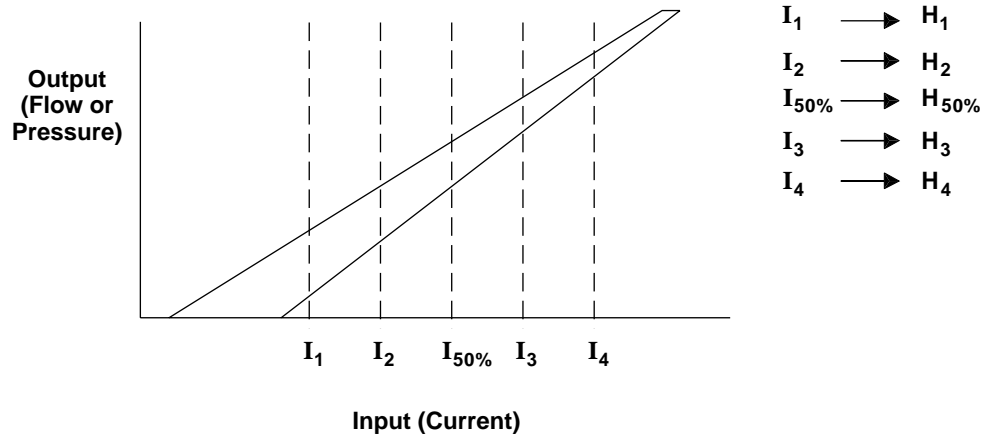
$O_{max}$  = output at maximum current



The previous graph is made by increasing the current applied to the solenoid until the maximum current is reached. The current is then decreased until the output drops to zero. The hysteresis in the graph is the difference between the measured output when the current is increasing and then decreasing. This number is then divided by the output range (flow or pressure). It is then multiplied by 100 because hysteresis is typically given in percentages. The following equation can be used to determine hysteresis.

$$\text{Hysteresis} = \left( \frac{O_{dec} - O_{in}}{O_{max} - O_{min}} \right) \times 100$$

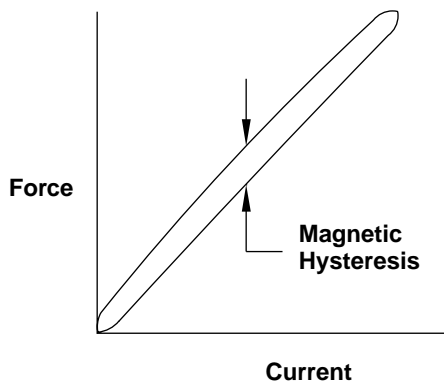
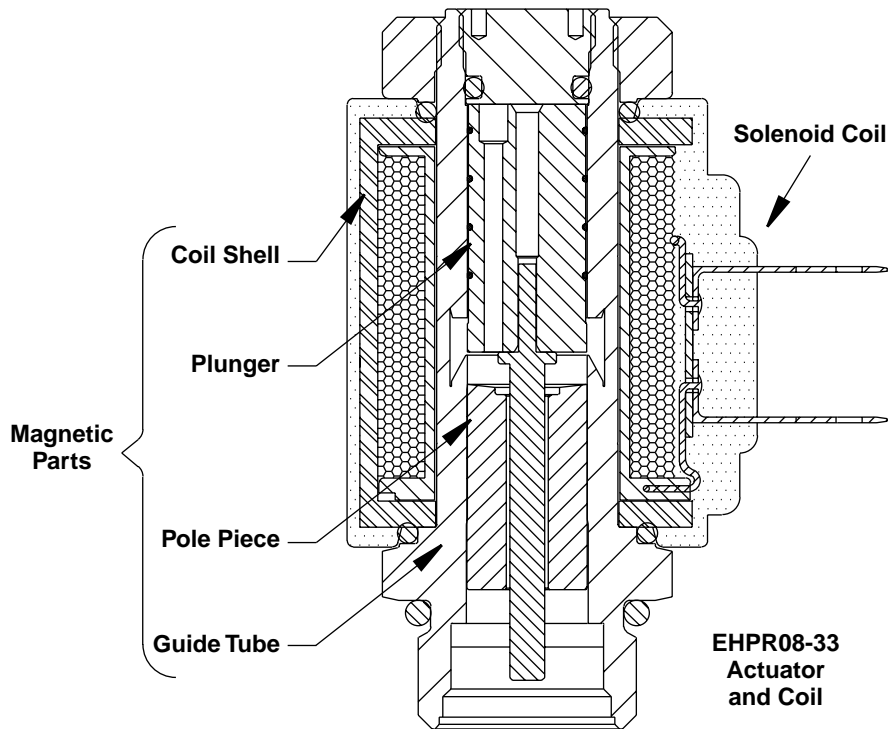
Since hysteresis can vary across the control range, as shown in the following graph, it is usually defined at 50% of the current range. In this example, hysteresis is higher at the lower end of the output. In addition, due to varying hysteresis, it is sometimes tabulated at different current levels.





**What Causes Hysteresis?**

Hysteresis is the result of several negative or adverse forces acting on the components of the proportional valve. These forces are categorized as magnetic forces and mechanical forces. The magnetic force at work is known as magnetic hysteresis, or residual magnetism. Magnetic hysteresis occurs in the magnetic parts of the solenoid actuator of a proportional valve. As current is increased, the parts become more magnetic, causing the magnetic force to increase. When the current decreases, the magnetic force decreases as well, but the current needs to be decreased further to reduce the magnetic force back to its original state. The following graph shows a hysteresis curve for the proportional actuator shown beside it. The graph was made by holding the armature in place while the current was varied.



Other forces which act to produce hysteresis in the valve are the mechanical friction and viscous forces. Both of these forces can occur when the plunger slides against the tube subassembly or guide tube. Mechanical friction exists between any two objects sliding against one another. The viscous force is caused by the fluid between these two sliding objects. As one object moves, the fluid attempts to hold it in place.

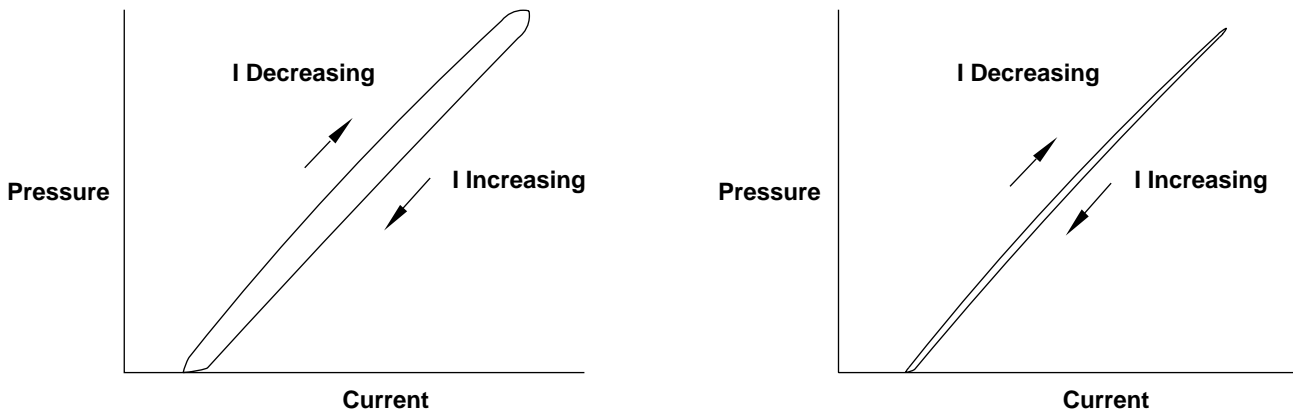
Of the forces involved in causing hysteresis, mechanical friction is usually the strongest and magnetic hysteresis is the weakest. Occasionally, viscous forces are greater than the mechanical friction forces. In this case, fluid viscosity is high, or in other words, the oil is thick which may occur in cold temperatures.

In the design of its valves, HydraForce minimizes the friction forces by machining the parts to provide a smooth finish and applying low friction coatings such as Teflon™, to make them slide easily. Magnetic hysteresis is minimized by selecting appropriate materials (low carbon steel) and annealing (softening of the material). The viscous forces are minimized by optimizing the clearances between parts and the size of the internal orifices.

### Effects of PWM and Dither

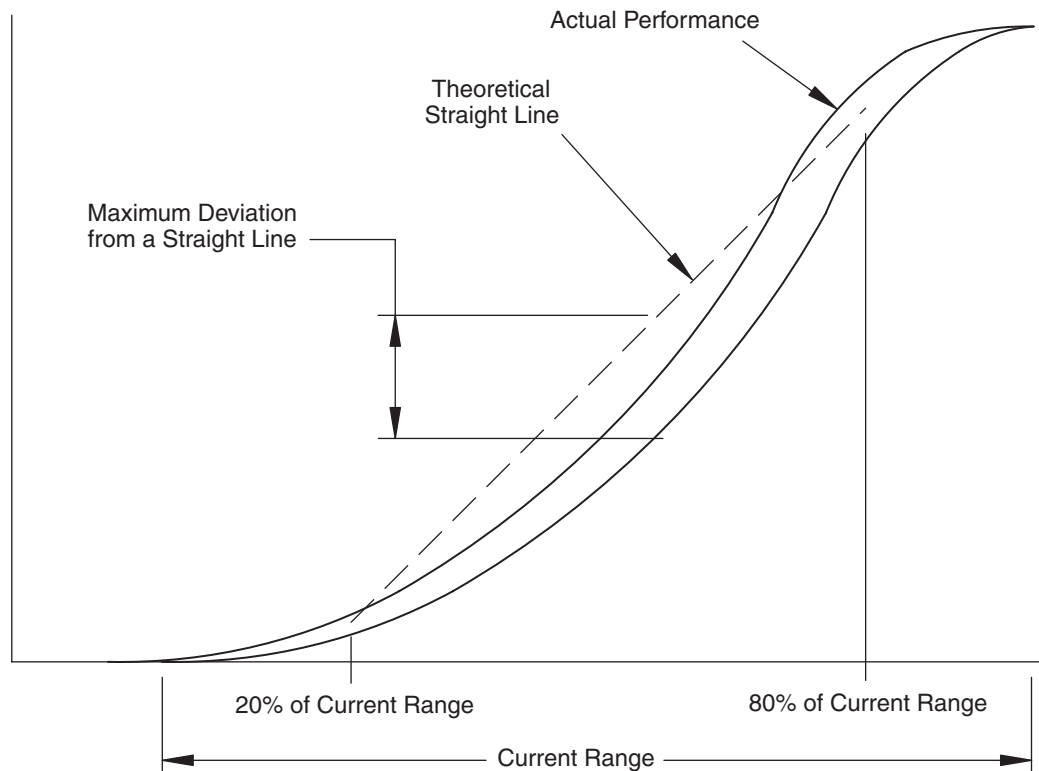
Despite the efforts of HydraForce to minimize hysteresis through valve design, it still exists to some degree. PWM or dither is applied to the proportional coil to reduce any hysteresis that may still be present. Refer to Chapter 2 for more information on PWM and dither.

PWM reduces hysteresis by keeping the armature in constant motion. This motion reduces the viscous force to a negligible amount, because the motion continuously breaks the bond between the plunger and guide tube. The magnetic hysteresis is reduced because the current continually rises and falls. A change in current is always accompanied by a change in magnetism. Notice in the second graph where PWM voltage was applied to the solenoid, that the hysteresis or distance between the increasing and decreasing pressure is reduced.



**Linearity**

Linearity is a measure of how well a curve approximates a straight line. The method of measurement for linearity is best demonstrated by the example below as shown in the graph. The linearity of a given characteristic is typically measured between 20 to 80% of the operating current range.



## Summary

In this chapter the following concepts were presented:

- The pressure and current levels that define the pressure control range.
- What hysteresis is, how to determine it from a graph and what causes it.
- How PWM reduces hysteresis.
- Valve performance characteristics were defined, specifically, pressure rise, droop and compensation.
- The definitions of repeatability, reproducibility and linearity.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. Define pressure rise. \_\_\_\_\_
2. Is pressure droop dependent on the flow through the valve? \_\_\_\_\_
3. True or False. Pressure compensation refers to the flow varying, regardless of the change in pressure. \_\_\_\_\_
4. How is the pressure control range identified? \_\_\_\_\_
5. Define hysteresis. \_\_\_\_\_
6. Does the magnetic force increase or decrease as current is increased. \_\_\_\_\_
7. What methods are applied to the proportional coil to minimize any existing hysteresis? \_\_\_\_\_
8. What causes hysteresis? \_\_\_\_\_
9. Define pressure range. \_\_\_\_\_
10. True or False. Hysteresis is only dependent on mechanical friction. \_\_\_\_\_

**Notes:**

# Chapter 4: Direct-Acting Pressure Controls

## Objectives

The objectives for this chapter are as follows:

- Learn about direct acting proportional pressure control valves.
- Become familiar with the operation of the manual override in the TS valves.
- Become familiar with the bleeding screw feature available on the TS valves.
- Recognize the difference between the force motor and linear proportional actuator.
- Understand the performance graphs associated with proportional pressure controls.
- Recognize the differences between the TS38-20 and the TS38-21 valves.
- Learn about the EHPR08-33.

## Introduction

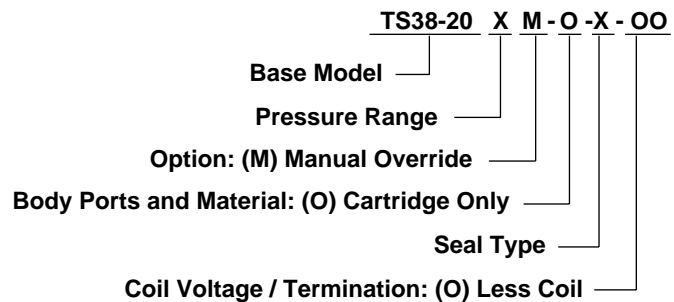
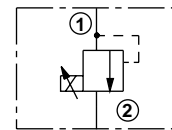
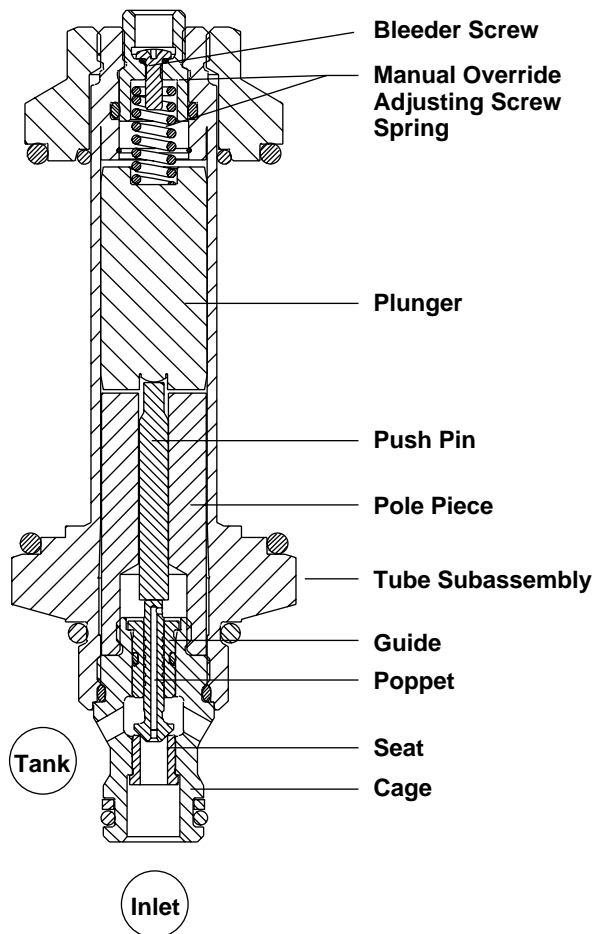
In this chapter, we will look at design, construction and operating parameters of the direct acting pressure control valves available from HydraForce.

The discussion begins with two different types of direct acting relief valves. One of which increases pressure as current is increased and the other where the control pressure decreases as current is increased. The final valve that we will look at is a direct acting pressure reducing and relieving valve.

### TS38-20

The TS38-20 is a direct acting pressure relief valve which is used to control the pressure of the hydraulic system. It is considered a direct acting valve because the poppet is the main hydraulic controlling element and directly controls the pressure. In addition, it can be used to control the pressure applied to a cylinder, motor or another valve.

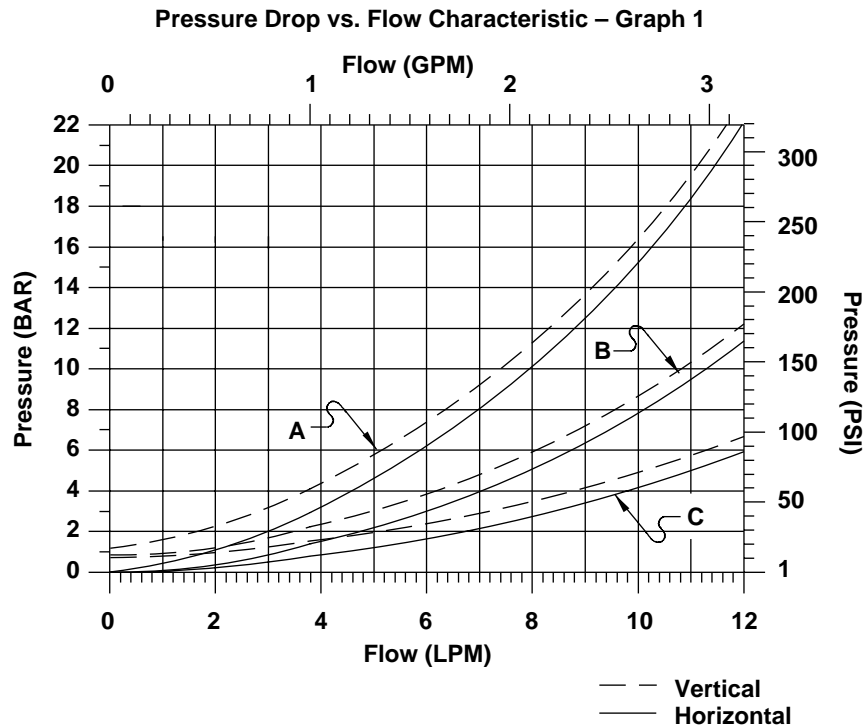
The cross section shown below is that of the TS38-20 valve. The following sections describe how the valve works, the forces acting on the components and the various operating parameters. Also included is a section on the manual override operation and bleeder screw operation. Note the TS38-20 is available without the manual override option.





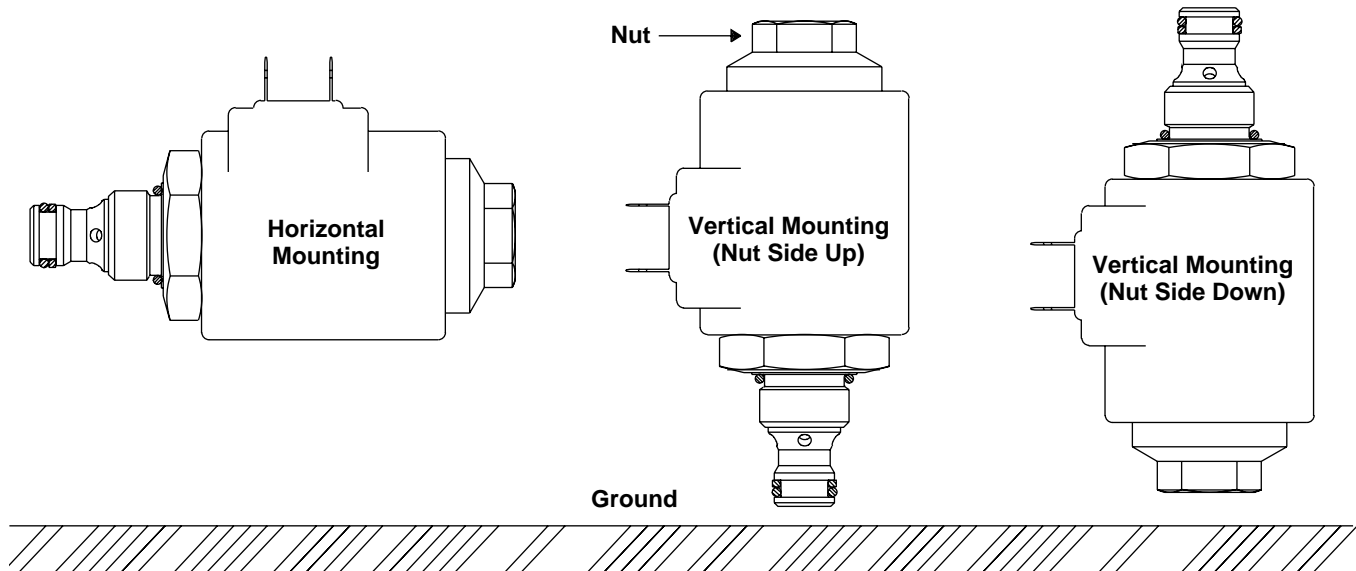
**Performance of TS38-20**

The following graphs illustrate typical performance of the TS38-20A with and without current applied. Graph 1 shows the pressure drop through the valve when no current is applied and the flow varies. Graph 2 shows the pressure dependency on the input of current. Graph 3 shows the relation between flow and pressure with fixed levels of current applied to the coil.

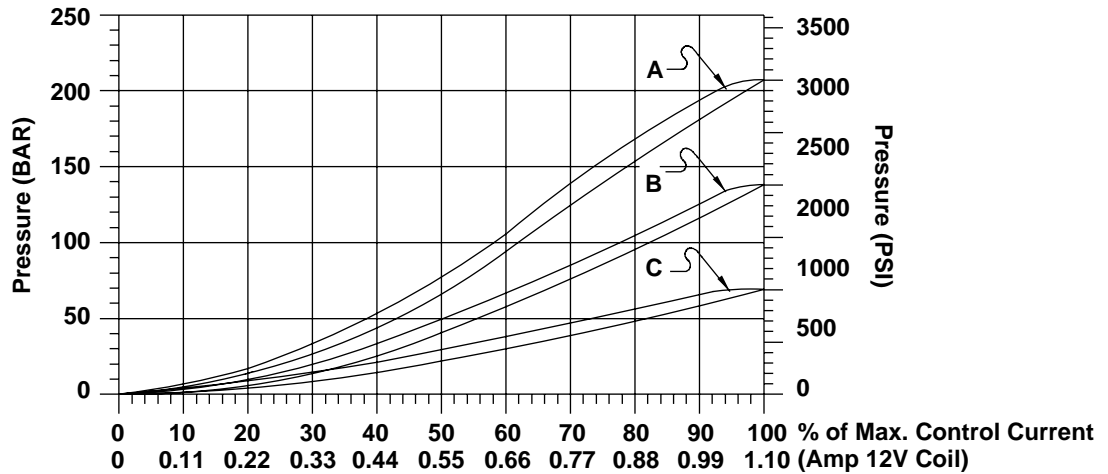


The graph above shows three sets of lines or pressure drop curves. Each set is designated by a letter, either A, B or C which indicates the pressure range. The pressure drop for A is higher than that for C because the seat for A is smaller than the one for C. A smaller seat or orifice means that the flow restriction is higher, increasing the pressure drop. There are two lines (solid and dashed) for each pressure control range. Two lines are shown because the pressure drop through the valve is dependant on the way the valve is mounted (positioned relative to the ground). A diagram of mounting positions is shown on the following page. If the valve is mounted vertically (nut side up), the weight of the plunger pushes down on the poppet. Pressure is required to push the poppet off its seat before oil can flow. This is reflected in the small offset shown between the solid and dashed lines.

The following diagram shows three positions in which valves may be mounted. The picture on the left shows the valve mounted parallel or horizontal to the ground. In this case, the cylindrical part of the coil and valve are laying on their side. The center picture shows the axis of the valve is perpendicular to the ground and is mounted with the nut furthest from the ground. The picture on the right also shows the valve mounted vertically with the nut side facing down.



**Relief Pressure vs. Current – Graph 2  
(Voltage Signal 250 Hz Dither)**



Shown above is the pressure vs. current graph for the TS38-20. Several pieces of information can be extracted from this graph. These include: threshold current, pressure range and hysteresis.

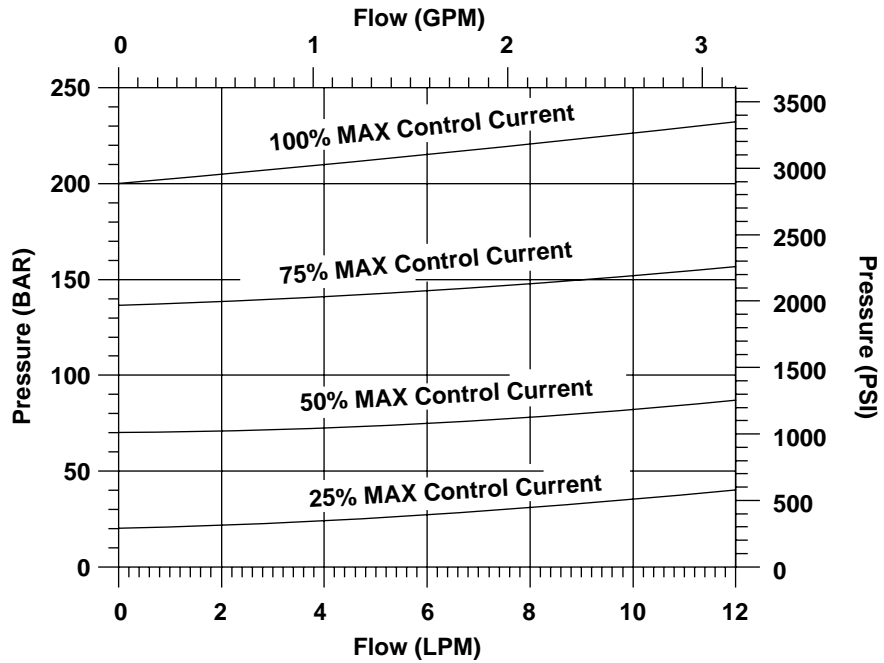
Notice in Graph 2 the threshold current is zero. This occurs because there is no force initially opposing the magnetic force developed by the current. A force which would initially work against the magnetic force is the spring. Since there is no spring opposing the plunger, there is no threshold current required. The spring shown in the manual override portion of the valve will be discussed later in this chapter. Recall that threshold current is defined as the current level which produces a change in the output, in this case a change in pressure. Graph 2 shows that there is an immediate change in pressure with the smallest amount of current.

Recall from Chapter 3 that the pressure range is the pressure between the threshold current and maximum current. There are three pressure ranges for the TS38-20: A: 0-3000 psi, B: 0-2000 psi and C: 0-1000 psi.

The hysteresis shown in the curves above is less than 3.3%. Recall that hysteresis is a measurement of how far apart the increasing and decreasing lines are from one another for a particular pressure range. Looking at 50% current for the A pressure range, shows there is a difference of 100 psi between the two curves, with a range of 3000 psi. This produces a hysteresis of 3.33%  $((100 \text{ psi} / 3000 \text{ psi}) \times 100)$ . As mentioned in Chapter 2, the use of dither decreases hysteresis. If dither or PWM is not used in this case, the hysteresis is 7% or less.

The following graph describes the pressure rise of the TS38-20. This graph shows how the pressure increases as the flow increases when a fixed level of current is applied to the coil. Pressure increases because of the pressure drop through the valve.

**Relief Pressure vs. Flow Characteristic – Graph 3  
For Pressure Range A**



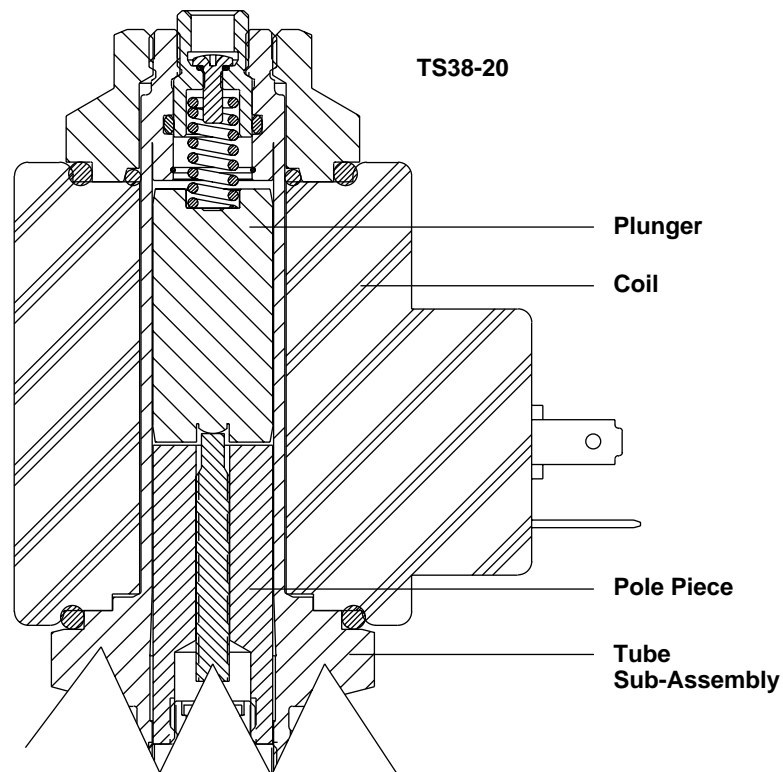
## Forces of TS38-20

The operation of this valve, as in all other valves discussed in this manual, is dependant on balancing the forces acting on the hydraulic components. Since forces such as viscous and mechanical friction, along with residual magnetic force or hysteresis have already been reviewed in Chapter 3, the remainder of the manual will focus on the dominant forces. These include the spring force, pressure force, magnetic force and flow force. Not all these forces are present or exist to a significant degree in all valves. The two main forces that will be covered on the TS38-20 are the magnetic force and the pressure force.

## Magnetic Force

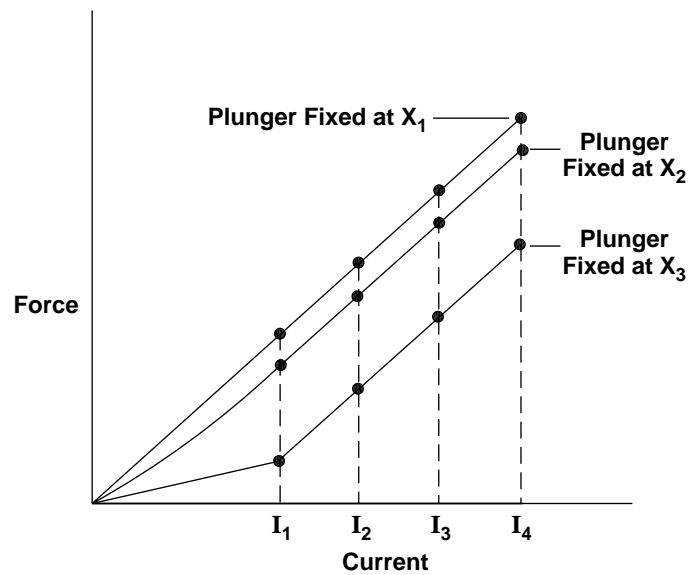
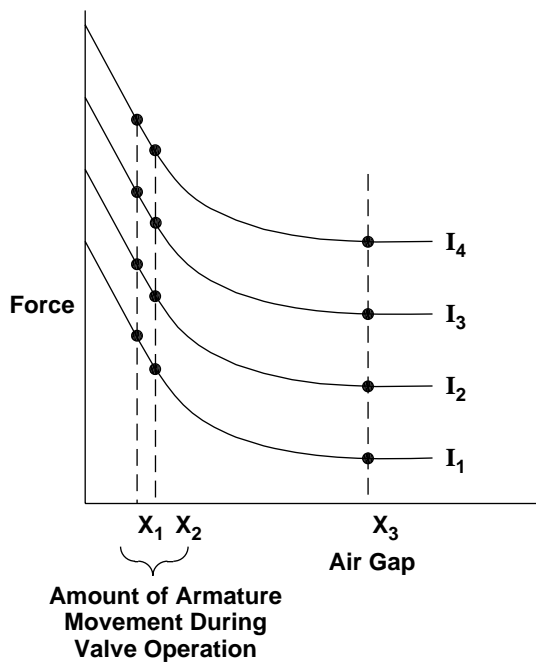
The type of solenoid actuator used in the TS38-20 is known as a flat face actuator or force motor. The part of the valve that makes up the actuator is shown below.

The actuator is termed flat face because of the flat surface between the two elements. The term force motor comes from the relation of the actuator elements to elements found in a common electrical motor. These elements are the winding, armature and pole piece. The armature and pole piece make up what is known as a *stator* in a motor. The stator is the part of the motor that becomes magnetic when the coil is turned on. A motor spins or rotates. The motor generates torque which is work that can be used to spin another device attached to the motor. In the case of a flat face actuator, it generates force that can move something in a straight line. In other words, the actuator is an electric motor that generates force instead of torque. Thus the term 'force motor'.



The graphs below show the force vs. air gap as well as the force vs. current characteristic. The graph on the left shows the force decreasing as the air gap increases. Because of this continual change in force, valves with this type of actuator typically use only a limited amount of air gap or stroke. The graph on the right shows how the force from the actuator increases proportionally with the increase in current.

If the change in air gap was not limited to a small amount (approximately 0.005 inches, or the difference between  $X_1$  and  $X_2$ ) a different force would result for the same amount of current. Compare the force vs. current graphs for air gap  $X_1$  and  $X_2$ . Notice that these differ only a small amount. Therefore, if the armature is at  $X_1$  and the same amount of force is required as the  $X_2$  graph, the current only needs to be increased slightly. However, if the armature was at  $X_1$  and the same force was required as the  $X_3$  graph, the current would need to be increased greatly.



**Pressure Force**

The pressure at the inlet acts on or pushes on the poppet, as shown in the diagram to the below. The type of poppet used in the TS38-20 is known as a *differential area poppet*. This means that the pressure is acting on two different areas. Differential area poppets are used because there is a limitation on the amount of magnetic force which the actuator can develop. For the poppet shown, the pressure force is found as follows:

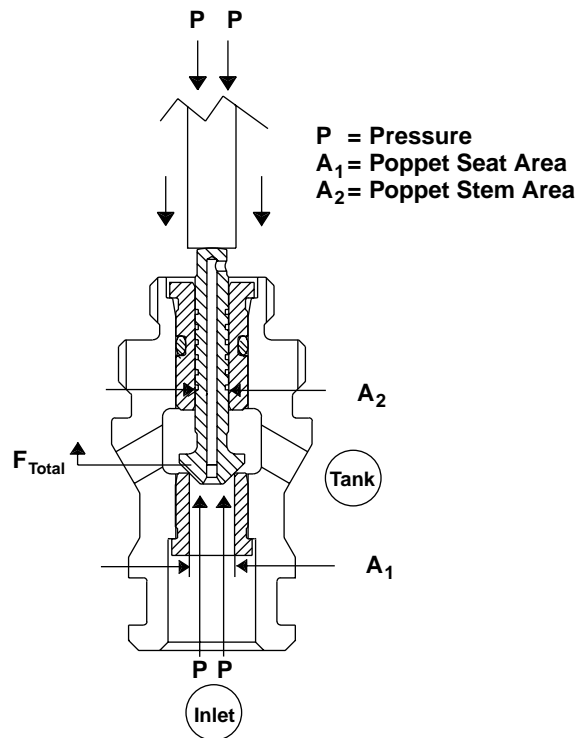
Force = Pressure x Area or  $F = P \times A$

The total pressure force is,  $F_P$ , is based on the difference between  $F_1$  and  $F_2$  where:

$F_1 = \text{Force acting on seat of poppet} = P \times A_1$

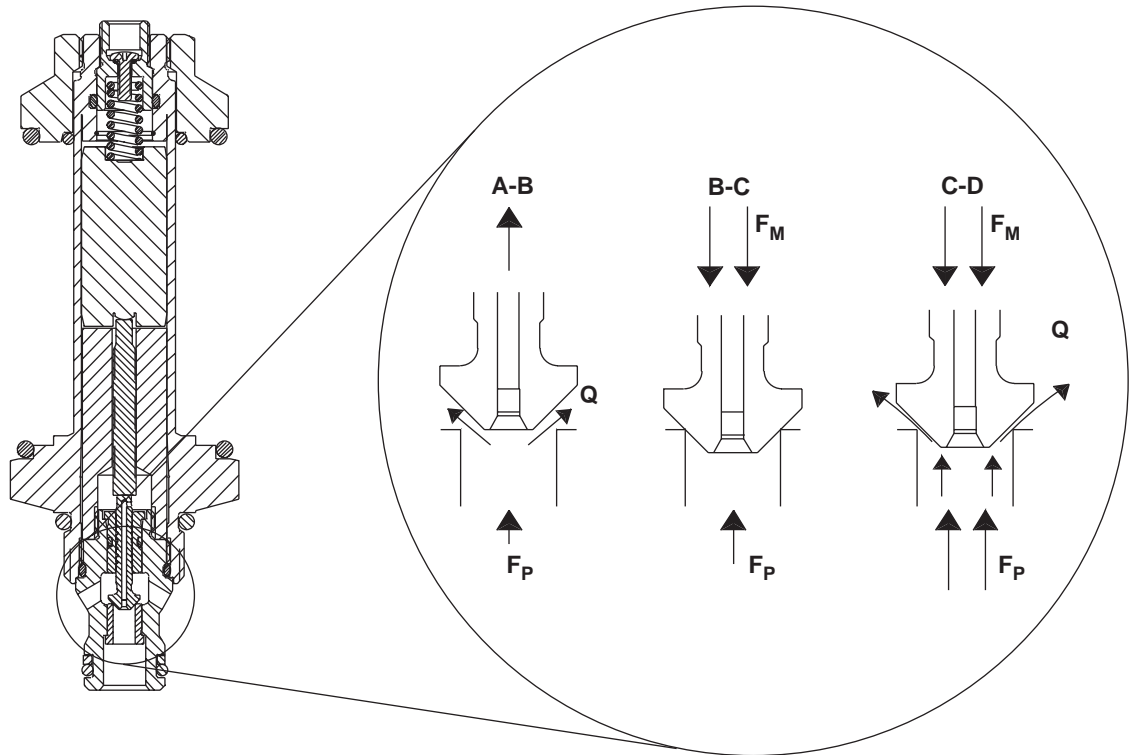
$F_2 = \text{Force acting on stem of poppet} = P \times A_2$

$F_P = F_1 - F_2 = (P \times A_1) - (P \times A_2) = P(A_1 - A_2)$



### Operation of TS38-20

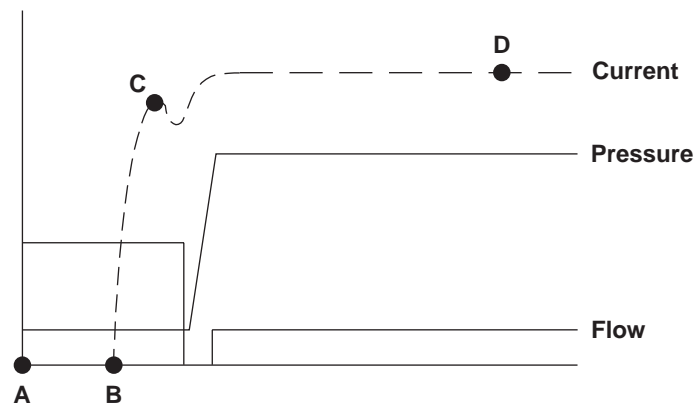
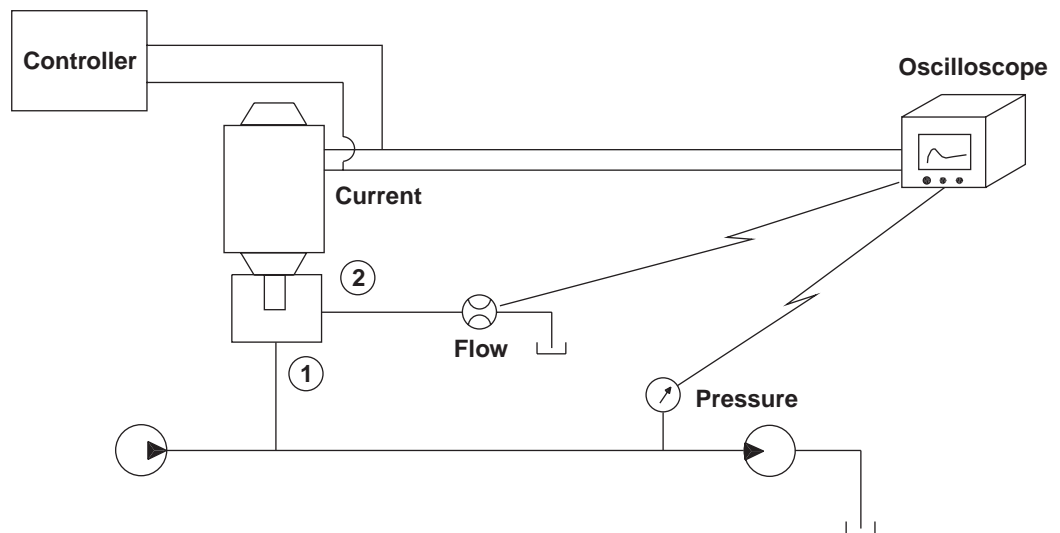
When current is applied to this valve, pressure builds in the system. An example of a system which could use this valve is shown on the following page. In this circuit, as the pressure increases, the torque developed by the motor increases. If the current is held at a certain level, the pressure and torque also remain at that level. When no current is applied, the pressure in the system is simply the pressure drop through the valve.



Where:  
 $F_M$  = the magnetic force  
 $F_P$  = the pressure force

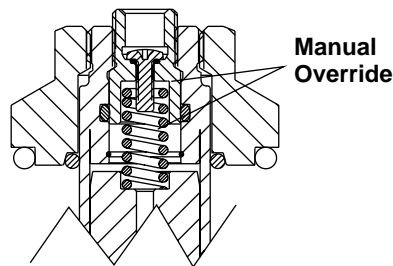


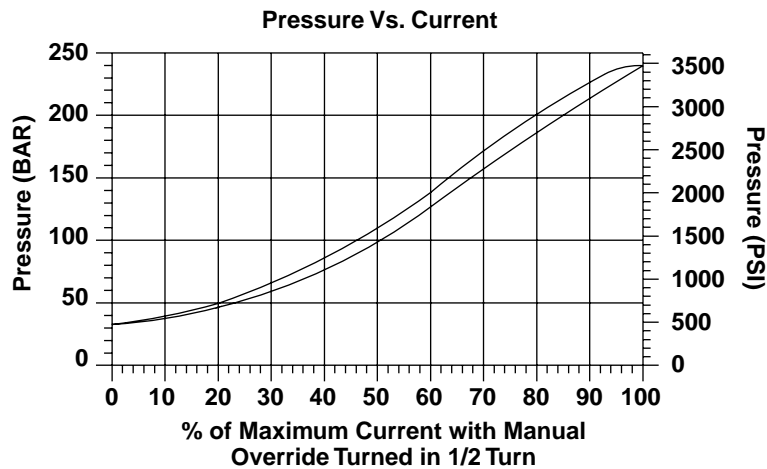
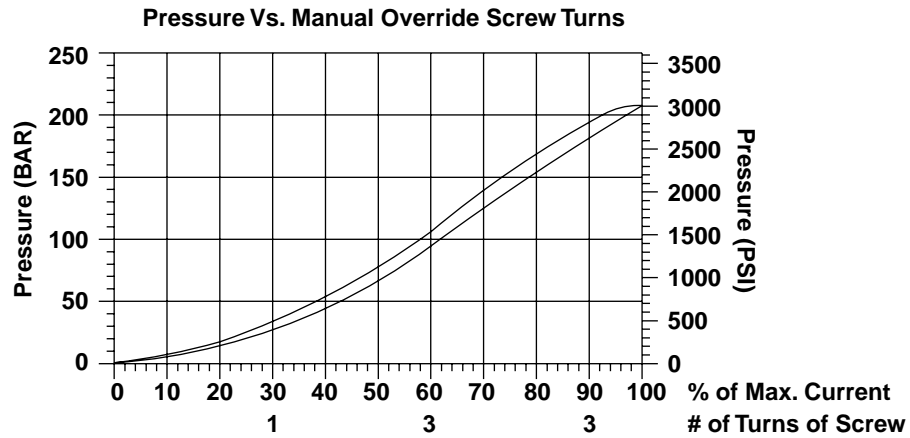
The graph below describes the pressure and flow of the hydraulic circuit below as well as the current applied to the coil. In addition, the diagram of the poppet on the previous page is related to the graph. From points A-B, there is no current applied to the coil. The pressure measured in the system is the pressure drop due to the flow through the valve. When current is applied (B-C portion of graph), the magnetic force ( $F_M$ ) or attraction between the pole piece and plunger builds. As a result, the poppet is pushed on its seat. This blocks the flow from the inlet of the valve to the tank. The pressure at port 1 builds until the pressure force ( $F_P$ ) equals or exceeds the magnetic force. Oil then begins to relieve to tank so that the magnetic force and pressure force remain in balance (C-D portion of the graph).



## Manual Override Operation

The manual override is to be used only in case of loss of electrical power to the solenoid or a failure of the solenoid itself. In other words, it is to be used in emergency situations. The manual override in the TS38-20 is known as a screw style. The TS38-20X is a version of the same valve with no manual override. Initially, the spring is free to float between the plunger and the manual override screw. As the screw is turned, it eventually comes in contact with the spring and it begins to compress the spring. The force applied to the opposing parts, in this case the plunger, increases linearly. The increase in force is directly related to the amount the spring is compressed. That is, force is equal to the distance of compression multiplied by the rate of the spring ( $F = D \times R$ ). The rate is a measure of how stiff the spring is. The stiffer the spring is, the greater the force. The spring force replaces the magnetic force. The number of times the adjusting screw is turned is superimposed on the pressure vs. current graph to the right.



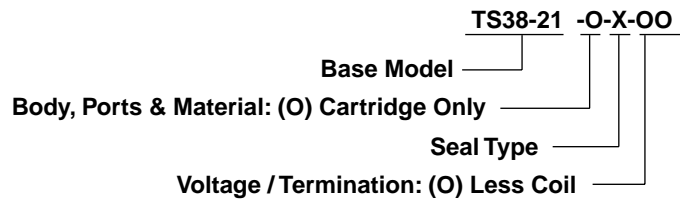
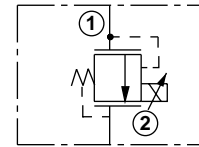
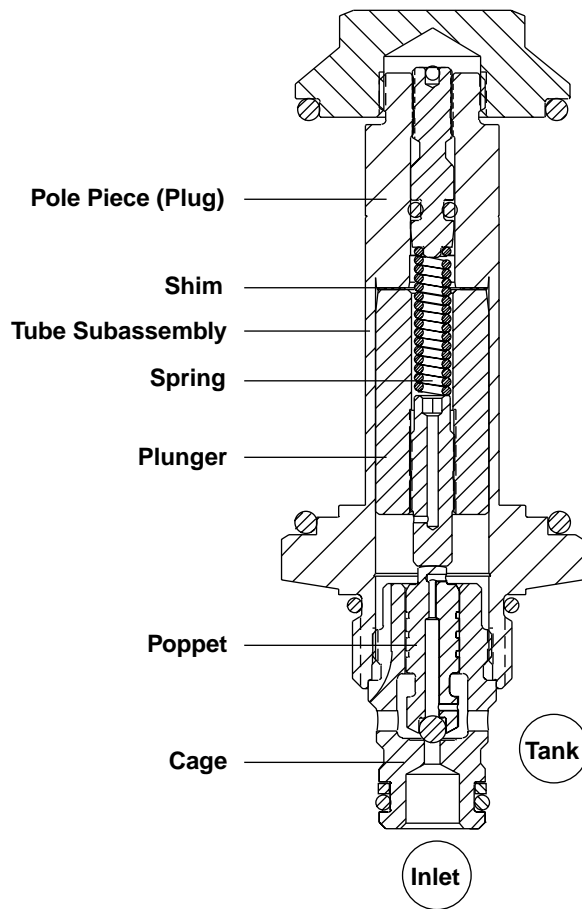


It is important to note that if the manual override screw is not fully backed out (turned full counter clockwise), then the amount the spring is compressed will add to the force developed by the solenoid. The lower graph illustrates this concept. Note that the pressure starts at 500 psi when no current is applied. This offset is due to the fact that the manual override screw is turned clockwise 1/2 to 1 turn. Since the spring force is added to the magnetic force, the manual override must be turned full counterclockwise during normal operation of the valve. If this is not done, the maximum pressure controlled by the TS38-20 may exceed the rating of the rest of the hydraulic system.

While the manual override should typically be used only in an emergency, it could also be used to set the minimum pressure. As noted in the graphs on the previous page, if the manual override is turned in approximately 1/2 turn, the minimum pressure setting of the TS38-20A will be between 200-400 psi, even when no current is applied. For every 1/2 turn from that point, the pressure will increase approximately 500 psi. In order to protect the other hydraulic components in the system,  $I_{MAX}$  from the controller would need to be limited to a level with the rating of these components. Another method which could be used to set the minimum pressure would be through the use of the  $I_{MIN}$  control. In this case, current would need to be applied to the coil to reach that minimum current.

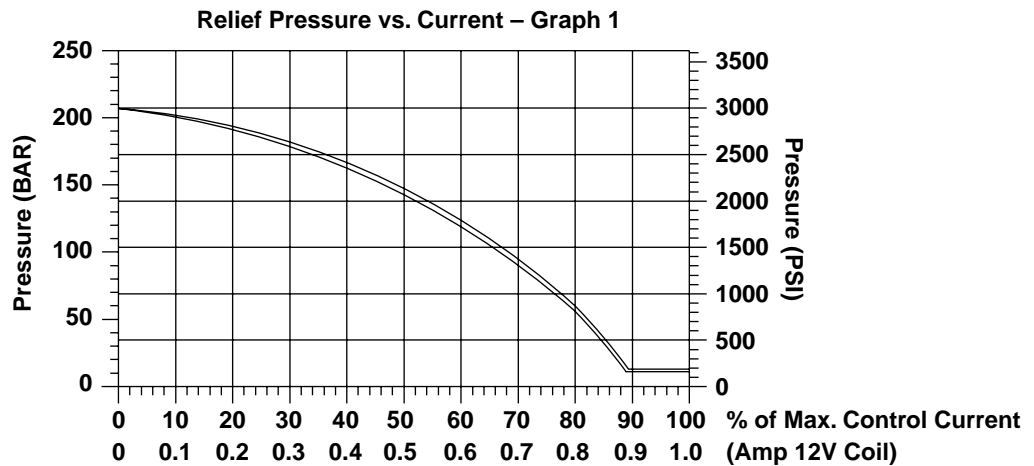
### TS38-21

The TS38-21 is a direct acting, single stage pressure control valve. Like the TS38-20, it can be used to directly control the pressure in a hydraulic circuit. However, because of its limited flow capability (0.3 gpm), it is typically used as a pilot element for larger spools. A cross section of it is shown in the diagram below. It works opposite to the way the TS38-20 valve works, meaning, as current is increased, the pressure decreases. The performance, forces and operation of the TS38-21 will be presented in the following sections.



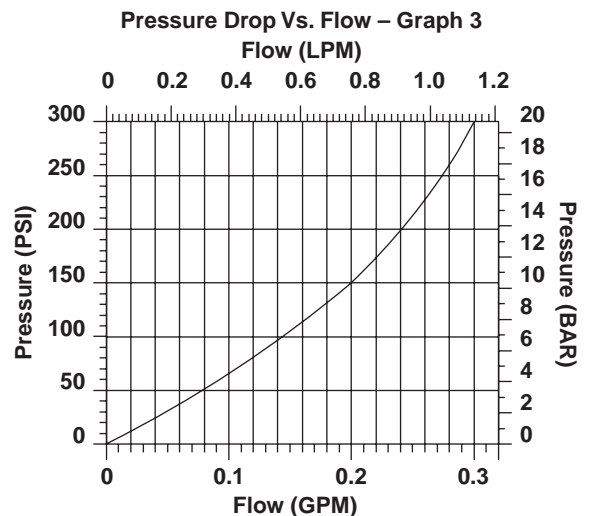
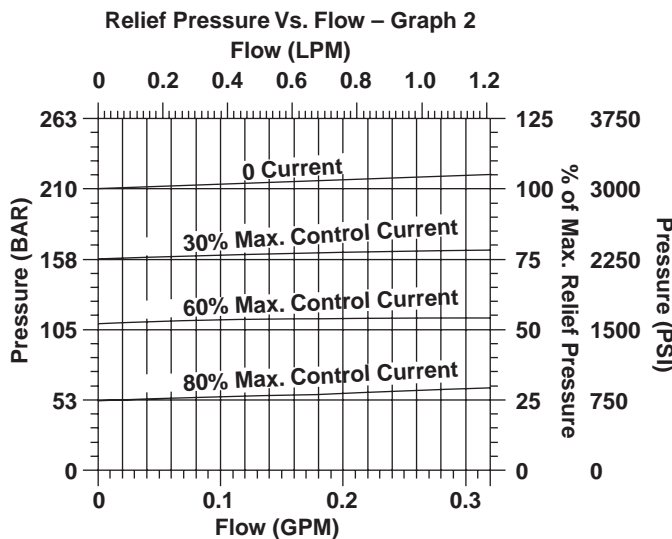
**Performance of TS38-21**

The first graph shown below is the pressure vs. current graph. Notice that the pressure at no current is high and as current is applied, the pressure decreases until approximately 89% of maximum current, at which point the pressure levels off. This will be further discussed in the operation of the valve.

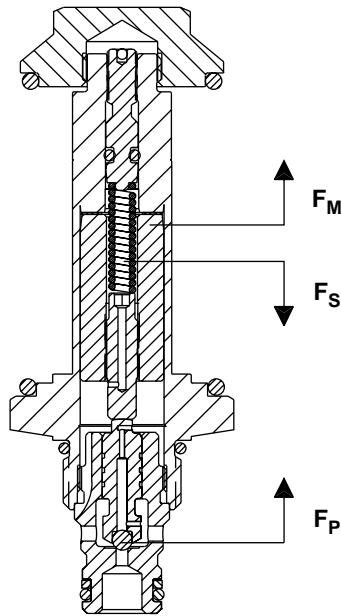


The pressure range for this valve is 3000 to 150 psi. The hysteresis shown above is 30% with 200Hz PWM or dither current applied to the coil. The threshold current is zero, even though there is a spring opposing the armature. This too will be further discussed in the operation section.

Graph two shown below depicts the change in pressure as the flow increases. This change in pressure is a direct result of the pressure drop through the valve as shown in Graph 3.



### Forces of TS38-21



The three dominant forces which will be reviewed are:

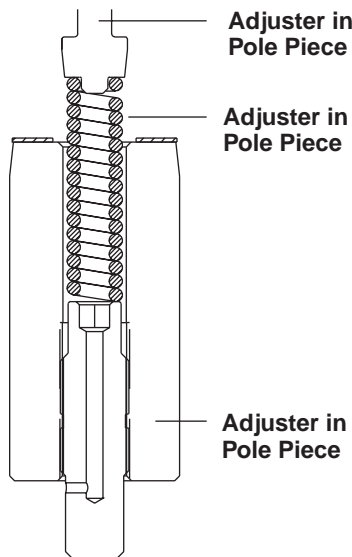
$F_M$  = Magnetic Force

$F_S$  = Spring Force

$F_P$  = Pressure Force

The magnetic force which is present in the TS38-21 is from a force motor or flat face actuator, as described in the TS38-20. The spring force opposes the armature (magnetic) force and at the same time, holds the poppet on its seat. The pressure force pushes the poppet up against the spring. These two forces will be described below.

### Spring Force

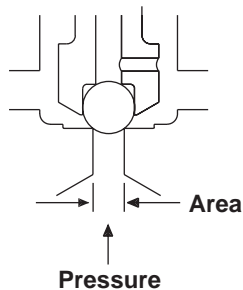


The type of spring inside the TS38-21 as well as the other valves in this manual is known as a compression spring. It is called this because the force of the spring increases as it is compressed. In the TS38-21, the spring is compressed between the adjuster in the pole piece and the plunger. The adjuster in the pole piece is used to consistently set the initial or installed force of the spring. This setting is only done in the factory. The following equation describes the spring force.

$$\text{Force} = \text{Rate} \times \text{Amount compressed}$$

The rate is a constant describing how stiff or difficult it is to compress the spring. The stiffer the rating, the higher the force is for a given amount of compression.

### Pressure Force

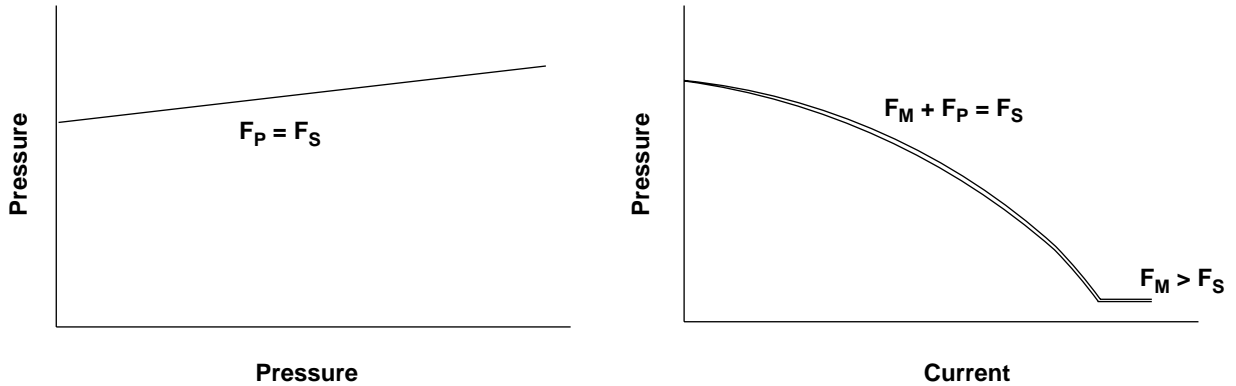


The pressure force acting on the poppet is similar to that of the TS38-20. However, in this case, the pressure acts directly on one area only rather than two. For this valve the pressure is described by the following equation:

$$\text{Force} = \text{Pressure} \times \text{Area}$$

**Operation of TS38-21**

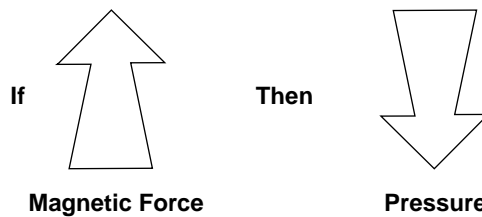
The two graphs below will be used in describing the operation of the TS38-21.

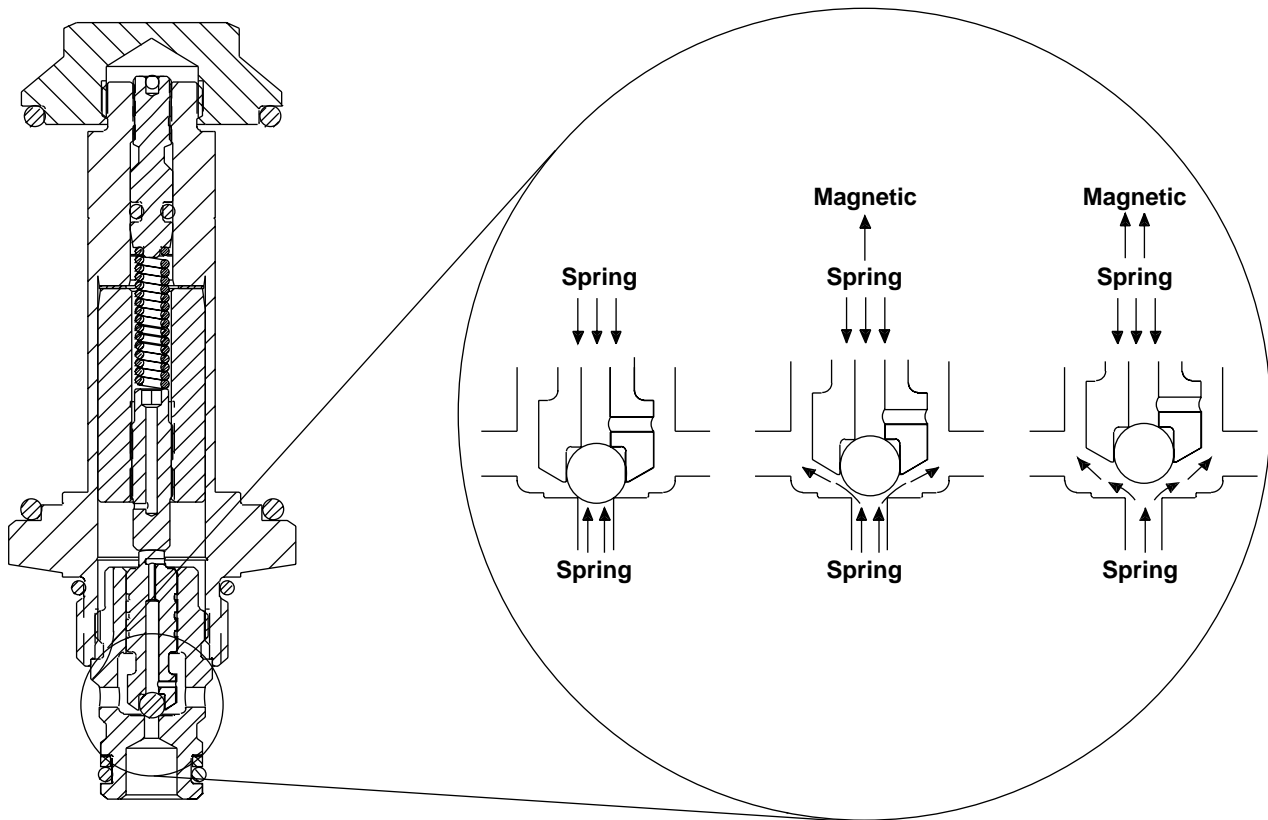


When no current is applied to the coil of the TS38-21, the valve acts as a mechanical relief valve. That is, it works like the relief valve described in Chapter 3. The pressure pushes up against the ball which in turn pushes on the plunger and spring. When the pressure force balances the spring force, the crack pressure is reached. If flow continues to increase, pressure across the valve begins to rise. This can be seen in graph on the left.

If current is applied to the coil, the magnetic force begins to build. As the magnetic force increases, it works against the spring force. Since the spring force is now acting against the poppet and the armature, the pressure begins to decrease. As the current is increased, the pressure continues to decrease until the magnetic force exceeds the spring and the pressure in the system drops to 150 psi.

**When Spring Force Is Constant:**





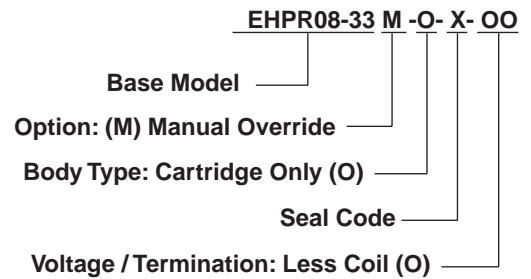
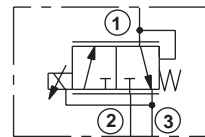
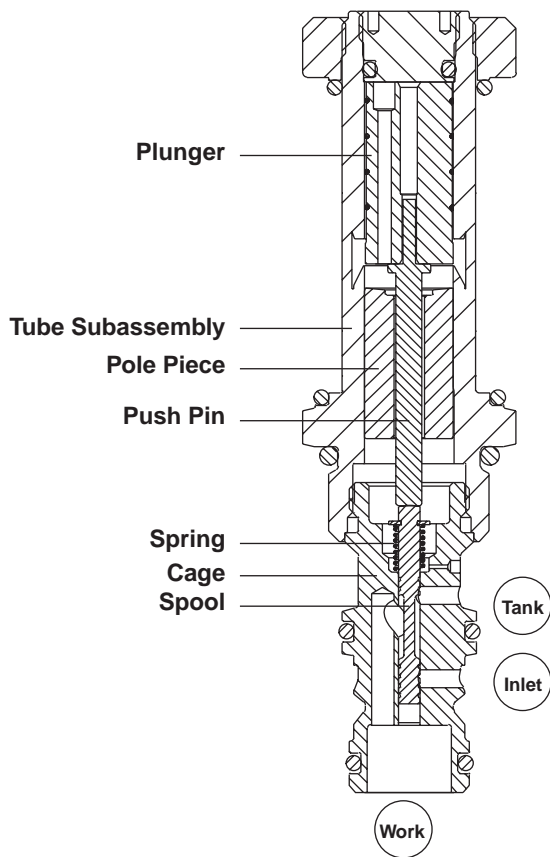
The three diagrams above further clarify the operation of the TS38-21. In the diagram on the left, pressure is pushing on the poppet. No flow is passing through the valve however, because the spring force is greater than the pressure force. In the center diagram, current is applied to the coil, creating the magnetic force. The magnetic force assists the pressure force and oil begins to flow. In the third diagram, the magnetic force is even greater. More oil is allowed to flow, which results in a decrease in pressure.

The lowest value to which the pressure can be decreased is 150 psi, regardless of the current that is applied to the coil. The reason for this is that the shim between the plunger and the pole piece limits the movement of the plunger. This also limits the movement of the poppet. Because the poppet is still close to the seat, the pressure drop through the valve is 150 psi with a flow of 0.3 gpm or the minimum setting of the valve.



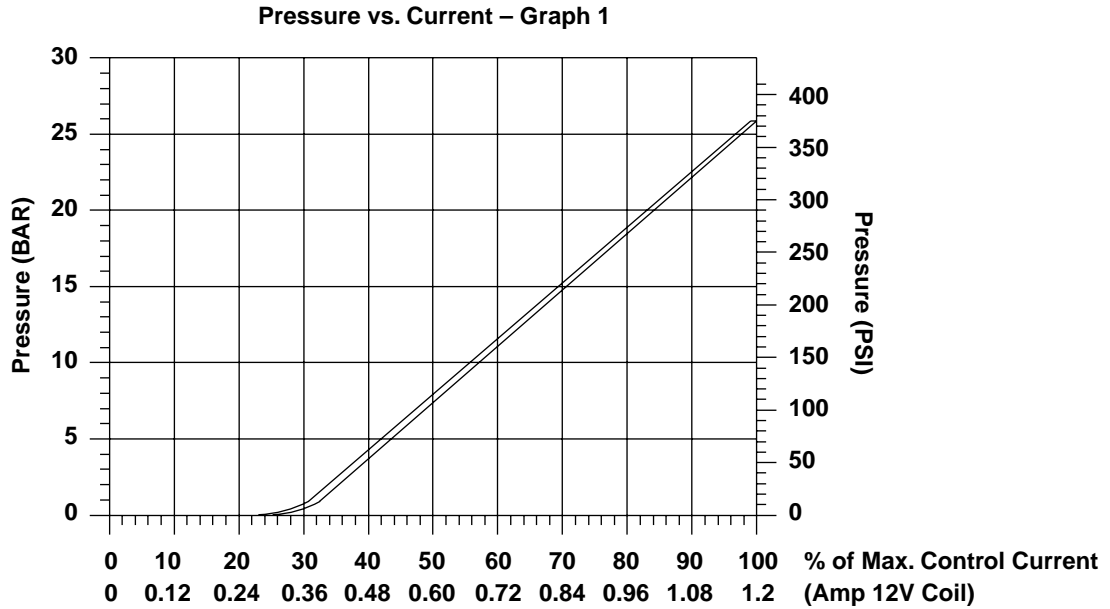
**EHPR08-33**

The cross section of the EHPR08-33 is shown below. It is a direct acting pressure reducing/relieving valve, used to control pressure at the work port (the port at which the load is connected), regardless of the inlet pressure. This is true as long as the inlet pressure is always greater than the control pressure. Direct acting means that there is one hydraulic component (the spool), within the valve controlling hydraulic pressure. This valve can be used to directly control pressure to a clutch. However, because the maximum operating flow is relatively small (1 gpm) for clutch applications, it is typically used to control pilot pressure applied to spools in larger valves. The performance, forces acting on the valve and operation of the valve will be discussed in the sections which follow.



### Performance of EHPR08-33

The following graphs illustrate typical performance of the EHPR08-33. Graph 1 shows the pressure vs. current performance. Graph 2 shows both the pressure rise performance when the valve works in the relieving mode and the pressure droop when the valve works in the reducing mode.



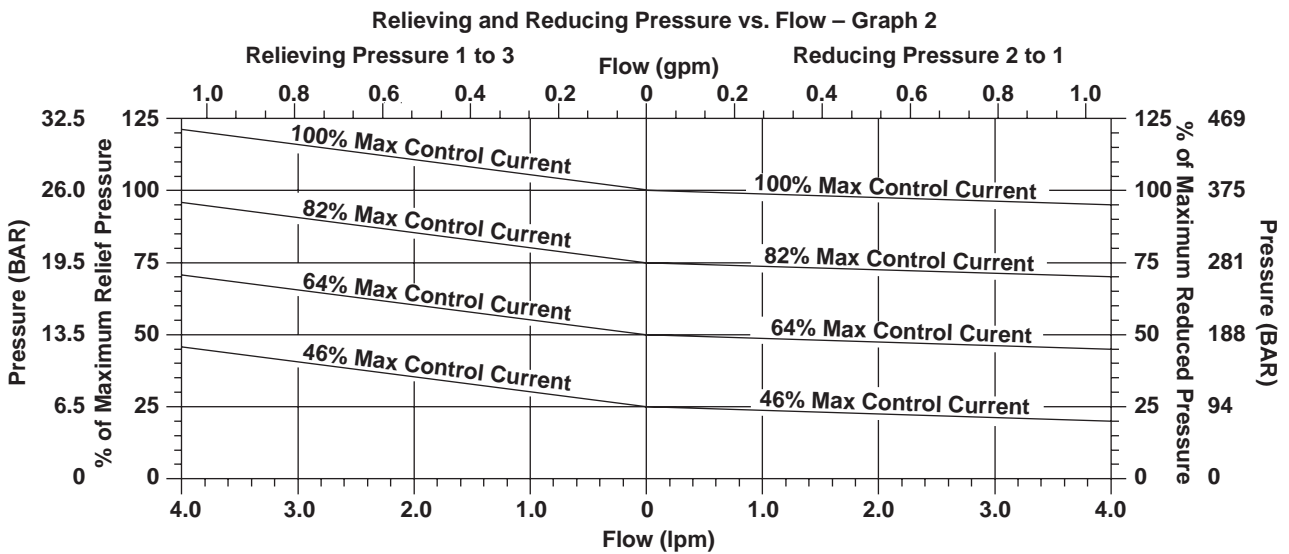
The graph above shows the pressure vs. current performance. Not only does it show how the pressure varies with current, it also gives other performance characteristics such as the threshold current. As the graph shows, the threshold current is 28% of the maximum current or 0.34 mAmp for a 12V coil.

The graph also shows the pressure range which is 0-375 psi for the EHPR08-33. Unlike the TS38-20, there is only one pressure range available.

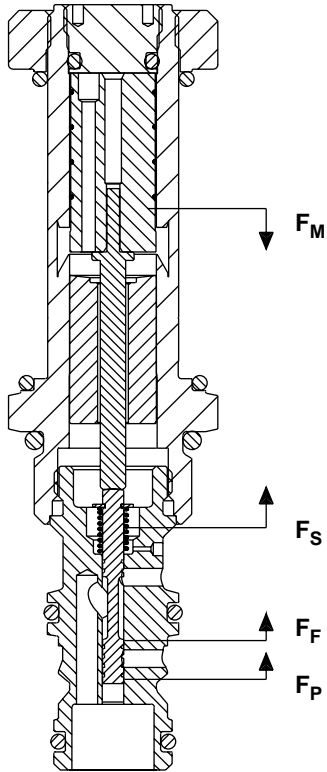
Another performance characteristic depicted by the curve is the hysteresis. The hysteresis for the EHPR08-33 is less than 3% when dither or PWM is applied to the coil. If smooth DC current is applied to the coil, the hysteresis is 10%. (This is not shown in the graph.)

The following graph shows the pressure rise and droop of the valve. The portion on the right shows the pressure droop or how the desired control pressure decreases as the flow increases. Pressure droop occurs when the flow is increased and the current is held fixed, while the valve is reducing pressure. For example, the “46% of maximum current” line shows that when there is no flow through the valve, the control pressure is 94 psi. When there is 1 gpm, the control pressure is approximately 75 psi.

The pressure rise is also shown on this graph, and is associated with the relieving mode of the valve. The change in the work pressure is a result of the pressure drop in the valve. The graph shows the pressure rises by approximately 20 psi for a change in flow from 0 to 1.0 gpm.



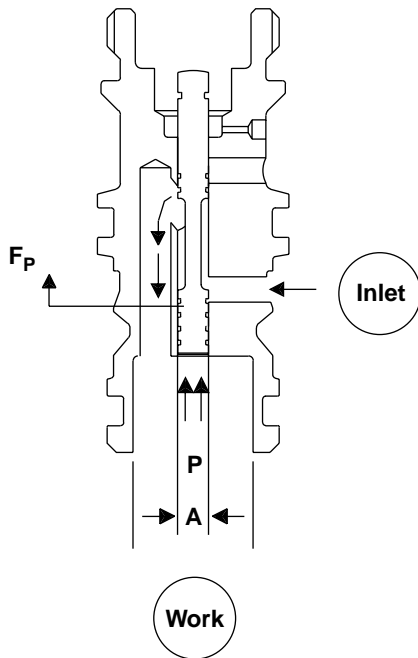
**Forces of EHPR08-33**



The forces acting on the valve components which will be considered in the following section are:

- F<sub>P</sub> = Pressure Force
- F<sub>M</sub> = Magnetic Force
- F<sub>F</sub> = Flow Force
- F<sub>S</sub> = Spring Force

These four forces and the way they act on the EHPR08-33 will be described in the following section. The spring force is the same as was described for the TS38-21 and will therefore not be discussed in this section.

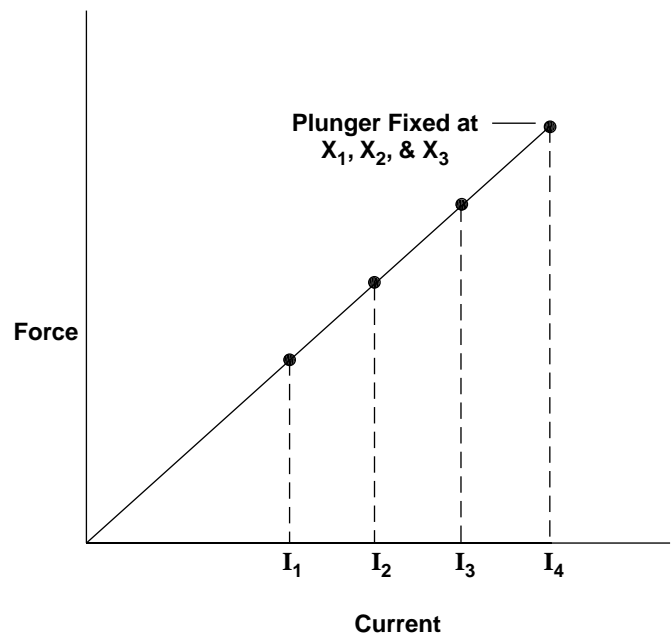
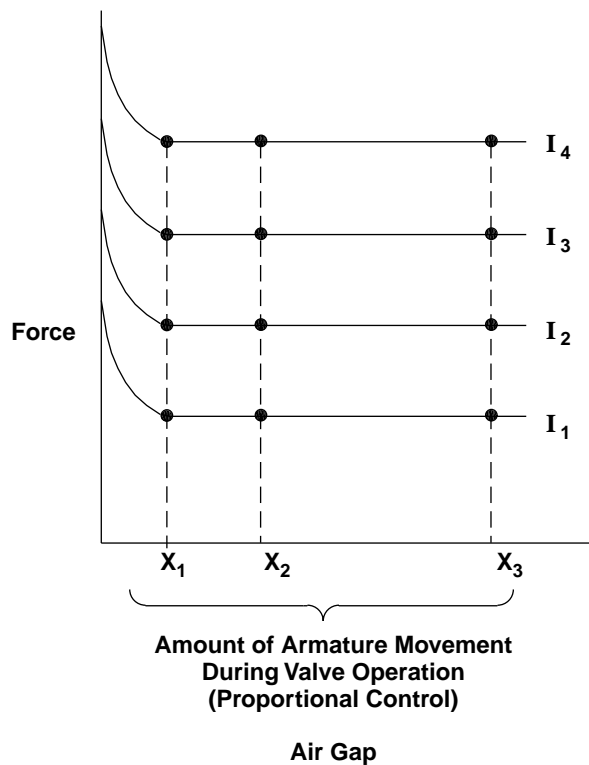


**Pressure Force**

When the coil is energized, the spool moves down, connecting the inlet to the work port. The oil flows from the inlet to the work port and pushes up on the bottom of the spool. This area, multiplied by the pressure, gives the pressure force ( $P \times A = F$ ).

### Magnetic Force

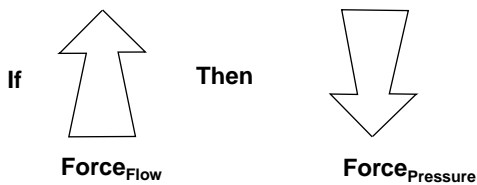
The type of actuator used in the EHPR08-33 is a linear proportional solenoid actuator. It is called this because the force builds proportionally to the current. In addition, it is called linear because the armature moves in a straight line as opposed to a rotary actuator which turns like a wheel. The following graphs show how the force varies with the movement of the plunger or change in air gap and how the force varies with current.



Both graphs show that the force remains constant regardless of the position of the armature. This allows the stroke of the hydraulic element, which the armature controls, to be much greater as compared to the flat face actuator. The drawback to this increased stroke is a decrease in force for the same size coil.

### Flow Force

In addition to the magnetic force, pressure force and spring force, a fourth force which acts on the spool would be the flow force. This force increases as the flow from port 2 to port 1 increases. As a result, the magnetic force is opposing two forces. In order for the forces acting on the spool to remain in balance, the pressure force needs to decrease. This is shown in the droop of the pressure in the reducing pressure vs. flow graph.



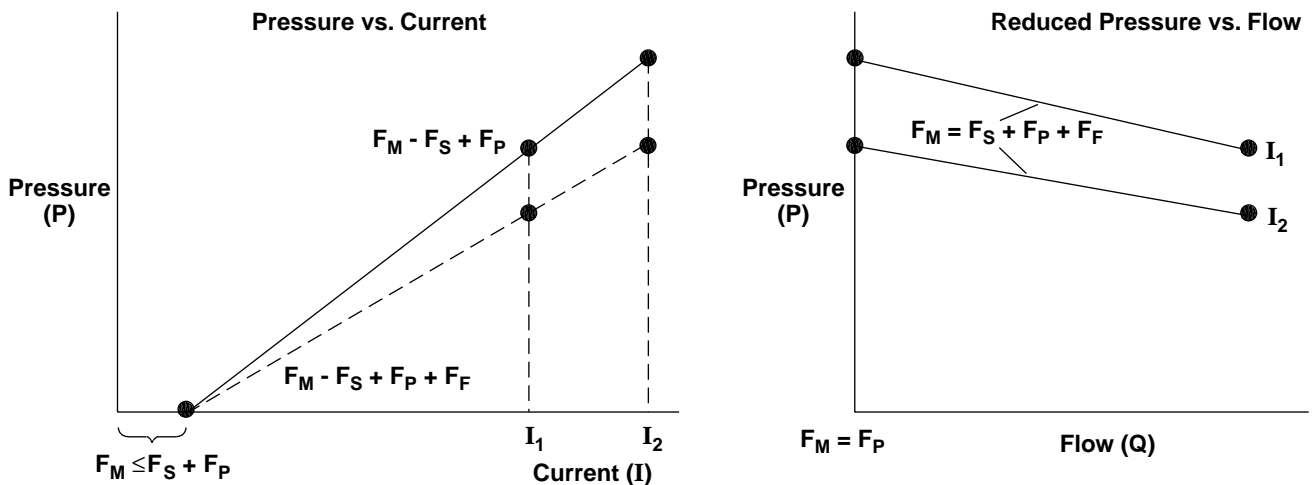
$$Force_{magnetic} = Force_{pressure} + Force_{flow}$$

This is true as long as the current to the coil or magnetic force remains unchanged.

### Summary of Forces

The force can be summarized by the following equation:  $F_M = F_S + F_P + F_F$

The interaction between the forces can be seen in the two graphs below. The graph on the left shows two curves representing the performance of the valve as current is varied. One curve gives the pressure when there is no flow or the oil is dead-headed (maintaining the pressure in a fixed volume of oil) at the work port. The other curve on this graph depicts the performance when oil flows out of the work port.

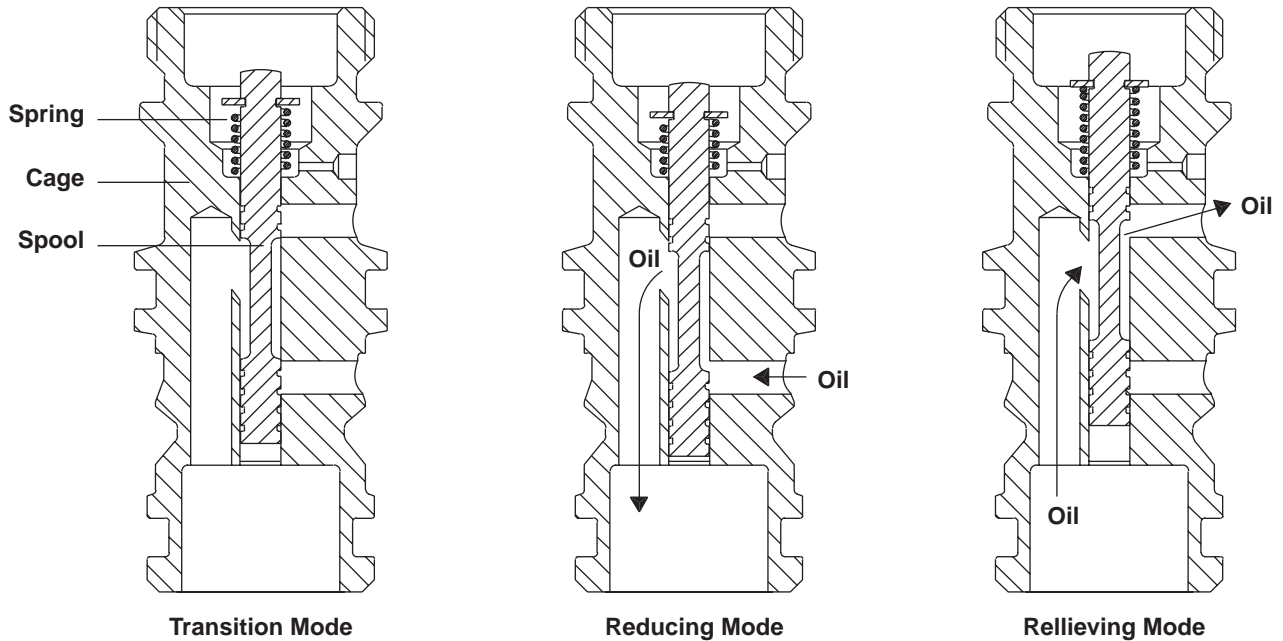


The graph above-right shows how the pressure droops from the no flow (deadheaded) condition to full flow. There are two curves on this graph as well. Each represents a fixed current level. Again, the pressure droop is caused by the fact that the magnetic force is held constant and the flow force is increasing as flow increases.

### Operation of EHPR08-33

Refer to the graphs on the previous page as well as the diagrams below for the following section.

The first diagram shows the spool in transition. This is the point where all ports are blocked. In this case, the current was just applied to the coil, causing the plunger to move down against the spool, compressing the spring. This is shown in the pressure vs. current graph as the flat part just before the threshold current is reached, or  $F_M \leq F_s + F_p$ . The next picture shows when the spool has been pushed down by the armature and opens a small passage between the inlet and the work port. Oil flows downstream of the work port. If no more oil is needed downstream of the workport, the system is considered to be dead-headed. The spool then moves between allowing oil from the inlet to the work port (reducing) and the work port to the tank (relieving). The relieving mode is shown in the diagram on the right.



## Summary

In this chapter the following concepts were presented:

- What a direct acting proportional valve is.
- The function of the TS38-20 as well as how it works and the forces acting on it.
- The positions in which valves can be mounted.
- The manual override option is and how it works.
- The bleeder screw and how it is used.
- The performance, forces and operation of the TS38-20.
- The performance, forces and operation of the EHPR08-33.
- What a force motor or flat face actuator is.
- The force vs. air gap characteristic of a linear proportional actuator.
- Why pressure droop and pressure rise occurs when using pressure reducing/relieving valves.



## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. Why is it important to assure that the manual override of the TS38-20 turned fully counterclockwise when power is applied to the coil? \_\_\_\_\_
2. Why is the stroke of the TS38-20 limited to 0.005 inches? \_\_\_\_\_
3. Describe the spring force. \_\_\_\_\_
4. What type of actuator is used in the EHPR08-33? \_\_\_\_\_
5. Does the force in the EHPR08-33 remain constant as the air gap changes? \_\_\_\_\_
6. What is the hysteresis of the TS38-20 with PWM? \_\_\_\_\_
7. What is the pressure range of the TS38-21? \_\_\_\_\_
8. What is the typical application for the TS38-21? \_\_\_\_\_
9. What causes the pressure droop in the EHPR08-33? \_\_\_\_\_
10. Why is the EHPR08-33 typically used to pilot larger spool valves? \_\_\_\_\_

**Notes:**

# Chapter 5: Pilot Operated Pressure Controls

## Objectives

The objectives for this chapter are as follows:

- Learn how pilot operated pressure controls work.
- Become familiar with the three different proportional pressure reducing/relieving valves.
- Understand the difference between the TS10-26 and the TS10-27.
- Learn about the hysteresis and pressure ranges of the pilot operated valves.
- Recognize the forces acting on the components of the pilot operated valves.

## Introduction

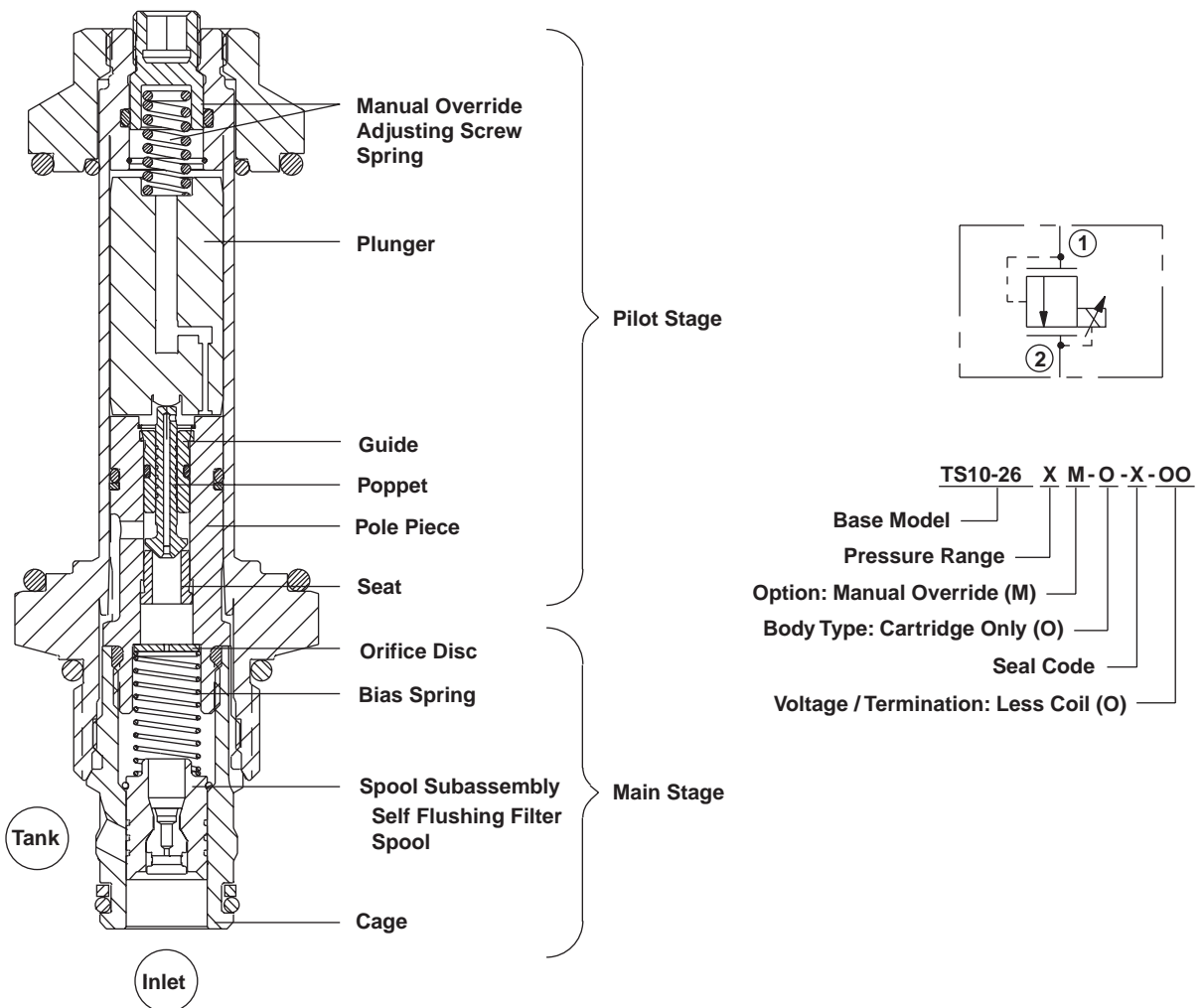
In this chapter, we will look at design, construction and operating parameters of the pilot operated pressure control valves available from HydraForce. The discussion begins with pilot operated relief valves. The relief pressure of one increases with increasing current, while the other decreases with current. The second set of valves to be discussed are the pilot operated pressure reducing and relieving valves. HydraForce offers three standard valves which meet this function. The differences between each will be described.

# HYDRAFORCE

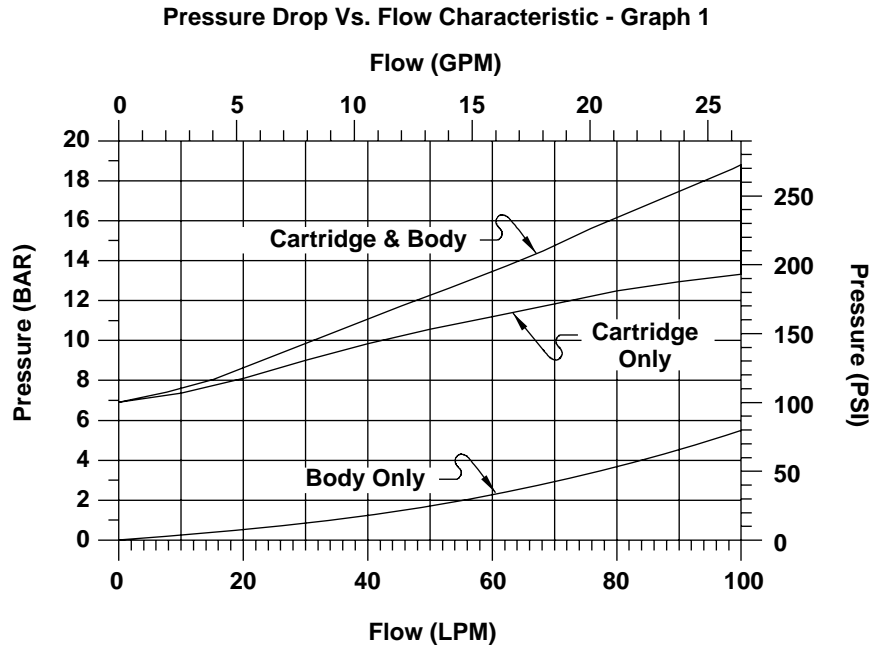
## TS10-26

The cross section shown below is the TS10-26XM. This valve is considered a two stage or pilot operated pressure control valve. It is a two stage device because there are two components used to control hydraulic pressure. The first one is the pilot stage which is made up of parts similar to the TS38-20 (see cross section below). The second stage is called the main stage. The pilot stage controls the main stage which in turn controls the hydraulic system pressure. Further explanation of how the TS10-26 works will be provided in the forces and operation sections following. Also included is an explanation of the performance characteristics. Note that the descriptions provided in the following sections also apply to the TS12-26. This valve is a larger version of the TS10-26 and is designed for higher flows. The TS10-26 and the TS12-26 are available without the manual override option. Refer to the TS38-20 section in Chapter 4 for the operation of the manual override and bleeder screw.

### Performance of TS10-26



An example of the typical performance characteristics used to describe the function and proper application of the valve are provided in this section. Graph 1 shows the pressure drop. Graph 2 shows pressure vs. current and Graph 3 shows the pressure rise as flow through the valve changes.

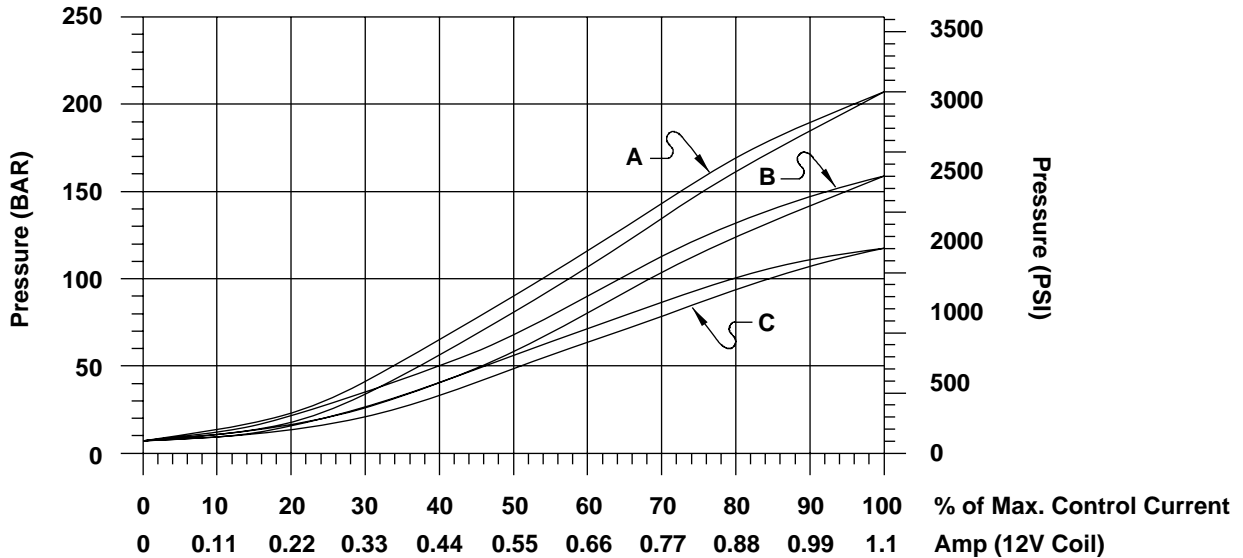


The graph above shows the pressure drop across the main stage portion of the valve. Notice that the pressure is offset by 100 psi or 7 bar when flow is first recorded. This indicates that some amount of pressure is required at the inlet of the valve before oil can flow. The pressure is required to overcome the spring which is acting on the main stage spool. Also notice, unlike the TS38-20, there is only one curve. This one curve represents the pressure for all three pressure ranges. This will be discussed in the operation of the valve.

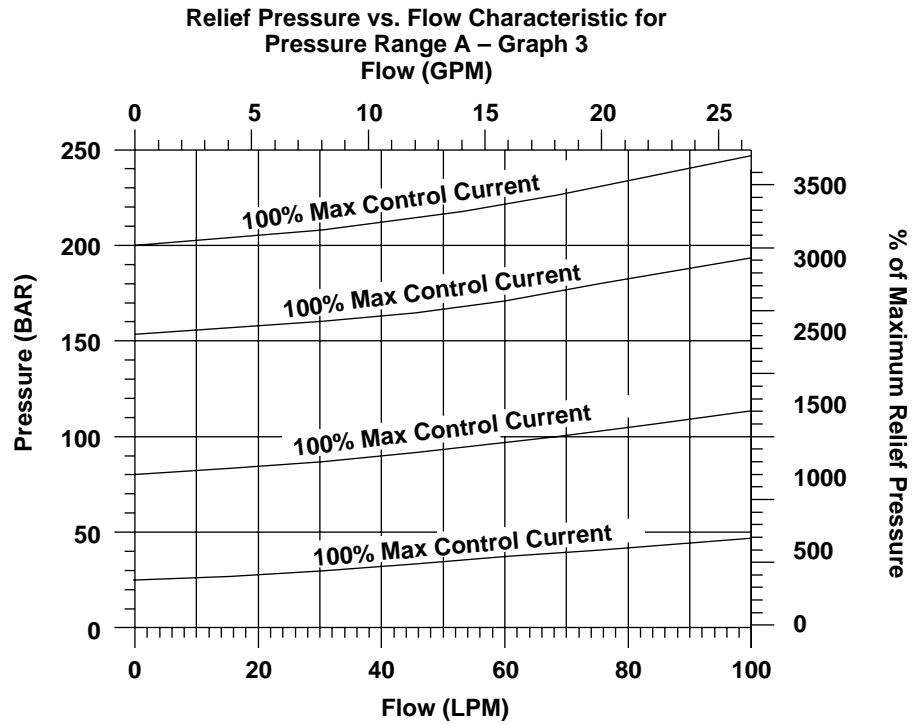
The next graph shows the pressure vs. current characteristic for the three different pressure ranges. The pressure ranges are:

- A = 100 to 3000 psi
- B = 100 to 2300 psi
- C = 100 to 1700 psi

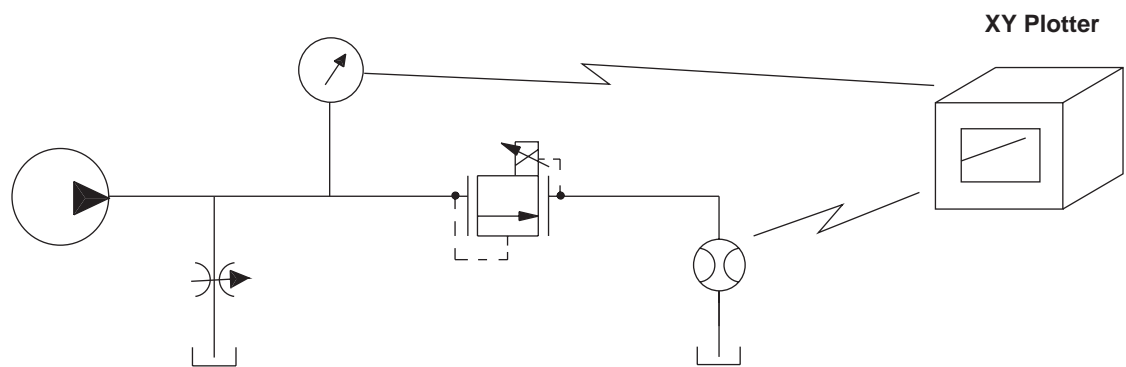
Relief Pressure vs. Current (DC) Characteristic – Graph 2



Like the TS38-20, the threshold current is zero because there is no spring force acting on the poppet in the pilot section. There is an immediate change in pressure with the smallest change in current. The hysteresis measured at 50% of  $I_{MAX}$  in each of the previous curves is less than 3%.

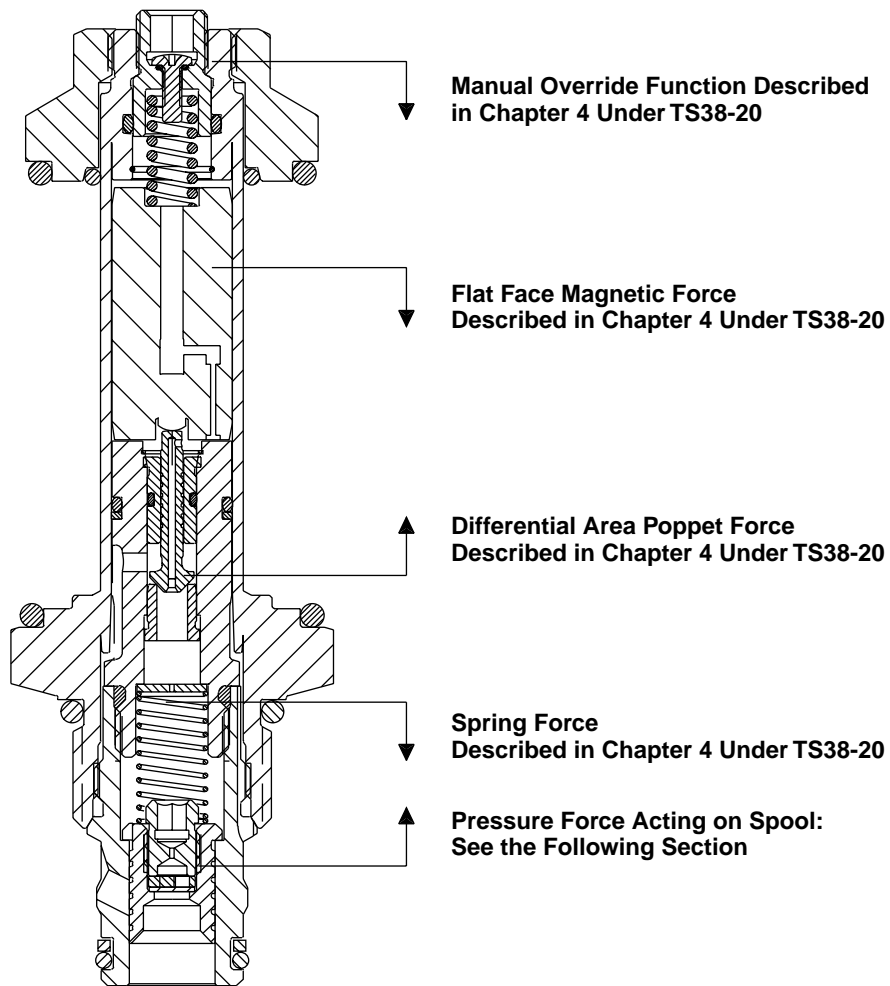


Graph 3 above shows the pressure rise or increase in pressure setting due to increasing flow. These curves were made by varying flow from zero to 26.5 gpm (100 lpm) and recording the change in system pressure. A schematic for this test is shown below.



### Forces on TS10-26

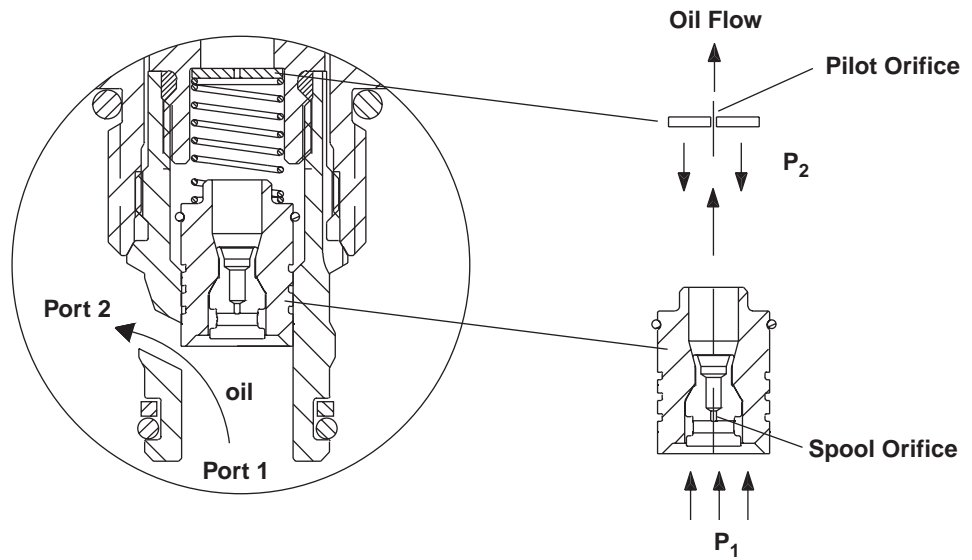
As with the previous chapter, the discussion of forces acting on the valve components will be restricted to the ones with the greatest magnitude. These are the magnetic force, pressure force acting on the poppet, spring force and the pressure force acting on the spool. The first three forces are exactly the same as the TS38-20. The remaining force (pressure force acting on spool) will be described in the following section. These four forces and the direction in which they act on the parts are shown in the diagram below.





**Pressure Force on Spool**

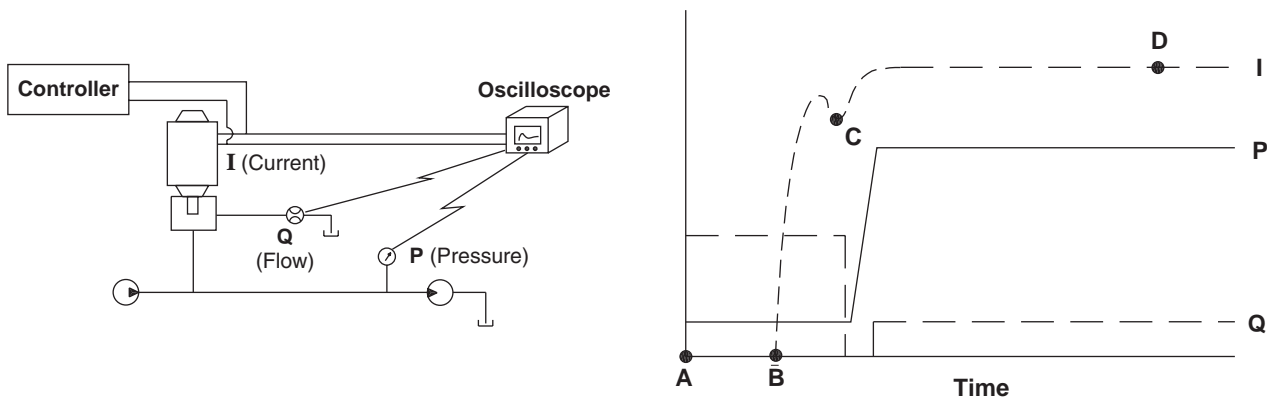
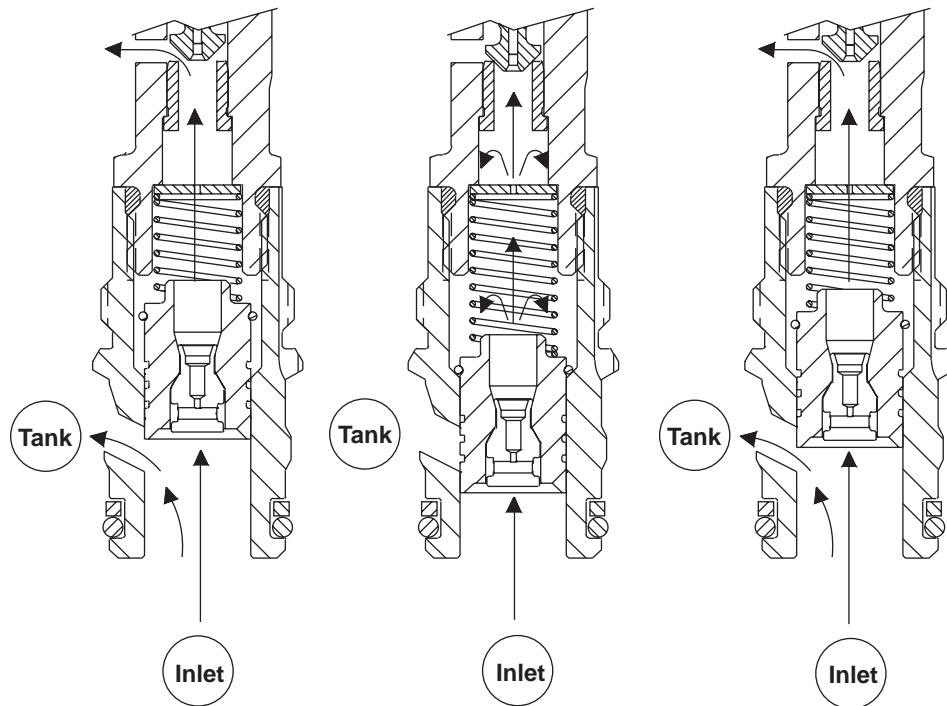
The diagram below shows the spool and the orifice disc. When the valve is in use, the pressure pushes up on the spool. When the poppet is off its seat (not shown here), oil flows through an orifice. There is a difference between  $P_1$  and  $P_2$  or  $P_1 > P_2$ . Therefore,  $P_1$  pushes the spool up, allowing oil to flow from port 1 to port 2.



### Operation of TS10-26

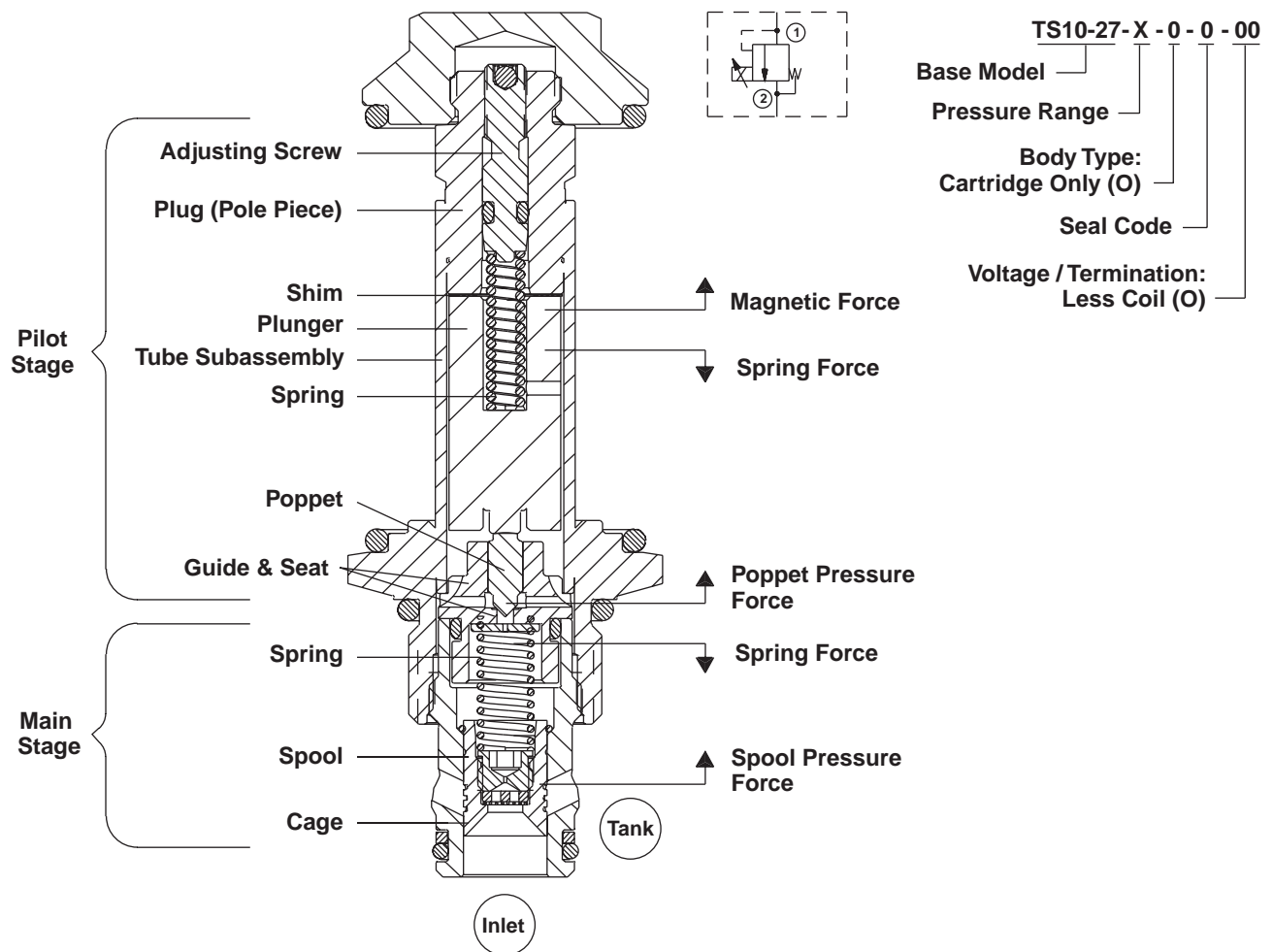
The graph below describes the pressure and flow of the hydraulic circuit as well as the current applied to the coil. From points A-B, there is no current applied to the coil. The pressure measured in the system is the pressure drop due to the flow through the valve. When current is applied (B-C portion of graph), the magnetic force or attraction between the pole piece and plunger builds. As a result, the poppet is pushed on the seat. This blocks the oil in the pilot chamber. The spool is then pushed down, closing off the inlet from the tank port. This is shown in the middle diagram. When the pressure at the inlet is high enough to overcome the magnetic force, the poppet is pushed off the seat. The spool then moves, allowing oil to flow from inlet to tank.

### TS10-27



The TS10-27 is a two stage pilot operated pressure control. It is made up of a pilot section from the TS38-21 and the main stage is similar to that of the TS10-26. The function of the TS10-27 is to maintain a high pressure setting when no current is applied. The difference between the TS10-26 and the TS10-27 from an application viewpoint, is that the TS10-26 increases pressure and the TS10-27 decreases pressure, with an increase in current. The components that make up the TS10-27 as well as the forces acting on these parts are shown in the cross section below.

A description of the performance graphs can be found in the following section. However, there will be no discussion of the force acting on the TS10-27 or a description of the operation. This will be left as an exercise in the study questions at the end of this chapter.



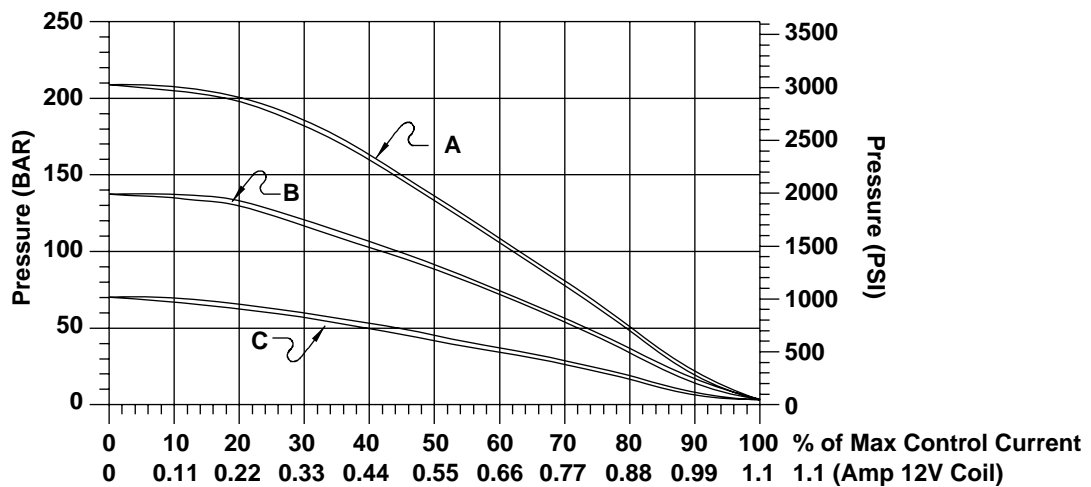
- The forces listed above have been described in previous sections. These are:
- magnetic force, described in Chapter 4 under TS38-20
  - spring force, described in Chapter 4 under TS38-21
  - pressure force acting on the poppet, described in Chapter 4 under TS38-20
  - pressure force acting on the spool, described in this chapter under TS10-26

### Performance of TS10-27

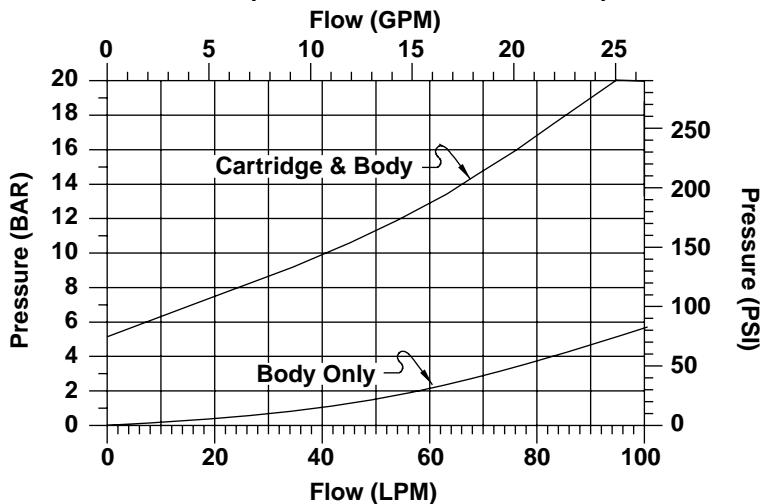
The three performance graphs used to describe the operating characteristic of the TS10-27 are shown below. The first graph shows the pressure vs. current characteristic for the three standard pressure ranges.

- A = 3000 to 100 psi
- B = 2000 to 100 psi
- C = 1000 to 100 psi

Relief Pressure vs. Current (DC) Characteristic – Graph 1



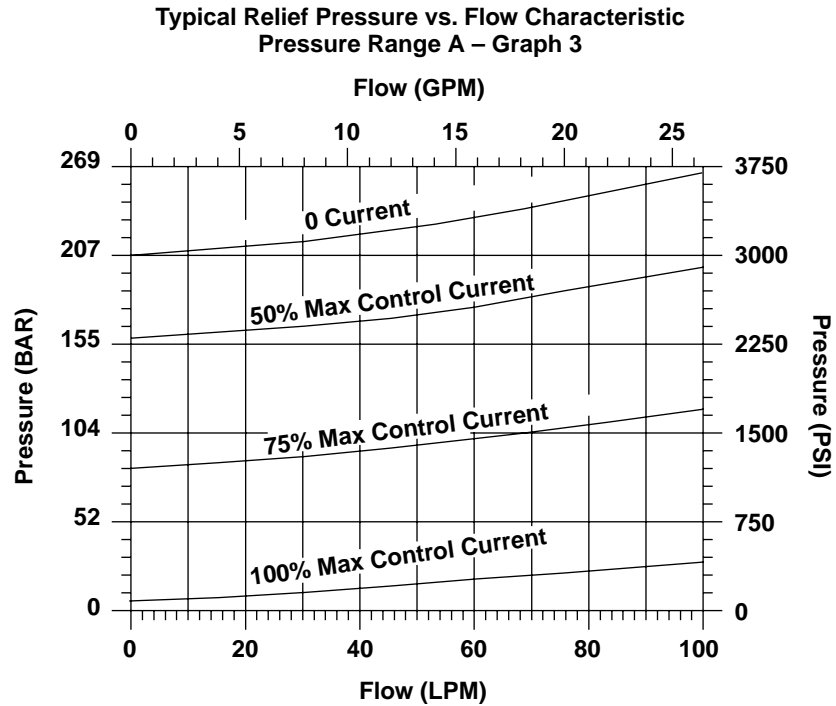
Pressure Drop vs. Flow Characteristic – Graph 2



The hysteresis for each shown is 5%. The threshold current is zero as in the TS38-21. As in the TS38-21, although there is a spring opposing the armature, this spring force is balanced against the pressure force acting on the poppet.

The second graph shows the pressure drop characteristic when I<sub>MAX</sub> is applied to the coil. This same curve is also shown in the third graph as the line labeled 100% of max control current.

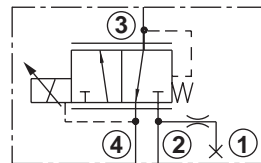
The third graph shows the change in the pressure setting as flow changes when current is held at a fixed level. The pressure rise is similar to that of the TS10-26 because the design of the main stage portion of the valve is nearly identical. At I<sub>MAX</sub>, the minimum pressure setting of the valve is 100 PSI. This level, as in the TS10-26, is due to the spring and orifice in the main stage spool.



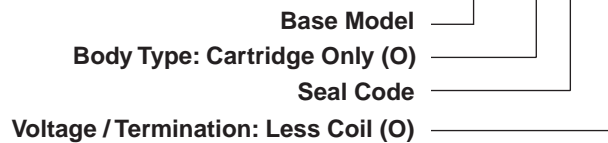
## TS98-30 & TS10-36

The TS98-30 and TS10-36 are the HydraForce standard pilot operated, proportional pressure reducing/relieving valves. There are some differences between each, either in performance or construction. Each section will begin with a description of the TS98-30, and then the differences between the valves will be discussed. The following section will show the cross section of each of the two valves and point out the construction differences between the three. The next section will discuss the typical performance characteristics associated with these valves, followed by a presentation of the forces acting on the valve and how the valves operate.

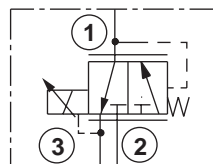
**TS98-30**



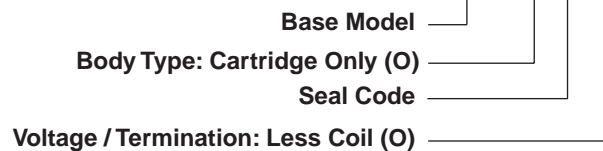
**TS98-30-0-X-00**



**TS10-36**



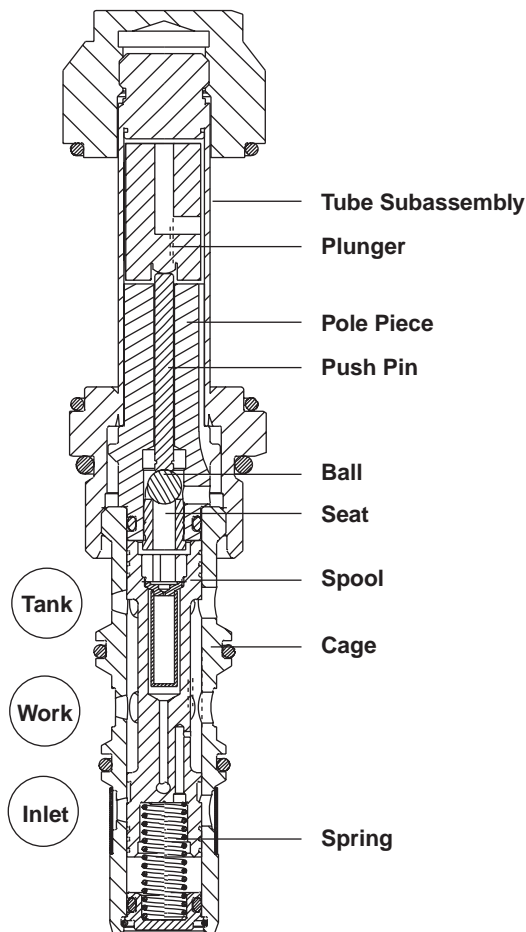
**TS10-36-0-X-00**



### Construction Differences

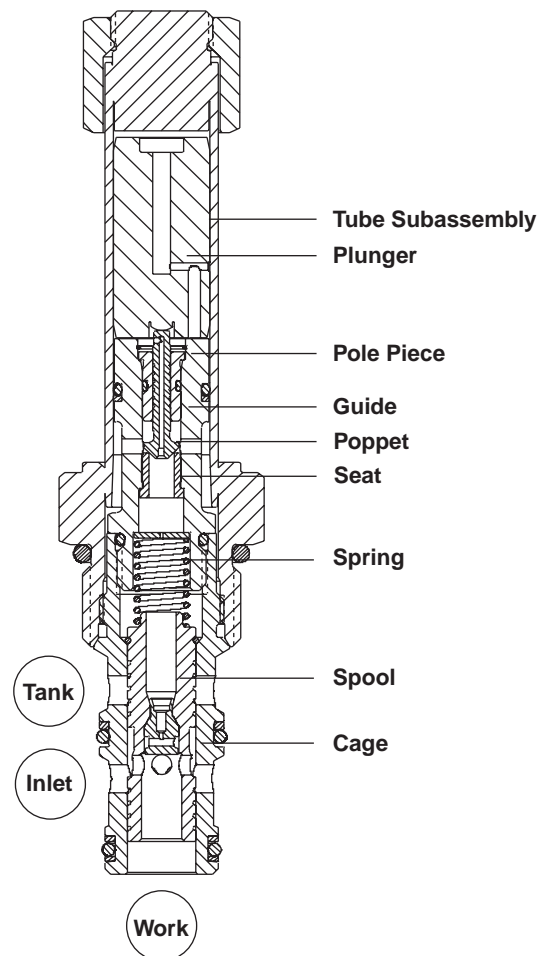
#### TS98-30

This valve uses an 08-size coil and is installed in a 10-size 4-way special cavity. The TS98-30 can reduce to zero psi. It is typically used with inlet pressures up to 500 psi. There are specials available which allow for inlet pressure up to 2000 psi. The tank and work port are connected when no current is applied to the coil. The pilot flow is supplied from inlet port 2. (Note: it is not recommended to connect the inlet at the bottom of the valve. Doing so would restrict the flow.)



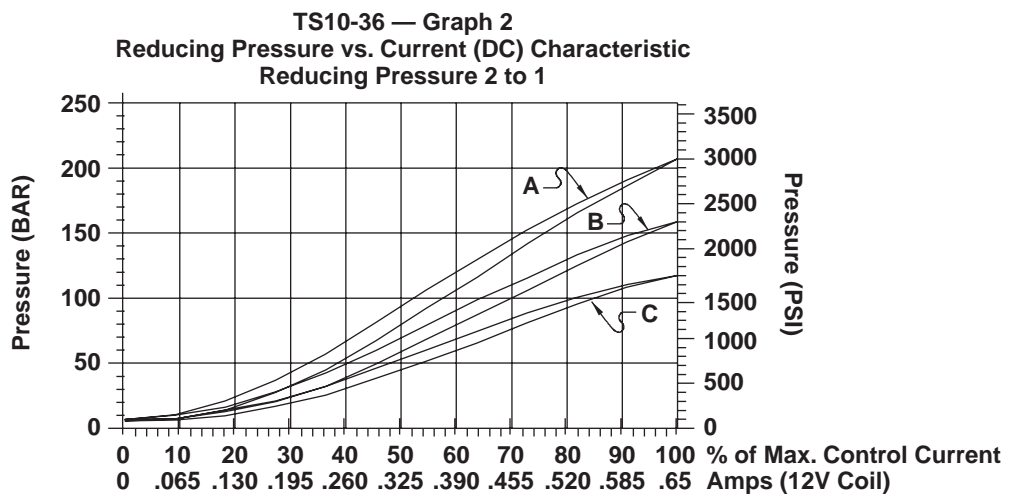
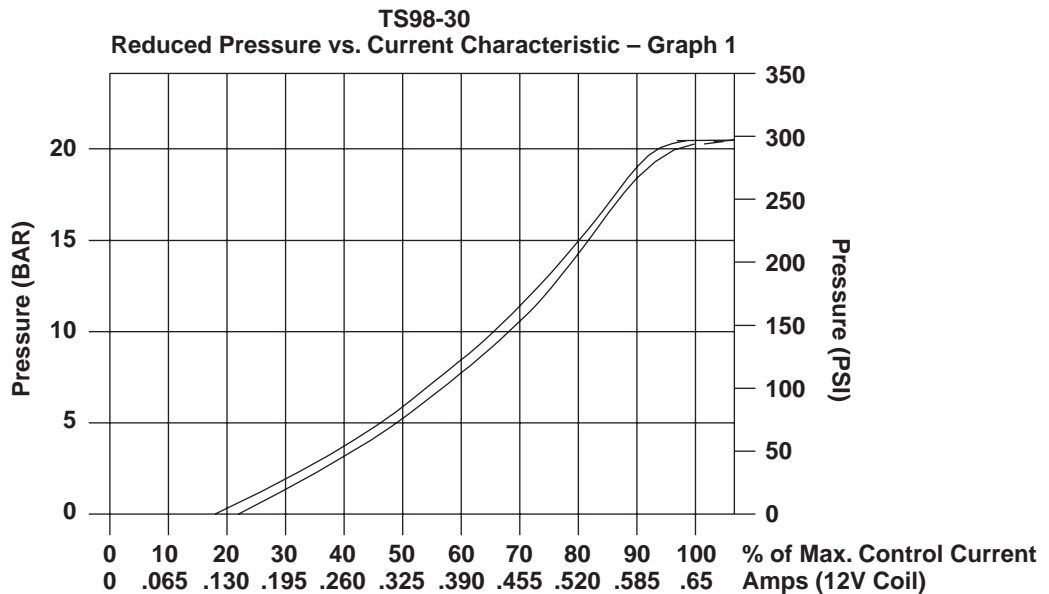
#### TS10-36

This valve uses a 10-size coil and is installed in a 10-size 3-way standard cavity. This valve is not able to reduce to zero psi. This valve can operate with inlet pressures up to 3000 psi. It has three optional pressure ranges. The inlet port is initially connected to the work port. The pilot flow is supplied from work port 1.



### Performance of TS98-30 / TS10-36

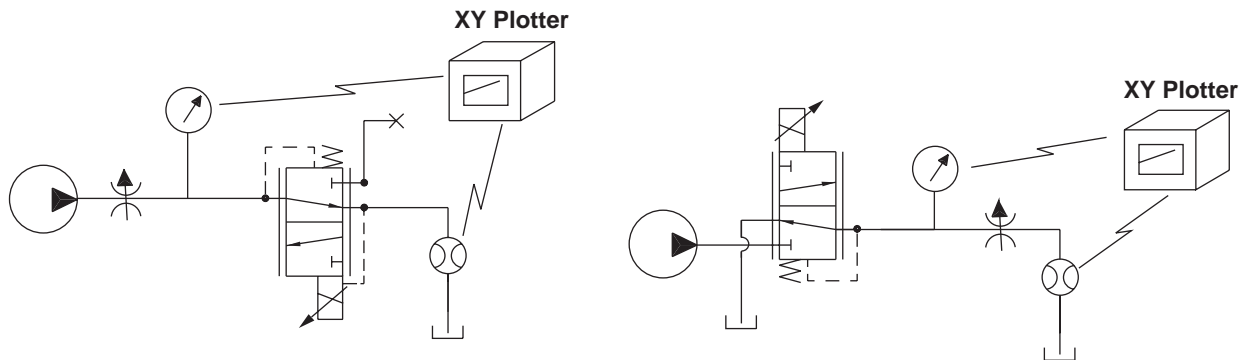
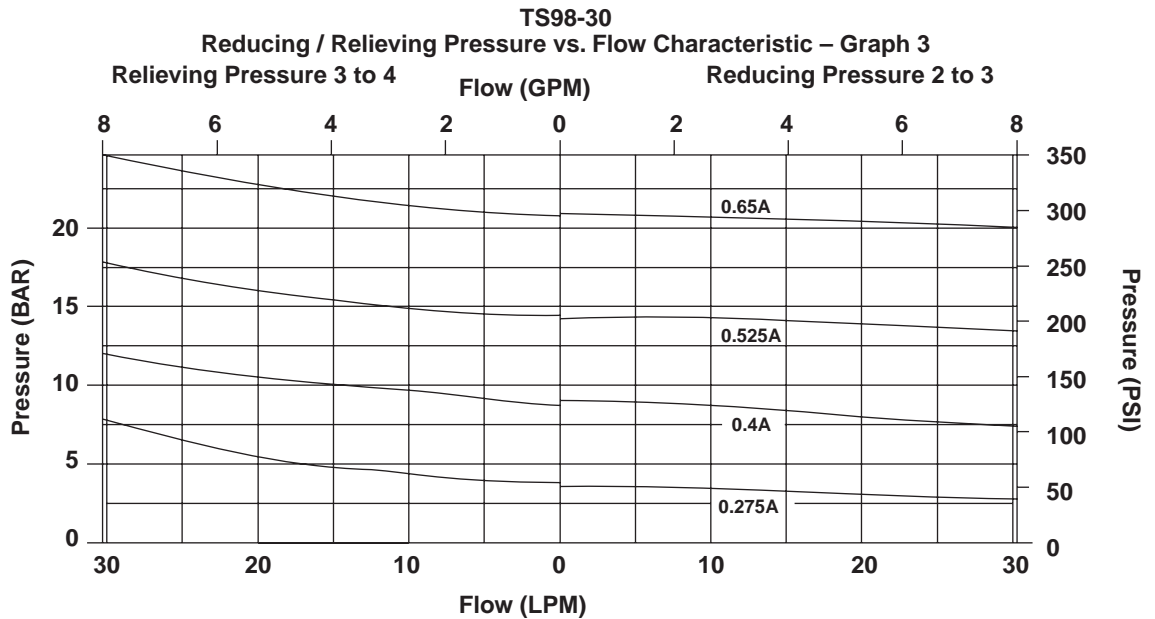
The following sections describe the performance graphs associated with the pilot operated pressure reducing/relieving valves. The first graph shown below is the pressure vs. current map or characteristic for the TS98-30. This graph gives important performance indicators such as pressure range, threshold current and hysteresis. This graph shows that the pressure range is from 0 - 300 psi. The threshold current is 20% of the maximum control current, where the maximum control current for a 12V coil is 650mA. The graph was developed by applying 200 Hz PWM or dither current to the coil. The hysteresis shown is 3%.





The pressure vs. current graph for the TS10-36, shown on the previous page is different from the TS98-30. This graph begins at 100 psi because the inlet and work port are initially connected. A more detailed explanation can be found in the operation section. Also note that there are three different curves indicating three different pressure ranges.

The graph below is the reducing and relieving pressure plotted against the flow, for the TS98-30. Below that is the hydraulic test circuit for both the reducing function and the relieving function.

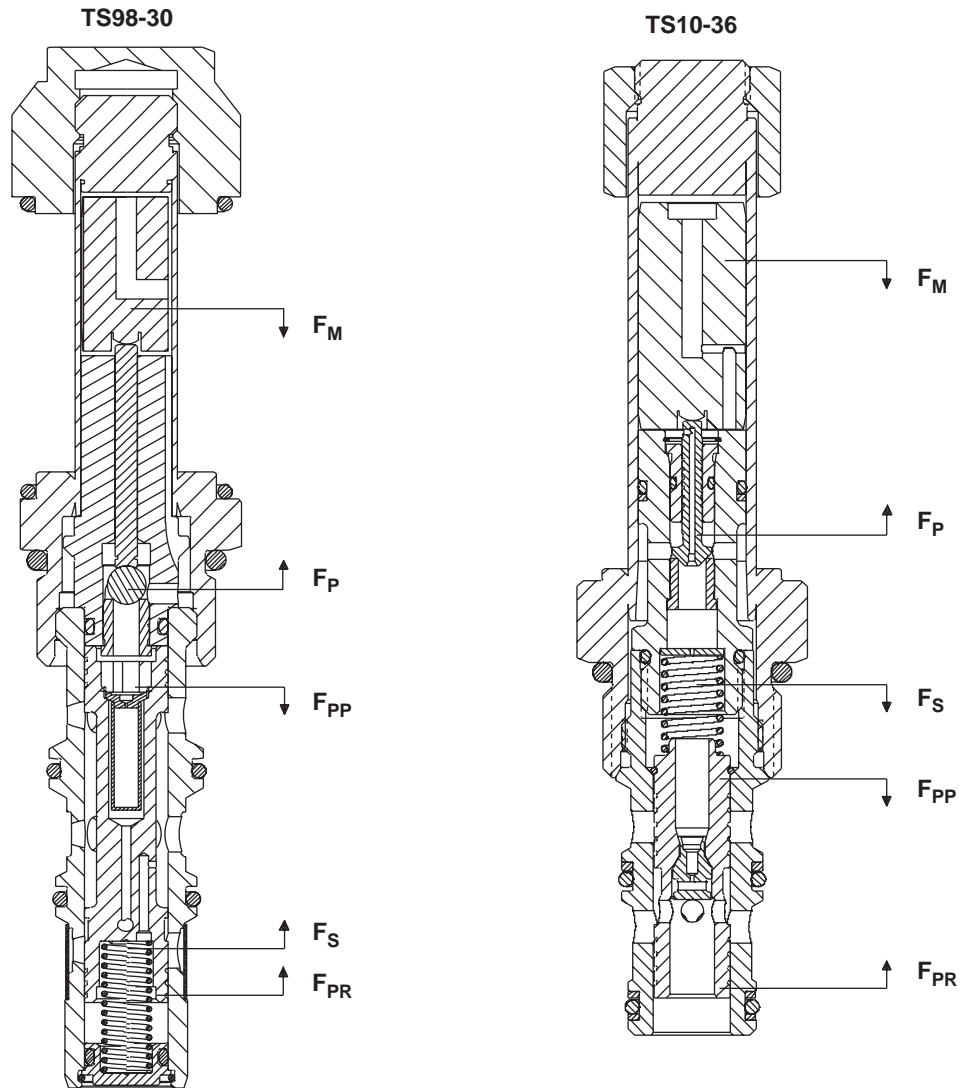


Each test was performed by setting the current applied to the coil, at a given percentage of the  $I_{MAX}$ . The needle valve in either circuit is slowly opened and the corresponding pressure vs. flow characteristic is graphed for that current level.

No chart is shown for the TS10-36. The characteristic is similar to that of the TS98-30, although the pressure levels differ.

### Forces on TS98-30 & TS10-36

The following section will not only detail the forces acting on the TS98-30, and the TS10-36. The cross sections shown below show the dominant forces acting on the components of each valve.

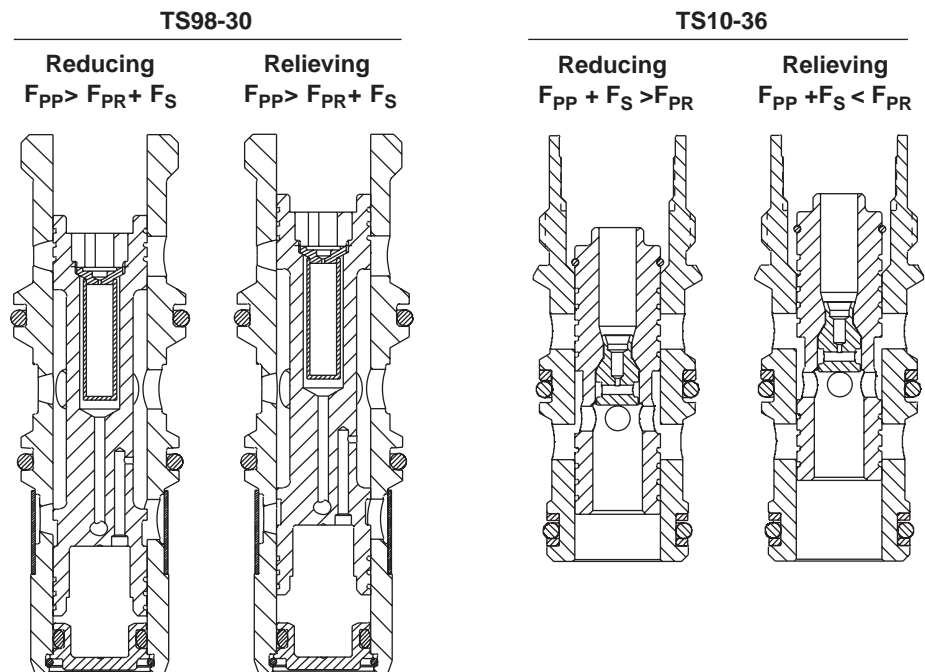


Where:

- $F_M$  = Magnetic force
- $F_P$  = Poppet pressure force
- $F_{PP}$  = Pilot pressure force acting on the spool
- $F_{PR}$  = Reduced pressure force acting on the spool
- $F_S$  = Spring Force

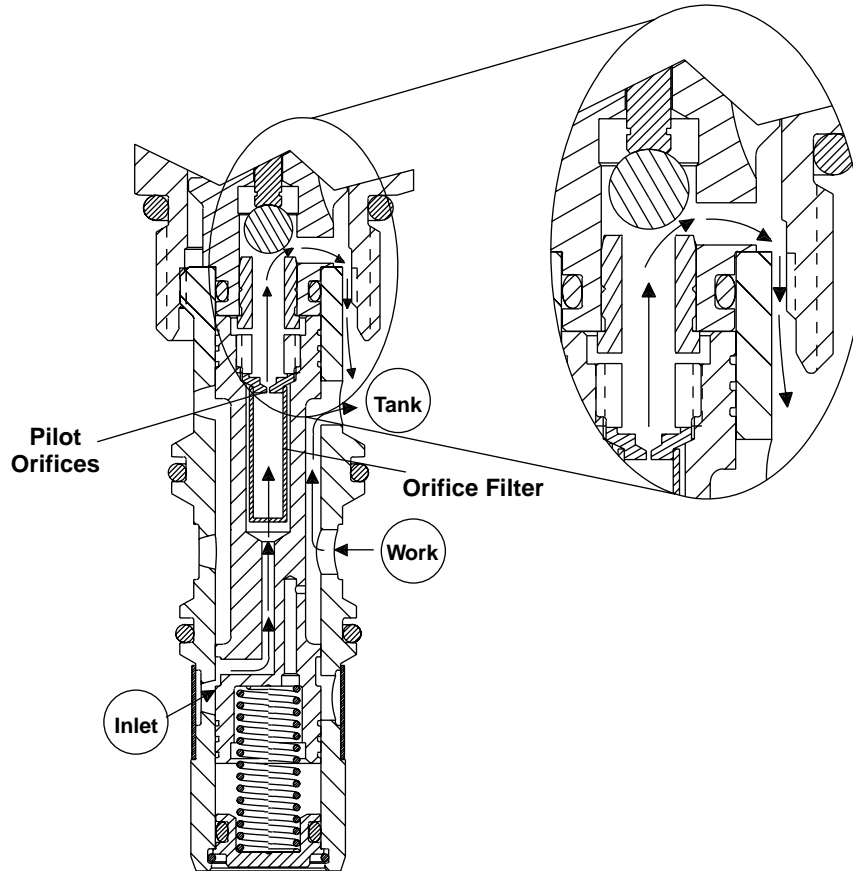
The magnetic force and poppet pressure force are the same as defined in the TS38-20. The spring is the same as the one described in the TS38-21 and can be categorized as a compression spring. In the TS10-36, the spring pushes down on the spool, assisting the pilot pressure (acts in the same direction). However, the spring in the TS98-30 pushes the spool opposing the pilot pressure or acting in the same direction as the pressure reducing force.

The cross section of the spool and cage for each of these pressure reducing/relieving valves is shown below. There are two cross sections shown for each valve. One depicts the spool in the relieving mode and the other in the reducing mode. Also given is the relative level of the forces in order for the spool to be in the given position.



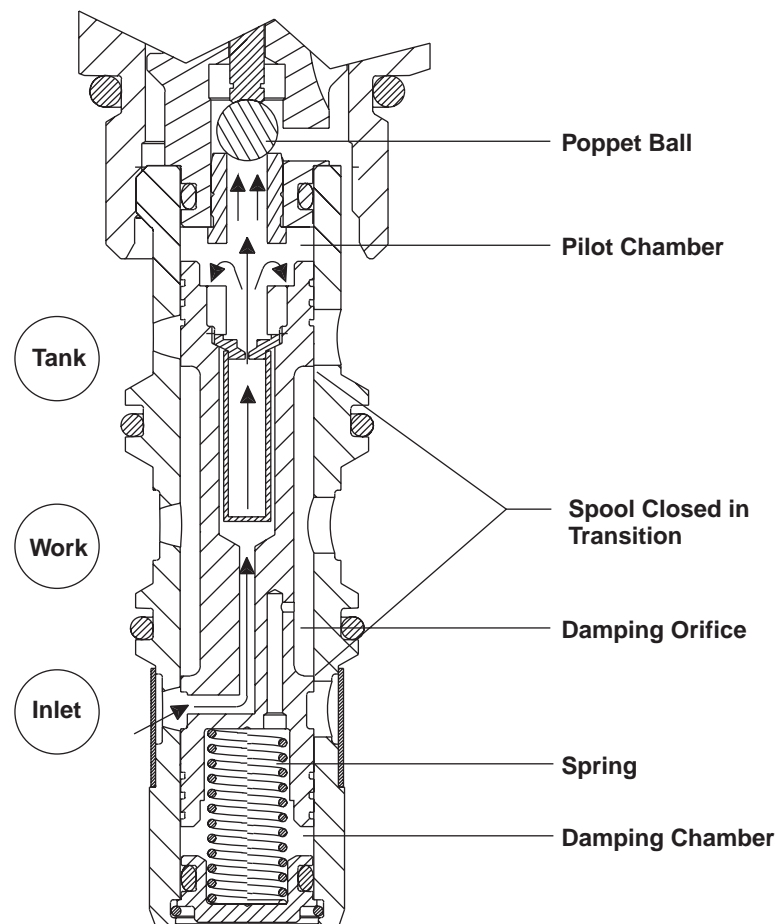
### Operation of the TS98-30

The following is a description of the operation of the TS98-30. This section will begin with a description of the valve when no current is applied to the coil, followed by the spool in a transition position where current is applied to the coil. The third and fourth diagrams show the spool regulating pressure at the work port when the coil is energized.

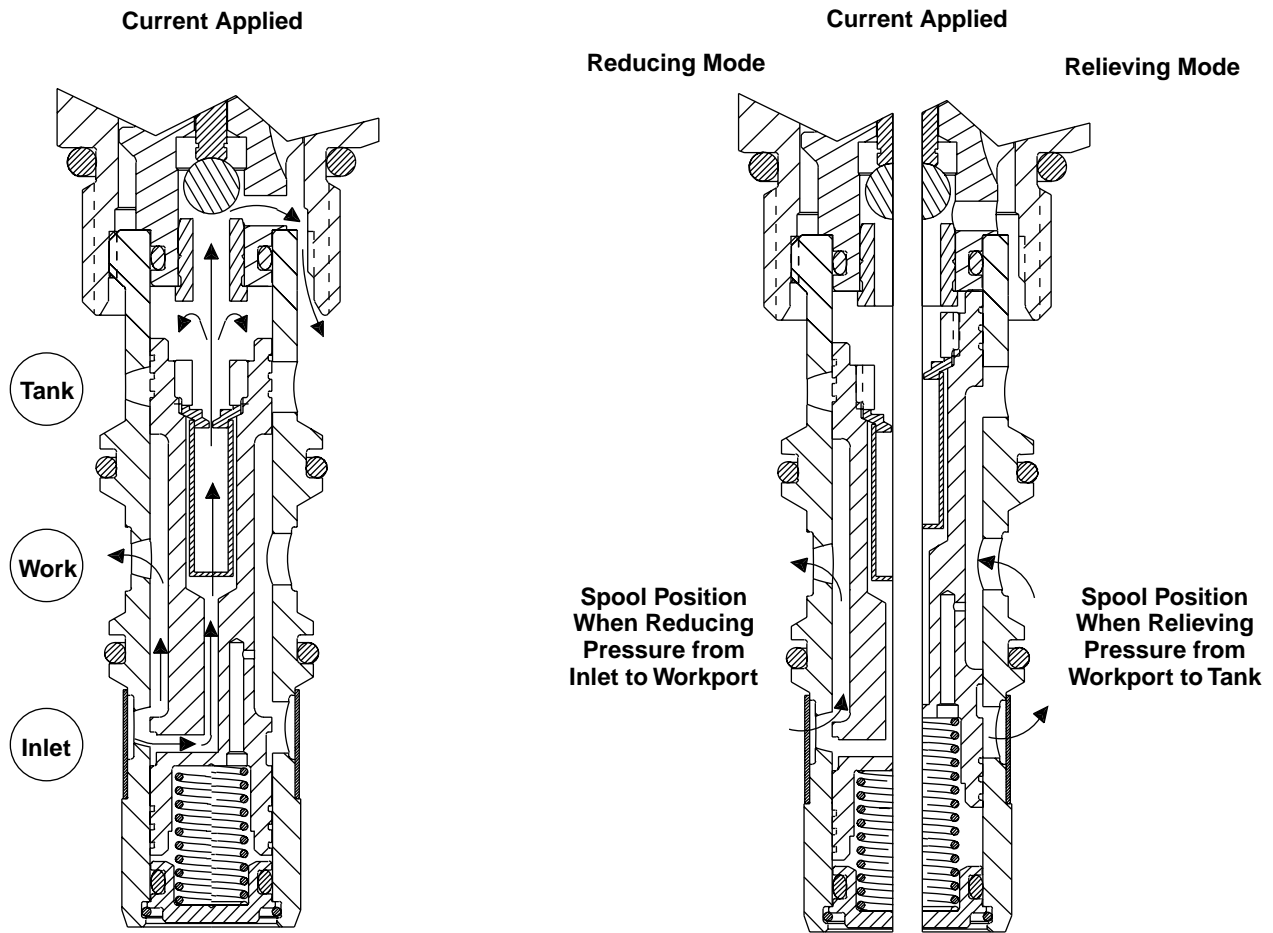


The drawing above shows the lower portion of the TS98-30 when no current is applied to the coil. Oil can flow from the inlet and out to the tank. The oil first passes through a filter on the outside of the cage, then through the center of the spool. It flows through a second filter screen which has holes that are 0.016 inches in diameter. This filter screen, similar to all the standard HydraForce two stage pressure control valves, is used to protect the orifices from contamination. After the oil is filtered and passes through the orifices, it then passes by the ball and out to tank. This flow from the inlet to tank is known as the pilot flow or leakage.

The next drawing shows the ball on its seat. In this case, the coil is energized. Oil does not flow past the ball. Instead, it begins to fill and pressurize the pilot chamber, causing the spool to move down and compresses the spring. In order to overcome the spring, current must be applied. Once current is applied, the ball closes and pressure builds under the ball. Pressure under the ball continues to increase and the spool continues to push on the spring. If sufficient current is applied, the spool will compress the spring to the point where the spool connects the inlet to work port. The amount of current required to build this amount of pressure under the ball is called the **threshold current**. Notice, in this drawing that the work port is isolated from both the tank and inlet. While the TS98-30 is able to control the reduced pressure to zero, it does so at the expense of the maximum control pressure. The TS98-30 must be used with an inlet that is at least 30 psi higher than the maximum reduced pressure. The reason for this is the bias spring holds the spool closed (blocking the inlet and work port) when no current is applied. This type of spool design is known as **closed in transition** or **positive (over)lap**. A closed-in-transition design is typically used on pressure reducing/relieving valves to ensure that the valve remains stable. Another feature which helps the valve remain stable is the damping transition passage and chamber. (See Appendix A for further discussion on damping). Damping of the valve is accomplished by the flow of oil into and out of the damping chamber being controlled by the damping passage.



The remaining two drawings show the inlet is connected to the work port and the poppet is off the seat. However, the coil is powered, which means the magnetic force is regulating the pressure in the pilot chamber, and in turn regulating the position of the spool. Once the pressure in the work port reaches the desired level, the spool will move between the reducing position and the relieving position. This is represented by the split view shown to the right.



### Operation of the TS10-36

The TS10-36 operates similar to the TS98-30. When no current is applied to the coil, the armature pushes down on the poppet. Pressure builds in the pilot section and the spool regulates the work pressure in the main stage accordingly. The significant difference is that the TS10-36 cannot control the reduced pressure to zero. The reason for this is that the bias spring is always pushing down on the spool, causing the work port and tank port to be connected. Because of this, there is always some pressure applied to the work port. The force exerted by the bias spring induces a pressure of 100 psi at the work port.

## Summary

In this chapter the following concepts were presented:

- The pressure ranges available for the TS10-26, TS10-27 and TS10-36.
- The differences between the two standard pressure reducing/relieving valves available from HydraForce.
- What the typical hysteresis is for the various pilot-operated pressure controls.
- What makes up the pilot stage of a two-stage pressure control.
- What makes up the main stage of a two-stage pressure control.
- The difference between the TS10-26 and the TS10-27.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. Describe the operation of the TS10-27. \_\_\_\_\_
2. What is the pilot section? \_\_\_\_\_
3. What is the main stage section? \_\_\_\_\_
4. True or False. The TS10-26 uses a linear proportional actuator. \_\_\_\_\_
5. Name three standard proportional pressure reducing/relieving valves available from HydraForce. \_\_\_\_\_
6. True or False. The hysteresis of the TS10-27 increases when PWM is used. \_\_\_\_\_
7. What causes the threshold to be required on the TS98-30? \_\_\_\_\_
8. How many pressure ranges are available with the TS10-36? What are they? (i.e. minimum and maximum pressure) \_\_\_\_\_
9. True or False. The TS98-30 has a damping chamber. If true, why. \_\_\_\_\_
10. Why are pressure reducing valves typically closed in transition? \_\_\_\_\_



# Chapter 6: Application of Pressure Controls

## Objectives

The objectives for this chapter are as follows:

- Learn about the proportional control of a clutch and the advantages of using the TS98-30.
- Discover how to control the position of a cylinder using a pressure control.
- Learn about a fan drive circuit.
- Understand how an EHPR08-33 can be used to pilot a larger spool.

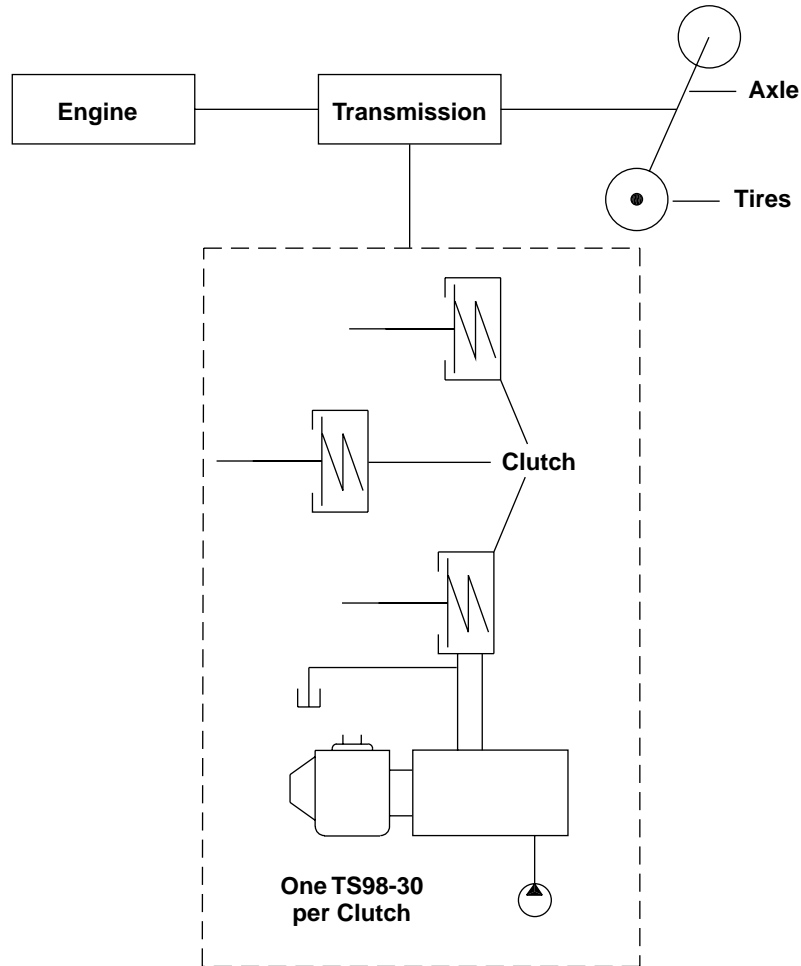
## Introduction

In this chapter, we will briefly look at four applications of the proportional pressure controls described in the previous chapters. The first one is control of the pressure applied to the clutch. The second is one possible use of the TS10-27 in control of a fan drive. The third is piloting of a spool. Finally, the fourth is position control of a cylinder.

# HYDRAFORCE

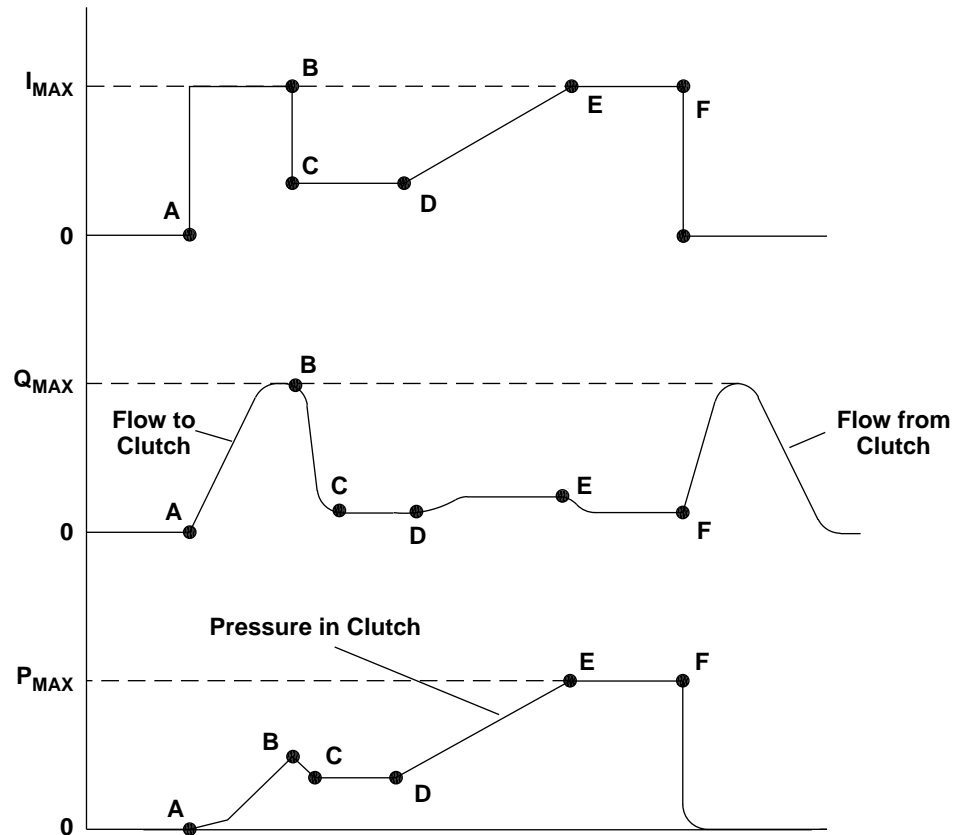
## TS98-30 Clutch Application

The following section describes the use of a TS98-30 in a clutch application. A clutch is a device which engages and disengages sets of gears inside the transmission to transfer the engine power to the drive shaft. Inside the transmission are several clutches for different speeds, as well as one to select forward and reverse.



The clutch is actuated in much the same way as a hydraulic cylinder. A spring holds the piston part of the clutch in the neutral position when the clutch is not engaged. When current is applied to the TS98-30, it allows pressure to be applied to the piston of the clutch. When this happens, the clutch is engaged. The clutch is said to be engaged when it is transferring engine power to the drive shaft.

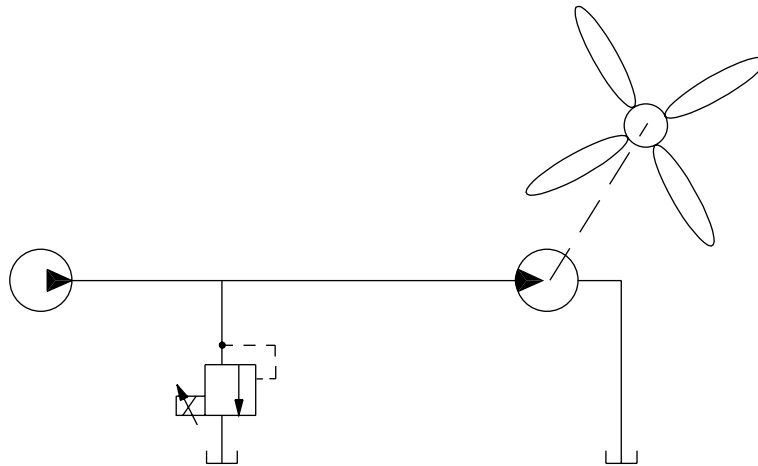
There are advantages to using the TS98-30 to regulate the pressure applied to the clutch. This is best described in the graphs below.



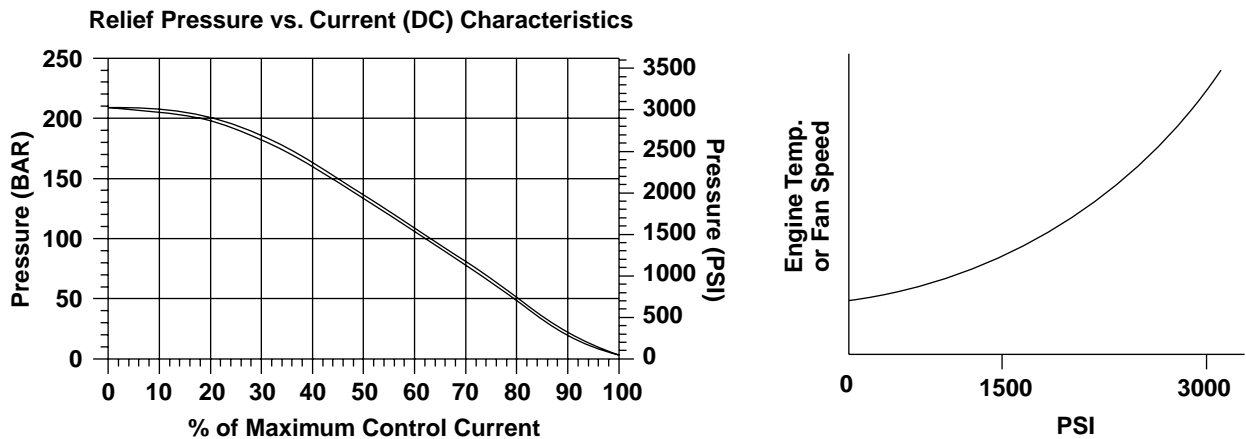
The first advantage, (shown by points A-B on the graph), is the ability of the TS98-30 to allow for a fast pre-fill of the clutch. The clutch prefill is the action where oil is quickly fed into the clutch and the pressure is allowed to reach the point where the piston just begins to compress the spring inside the clutch. Notice that during this prefill action, the flow is relatively high. However, this only occurs over a short time. Typically the flow may reach as high as 8 gpm for about 0.1 or 0.2 seconds. The current is then dropped down to point C. From point C to D the clutch is waiting to be engaged. The current and pressure is ramped from point D to E. A second advantage of the TS98-30 is that it allows for this ramping of the pressure to slowly move the piston. The slow movement of the piston gives a smooth, soft engagement of the clutch. This is beneficial because power from the engine is gradually applied to the drive shaft, allowing the vehicle to accelerate smoothly rather than jumping or jerking. When the drive shaft no longer needs to be connected to the engine (driver stops), the current to the valve is turned off. The flow discharges from the clutch immediately and pressure inside the clutch drops.

Note the graph shows that the valve must be capable of two extreme modes of operation. The first is that it must be able to create a very large opening to allow high flow to occur at low pressure. Secondly, when current is controlling the valve during the ramping mode, only a small amount of fluid needs to pass through the valve. At this time, the opening created by the valve is small. These two modes of operation are difficult to achieve, but have been accomplished in the TS98-30. There are applications where the valve may become unstable (pressure pulses will show up on the graph). The driver may feel the clutch engaging and disengaging slightly. This may be due to differences in the clutch design or air entrapped in the system, among other factors. If this occurs, consult HydraForce.

**TS10-27 and Fan Control**



A growing application in hydraulics is that of a variable speed, hydraulically driven fan used to cool fluid passing through a radiator. A pressure control like the TS10-27 or a variation of it such as the TS38-21 or TS12-27, depending on the flow, can be used. As a reminder, the pressure controlled by these valves decreases as the current increases. The reason to use such a valve is in case the solenoid or electronics controlling the solenoid fail. If such a failure occurs, the system pressure will be at the high setting of the valve, which is maximum fan speed. Recall the performance graph showing the pressure vs. current characteristic of the TS10-27.

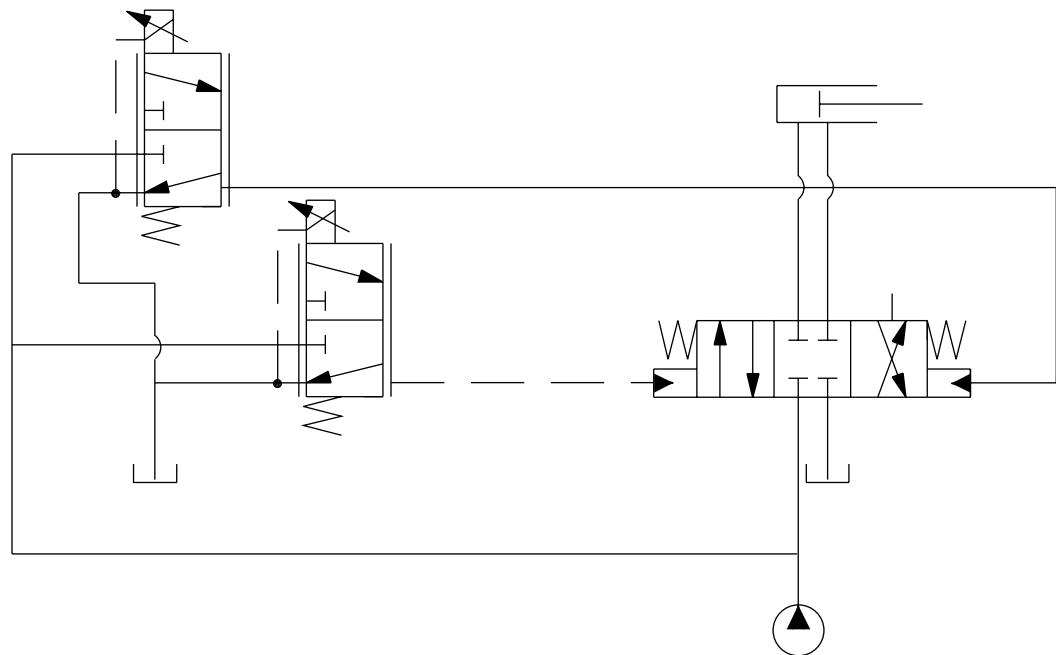


A variable speed hydraulic fan can be used to reduce noise and to improve fuel efficiency. The fuel efficiency is improved because the motor is not always running at high speed or full load. Instead, when the engine temperature is low, the current in the TS10-27 is high and the pressure and motor speed are low. Since the pressure is low, the demand on the hydraulic pump and therefore the engine, is low. The decrease in noise is basically due to the same reason. With the fan not running continuously at high speed, the noise heard by the operator is reduced.

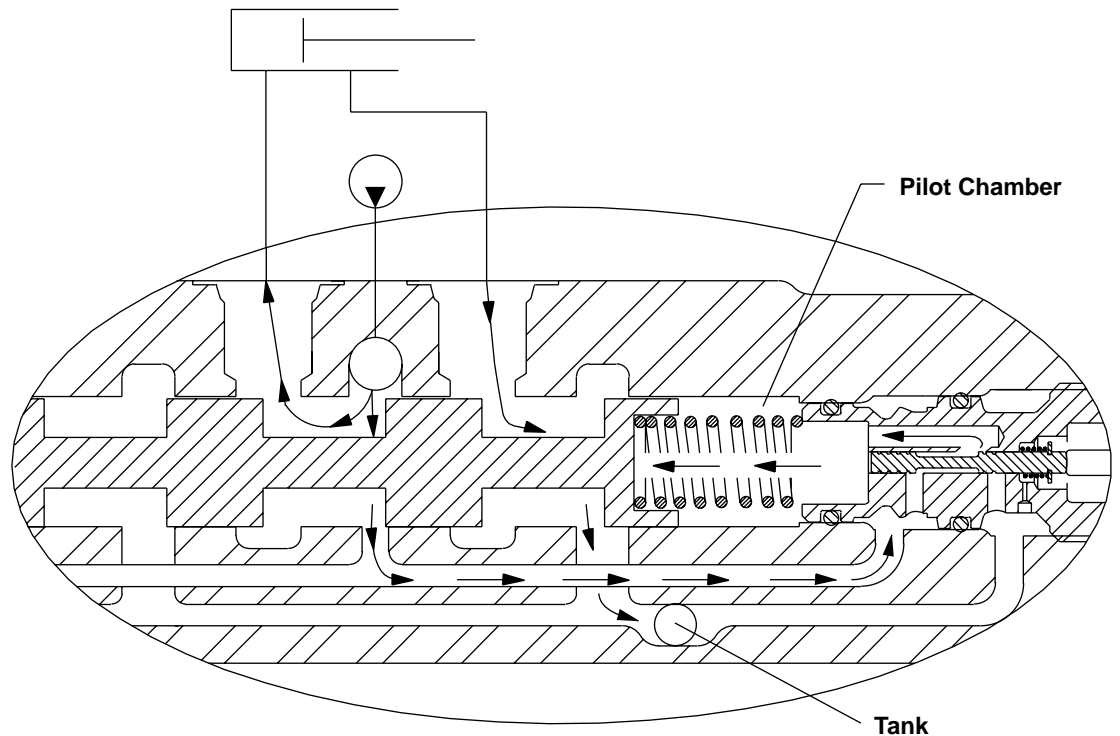
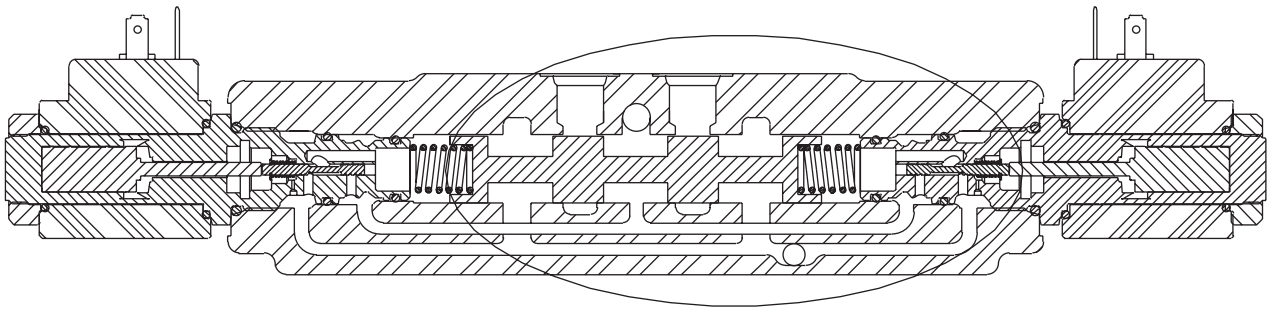
### EHPR08-33 Directional Control Valve

Large construction equipment such as excavators use directional control valves, which are known as stack valves. Multiple valves are stacked together to form a hydraulic circuit rather than screwing valves into a manifold. These valves are typically larger than cartridge valves and handle larger flows. Because of the size of the valves, they are actuated hydraulic pressure, rather than a solenoid.

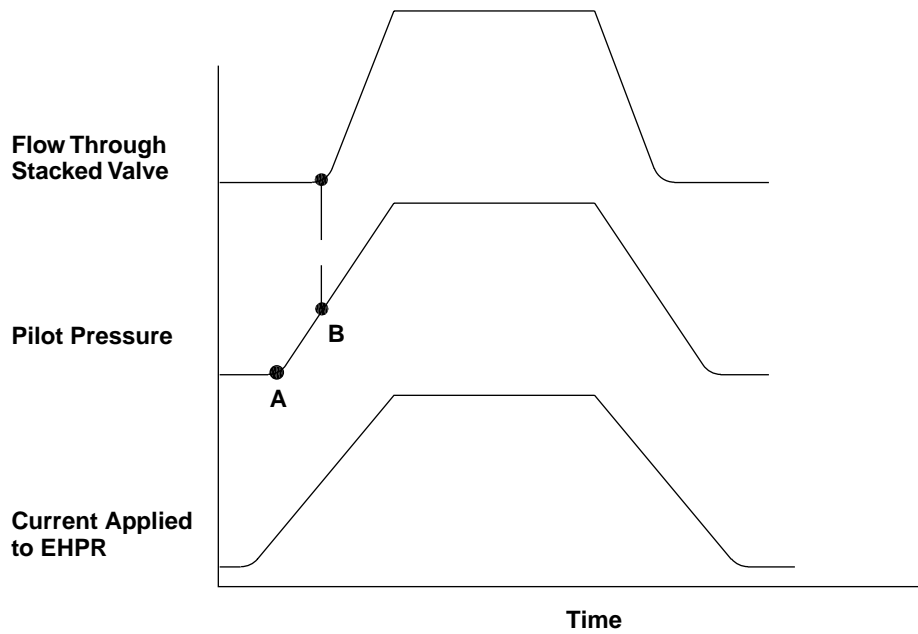
Shown below is the schematic of two EHPR08-33 valves and the directional control of the stack valve. On the following page is a cross section of a stack valve which includes the two EHPR08-33 valves, the spool of the stack valve and springs which center the spool.



The first cross section below shows the spool centered by the two springs. In this condition, no current is applied to the EHPRs. The enlarged sectional view shows the large spool shifted in one direction, allowing oil to flow to and from the actuator. In order for this to occur, current is applied to one of the EHPRs. The reason to use a proportional valve instead of an on/off solenoid is to slowly shift the directional valve, thus removing the shocks from the hydraulic system.



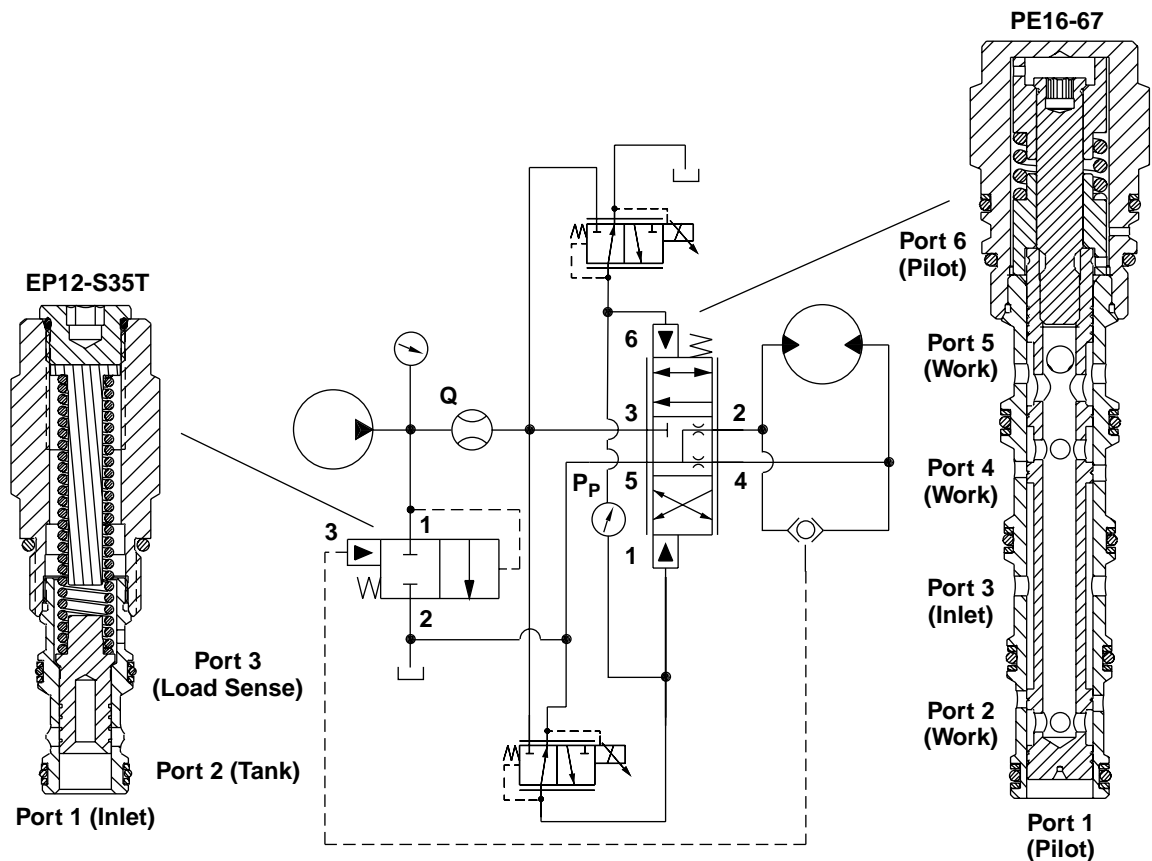
The EHPR08-33 valves are used to modulate the pressure applied to either end of the spool. When current is applied to the EHPR, pressure feeds into one of the pilot chambers of the large spool. Pressure continues to build until the pressure force exceeds the force of the spring in the opposite pilot chamber (shown as points A-B on the graph below). At point C, oil begins to flow through the spool of the stack valve. The reverse occurs as the pilot pressure from the EHPR increases, so does the flow.

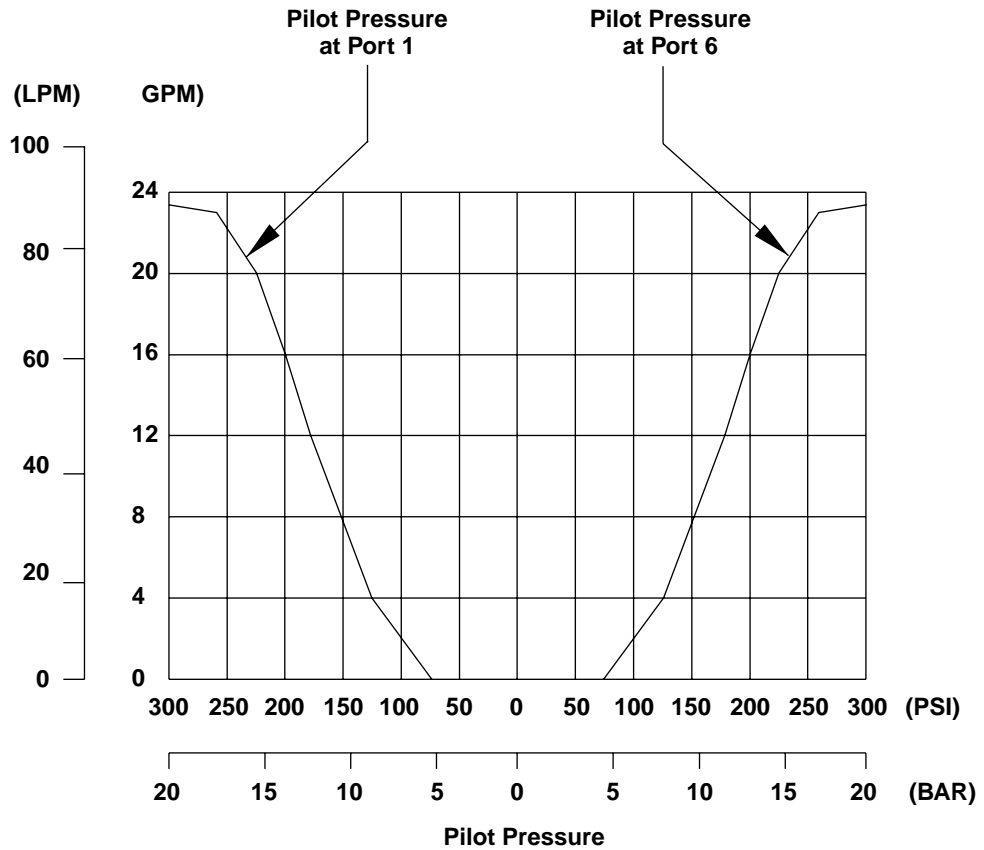




**EHPR08-33 & PE16-67**

Another type of directional valve which can be piloted using the EHPR08-33 is a PE16-67D-0-N. This is a 6 ported, piloted directional control which proportionally controls flow from the inlet to the cylinder ports and back to tank. The reason to use this directional control is due to its ease of integration into a cartridge manifold system. The schematic below shows this valve, along with two EHPR08-33s and an EP12-S35T (in-line pressure compensated flow control). The cross sections of these two valves are also shown below.

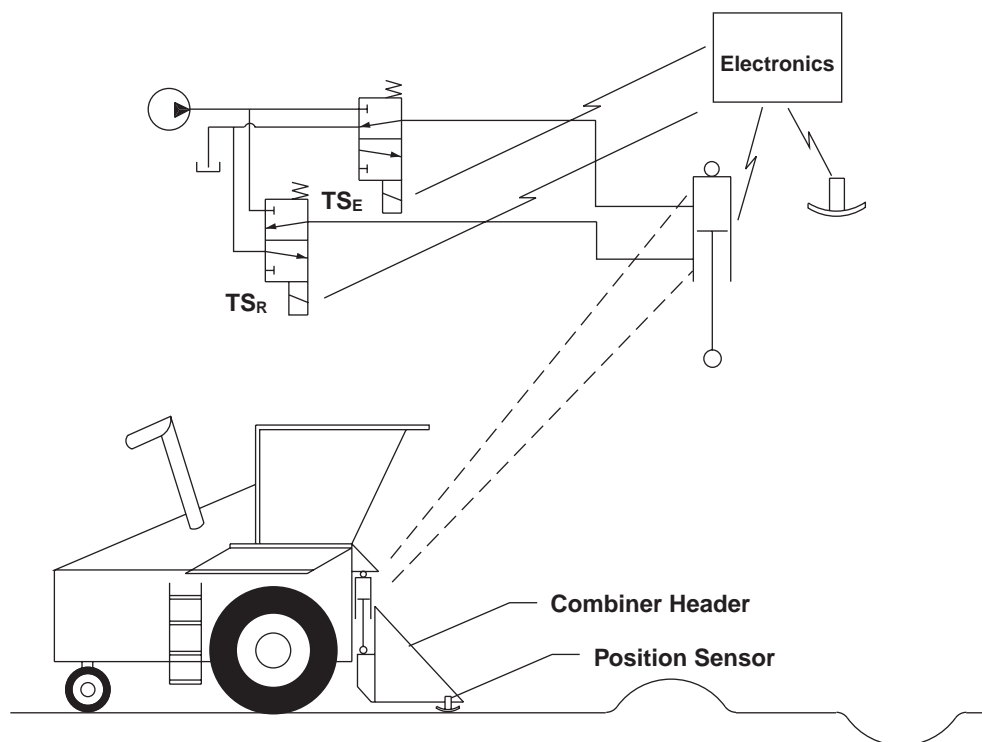




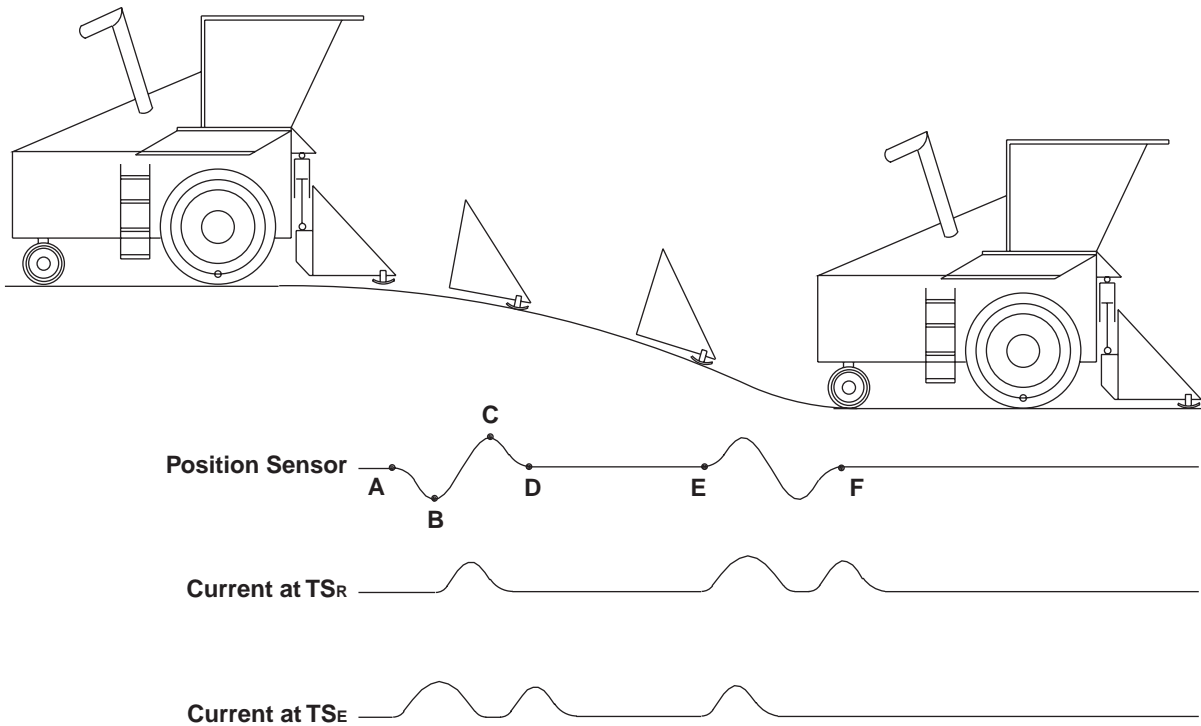
The graph above shows the typical flow vs. pilot pressure characteristic of the PE16-67. This graph was developed using the EP12-S35T. However, the PE16-67 can be used without a compensator. While the valve will remain stable without the compensator, the flow may change as a result of load pressure.

## TS10-36 & Cylinder Control

The diagram below is a schematic for the positioning of a hydraulic cylinder. The cylinder is held in a fixed position by pressure controlled by the two TS10-36 valves. Notice that because the cylinder is a double acting single rod cylinder, the piston area exposed to pressure is larger on the blind side, compared to the rod side. In order to hold the cylinder in place, the pressure supplied by the TS10-36 on the rod side (noted as  $TS_R$ ) must be higher than the blind side. Once the cylinder is in a fixed position, if for example, it needs to be retracted, the current applied to the  $TS_R$  must be increased. The pressure increase is maintained until the new position of the cylinder is reached.



One application for this circuit is to control the height of a tractor implement or a combine header (shown above). The combine header is the part of the combine that pulls the cotton from the cotton plant during harvesting. During harvesting, the farmer may need to continually adjust the height of the header in order to maintain the optimal position. If there is a mound of dirt, he needs to raise the header, or damage to it could result. If there is a dip in the ground level, the farmer may not strip all cotton possible from the plants. To take advantage of the functionality of the combination of the two TS valves above, a position sensor must be mounted to the front of the header. By measuring the change in the distance of this sensor, relative to the ground, the optimal position of the header can be maintained. This is further clarified in the following graph.



- A-B) The position sensor is extending because the combine is going down a hill. Current to the TS<sub>E</sub> is increasing so that the header will stay close to the ground.
- B) At this point, the current applied to TS<sub>E</sub> is reduced back to its steady state level. This occurs because the pressure caused the cylinder to extend and at the same time, the rest of the combine has begun to go down the hill.
- B-C) The header and position sensor overshoots the desired position. The current applied to the TS<sub>R</sub> increases to move the header back to the level position.
- C-D) The movement of the header is too much. The current applied to TS<sub>E</sub> increases slightly to correct for the overshoot.
- D-E) The desired level position of the header is reached.
- E-F) The position of the header goes through similar correction as when the combine began to go down the hill.

## Summary

In this chapter the following concepts were presented:

- The function of a clutch and how the TS98-30 can be used to control the engagement of the clutch.
- The use of a TS10-27 in a fan control circuit.
- A stack valve and how an EHPR08-33 can be used to control flow through a larger spool valve.
- A proportional circuit that could be used to control the position of a cylinder.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. What is one reason to use a proportional pressure control to regulate pressure applied to the clutch? \_\_\_\_\_
  
2. Why is the TS10-27 used to control pressure in a fan drive circuit instead of a TS10-26? \_\_\_\_\_
  
3. Where is an EHPR typically used? \_\_\_\_\_
  
4. Describe how two pressure reducing valves could be used to control cylinder movement. \_\_\_\_\_

# Chapter 7: Flow Controls – Introductory Concepts

## Objectives

- Learn the definition of a fixed and variable orifice.
- Learn the basics of a compensated and restrictive flow control.
- Understand the operation of a priority/bypass flow control.
- Learn about pressure compensation characteristics.

## Introduction

This chapter is an introductory chapter to proportional flow controls. It will introduce basic, non-electrical flow controls. This will act as a foundation in understanding the terminology associated with the construction and operation of the proportional flow valve. In addition, terminology to describe the performance will also be introduced. Other terms that are specific to proportional valves such as hysteresis, saturation, control range etc. can be found in Chapter 3.

# HYDRAFORCE

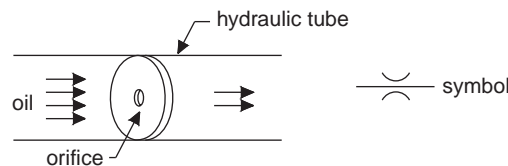
## Flow Control – Introductory Concepts

A flow control valve is one which specifically regulates the volume of oil passing through a hydraulic system.

The following sections describe two types of flow controls, non-pressure-compensated and pressure-compensated. The first two flow controls described are the fixed orifice and the variable orifice, or a needle valve. The third and fourth flow controls presented are pressure-compensated flow controls. All four are described in the following sections to give a basic understanding of how the proportional flow controls operate.

### Fixed Orifice

A flow control valve is one that is used to meter the quantity of fluid passing through a hydraulic system. The simplest type of flow control, a fixed orifice, is shown below.



Since the hole or orifice is smaller in diameter than the tube, the oil flowing through the tube, downstream of the orifice, is restricted compared to the flow upstream.

The amount of oil flowing through the orifice is dependent on the area of the orifice, the pressure drop across the orifice and the viscosity or thickness of the oil. The relationship below can be used to determine the flow through an orifice.

$$\text{Flow} = 31 \times \text{Orifice Area} \times \sqrt{\text{Pressure drop}}$$

The equation above is a simplified version of one used in engineering fluid dynamics. The equation is based on one for an orifice as shown above, where the edges of the hole are sharp. The simplifications or assumptions that are built into this equation are:

- oil viscosity: 32 cSt
- the pressure is measured in psi
- the flow is measured in gpm
- the area is measured in inches squared
- the length of the orifice does not exceed the diameter

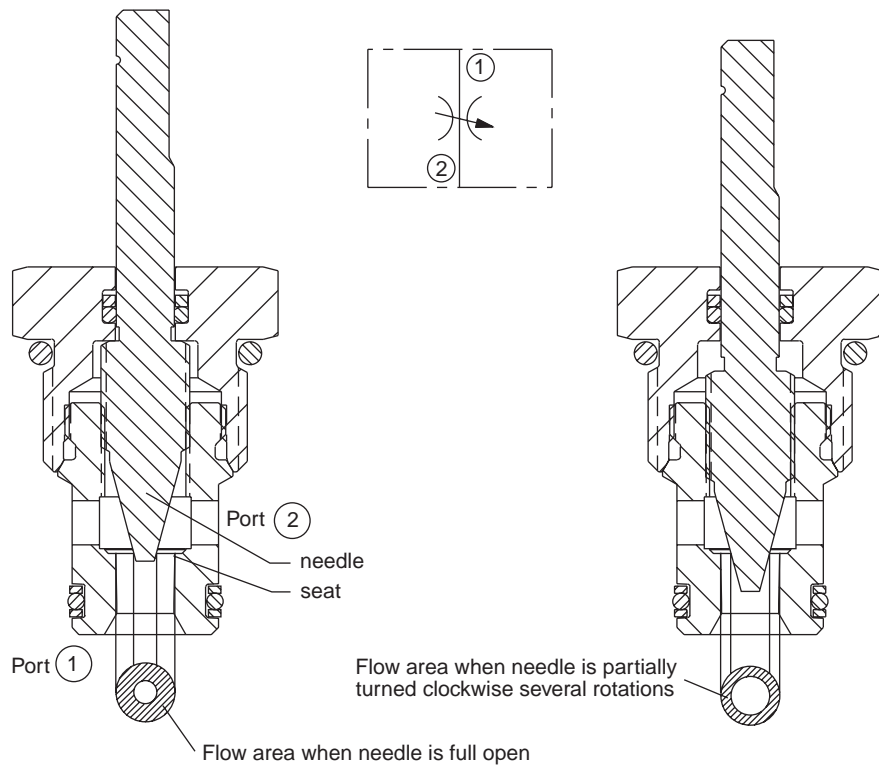
A centistoke or cSt is simply a measure of the oil viscosity as an inch is a measure of distance. The viscosity represents how thick the oil is. 32cSt was selected because this is the typical viscosity at which valves are tested when performance data is presented in a catalog.

The number 31 in the equation above takes into account the viscosity and the adjustment for non-sharp orifice.

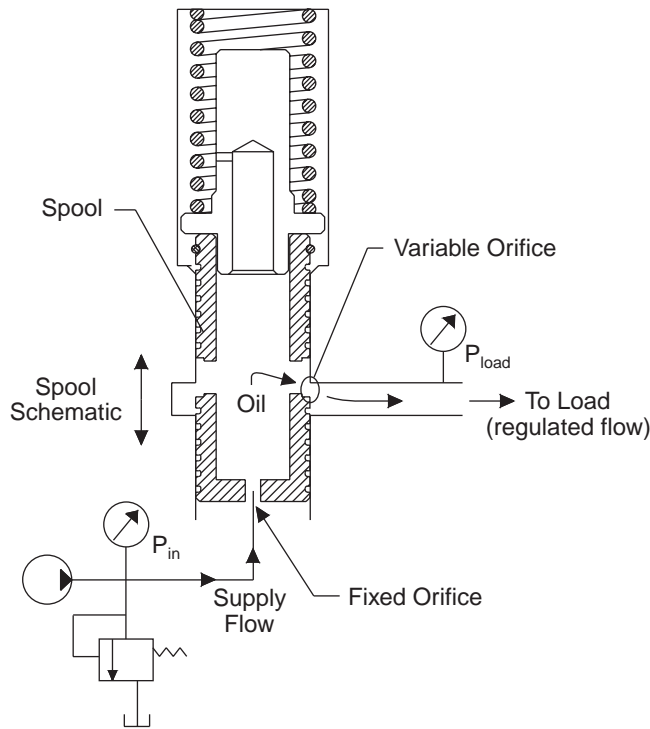


### Variable Orifice

A needle valve is similar to a fixed orifice in that it meters flow through a set area defined by the needle and seat. The difference is that the needle can move up or down to increase or decrease the size of the orifice. This is why it is known as a variable orifice. The needle valve can open or close the same way a faucet does, allowing more or less flow, depending on the open area. The flow through the needle valve can be determined by using the equation from the previous page.

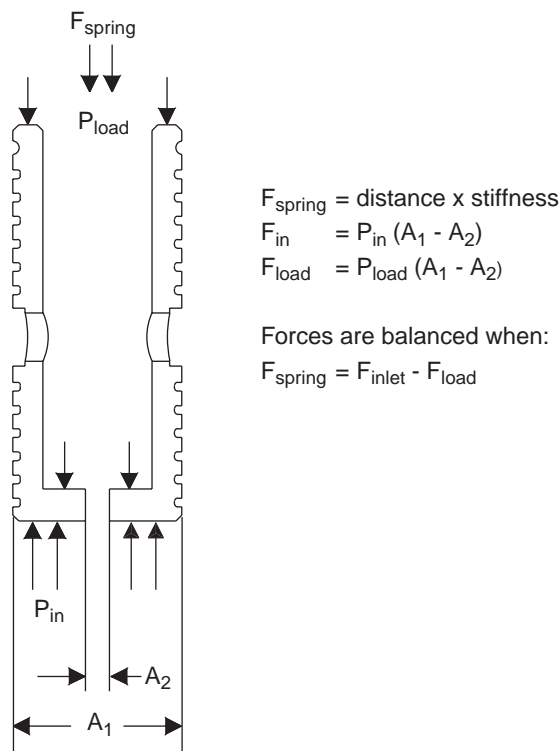


While both the fixed and variable orifices control flow, there is one drawback to using them. If the pressure across the flow area changes, so does the flow. This occurs when the pressure in a hydraulic system changes as the load changes. For example, there is a difference in the hydraulic pressure required to lift an empty excavator bucket versus a full one.



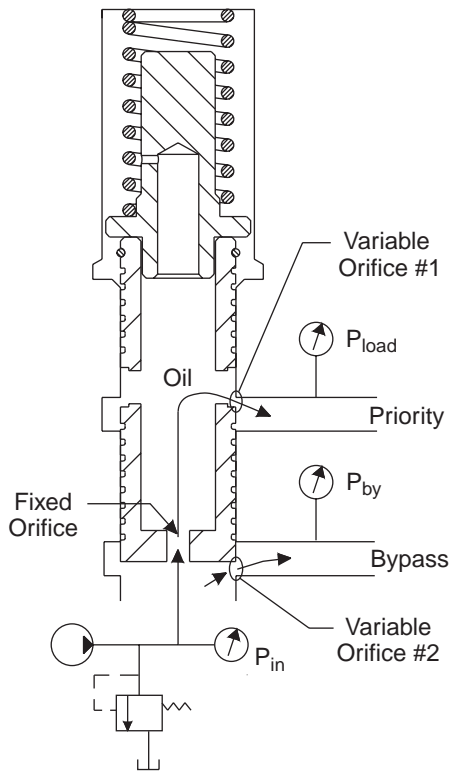
### Restrictive Compensated Flow Control

The previous section described a fixed orifice flow control where the flow varied when the pressure varied. In order to eliminate or at least minimize how pressure affects the flow, a valve known as a pressure compensated flow control can be used. A simple representation of a pressure compensated flow control is shown to the left. This particular type of flow control is known as a restrictive flow control because, typically, it restricts the downstream flow to a value lower than the supply or pump flow. Notice that this valve is made up of a fixed orifice and a variable orifice. The variable orifice is created by the spool moving up and down. As the spool moves up or down the flow area of the variable increases or decreases. This type of device is typically used to control flow into or out of a cylinder, thus controlling the speed of the cylinder extending or retracting.



The valve works by maintaining a constant difference between the inlet pressure  $P_{in}$  and the pressure at the load,  $P_{load}$ . This is accomplished by the fixed orifice, variable orifice, and the spring working together. First the fixed orifice regulates the flow as mentioned in the previous section. That is, the flow across this orifice is still dependent on the area, pressure drop and viscosity. The flow across the variable orifice is also dependent on the pressure difference across the fixed orifice. When the pressure difference across the fixed orifice increases, the force acting on the spool increases. The equation and diagram to the left describe this force.

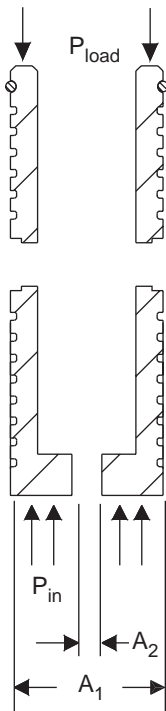
Initially, oil flows across the fixed orifice and the position of the spool remains unchanged. This is true as long as the pressure force ( $P_{in}$ ) is less than the spring force. As the pressure force increases the spool begins to push against the spring. The spool will continue to move until the spring force equals the pressure force. (Note the spring force is based on the amount the spring is compressed, multiplied by the stiffness of the spring, or the rate.)



### Priority / Bypass Compensated Flow Control

Another type of pressure compensated flow control is known as a priority/bypass flow control valve. It is known as a priority/bypass flow regulator because flow is regulated at the priority port like the previous valve and the excess is diverted (bypassed) to another portion of the circuit. As with the first type of flow control, it is also made up of a fixed and a variable orifice. These two orifices make up the priority portion of the flow control. A third orifice which is also variable, is the bypass portion of the valve. This type of valve is shown to the right.

This flow control has two advantages over the fixed orifice. First, it maintains a constant flow rate regardless of the pressure. Second, the excess system flow can be used in a secondary hydraulic circuit or it can be dumped to tank through the bypass line, thus reducing the amount of wasted energy.

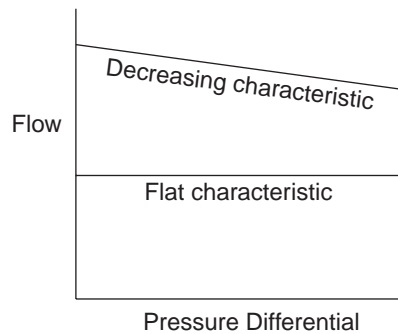


As with the restrictive type flow control, the spool position remains unchanged until the pressure force exceeds the spring force. The spool will then move in response to the change in pressure at the priority port. When there is a small difference between the inlet pressure  $P_{in}$  or  $P_{by}$  and  $P_{load}$ , the variable orifice #2 is opened less compared to variable orifice #1. Conversely, when the inlet pressure and the bypass pressure begin to increase compared to the priority pressure, the variable orifice #1 begins to decrease. Yet, in either case, the flow rate at the priority port is maintained at a constant level. Further, this implies the flow at the bypass port is also compensated. That is, if the pump flow and the flow at the priority port remain fixed, the bypass flow will remain unchanged regardless of changes in pressure. Again, the spool will move when the pressure force exceeds the spring force. When this occurs, flow at the priority port will be restricted until the force due to  $P_{load}$  plus the spring force equals the force due to the inlet pressure. In other words, the spool position is in equilibrium (held in place) when  $F_{spring} = F_{inlet} - F_{load}$ .

For a more in-depth review of force balance equations, refer to the HydraForce Solenoid Course Manual.

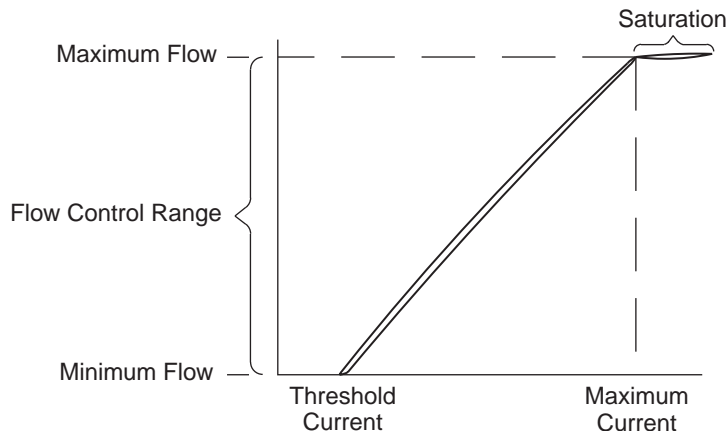
### Compensation Characteristic Curves

The performance characteristic that represents the ability of the flow control to maintain a desired flow value, is the compensation characteristic. As the curves show, there is no set rule-of-thumb regarding a general trend for a pressure compensation characteristic. The characteristic can be flat or varying, depending on the differential pressure between the inlet and the work port of the flow control. It is also dependent on the spring force valve design in general, and the geometry of the valve components. Typically, when the flow is at the maximum operating level, the characteristic is a decreasing trend. This decreasing trend is known as *droop*. In this case the flow control does not fully compensate for the change in pressure.



### Saturation

Saturation is indicated by the flat portion of the flow vs. current characteristic. Saturation occurs when there is no relative change in flow, even with a change in current. The saturation current of a normally closed valve is the level at which any increase in current does not increase the flow. The saturation current for a normally open valve is the level where the flow decreases to the expected leakage flow rate. Note: The maximum current may or may not be the same value at which saturation occurs. In other words, the saturation flow is typically at or above the maximum current required for most HydraForce flow control valves and directional control valves.



## Summary

In this chapter the following concepts were presented:

- An equation to determine flow through an orifice.
- Fixed and variable non-pressure compensated flow controls.
- Introduction to pressure compensated flow controls.
- Pressure compensation characteristic.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. The pressure drop across an orifice is 100 psi. The area is 0.005in<sup>2</sup>. Determine the flow. \_\_\_\_\_
2. What is the disadvantage of a fixed orifice? \_\_\_\_\_
3. What is one typical application for a restrictive style flow regulator? \_\_\_\_\_
4. What is the typical viscosity used when valves are tested for presentation of performance in the HydraForce catalog? \_\_\_\_\_
5. True or False. The general trend of the flow characteristic for a pressure compensated flow control is decreasing with increasing pressure. \_\_\_\_\_
6. How many orifices are active during the operation of a priority/bypass type of flow control? \_\_\_\_\_
7. Describe saturation. \_\_\_\_\_
8. Define flow range. \_\_\_\_\_

# Chapter 8: Non-Compensated Flow Control

## Objectives

- Obtain general knowledge of the non-compensated proportional flow controls.
- Learn about the forces acting on the valve components, specifically the flow force, spring force and magnetic force.
- Learn methods used by HydraForce to minimize flow forces.
- Discover how these proportional flow controls can be used with the external pressure compensator valves.
- Learn about the manual override used on the proportional flow control valves.
- Learn how the external pressure compensators operate.

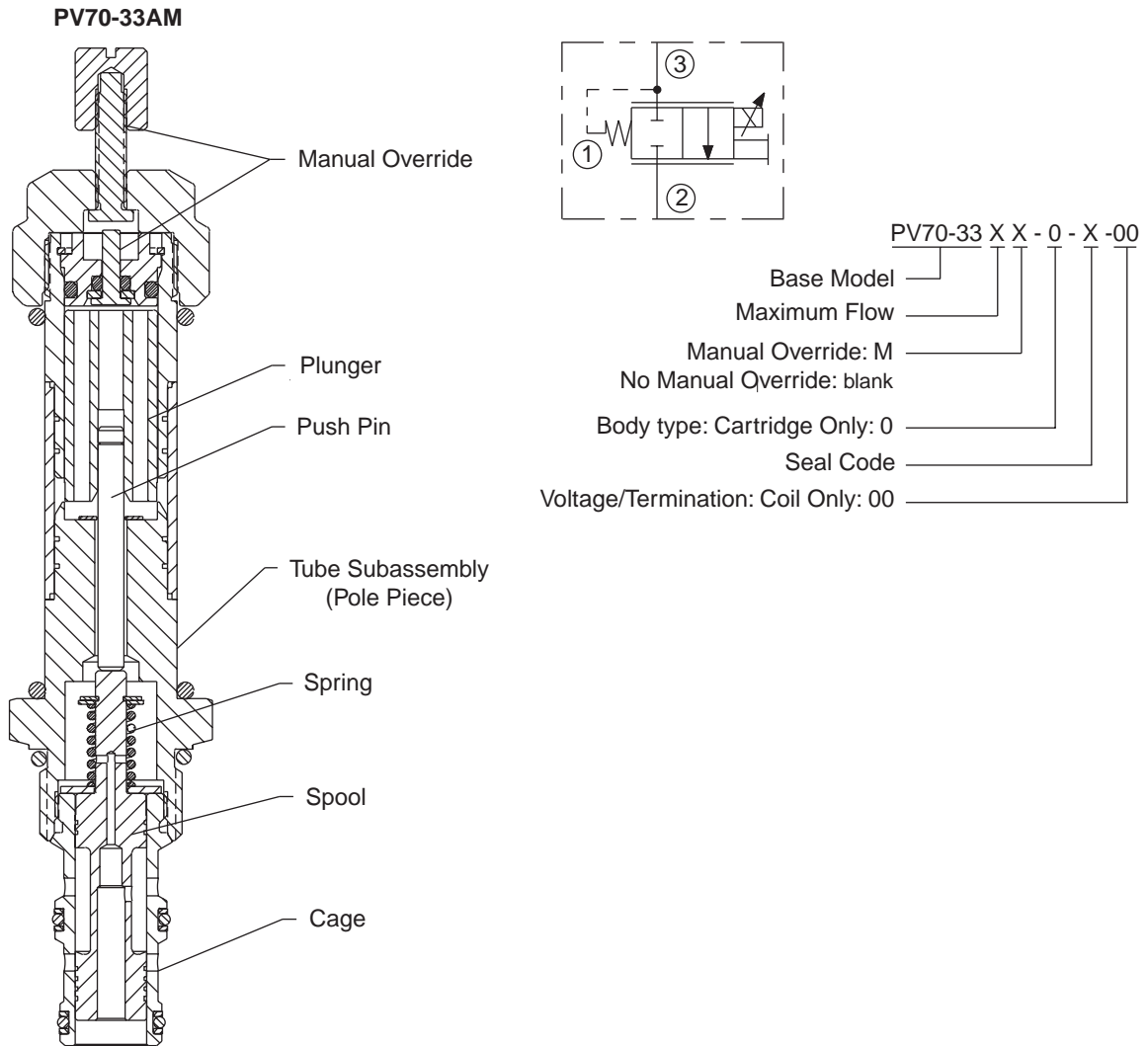
## Introduction

In this chapter we will learn about two types of direct acting, non-compensated, proportional flow control valves. We will review the construction and forces acting on the components. In addition, the performance of these valves used in conjunction with external compensating valves will be presented.

# HYDRAFORCE

### PV70-33

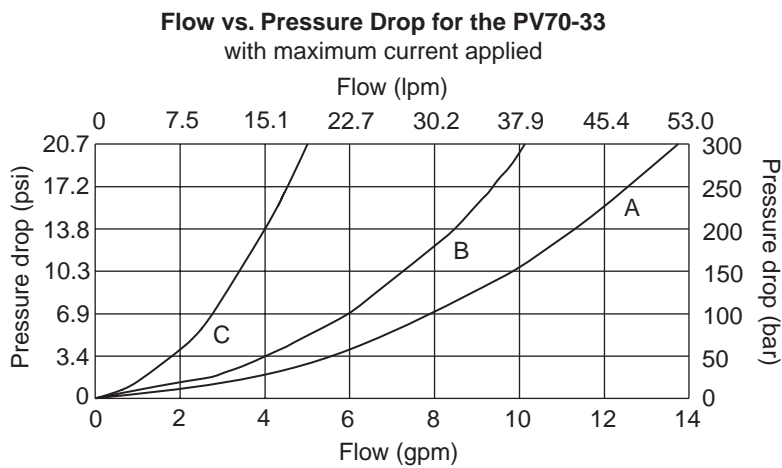
The first valve to be reviewed is the PV70-33. This valve is a direct acting, normally closed proportional flow control. The pressure drop curve, forces acting on the valve and the operation of the valve will be discussed in this section. The operation of the optional manual override available for the PV valves will be discussed at the end of this section. A cross section of the PV70-33AM is shown below.





**Performance of PV70-33**

The flow vs. pressure drop characteristic is shown in the graph below. Although the performance of a proportional valve is dependent on the current, a current vs. flow characteristic is not shown for the PV70-33. The reason for this is that the PV70-33 should be used with an external compensator. The current vs. flow characteristic for the PV70-33 when used with an external compensator, is presented in a later section. The PV70-33 and PV72-33 valves are normally used in conjunction with an external compensator because the differential pressure across these valves must be limited to 300 psi. This will be explained further in the section which details the forces acting on the PV70-33.



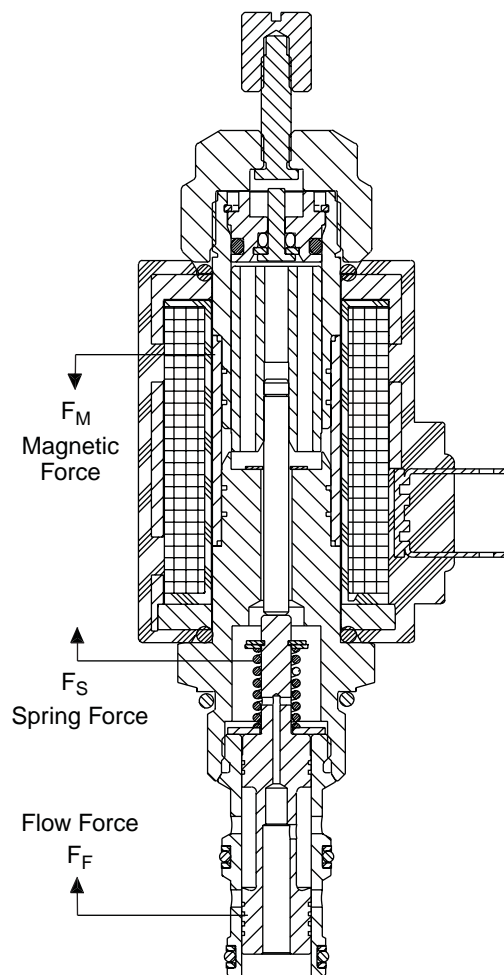
The three pressure drop characteristics shown above represent the three different maximum flow variations available from the PV70-33. Three different flow characteristics are made possible by changing the number of cross holes in the cage. This too will be covered further in the operation section.

### Forces on PV70-33

There are two significant forces which act on the valve components of the PV70-33. These two forces are the magnetic actuator force and the spring force. This holds true as long as the pressure differential across the valve is below 300 psi as shown in the previous graph. Both of these forces are briefly defined below. A more detailed explanation can be found in the Solenoid Valves Manual and the Proportional Pressure Control Manual from HydraForce.

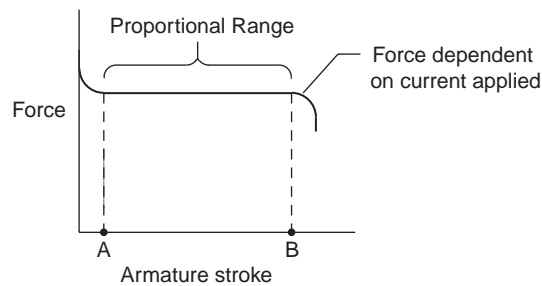
A third force which can influence the performance of the flow control valves is known as a flow force or Bernoulli force. A description of this force will also be presented in the following sections.

Further, three other forces that act on the components of the flow control valves. These include magnetic hysteresis, mechanical friction and viscous friction. These combine to create the hysteresis that is shown on the flow vs. current graphs. A full explanation of these can be found in the Proportional Pressure Controls Manual and the Solenoid Valves Manual.



### Magnetic Force

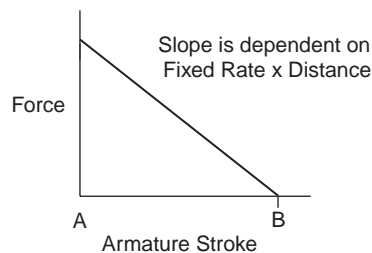
The magnetic force developed by the coil and armature shown on the previous page is proportional to the current applied to the coil. The actuator assembly (coil, armature and pole piece) is known as a linear proportional actuator. The force vs. stroke characteristic is shown below. Notice that the force remains unchanged regardless of the position of the armature listed on the x-axis.



A = Armature closest to pole piece  
 B = Armature furthest from pole piece

### Spring Force

The type of spring force used in the PV valves is known as a compression spring. As the spring is compressed, the force increases. The force is proportional to the stiffness (rate) of the spring multiplied by the distance the armature moves. The spring force increases as the armature or plunger moves closer to the pole piece.

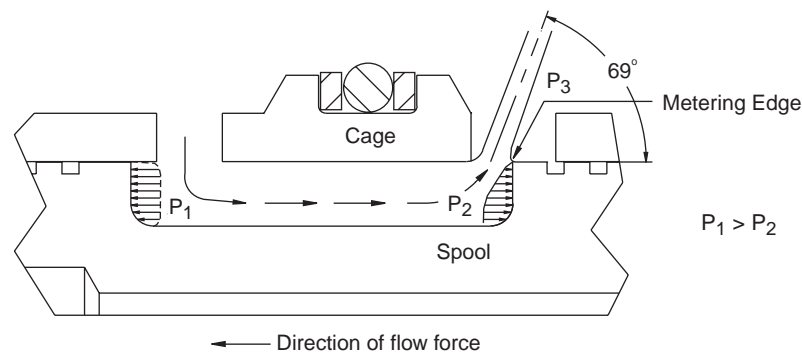


A = Armature closest to pole piece  
 B = Armature furthest from pole piece

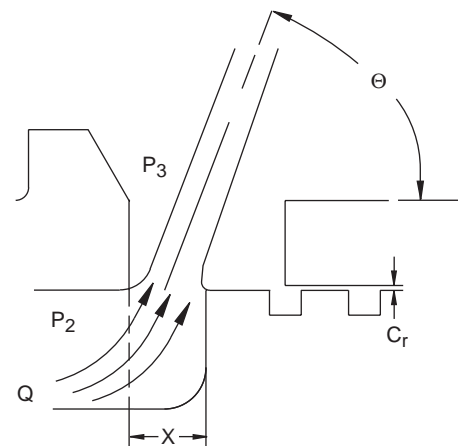
### Flow Force

The following section provides a brief description of the flow force. An introduction to the flow force is given in the Solenoid Valve Course Manual. The **Bernoulli force** or flow force as it is referred to in industry, acts on the spool. There are two components that make up the flow force, these are the axial and lateral flow forces. We are only concerned with the axial force because the lateral force acts evenly around the spool and does not affect the operation of the valve.

As oil flows past the edge of the spool which is metering or restricting the flow, the oil accelerates in this restricted area. This occurs because the same amount of oil that flows into a valve must flow out. If the flow passage in a valve decreases in size, the velocity or speed of the oil must increase to ensure the same amount of oil continues to move through this passage. (Note: There must be sufficient pressure to push the oil through the metering area to maintain the flow rate.)



As the oil moves past the metering edge, the pressure decreases. That is, the pressure noted by P<sub>2</sub> in the diagram above is higher than P<sub>3</sub>. Further, the pressure acting on the surface closest to the metering edge P<sub>2</sub>, is lower than P<sub>1</sub>. Since there is a difference between P<sub>1</sub> and P<sub>2</sub>, there is a force acting on the spool in the direction of P<sub>1</sub>, or towards the left. This is the flow force. The flow force described is governed by the following equation:



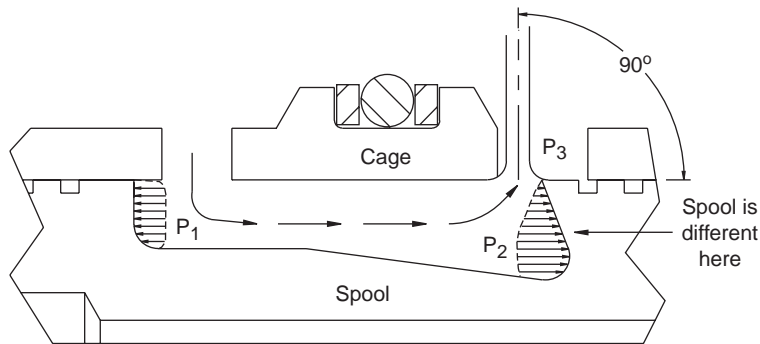
$$F = \frac{Q \sqrt{2\rho \Delta P (X^2 + C_r^2)} \cos \Theta}{X}$$

Where:

- Q = Flow rate of oil
- X = Amount the cross holes are open
- C<sub>r</sub> = Clearance between cage and spool
- ΔP = Pressure drop through the valve
- ρ = Density of fluid
- Θ = Angle at which the fluid exits

### Compensation for Flow Force

One part of the equation given in the section describing the flow force, which can be influenced through valve design, is the angle the fluid exits the valve. As the previous diagram shows, this is typically close to  $69^\circ$ . The diagram below shows one possible configuration to change the angle at which the flow exits. The difference between this diagram and the previous is that an angle or hook type feature has been added to the spool in the area where  $P_2$  acts. The ideal angle for the flow to exit would be  $90^\circ$ . This is because the cosine of a  $90^\circ$  angle is 0 and the force described by the previous equation would go to zero. In actual valves, the change below does not change the angle to  $90^\circ$ . Typically, it is less than  $90^\circ$ , therefore, there is always a force acting on the spool.



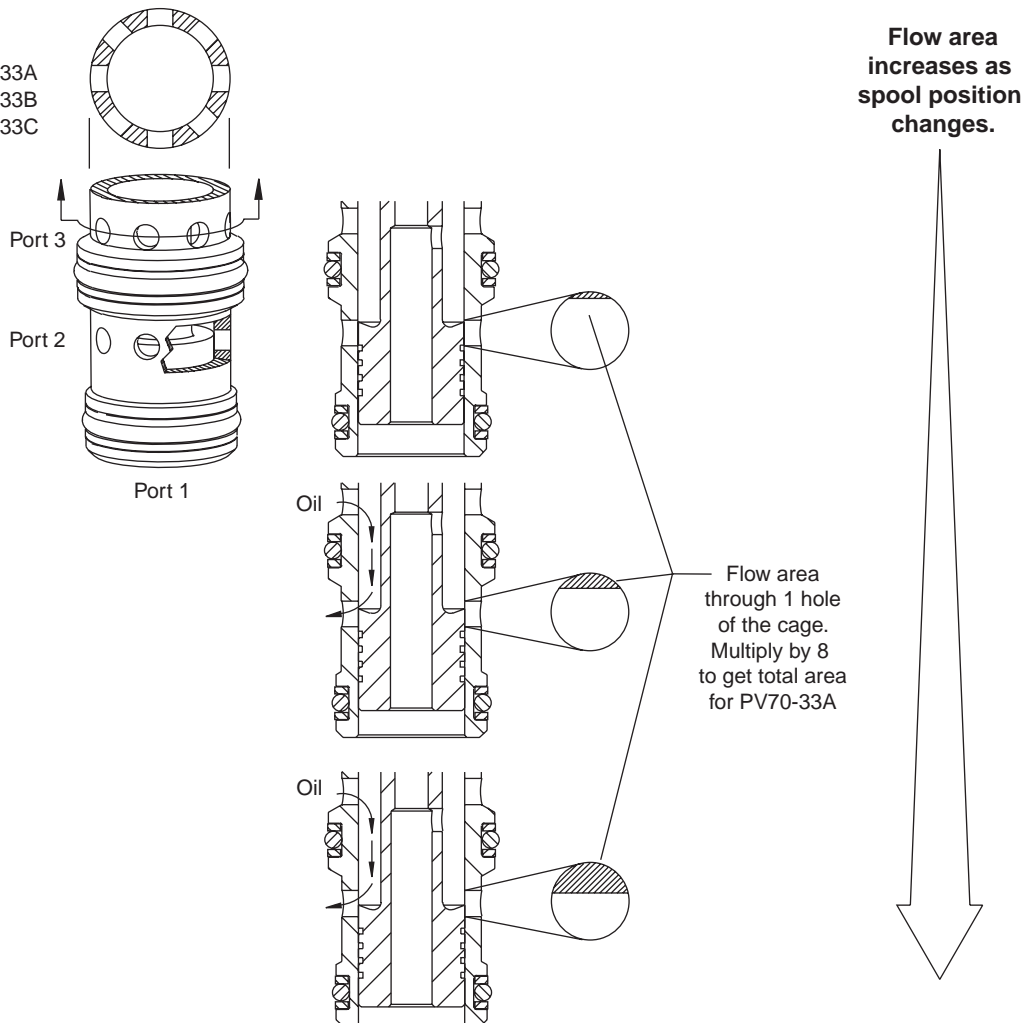
Other design features used by HydraForce to minimize the flow force include assuring that the metering edge is sharp, and that the clearance ( $C_r$ ) between the spool and the cage is as small as possible.

### Operation of PV70-33

The operation of the PV70-33 is simply based on the armature pushing on the spool and spring. Initially, the spool is blocking the flow of oil from port 3 to 2. (Note: port 1 is not used as a flow path. The cavity is “blind,” that is, no oil can flow through this port.) As current is applied to the solenoid, the armature is attracted to the pole piece. In turn, the armature pushes down on the spool/spring assembly. Oil then begins to flow from port 3 to 2. When more current is applied, the armature moves the spool further. The maximum flow through the valve is dependent on the number of cross holes in the cage and the pressure differential from port 3 to 2. The flow area through each cross hole can be added to one another to give an overall flow area. The amount of flow through the total area is determined by the same equation given for the fixed orifice in the previous chapter ( $Q = 31 \times A \times (DP)^{1/2}$ ). Also, by increasing the number of holes and leaving the pressure differential constant, the flow through the valve will increase. The same is true in regards to the amount the spool opens the flow area through the cage cross holes. The more the spool opens, the larger the flow area. Again, if the pressure drop remains constant, the flow will increase. The diagrams below show how the flow area is increasing as the spool moves to open more of the cage cross holes.

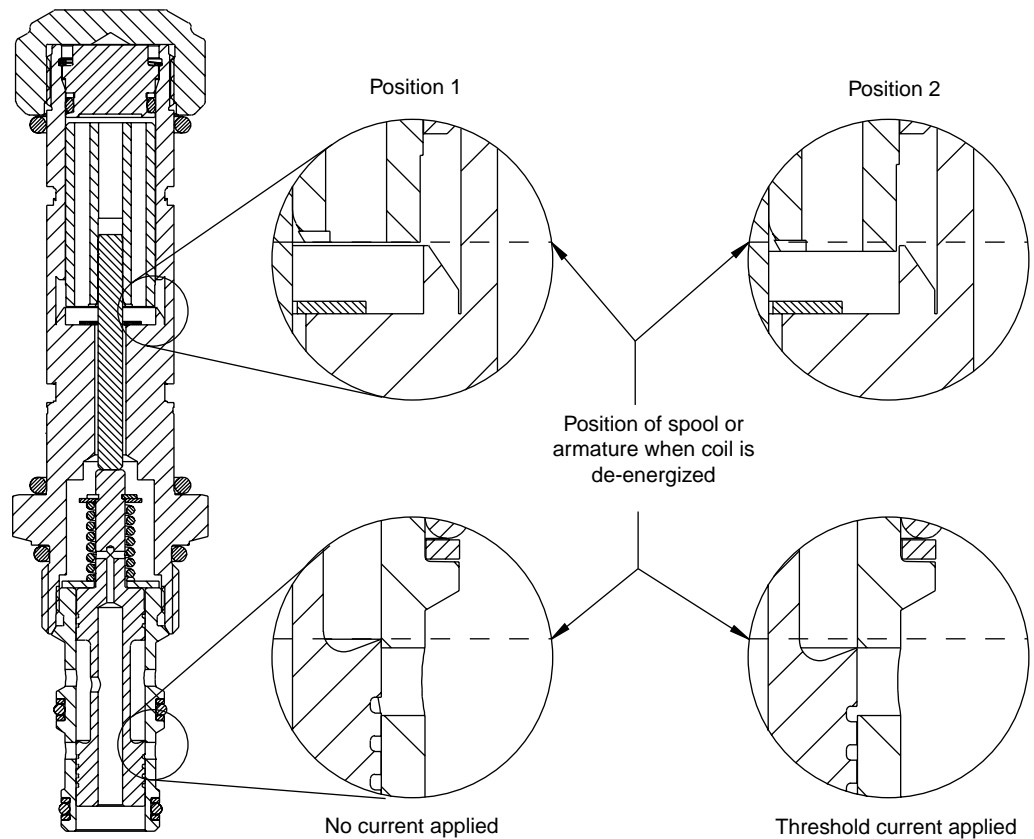
**Cross Section through Cage:**

8 holes for PV70-33A  
6 holes for PV70-33B  
4 holes for PV70-33C

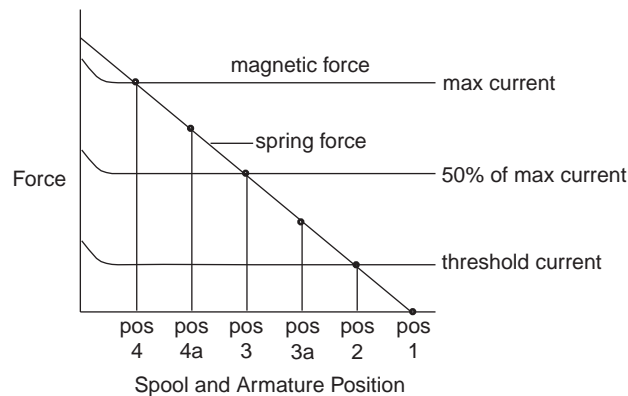
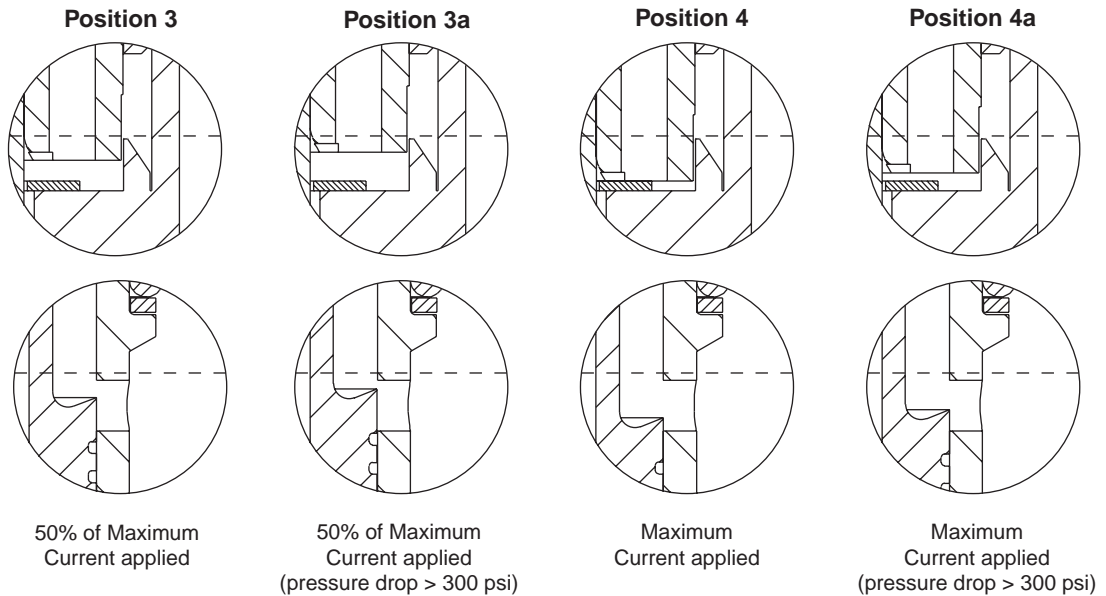


**Effects of Flow Force**

The flow force begins to influence the performance of the PV70-33, PV70-35, PV72-33 and the PV72-35 when the differential pressure exceeds 300 psi. A description of how the flow force influences the amount the spool can move is shown below. The diagrams show relative positions of the armature and spool at various current levels applied to the coil. The cross section of the valve and the enlarged view in the middle of the picture represent the spool and armature position when no current is applied to the valve. The view on the right shows the position of the spool and armature when the threshold current has been applied to the coil. Notice that the spool edge is in line with the edge of the cross. Again, the definition of the threshold current is the level of current where the flow begins to change, relative to a change in current. Since there is no flow, this current and relative spool position are the same, regardless of the differential pressure.



The next set of diagrams show various spool positions as the current is increased. The four pictures to the left show the position of the armature and spool when 50% of the maximum current is applied. Notice there is a difference in the position of the armature and spool when the pressure exceeds 300 psi. When the pressure drop is lower than 300 psi, the spool opens almost half of the cage cross hole. However, when the pressure drop exceeds 300 psi, (or the flow force influences the valve performance) the cage hole is only open about 25%. This same analogy applies to the four pictures to the right where maximum current has been applied. When the pressure drop across the PV70-33 is less than 300 psi, the spool can uncover more than 75% of the cross hole. However, if the pressure drop exceeds 300 psi, the magnetic force is only strong enough to push against the spring and flow force to a point where 50% of the cage cross hole is open.



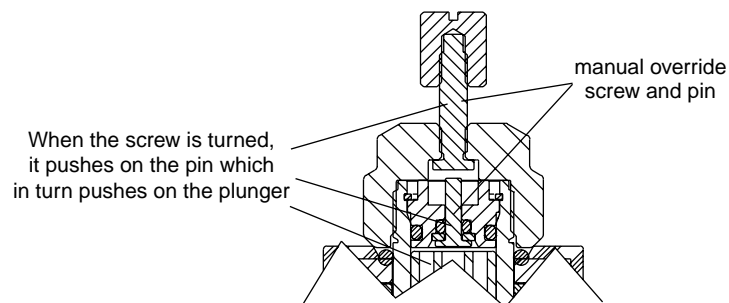


The different spool positions, spring force and magnetic force are summarized in the graph on the previous page. Again, position 1 represents the forces when no current is applied. Position 2 is the point where flow begins to pass through the valve. This represents the threshold. Notice how the spring force is exactly equal to the magnetic force at the threshold current. This is also true at the 50% current level and at maximum current. Point 3a and 4a show that the spool moves towards position 1 even though the current should create magnetic force sufficient to move the armature to position 3 and 4 respectively. This is because of the influence of the flow force.

The effect of the flow force can also be summarized in a force balance equation. For example, the force balance equation at position 3 or 4 would be  $F_M = F_S$ . However, at position 3a and 4a the force balance equation would be  $F_M = F_S + F_F$ . Of course, both equations have been simplified. In fact, there is always a flow force acting on the spool. In other words, as the equation describing the flow force shows, the flow force is proportional to the amount the spool uncovers the cross hole and the difference in pressure. When this pressure differential exceeds 300 psi, the flow force is sufficient to influence or disrupt the proper operation of the valve. When the flow force is sufficiently large enough to influence the position of the spool, this becomes a form of compensation. That is regardless of how large the pressure drop is, the flow does not increase. Further, if the pressure drop becomes sufficiently large, the spool could close off the cage cross holes completely. This is known as the valve *washing shut*.

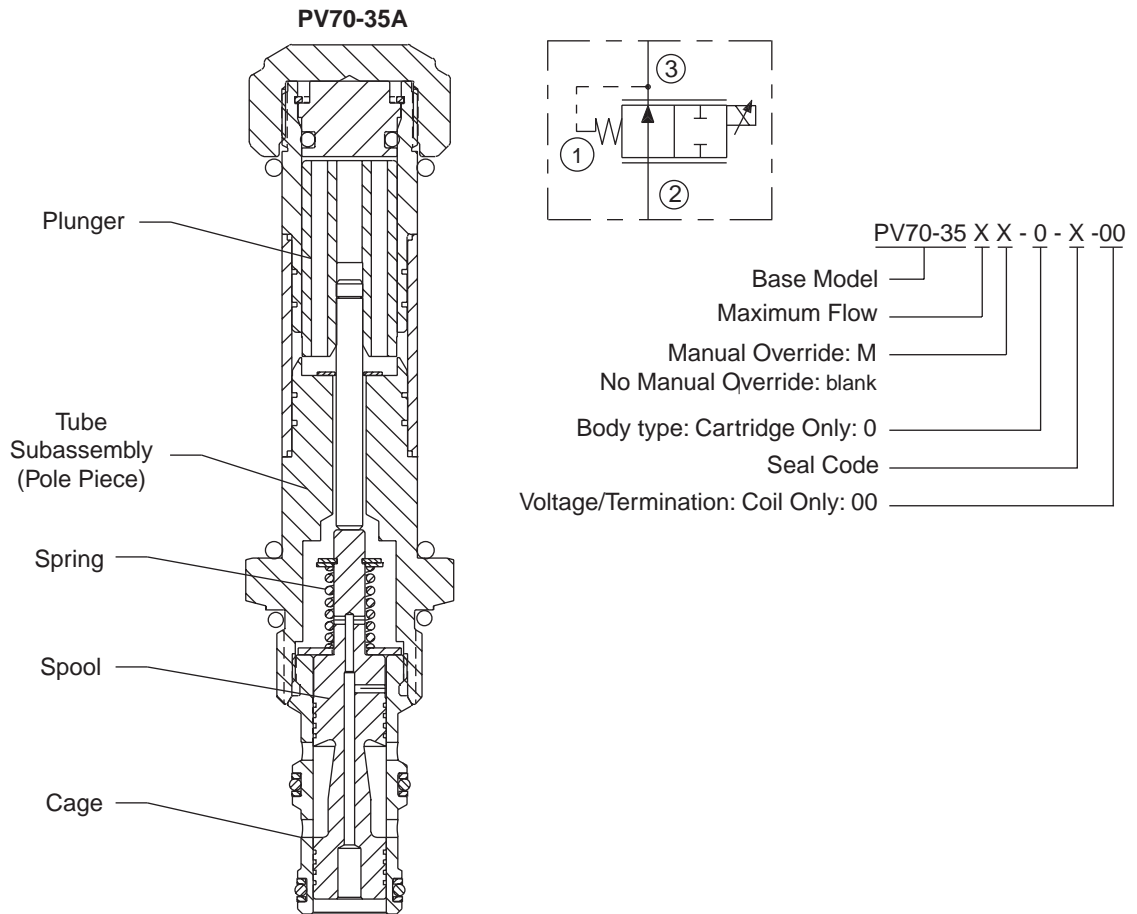
## Manual Override Operation

The manual override used with the PV type valves is known as a screw style override. When the manual override screw is fully “backed-out” (turned counterclockwise until it stops), the manual override is disengaged. As the manual override screw is turned clockwise, it begins to push down on the armature. This movement of the manual override pushing down on the armature replaces the magnetic solenoid force. The more the manual override is turned clockwise, the more the flow increases.



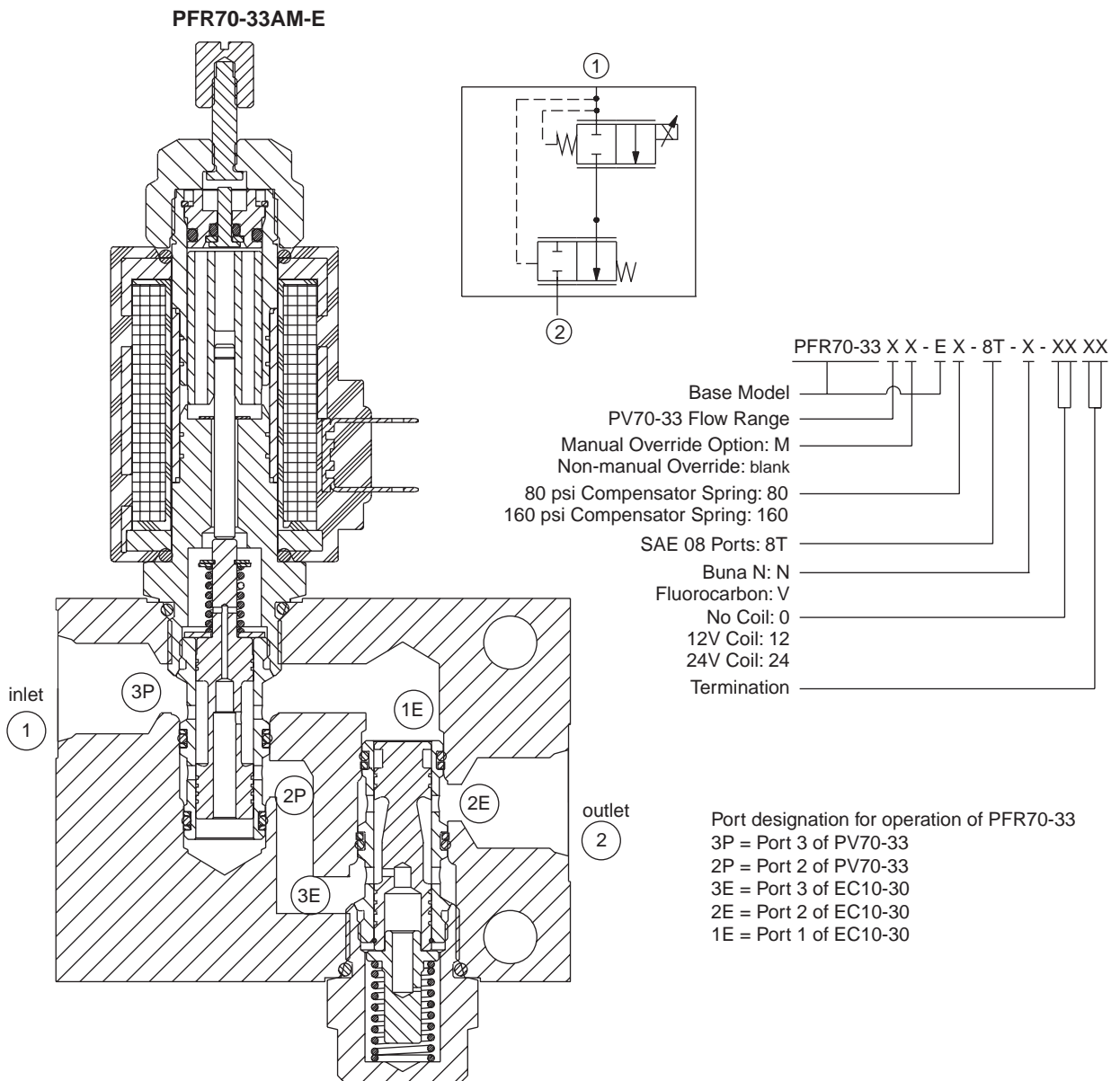
### PV70-35

Shown below is the PV70-35. It is a normally open, direct acting, non-compensated proportional flow control. It is noted here simply to demonstrate that the normally open product is available. The performance, forces acting on the components and operation of this valve are similar to the PV70-33. The only difference is that the inlet is at port 2, the outlet is at port 3 and the inlet and outlet are normally connected when no power is applied to the coil (de-energized). Port 1 is not used, therefore no hydraulic connection is required. Since this valve is similar to the PV70-33 no further information will be presented, except for the performance of the valve when used with an external compensator. This will be presented in a later section.



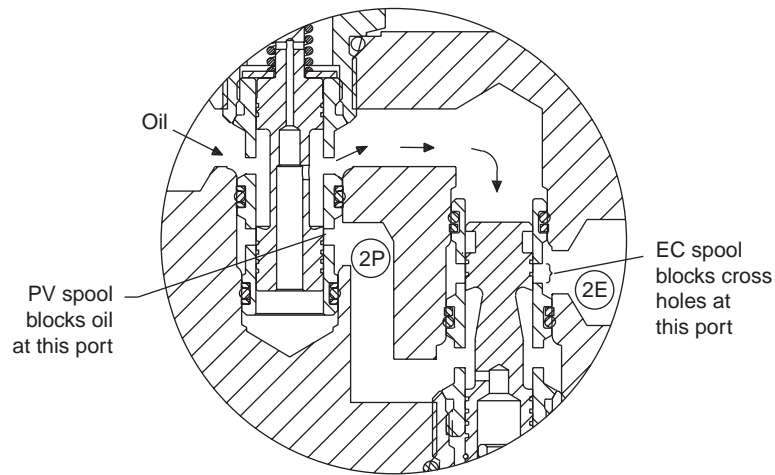
**PFR70-33-E**

The combination block shown in the cross section below is the PFR70-33AM-E. It consists of a direct acting proportional flow control valve, PV70-33 and a restrictive style pressure compensator valve, EC10-30. Both are installed in a combination manifold. The PV70-33 is similar to an electric version of the needle valve described in Chapter 2. The EC10-30 acts like a restrictive type compensator also described in Chapter 2. The operation of these two valves in combination, as well as the flow vs. current characteristic and compensation curve, are described in the following sections.

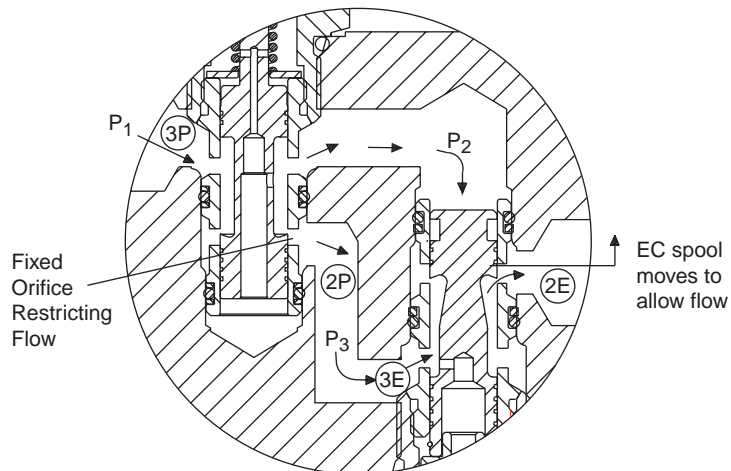


### Operation of the PFR70-33

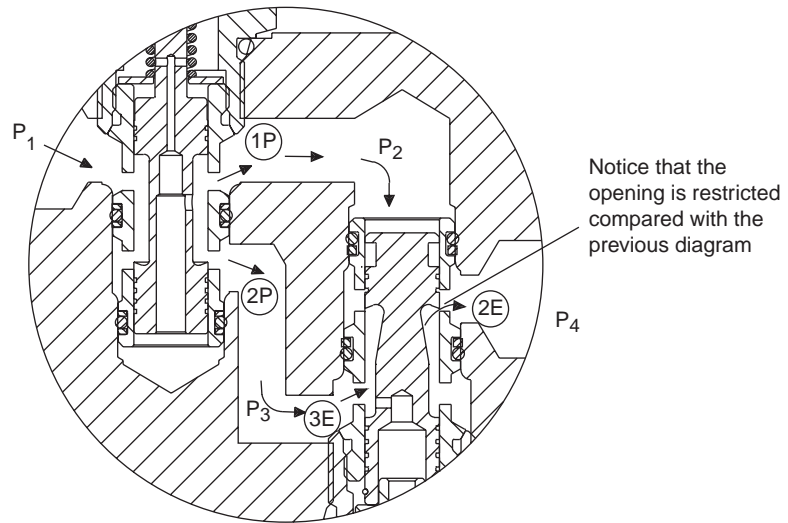
The operation of the PFR70-33-E is described in the following section. The first operational diagram shows the combination valve when the oil is pressurized and no current is applied to the coil. Notice the spool of the PV70-33 blocks port 2P completely. Again, this is because no current is applied to the coil. Also, the spool of the EC10-30 is at the extreme shifted state. It has moved from the 'at rest' position shown in the previous diagram. Notice the spool completely blocks the cross holes at port 2E.



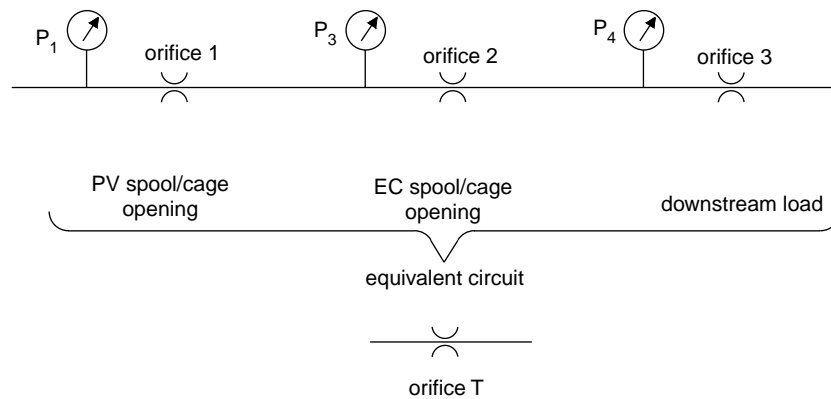
In the next cross section current has been applied to the coil. This causes the armature to be attracted to the pole piece. At the same time, the armature pushes down on the spring/spool assembly. Oil then flows from port 3P to 2P of the PV. The oil continues to flow into port 3E of the EC10-30. Even though the pressure,  $P_3$ , is lower than  $P_2$ , because of the restriction of the fixed-orifice of the PV spool, the EC10-30 spool moves, allowing oil to flow out of 2E. This occurs because the pressure force developed by  $P_3$  acting on the spool, is assisted by the spring in the EC10-30. The forces acting on the EC10-30 spool are summarized in the following section.



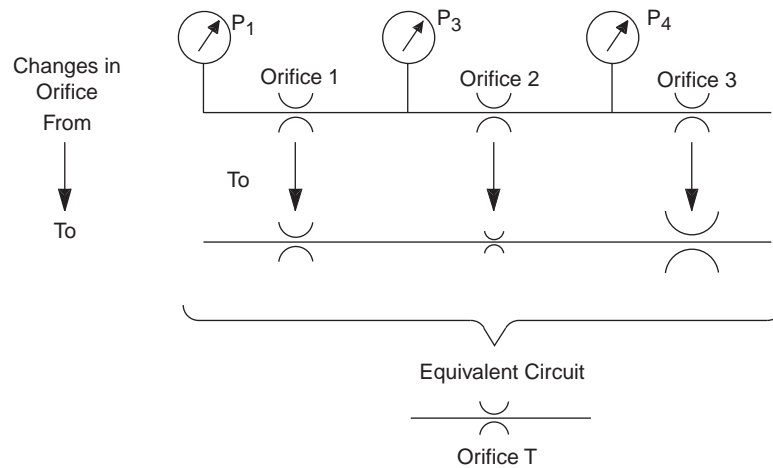
The next cross section shows how the EC spool compensates for changes in load pressure downstream of the outlet. As current applied to the PV coil is increased, the PV spool opens more of the cage cross holes at port 2P compared to the previous diagram. However, notice that the opening of the cage cross holes at port 2E of the EC is less than the previous diagram. This is because there was a decrease in the required pressure downstream of the outlet. If the EC spool did not move to decrease the opening, the flow would increase. This is further described by the orifice diagrams below.



The circuit below represents the PFR70-33-E and the load downstream of the manifold. It will be used to demonstrate the EC spool compensating for the change in load.

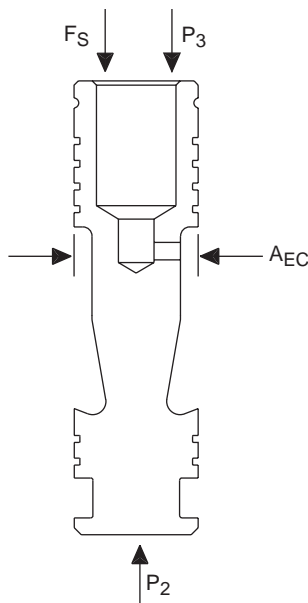


Recall that the flow through an orifice can be represented by the simplified equation  $Q = 31 \times A \times (DP)^{1/2}$ . In this case, A is the area of orifice T. Also, keep in mind that the EC is in the circuit to maintain the pressure drop across orifice 1 or the difference between  $P_1$  and  $P_3$ . If this is true and orifice 3 becomes larger, then orifice 2 must decrease. The schematic below represents this change in load pictorially.



To summarize these points, as the load changes, the orifice created by the EC spool and cage cross-hole changes accordingly to compensate. While at the same time the orifice created by the PV spool and cage remains the same. Also notice that orifice T remains unchanged because the compensator works to maintain a constant flow rate.

### Forces Acting on the EC Spool



The diagram to the left and the variables below are used to describe the forces acting on the EC spool.

- $A_{EC} P_2$  = Pressure force acting on spool at Port 1
- $A_{EC} P_3$  = Pressure force acting on spool at Port 3
- $F_s$  = Spring force

The following equation describes the balance of forces acting on the EC:

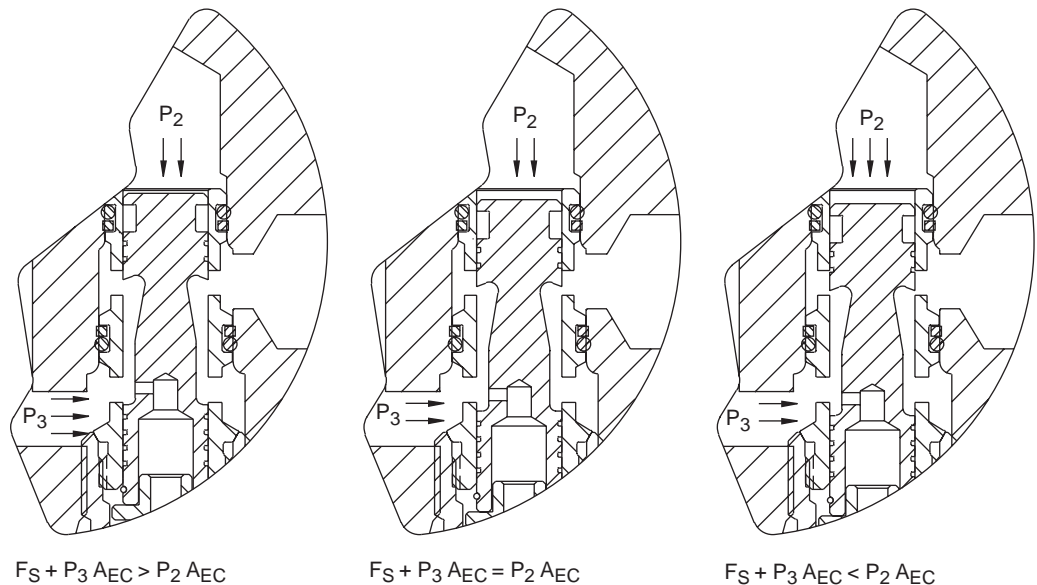
$$A_{EC} P_2 = A_{EC} P_3 + F_s$$

The equation above is balanced whenever the inlet pressure force ( $A_{EC}P_2$ ), minus the outlet pressure force ( $A_{EC}P_3$ ), is equal to the spring force. This is summarized as follows:

$$(P_2 - P_3) A_{EC} = F_s$$

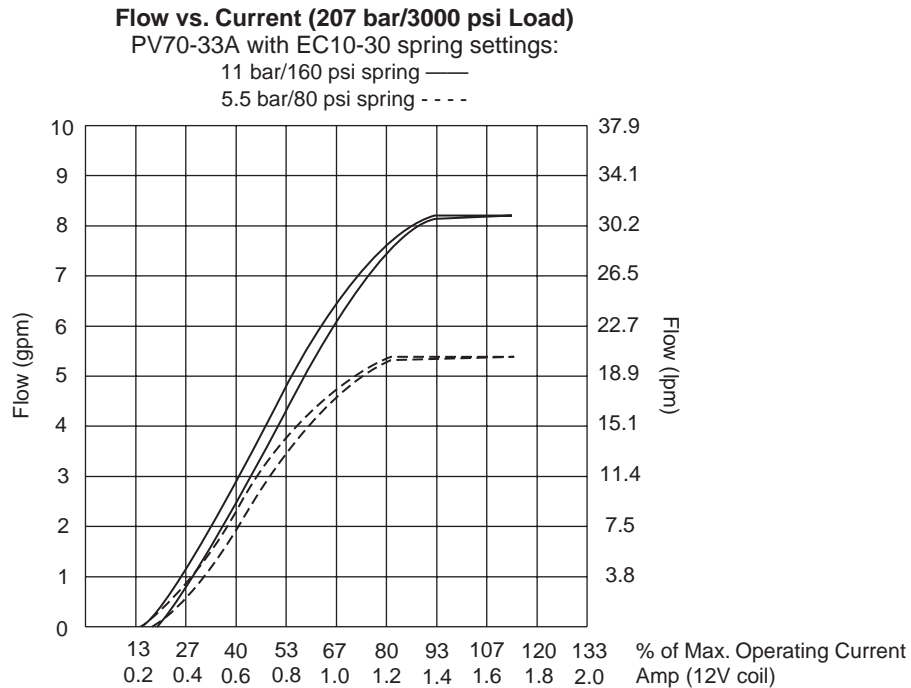
As noted on the previous page, the spool of the EC10-30 moves to maintain a pressure differential so that the spring force is balanced. This differential pressure value is appropriately known as the compensator value. If this compensation pressure is not reached, then the flow will be dependent only on the size of the orifice between the spool and the cage cross holes of the PV. This means that the flow will be dependent on the pressure drop across this orifice. Therefore the resultant flow will vary based on the fixed orifice equation described in Chapter 7

The equation  $(P_2 - P_3) A_{EC} = F_s$  shows that the spool will move whenever the left side of the equation does not equal the right side. The spool position changes when  $P_2$  is greater than  $P_3$  by an amount over the rated pressure compensation value of the EC. For example, assume the pressure compensation value of the EC is 80 psi. Then, if the difference between  $P_2$  and  $P_3$  is greater than 80 psi, the left side of the equation above is greater than the right side. The effect on the EC is that the spool metering edge will close off the cage cross-holes. This will cause the pressure  $P_3$  to increase until the equation is balanced again. As expected, if the difference between  $P_2$  and  $P_3$  is less than 80 psi, the spool metering edge will move to increase the opening of the cage cross holes at port 2E. These concepts are summarized in the drawings below.



### Performance of PFR70-33-E

The flow vs. current characteristic for the PFR70-33-E is shown below.

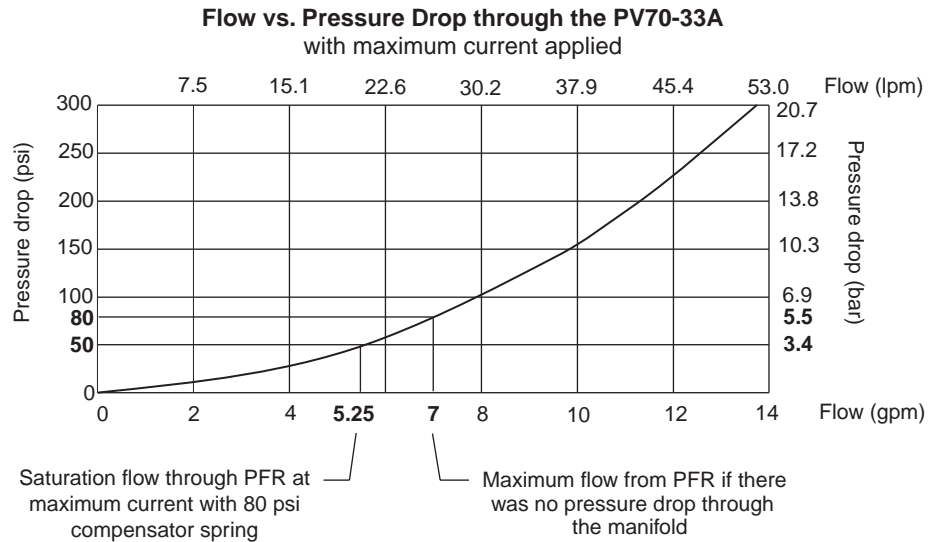


This graph shows the performance of the PFR70-33A-E with two different compensator springs in the EC10-30. From the graph, the flow range and the hysteresis can be determined, as well as the threshold current and the saturation current. The flow range shown above is from 0 to 5.25 gpm when the PV70-33 is used with the EC10-30, which has an 80 psi compensating spring. If the EC10-30 has a 160 psi compensating spring, then the flow range is 0 to 8.25 gpm. The hysteresis for either flow range shown above is less than 6%. This is true as long as PWM or dither is applied to the coil. (Refer to the HydraForce Proportional Pressure Control Course Manual for a definition of PWM, dither, and how these are used to reduce hysteresis.) The threshold current is 13% of the maximum operating current. The maximum operating current for the 12V coil is 1.5 Amp when used with the PV70-33. (The maximum current draw limitation will be defined in Chapter 14). As mentioned above, the saturation flow for the PFR70-33A-E is 5.25 gpm when the 80 psi spring is used. The saturation current for the combination, which has an EC10-30 with the 80 psi compensation spring, is 75% of the maximum current and 95% of the maximum current for the 160 psi compensation spring.

As noted on the graph above, there are two different maximum flow rates depending on the compensator springs. Recall from the previous section that the EC10-30 should maintain a constant pressure drop across the PV70-33 that is equal to the compensating spring value. Further, when the spool of the PV70-33 is fully shifted (I<sub>MAX</sub> applied), the maximum flow should be predicted from the pressure drop curve given in the performance section of the PV70-33.

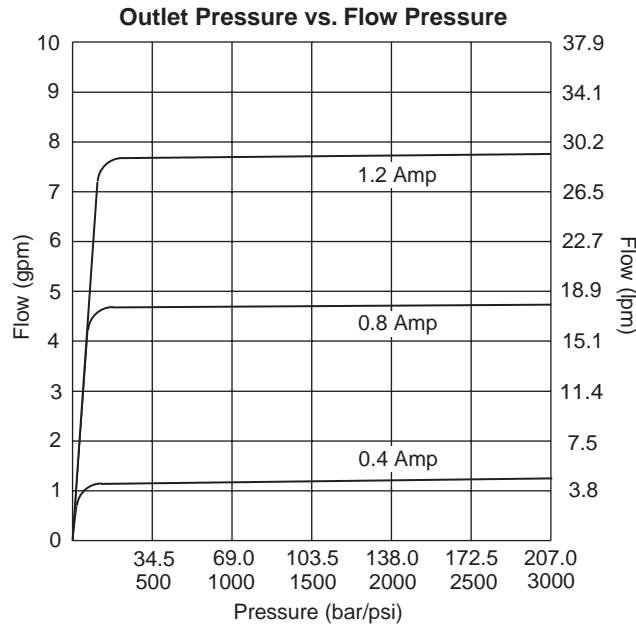


The following graph is used to depict this concept. It is the same one used to describe the pressure drop through the PV70-33. The difference is that only the curve for the PV70-33A is shown and two flow values and two pressure values are highlighted.

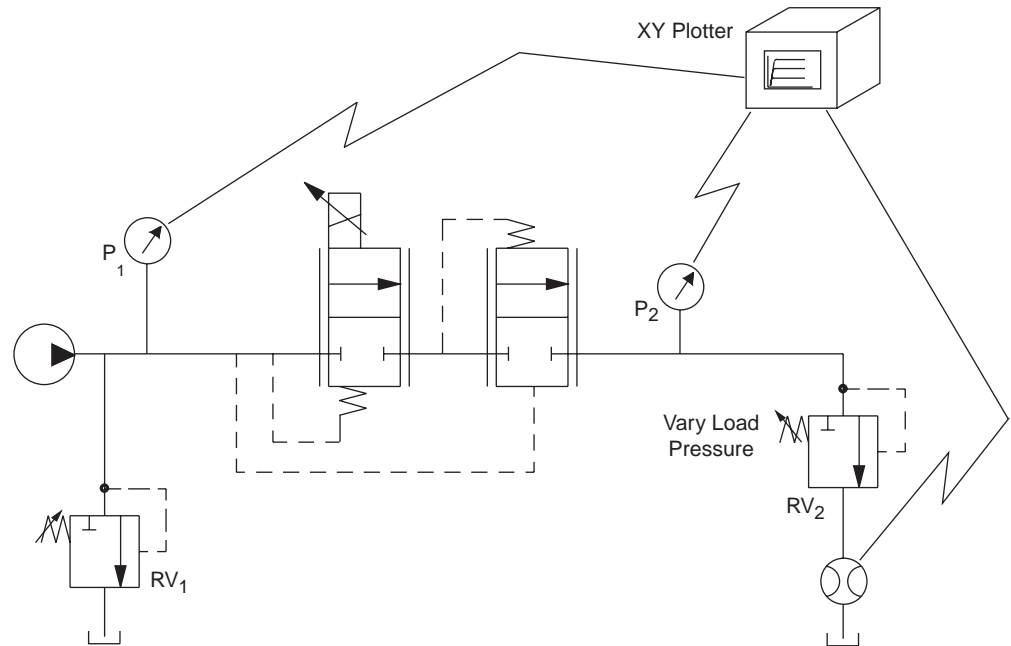


The first flow highlighted in the graph above is the maximum flow for the PFR with an 80 psi compensator spring. The second flow highlighted is the flow that creates a pressure drop of 80 psi across the PV70-33. These were selected to show the effect of pressure drop through a manifold. As noted on the graph, if there was no pressure drop through the manifold, the maximum flow the PFR would control is 7 gpm with the 80 psi compensator spring in the EC. However, as the graph also shows, the maximum flow is 5.25 gpm but the pressure drop for the PV70-33A at this flow is only 50 psi. This implies that the pressure drop between PV70-33A and the EC10-30 is 30 psi. This is the pressure drop of the manifold.

The other characteristic that defines the performance of the PFR70-33A is the compensation characteristic. The curves below depicts the performance or compensation characteristic of the EC10-30 with the 160 psi compensating spring.



Three different curves are shown on the graph. Each curve represents a different orifice or spool/cage opening of the PV70-33A. Recall that each orifice size is dependent on a given amount of current. This current level can be related to the previous graph showing flow versus current. The current level is written next to the corresponding compensation curve. One other characteristic worth noting on the graph is the sloped line starting at zero and joining the three compensation curves. This line represents the flow characteristic (pressure drop) of the compensator before the spool of the EC10-30 actually moves. In other words, the flow is not being compensated for changes in the pressure downstream of the outlet. The reason for this is that the pressure differential force across the EC10-30 spool does not exceed the spring force. Recall the equation  $(P_2 - P_3) A_{EC} = F_s$  which determines at what pressure differential the spool compensates or moves. When the difference in these two pressures does exceed the spring force, the spool begins to compensate for changes in pressure. This is shown by the fact that the three curves are relatively flat, even though the pressure is increasing.



The method or test used to develop these curves is shown above. RV<sub>1</sub> is set to a maximum setting to protect the circuit. RV<sub>2</sub> represents a change in load pressure. The curves are generated by recording the flow and difference in pressure between P<sub>1</sub> and P<sub>2</sub> on an x-y plotter. The flow available from the pump is set to a level that exceeds the maximum rating of the PFR70-33. The excess flow discharges across RV<sub>1</sub> depending on the current level applied to the PV70-33. Initially, RV<sub>2</sub> is set to the lowest pressure setting. The current applied to the PV70-33 is then set to one of the levels given on the graph on the previous page. Finally, the setting of RV<sub>2</sub> is slowly changed and the x-y plotter records the change in flow and pressure.

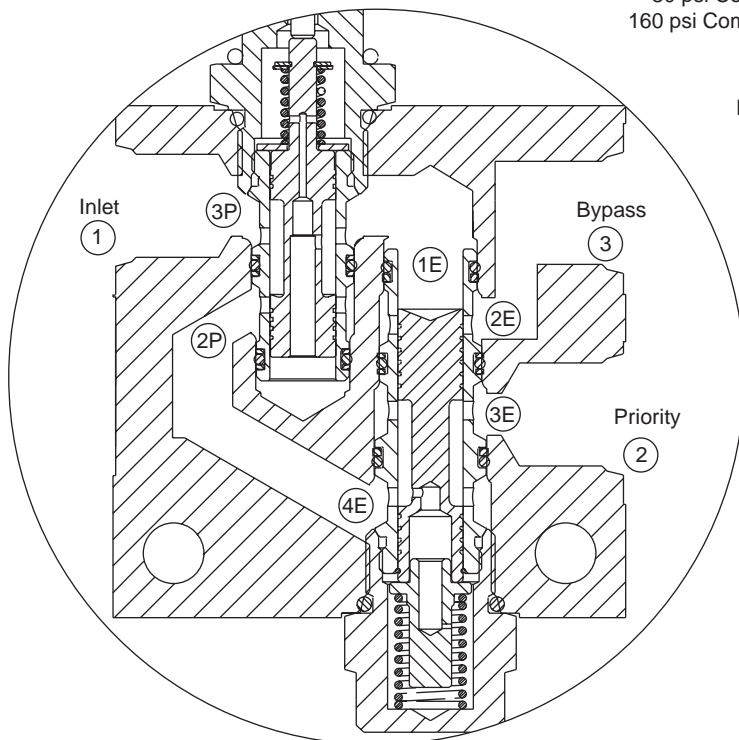
### PFR70-33-F

The PFR70-33-F is a combination valve similar to the PFR70-33-E described in the previous section. The difference between the two is the compensator. The compensator used in the PFR70-33-F is an EC10-40. This compensator regulates flow similar to the EC10-30 but it also has an extra port to bypass excess flow to other hydraulic functions. In principle, the operation of this valve is the same as the priority/bypass flow regulator described in Chapter 7.

The cross section of the PFR70-33-F is shown below (the coil and the proportional actuator were omitted to show the hydraulic components more clearly). A description of the operation of this valve follows as well as a typical flow vs. current characteristic and pressure compensation characteristic.

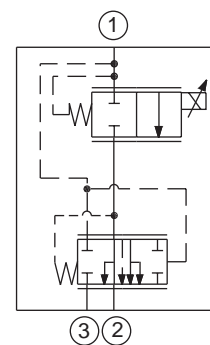
Port designation for operation of PFR70-33-F

- 3P = Port 3 of PV70-33
- 2P = Port 2 of PV70-33
- 4E = Port 4 of EC10-40
- 3E = Port 3 of EC10-40
- 2E = Port 2 of EC10-40
- 1E = Port 1 of EC10-40



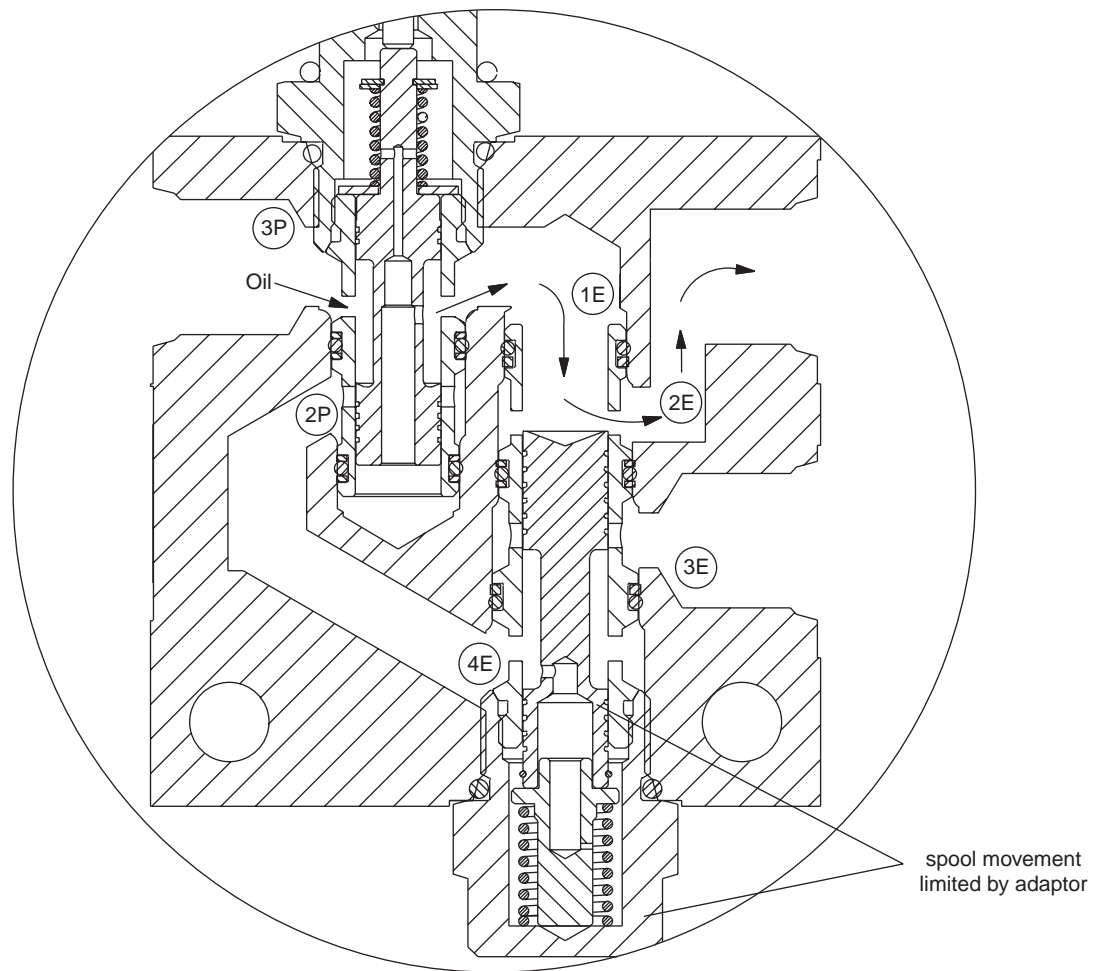
PFR70-33X X - F X - 8T - X - XX - XX

- Base Model
- PV70-33 Flow Range
- Manual Override Option: M
- Non-manual Override: blank
- 80 psi Compensator Spring: 80
- 160 psi Compensator Spring: 160
- SAE 08 Ports: 8T
- Buna N Seals: N
- Fluorocarbon Seals: V
- No coil: 0
- 12V Coil: 12
- 24V Coil: 24
- Termination

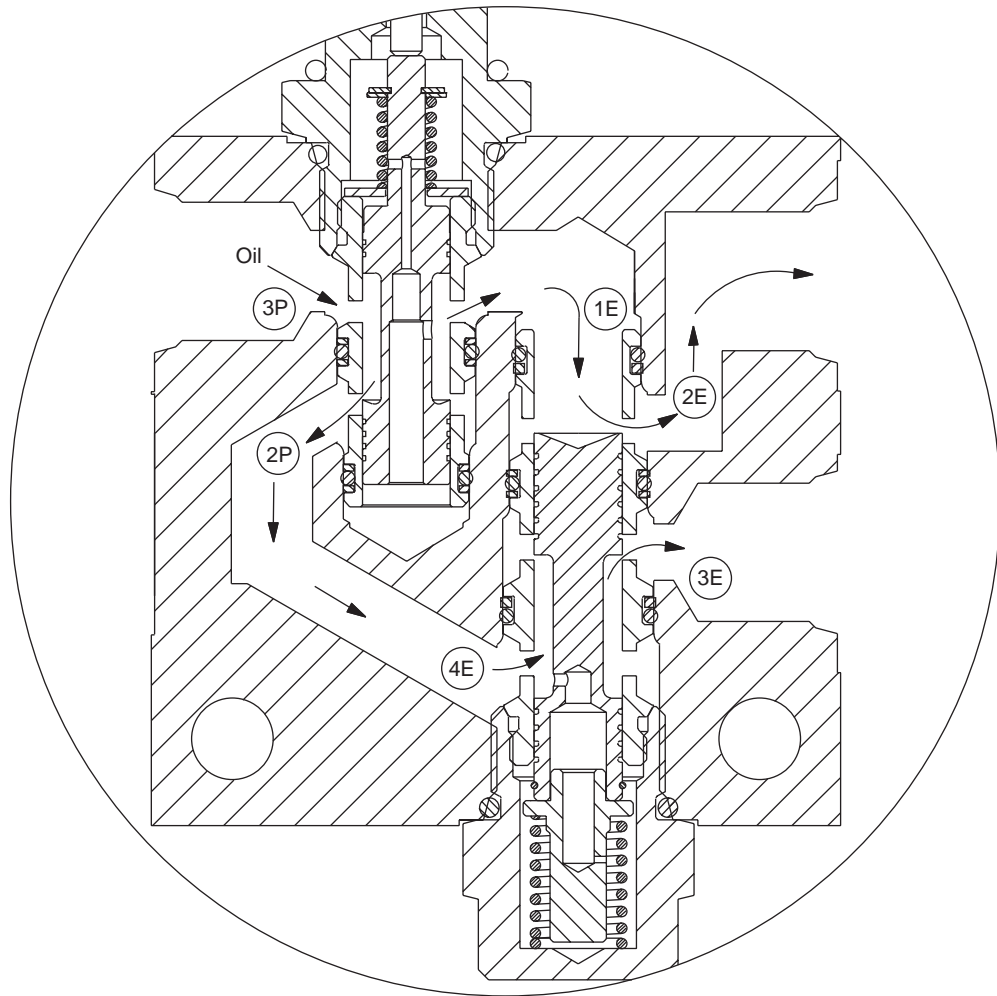


**Operation of PFR70-33-F**

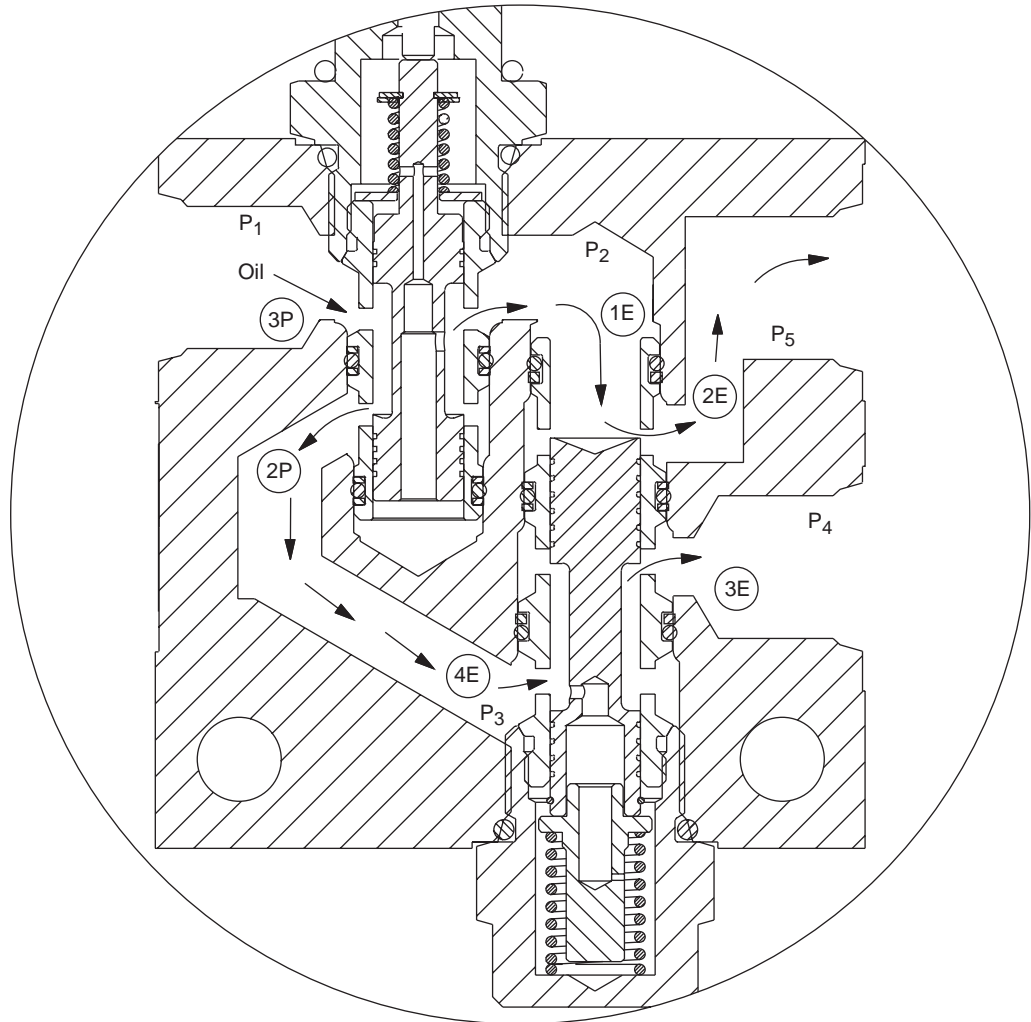
The first diagram used to describe the operation of the PFR70-33-F is shown below. As with the PFR70-33-E, when no current is applied, oil does not flow through port 3P to port 2P of the PV70-33. The oil flows in and around port 3P to port 1E of the EC10-40. The pressure at port 1E exerts a force greater than the spring force, causing movement of the EC10-40 spool. The spool compresses the spring until the spool movement is stopped by the adaptor. This is the extreme shifted position as also seen on the EC10-30 spool in the PFR70-33-E. The oil at 1E then exits through port 2E and out through the bypass port.



The next diagram shows the spool of the PV70-33 has moved because current was applied to the coil. Oil flows from port 3P to 2P of the PV70-33 and then to port 4E of the compensator. The oil continues to flow through the compensator, exiting port 3E or the priority hydraulic function. Any excess flow is diverted through port 1E and out of port 2E of the EC10-40 to the bypass or secondary hydraulic function.



The last diagram depicting the operation of the PFR70-33-F is shown below. In this diagram, the spool of the PV70-33 has uncovered more of the cage cross holes at port 2. This is because an even higher level of current has been applied to the coil than in the previous diagram. Corresponding to this increased opening is an increased opening at port 3E of the EC10-40. This implies that a lower amount of flow is bypassed downstream of port 2E of the EC10-40.



### Forces Acting on EC10-40 Spool

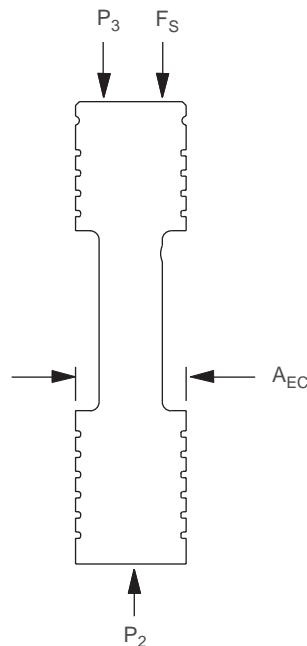
The amount the spool of the EC10-40 opens the cage cross holes at port 3E or the position of the spool is determined by the same equation given for the EC10-30. Again, the position is dependent on maintaining a differential pressure force equal to the force exerted by the spring. This is summarized in the diagram below.

The variables below are used to describe the forces acting on the EC10-40 spool:

$A_{EC} P_2$  = Pressure force acting on spool at Port 1  
 $A_{EC} P_3$  = Pressure force acting on spool at Port 4  
 $F_s$  = Spring force

The following equation describes the balance of forces acting on the EC10-40 spool:  $A_{EC} P_2 = A_{EC} P_3 + F_s$

The equation above is balanced whenever the inlet pressure force minus the outlet pressure force is equal to the spring force. This is summarized as follows:  $(P_2 - P_3) A_{EC} = F_s$ .





**Performance of PFR70-33-F**

The following graphs are used to describe the performance of the PFR70-33-F. These include the flow vs. current characteristic and the compensation characteristic of the priority port of the EC10-40.

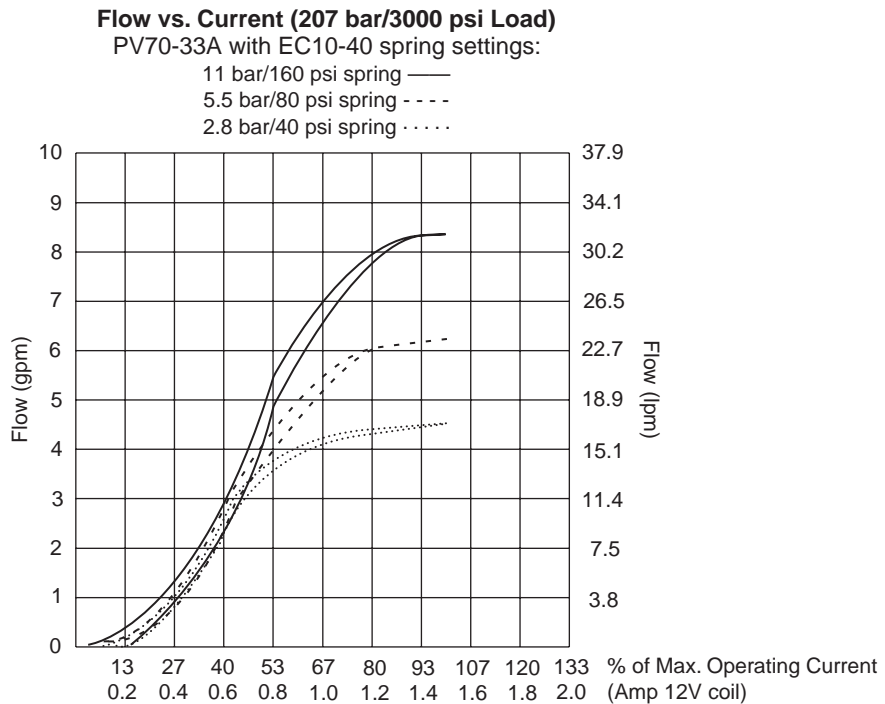
The flow vs. current characteristic is shown below. The following information can be determined from this graph:

- Flow Range: 0 - 8.25 gpm with EC10-40 160 psi compensation spring
- 0 - 6.25 gpm with EC10-40 80 psi compensation spring
- 0 - 4.25 gpm with EC10-40 40 psi compensation spring

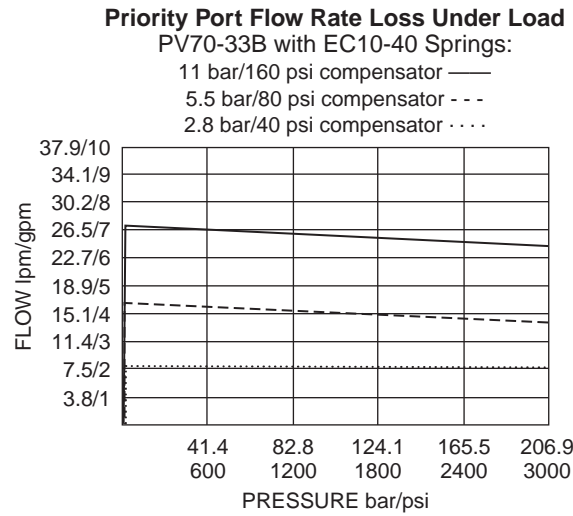
Hysteresis: < 6%

Threshold Current (I<sub>MIN</sub>): 13% of maximum operating current

- Saturation Current: 1.40 Amp with EC10-40 160 psi compensation spring
- 1.30 Amp with EC10-40 80 psi compensation spring
- 1.10 Amp with EC10-40 40 psi compensation spring



The flow vs. pressure characteristic for the EC10-40 when used in the PFR70-33-F is shown below.

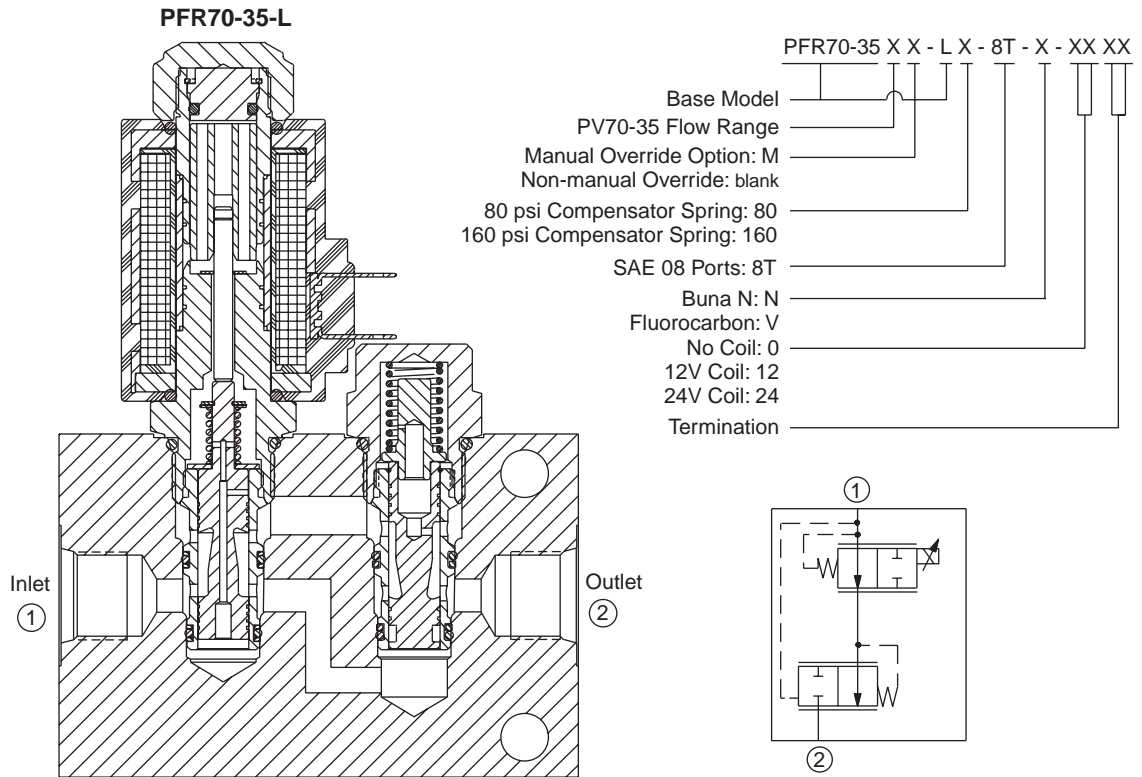


The curves above show *droop*. That is, they do not remain flat. This occurs because of a flow force acting on the compensator spool. Instead of the force balance equation consisting only of the spring force and pressure force, the flow force is added in as well. The new equation is  $F_s = (P_2 - P_3)A_E + F_{Flow}$ . This shows the spring force needs to work against the flow force as well.

**Other PFRs**

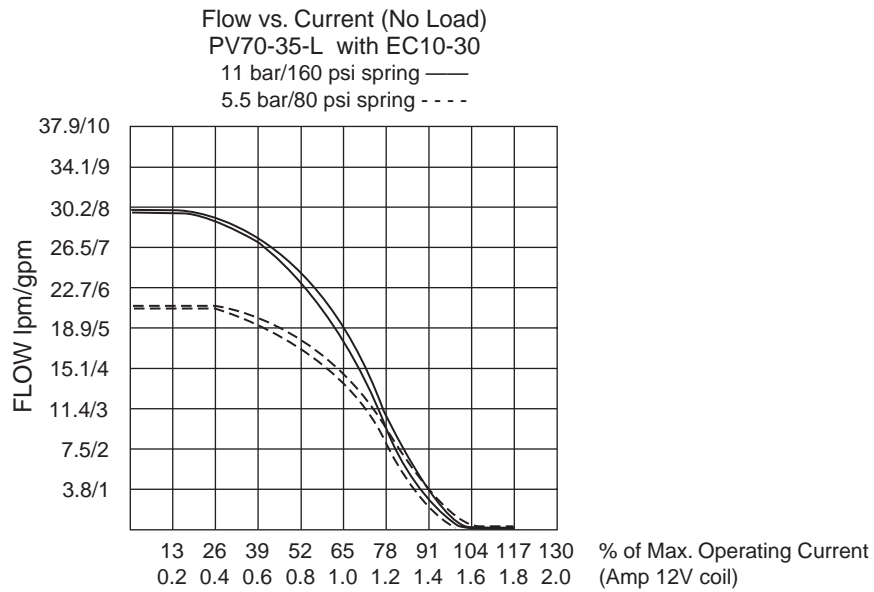
**PFR70-35-L**

Shown below is the cross section of the PFR70-35-L. It is considered to be a normally open, restrictive type proportional flow control valve. The flow vs. current performance is given on the following page. Since this combination valve is similar to the PFR70-33E, the operation can be obtained in the section describing it.



### Performance of PFR70-35-L

Shown below is the flow vs. current characteristic for the PFR70-35-L. When a 160 psi spring is used in the EC10-30, the threshold current for this assembly is 13%. When the 80 psi spring is used, the threshold current is 32%. The hysteresis, as in the PFR70-33, is less than 6%. The operation is similar to the PFR70-33-E. The only difference is that the current is increased to decrease the flow.



**PFR70-33-J**

This combination valve is similar to the PFR70-33-F in that it uses a PV70-33. However, the compensator is the larger EC12-40. With this combination, the bypass flow that can be reached is 11 gpm when the compensator spring is either 100 psi or 160 psi. This is in comparison to the maximum bypass flow of 7 gpm with the EC10-40 and either compensator spring value of 80 psi or 160 psi. Also, the larger compensator spool may provide improved system stability in some applications.

**PFR72-33 and PFR72-35**

There are similar combinations using 12 size direct acting proportional valves. The advantage to these valves is that they allow for a higher regulated or priority flow as compared with the PV70-33 or PV70-35. The operation is similar to that of the 10 size product. The typical maximum regulated flow is 17 gpm with the 160 psi spring in the compensator. Again, the maximum bypass flow with the EC12-40 is 11 gpm.

## Summary

In this chapter the following concepts were presented:

- A description of the operation of a direct acting, non-compensated flow control.
- Why a flow vs. current characteristic is not given for the non-compensated flow controls.
- A definition of the flow force.
- How HydraForce designs to compensate for the effect of flow forces.
- Why the non-compensated flow controls require an external compensator.
- A definition of compensating pressure.
- How the PV70-33 works with either an EC10-30 or EC10-40.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. What are the forces acting on the components of the PV70-33? \_\_\_\_\_
2. Why is the PV70-33 typically used with an external pressure compensated flow control? \_\_\_\_\_
3. What maximum differential pressure can be applied across a PV70-33 or PV70-35? \_\_\_\_\_
4. Define the compensation value. \_\_\_\_\_
5. Describe the operation of the manual override available for the PV style of valves. \_\_\_\_\_
6. What two factors determine the maximum flow rating of a PV70-33? \_\_\_\_\_
7. What are some of the reasons to use an EC12-40 priority/bypass compensator over an EC10-40 when a PV70-33 is used? \_\_\_\_\_
8. What is the principle difference between a PV70-33 and a PV70-35? \_\_\_\_\_
9. If the load down stream of the PFR70-33- E suddenly changes from a requirement of 2000 psi to 1500 psi and the inlet pressure is 3500 psi, what happens to the flow seen by the load? (Assume that the current remains unchanged.) \_\_\_\_\_

**Notes:**



# Chapter 9: Internally Compensated Flow Control

## Objectives

- Learn about the restrictive internally compensated flow controls.
- Learn about priority bypass internally compensated flow controls.
- Discover how a priority bypass flow control can be made into a restrictive type.
- Learn how the flow force affects the compensation characteristic.
- Learn about the reverse flow check valve built into one of the restrictive style valves.

## Introduction

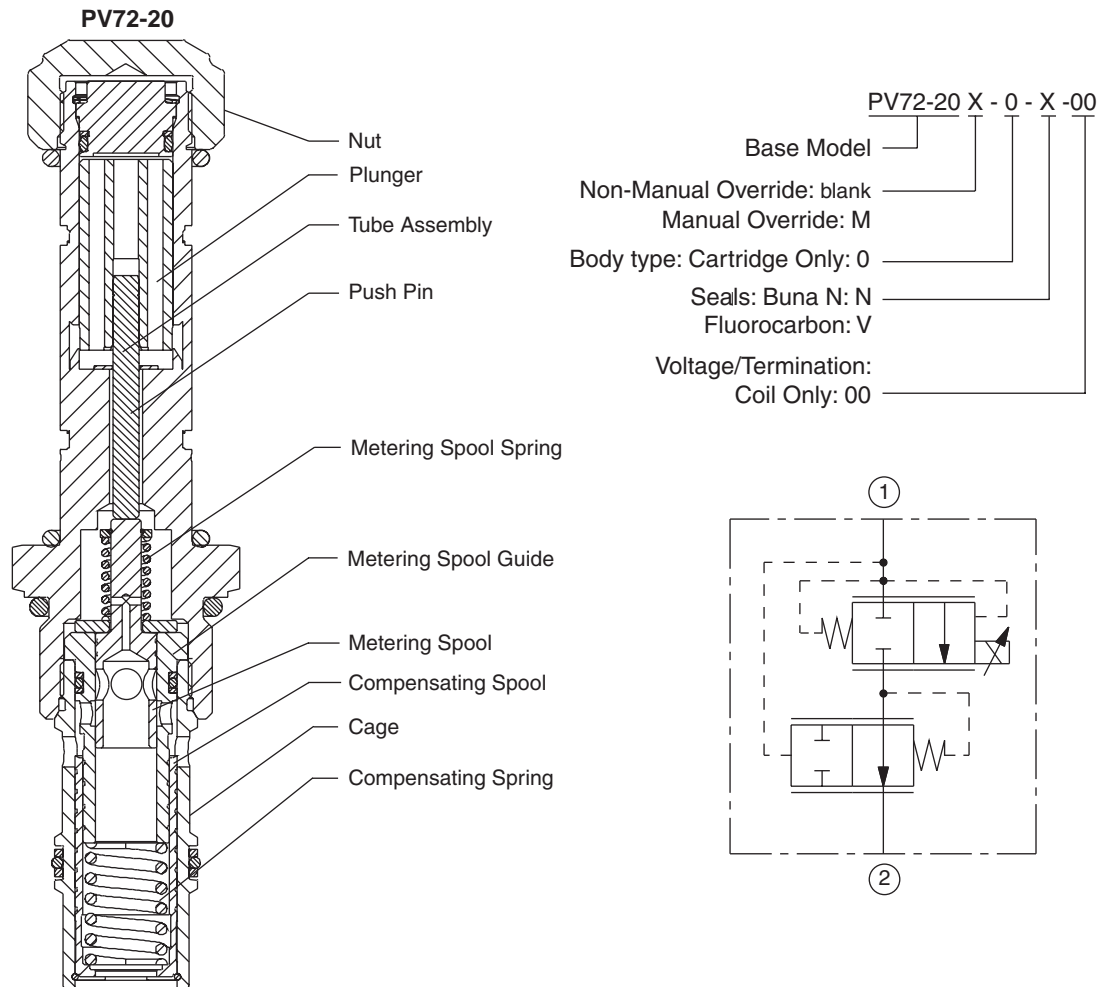
In this chapter we will learn about the proportional flow control valves that are internally compensated. That is, there is a compensating element or spool/spring combination that is built into these valves. This combination works similar to the one in the EC10-30 and EC10-40 which are described in the previous chapter. The operation, construction and function of both restrictive and priority/bypass type internally compensated proportional flow controls will be presented in this chapter.

# HYDRAFORCE

### PV72-20

The PV72-20 is a direct acting, internally compensated, normally closed, proportional flow control. The normally open version is the PV72-21. The valves are virtually identical except for the metering spool. The difference between a normally open and normally closed metering spool will be discussed in the PV70-30 section. The performance, forces acting on the valve, components and operation are described in the following sections.

HydraForce offers restrictive type, internally compensated flow controls in several cavity sizes. However, the PV72-20 is unique because it is the only valve in the range that is a 2-ported valve. In order to accomplish this function in other cavity sizes, one of the three ports in the PV70/76-30 style valves would be blocked. This will be further explained in the section describing the PV70-30. The PV72-20 is a two way valve designed specifically for restrictive flow control (which is a 2-way function). This implies that the valve was optimized for maximum flow handling capability.

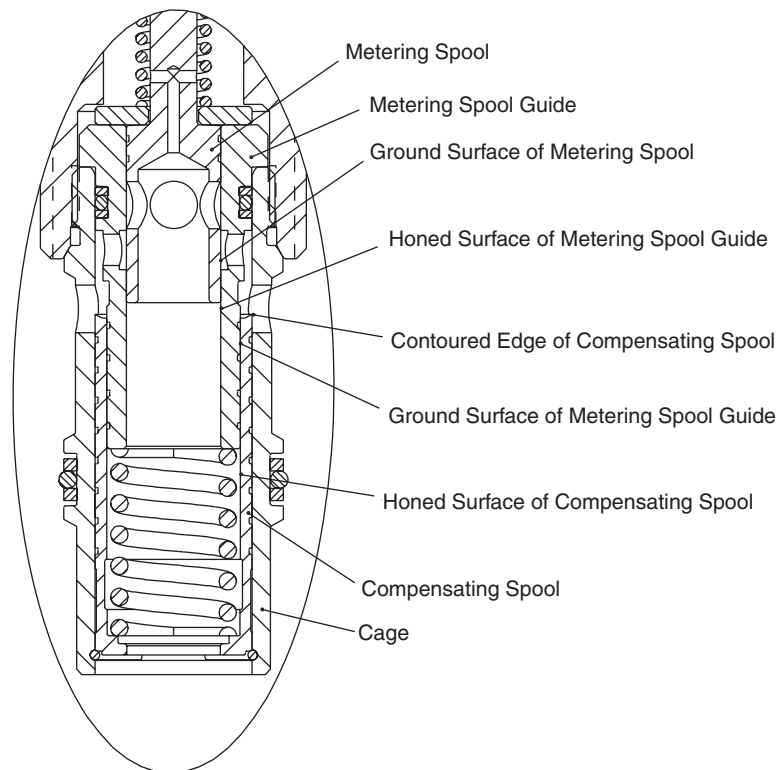


### Construction of PV72-20

The drawing below shows an enlarged view of the hydraulic components of the PV72-20. There are a few interesting features that are common to all the internally compensated flow controls. First, as expected, there are two spools. This is because the metering spool of the PV72-33 has been combined with the compensating spool of the EC12-30 into one valve.

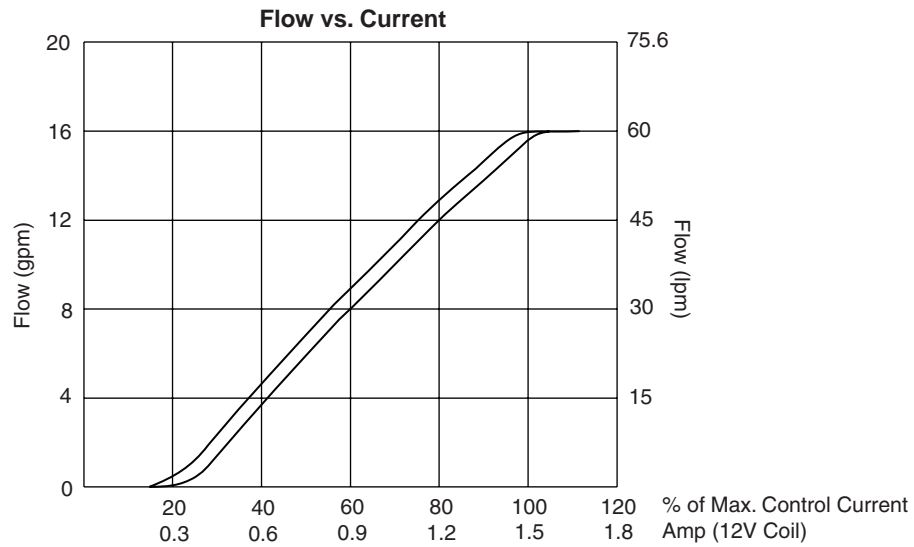
All of the hydraulic components described below have been heat treated. This provides a hard surface for wear resistance. The compensating spool, which rides inside the cage, has a ground outer surface and a honed inner diameter. The cage itself is also honed. The outer surface of the metering spool guide is ground, as is the metering spool. The inside surface, where the metering spool rides, is honed. These processes are done to allow a close fit between the parts to minimize the leakage from the inlet (port 1) to the tank (port 2).

There are two other features worth noting in the construction of the internally compensated flow controls. The first is that the metering spool guide floats in the assembly. This is to reduce the mechanical friction between the guide and compensating spool. If it did not float, it may rub on one side of the compensating spool rather than being centered. The last feature to note is the contoured edge of the compensating spool which minimizes the affects of flow forces. Without it the maximum flow may be limited to a lower value.



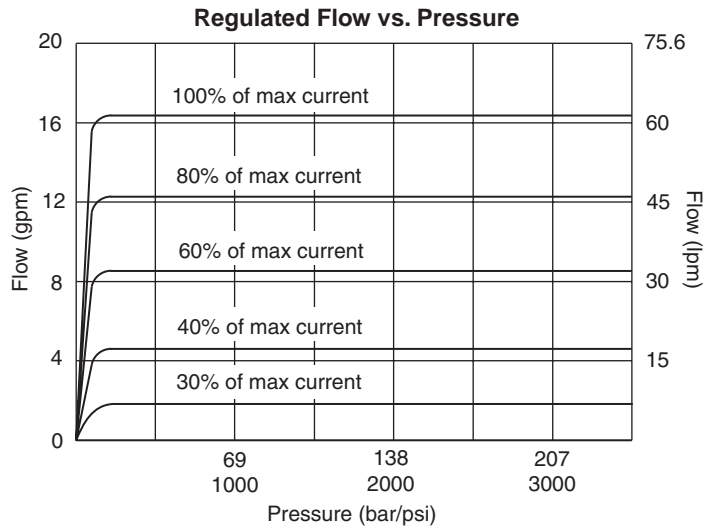
### Performance of PV72-20

Two primary characteristic graphs are presented in this section: the flow vs. current performance and the pressure compensation characteristic. The first graph shown below is the flow vs. current characteristic. The inlet pressure at port 1 was 3500 psi and the load pressure down stream of port 2 of the PV72-20 was 3000 psi. A current regulated electronic driver with the dither frequency set to 100 Hz was used to drive the current across the coil.

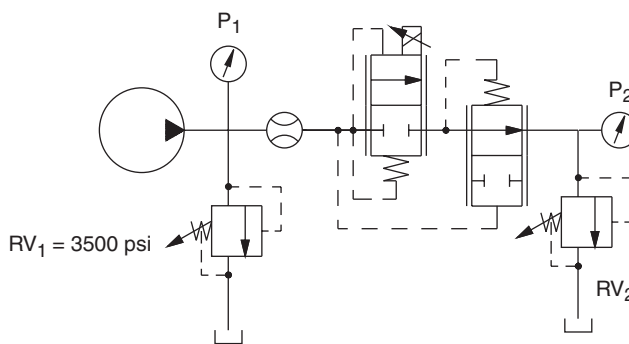


The graph above shows that the hysteresis is less than 6%. The flow range is 0 to 16 gpm. Saturation of the flow occurs at 1.5 amp or 100% of the rated current. Note that for the PV72-20 and PV72-21 the maximum control current is also the saturation current. For these two valves this value is 1500 mA for a 12V coil.

The graph below defines the compensation characteristic. Notice that the curves are flat or horizontal. This implies that regardless of a change in the upstream or downstream pressure, the flow remains constant based on the current applied. This is true as long as the difference in pressure between the inlet and outlet is above the compensation spring value, or approximately 245 psi. This is indicated by the bend in the curves where the flow characteristic changes from flat to a steep decrease in flow to zero.

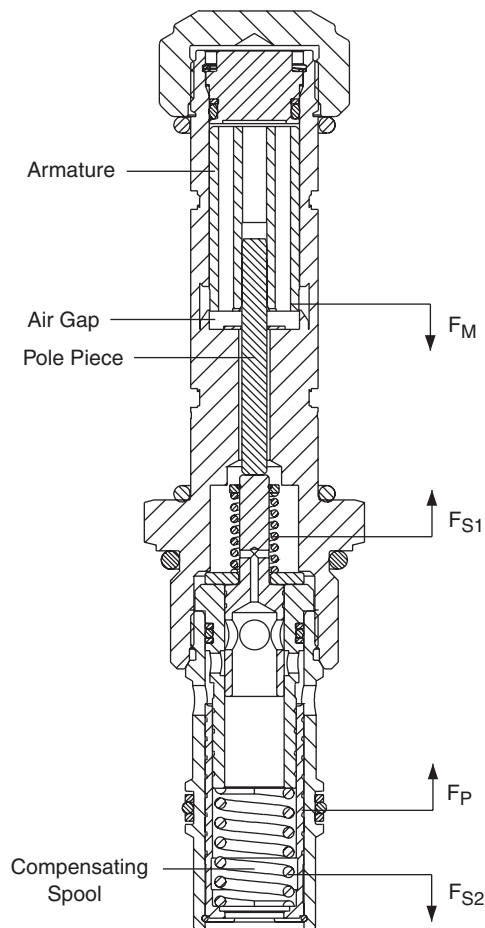


The following schematic depicts the hydraulic circuit used to develop the graph. The flow and pressure was recorded using an x-y plotter. The inlet pressure (RV<sub>1</sub>) was set to 3500 psi and the flow is set to one of the levels shown on the graph. The load relief valve (RV<sub>2</sub>) was initially set to the minimum setting. This gives the largest differential between P<sub>1</sub> and P<sub>2</sub>. The setting of RV<sub>2</sub> is slowly increased thus decreasing the difference between P<sub>1</sub> and P<sub>2</sub> until the flow and pressure go to zero.



### Forces on PV72-20

Shown below is the cross section of the PV72-20. The forces acting on the components are also labeled on the drawing. The direction the arrowheads face indicates the direction in which the force is acting. The spring forces  $F_{S1}$  and  $F_{S2}$ , like all the compression springs, oppose the movement of the components in the valve. In this valve these are the armature and compensating spool, respectively. Also, a description of each force is provided below.



Where:

$F_M$  = Magnetic force

$F_{S1}$  = Metering spool spring force

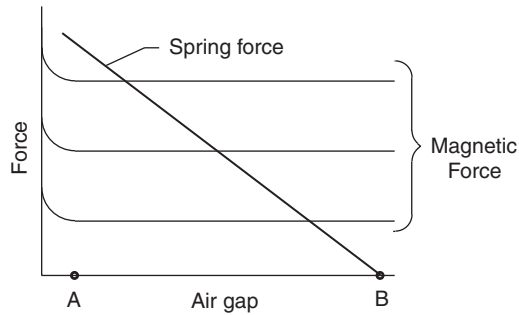
$F_P$  = Compensator spool pressure force

$F_{S2}$  = Compensation spool spring force

As with the non-compensated proportional valves presented in Chapter 8, the consideration of friction and viscous damping forces have been omitted because these forces are typically much smaller.

The magnetic force and the spring force acting on the metering spool are the same as those defined in Chapter 8. As for the spring and pressure force acting on the compensator spool, the forces defined for the EC10-30 are similar. All four of these forces will be briefly reviewed in the following section.

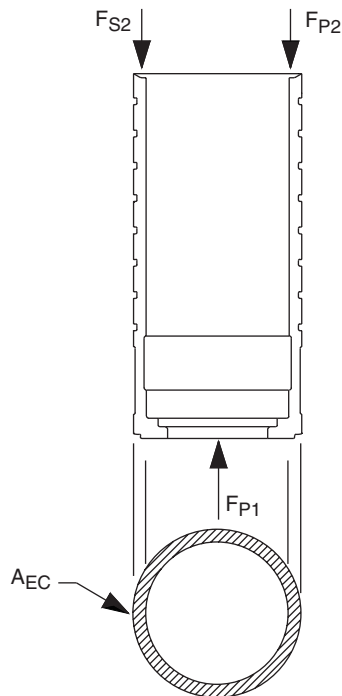
The metering spool spring force and the actuator magnetic force are briefly viewed in the graph below.



A = Position of armature closest to pole piece  
 B = Position of armature furthest from pole piece

The spring force is dependent on the stiffness of the spring and how much the spring is compressed. The magnetic force is dependent on the amount of current applied to the coil, but not on the position of the armature. This implies that the force balance equation for the metering spool is  $F_M = F_s$ . In other words, the magnetic force is balanced by the spring force, or, for each current level which creates a given level of magnetic force, there is an equal and opposite spring force.

The other two forces of concern in the PV72-20 are the compensator spring force and pressure force acting on the compensation spool. The compensator spool is shown below with the direction these forces act, as well as the area on which the pressure acts.



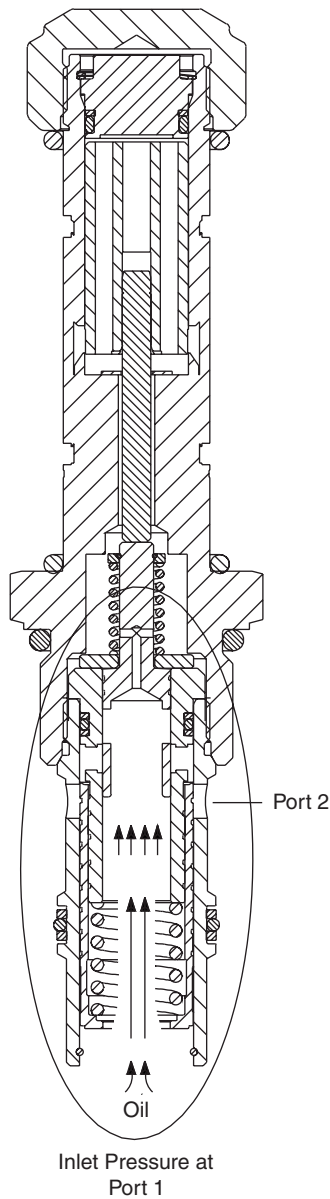
Where:

- $A_{EC}$  = Area affected by pressure
- $F_{S2}$  = Compensator spring force
- $F_{P1}$  = Force due to the inlet pressure acting on  $A_{EC}$
- $F_{P2}$  = Force due to the load pressure acting on  $A_{EC}$

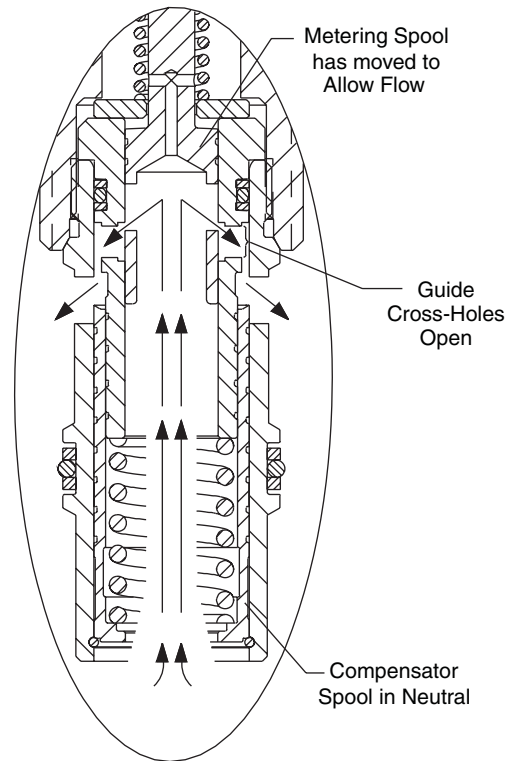
As with the compensator spools in previous chapters, the force balance equation is  $F_s = F_{P1} - F_{P2}$ . Again, this equation shows that the spool will not move or compensate for changes in load pressure until the difference between the two pressure forces exceeds the spring force.

### Operation of PV72-20

The operation of the PV72-20 is similar to the PV70-33-E. The function of the valve is to give a constant flow for a given current level regardless of the load pressure. The first drawing to the left shows the metering spool in the neutral position. In this case the current has not been applied to the coil. Notice that the compensator spool has moved to block off the cage holes at port 2. This occurs because there is no load pressure to oppose the force created by the inlet pressure.



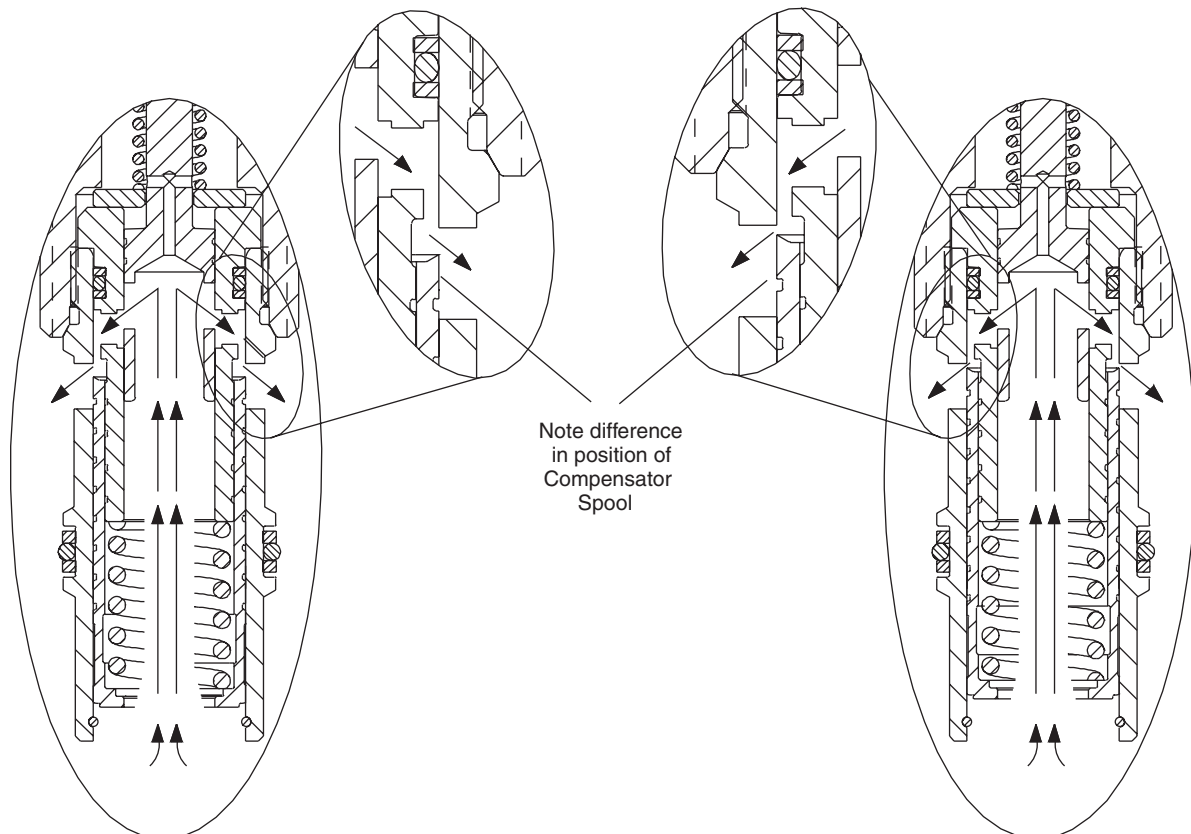
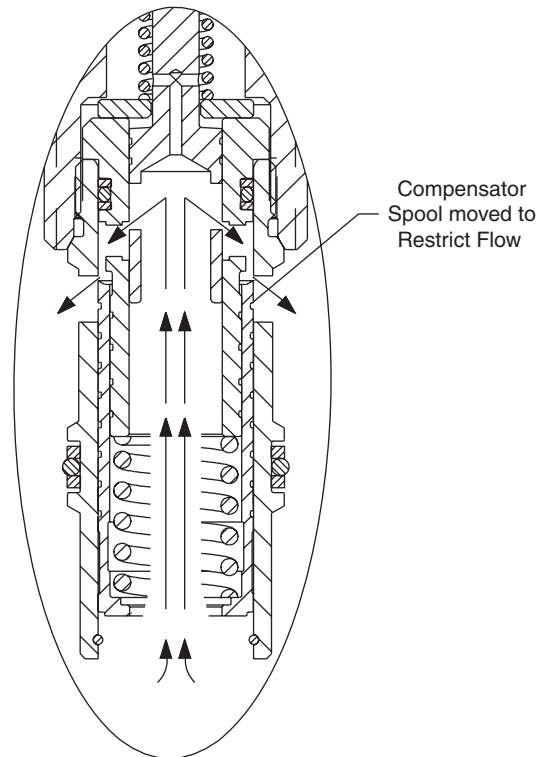
The next diagram shows that the metering spool has opened or uncovered some of the cross holes in the guide. Notice however that the compensating spool has moved back to the neutral position. This occurred because the pressure drop (difference between the inlet and load) is less than 240 psi or less than the compensating value.





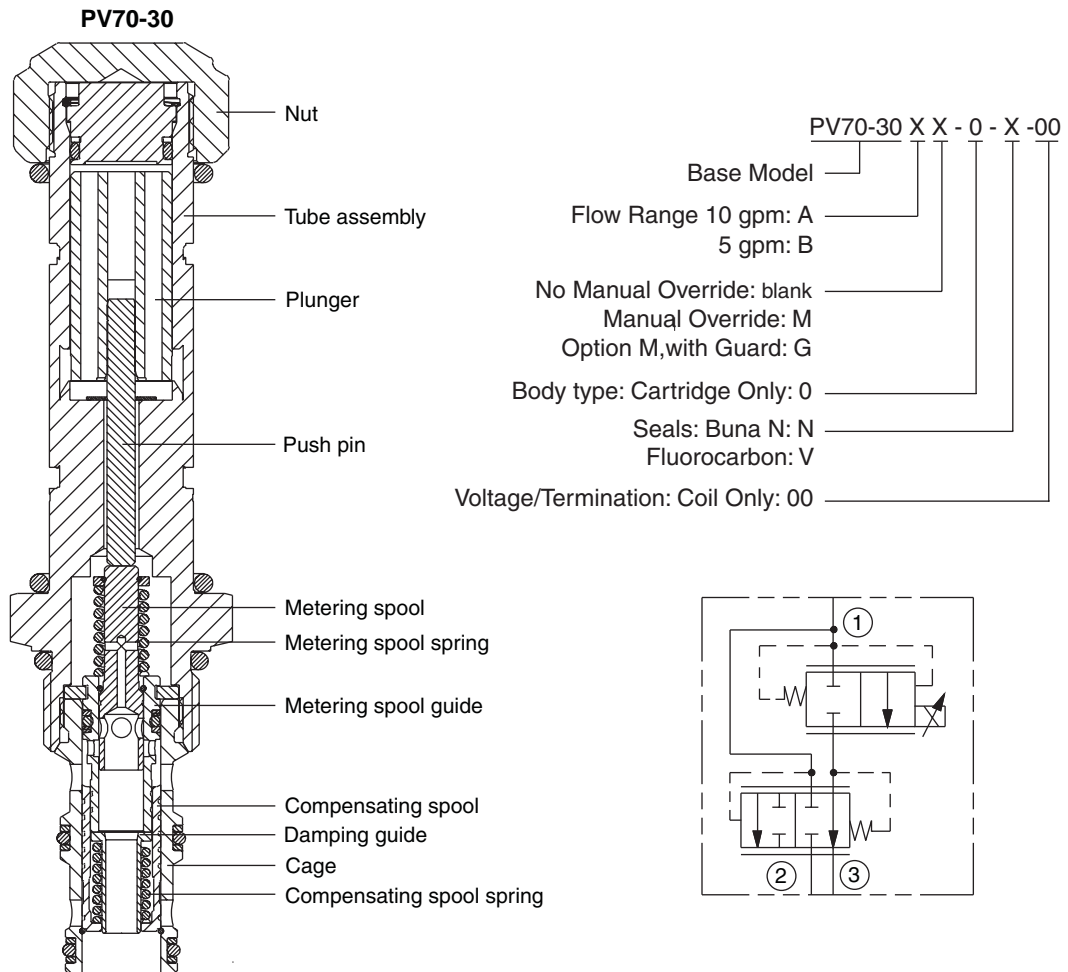
The diagram to the right shows that the compensating spool has moved to restrict the flow of oil to the load. As expected, this happened because the pressure drop across the inlet and outlet has exceeded the compensating spring value.

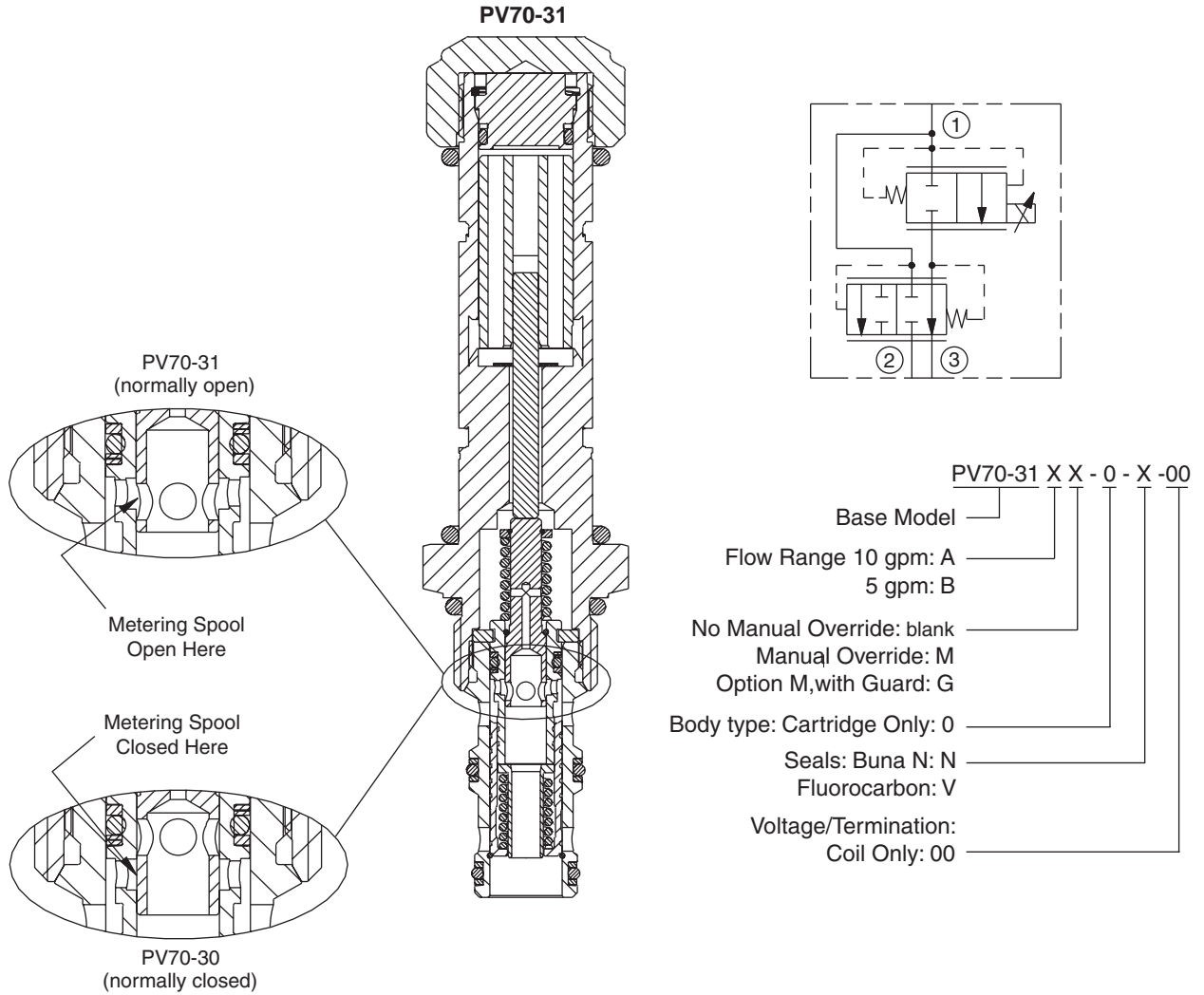
In the next two drawings shown below, the metering spool has moved even further to allow more flow. The difference between the two diagrams is the position of the compensating spool. The diagram to the left shows the compensating spool moved down further, opening more of the cage cross hole compared to the drawing on the right. The opening of the cage cross holes created by the edge of the compensating spool is less because the load downstream is less than the one to the left. This illustrates that the operation of the PV72-20 is the same as the PFR70-33-E. That is, if the load decreases, the compensating spool moves to assure the flow rate is maintained.



### PV70-30

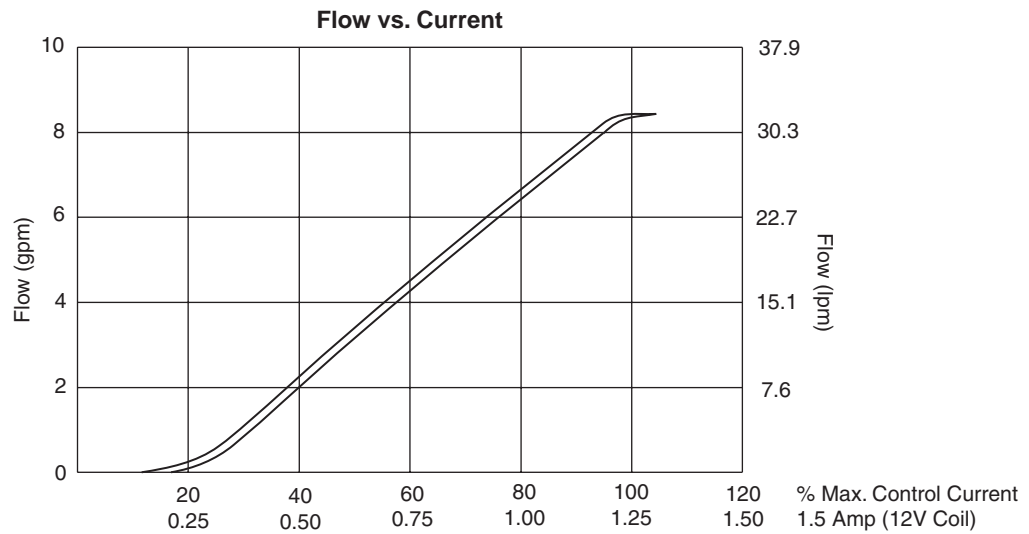
The PV70-30 is a normally closed, internally compensated, proportional flow control valve. The operation of this valve is similar to the PFR70-33-F described in Chapter 8, however it is one cartridge rather than two. The cross section of the valve is shown below. The normally open version, PV70-31, is shown on the following page. The difference between the two is the metering spool. The comparison of these two spools is also shown on the following page. Both the PV70-30 and PV70-31 can be used as either a restrictive type flow control or a priority/bypass style. These valves can be used as restrictive type flow controls by blocking port 2. The performance difference between the 2-way mode and 3-way mode, as well as forces acting on the components and operation of the PV70-30, are presented in the following sections.





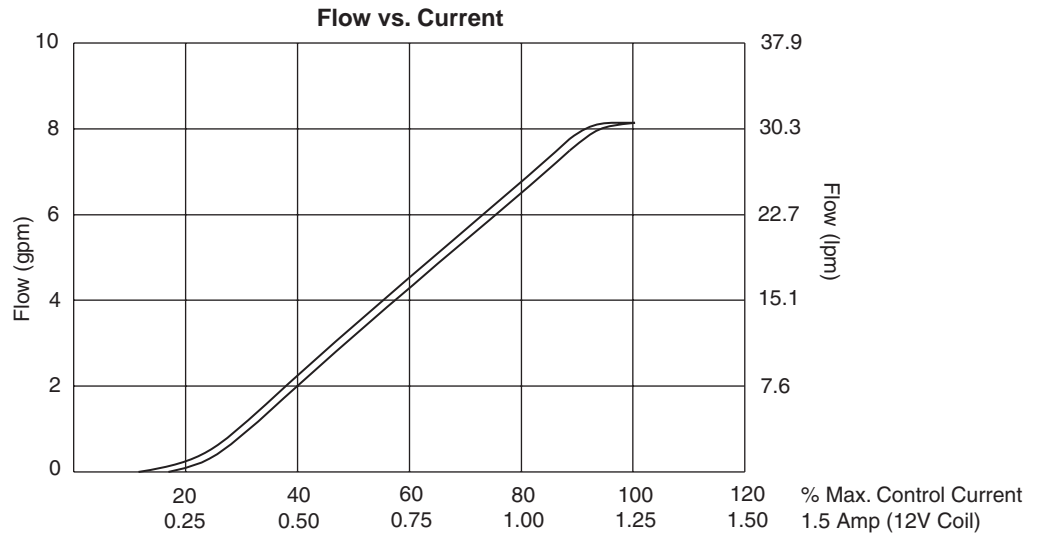
### Performance of PV70-30

The graphs below represent the performance of the PV70-30. The first graph is the flow vs. current characteristic when the valve is used as a restrictive type flow control. Recall that at a restrictive type flow control is a two port device with an inlet and a regulated work port. The load pressure was held at 3000 psi while the current was varied in the graph below. Also, the dither frequency was set to 110Hz.



The graph above shows that the regulated flow range of the PV70-30A when used as a restrictive flow control is 0 to 8.5 gpm. The threshold current is 20% and the saturation current is 95% of the maximum current. The hysteresis is 6%.

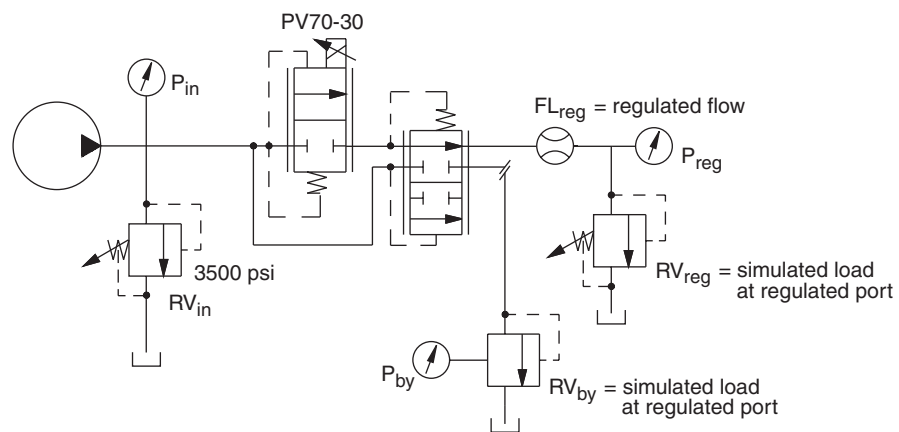
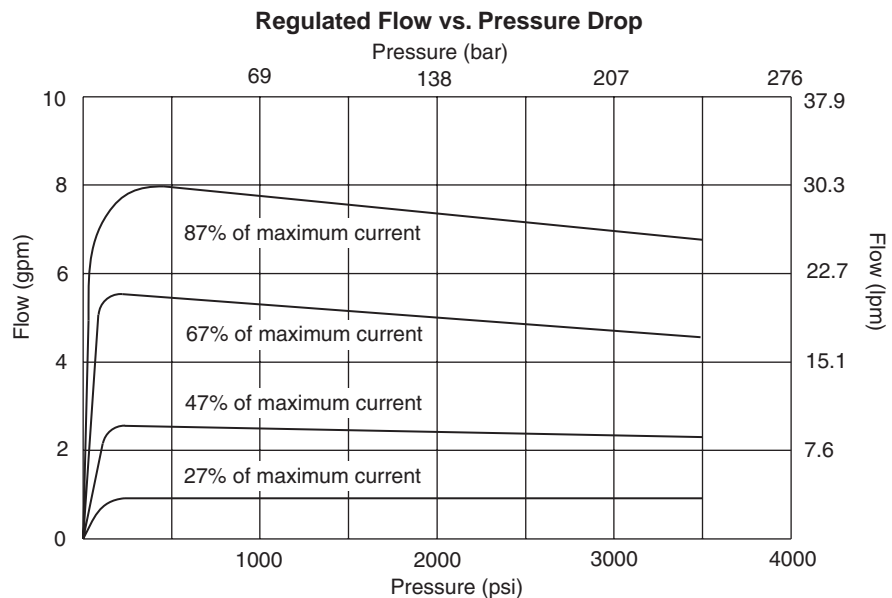
The next graph shows the flow vs. current characteristic of the PV70-30A when used as a 3-port priority/bypass flow control valve. As with the previous graph, the load pressure was 3000 psi and the dither frequency was 110Hz while the current was varied.



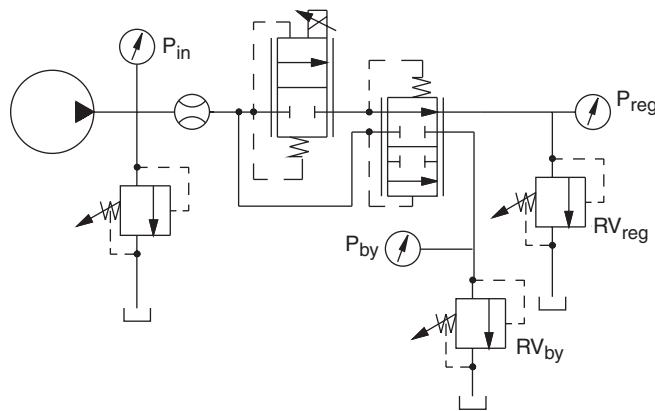
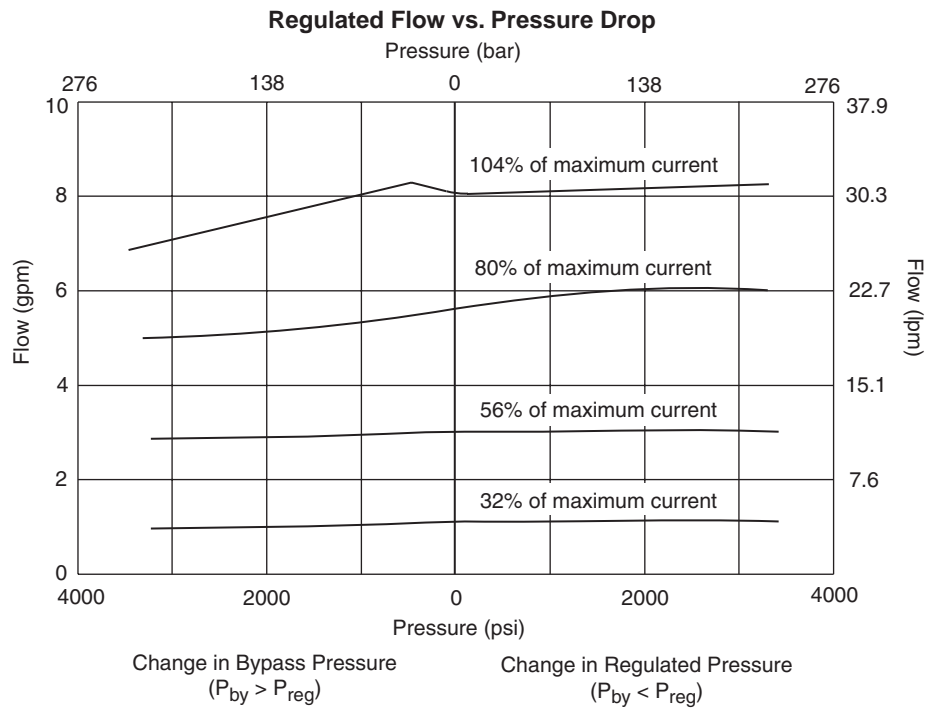
The hysteresis and threshold current are the same as the previous graph. However, the saturation current is 90% of the maximum control current. Note that for the PV70-30 and the PV70-31, the maximum control current corresponds to the saturation current when the valve is used as a two port restrictive type valve. That is, when the valve is a 3-way bypass style, the flow saturates at a lower current. The saturation current is 1250mA for the 2-way mode and 1200mA for the 3-way mode. This is less than the 1500 mA given for the PV72-20. A further explanation of the maximum allowed current for the proportional flow controls will be given in Chapter 14.

The following graphs represent the compensation characteristic. The first one shows the compensation characteristic for the PV70-30A when it is used as a two-port restrictive type valve. The test schematic used to develop these curves is also shown below the graph. It is similar to that of the one used for the PV72-20.

The relief valve used to simulate the load,  $RV_{REG}$ , is initially set to the lowest possible setting. This would cause the difference between  $P_{REG}$  and  $P_{IN}$  to be approximately 3500 psi. This is indicated by the pressure recorded to the far right of the graph. The setting of  $RV_{REG}$  is increased until the flow at  $FL_{REG}$  is reduced to zero. This is because there is insufficient pressure to overcome the setting of  $RV_{REG}$ . The droop shown in the top curve from 8.00 gpm to 6.5 gpm will be explained further in the operation section.



The next graph is the compensation characteristic for the priority/bypass function of the PV70-30A. The difference between this and the previous schematic is the additional load relief valve  $RV_{BY}$ , connected to port 2. The graph below shows how the regulated flow varies with a change in pressure. The left side shows how well the desired regulated flow is maintained, while the pressure in the bypass line changes. The right side shows the change in regulated flow due to varying pressure in the regulated line.

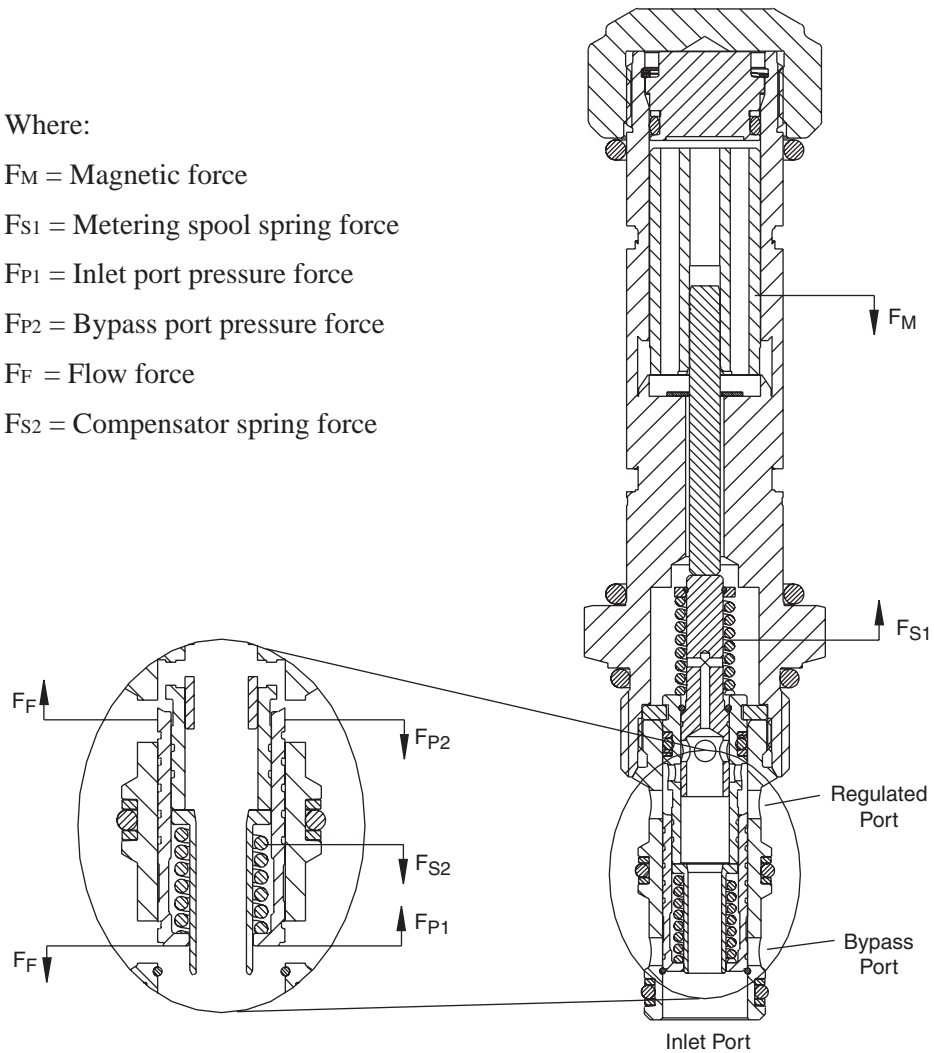


### Forces of PV70-30

The diagram below shows the cross section of the PV70-30A and the forces acting on the components. Notice that these are all similar to the ones described for the PV70-20 except for the flow force and damping force indicated on the compensation spool. For a description of all other forces except the flow force and damping force, refer to the PV72-20. The damping force and flow force are described in previous chapters.

Where:

- $F_M$  = Magnetic force
- $F_{S1}$  = Metering spool spring force
- $F_{P1}$  = Inlet port pressure force
- $F_{P2}$  = Bypass port pressure force
- $F_F$  = Flow force
- $F_{S2}$  = Compensator spring force

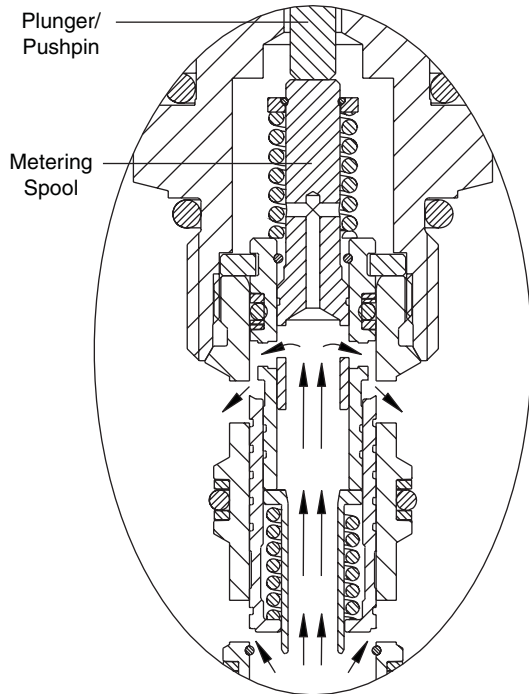




### **Forces acting on Compensating Spool**

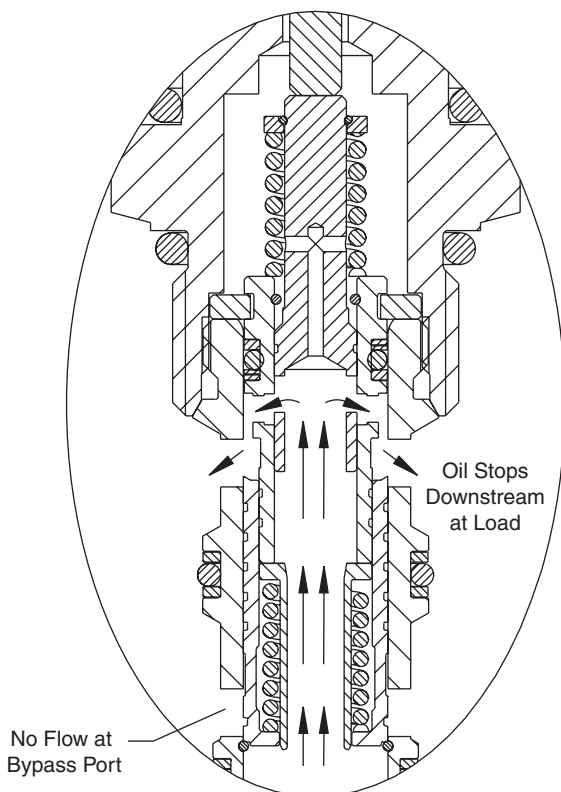
The diagram on the previous page shows a cross-section of the compensating spool and cage. The forces acting on the spool are labeled. As with the compensating spool in the PV72-20, there is a spring force and a pressure differential force. A third force shown is the flow force. A full description of the flow force was presented in the previous chapter. The flow force is noted here in order to describe why the compensation curve is not flat, or droops. This is when the current applied is greater than 65% of  $I_{MAX}$ . As with the other compensating spools, its position relies on the compensator spring force being balanced by the pressure force or  $F_{S2} = F_{P1} - F_{P2}$ . However, the flow force can also influence the position of the compensating spool. In most compensators, the flow force is not as large as the other two and therefore is omitted for simplicity. In the case of the PV70-30, where HydraForce tried to maximize the available regulated flow, the flow force is almost as large as the other two forces. Notice that the flow force acts on the spool where oil exits at both port 2 and port 3. In other words, there is a flow force acting on the spool when oil flows through the regulated port and the bypass port. The force balance equation is  $F_{S2} = (F_{P1} - F_{P2}) + F_F$ . This equation shows that as the flow force increases, the compensator spring will need to increase. Recall that the spring force is determined by how much it is compressed. When the compensator spool moves, the compensator spring is compressed. Thus balancing the flow force. As this occurs, the compensating spool closes off the cage cross holes at port 2, thus restricting or reducing the flow expected. This is shown on the graph as droop.

### Operation of PV70-30



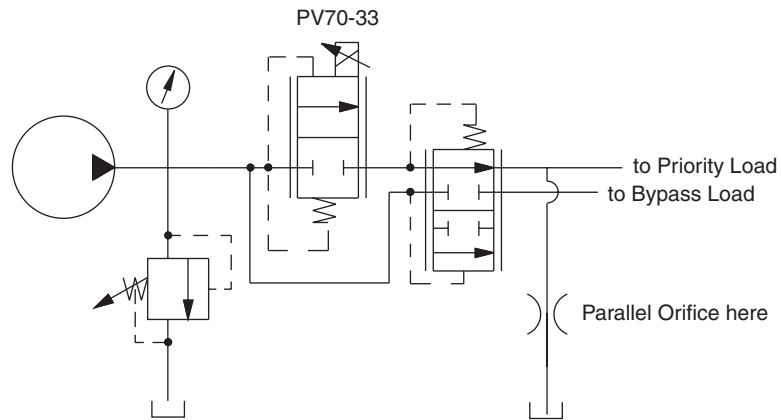
In this section, various operating modes of the PV70-30 will be described. The first diagram shows that the plunger/pushpin is pushing down on the metering spool because current is applied to the coil. As with the PV72-20, the compensating spool is actively metering the flow at the regulated port. Also, notice that the excess flow is being bypassed through port 2.

This is the typical operation of the PV70-30. That is, a current level is selected causing the metering spool to move to a defined position. The compensating spool then moves in response to a change in pressure. This assures a constant pressure drop across the metering spool. The compensating spool of the PV70-30 begins to regulate at a differential pressure of 190 psi. The oil is flowing out of the regulated and bypass port. In order to achieve the maximum rated regulated flow given in the performance section, the inlet flow must be 10-20% greater.



The next diagram shows what occurs if the flow downstream of port 3 suddenly stops. This could happen if a cylinder reaches the end of its stroke. Notice the metering spool is still in the shifted position, as shown in the previous diagram. The compensating spool has moved back to its original at rest position, closing off the bypass port. The reason is related to the forces acting on the compensating spool. As noted in the previous section, the compensating spool moves in response to changes in the inlet pressure, outlet pressure, flow force and the spring force. Recall that the force balance equation is  $F_s = (P_{IN} - P_{OUT}) A_{SP} + F_F$ . When flow is stopped, because the load is no longer moving, the difference between  $P_{IN}$  and  $P_{OUT}$  is zero. This implies that there is no force to balance the spring force. Therefore, the spring pushes the compensating spool until it stops on the cage. This in turn causes the bypass flow to go to zero and all the flow will then pass through the system relief.

The operation of the compensating spool closing off the bypass port will also occur with the EC10-40 when used with the PV70-33 as discussed in Chapter 8. If continuous flow is required to the bypass, regardless if the flow to the regulated port ceases, it is recommended to install a small orifice in parallel with the regulated load, as shown in the schematic below.

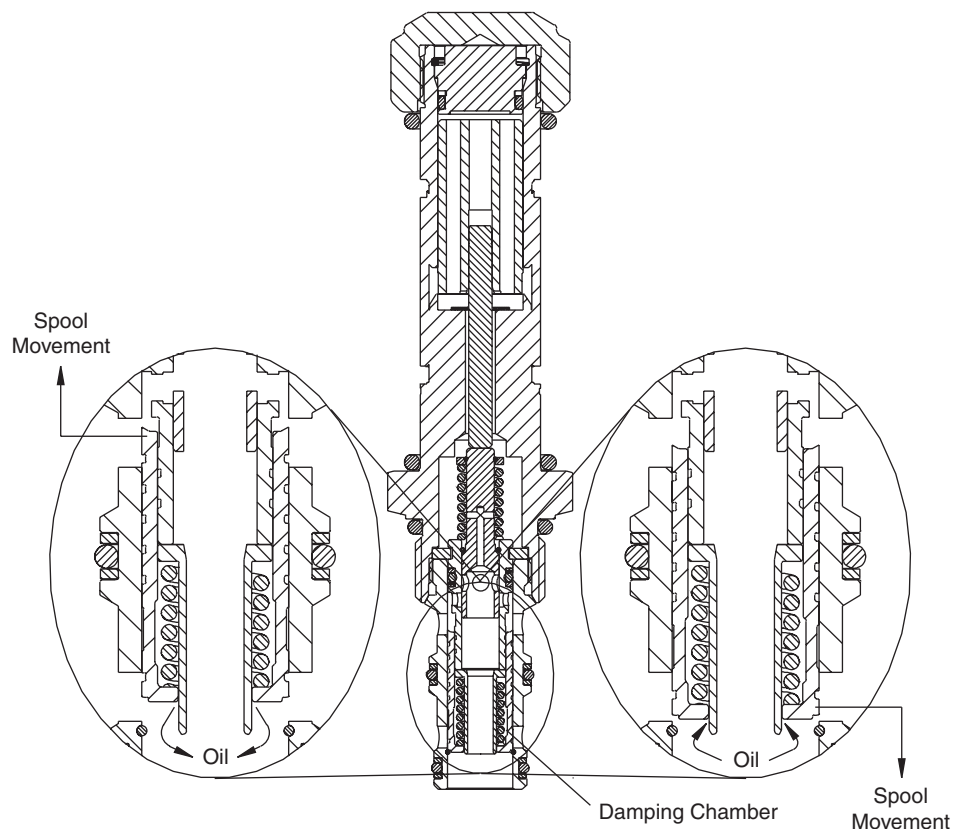


An 0.020 orifice is recommended to be used with the PV70-30, PV70-31 and PV70-33. An 0.031 orifice is recommended to be used with the PV72-30, PV76-30, PV72-31, PV76-31 and PV72-33. Alternatively, a relief valve could be installed in place of the parallel orifice. The setting of this relief valve should be lower than the setting of the system relief, but higher than the pressure required to move to load.

## Damping Chamber Operation

Inside the priority/bypass internally compensated flow control valve is a damping chamber. The damping chamber is also the area that holds the compensating spool spring. This chamber dampens or slows the movement of the compensator. Its function is to prevent the spool from oscillating, which can occur due to fluctuations in the output of flow from the pump (known as *pump ripple*). The fluctuations in flow may also occur because of the stopping and starting movement of an actuator. This is caused by the mechanical friction of the piece of equipment in which the valve and actuator are installed.

When the compensator spool moves to regulate flow the damping chamber is in operation. As it moves, oil is pumped in and out of the damping chamber. When the compensating spool moves up, as shown in the left hand diagram below, oil is pushed out of the damping chamber. When the damping spool moves down, oil is drawn into the damping chamber as shown on the right.



The clearance between the compensating spool and guide determines the amount of damping. This clearance is balanced to allow the compensator to react to changes in load pressure, but to ignore small oscillations due to pump ripple.

## Summary

In this chapter the following concepts were presented:

- The performance of restrictive style internally compensated flow controls.
- The performance of the priority/bypass internally compensated flow controls.
- The reason for droop in a compensation curve.
- Basic construction of an internally compensated flow control.
- How a priority/bypass flow control can become a restrictive style flow control.
- Bleed orifice requirements for no flow priority/regulated port such as at the end of a stroke condition.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. What type of flow control is the PV72-20? \_\_\_\_\_
2. Which port of the PV70-30 is blocked to make it a restrictive type flow control? \_\_\_\_\_
3. What does the compensating spool do if the load pressure suddenly changes? Assume that the difference between the inlet pressure and load pressure is greater than the compensating pressure and the current remains constant. \_\_\_\_\_
4. What is the difference between the PV70-30 and PV70-31? \_\_\_\_\_
5. How many spools are inside the internally compensated flow controls? \_\_\_\_\_
6. True or False. Inside the internally compensated flow controls, the magnetic force is balanced by the metering spool spring force. \_\_\_\_\_
7. What purpose does the damping chamber serve? \_\_\_\_\_

# Chapter 10: Bidirectional Flow Controls

## Objectives

- Learn about the patented, bidirectional flow control valves.
- Understand how the flow force affects the performance of the ZL70-36.
- Learn why there is a small pressure sense tube used in the ZL70-30.
- Understand the differences between the ZL70-33 and the ZL70-36 valves.

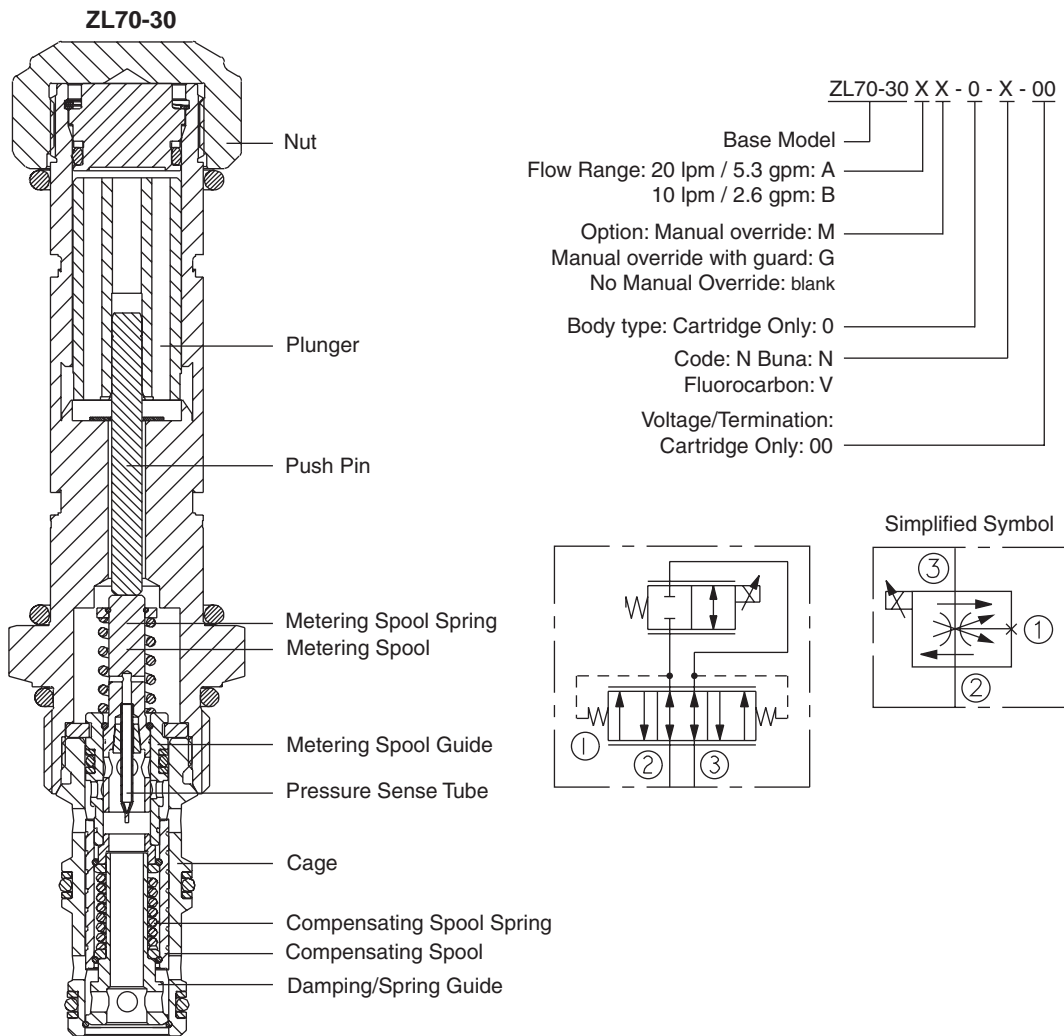
## Introduction

This chapter introduces the proportional bidirectional flow control valve for which HydraForce has obtained two patents. The first patent protects the invention of the bidirectional, internally compensated flow control. This is a restrictive style two-port device. The second patent protects the invention of a unique, internally compensated flow control specifically designed for use with a single acting cylinder. In one direction the valve acts as a priority/bypass flow control and in the other it acts as a restrictive type. The operation, performance and forces acting on the valve components will be presented. The application of these valves will be presented in the following chapter.

# HYDRAFORCE

### ZL70-30

The ZL70-30 is a bidirectional, normally closed internally compensated, proportional, restrictive type flow control. Like the PV70-31, the ZL70-31 is the normally open version of this valve. The cross section, model code structure and two symbols are shown below. The first symbol separates the compensating element and the variable metering orifice. The second symbol was created for ease in drawing of a hydraulic circuit. The performance, forces acting on the valve components and operation will be presented in the following sections.

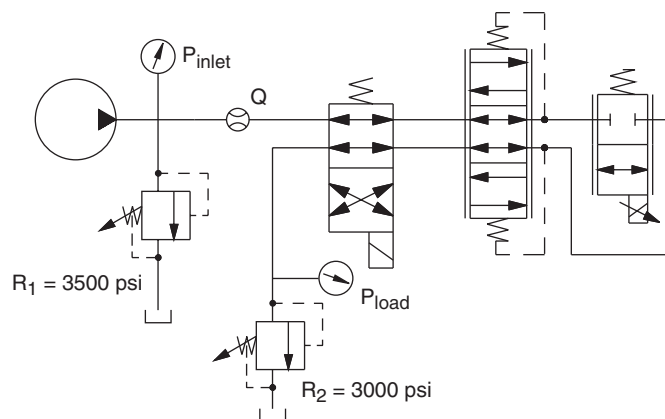
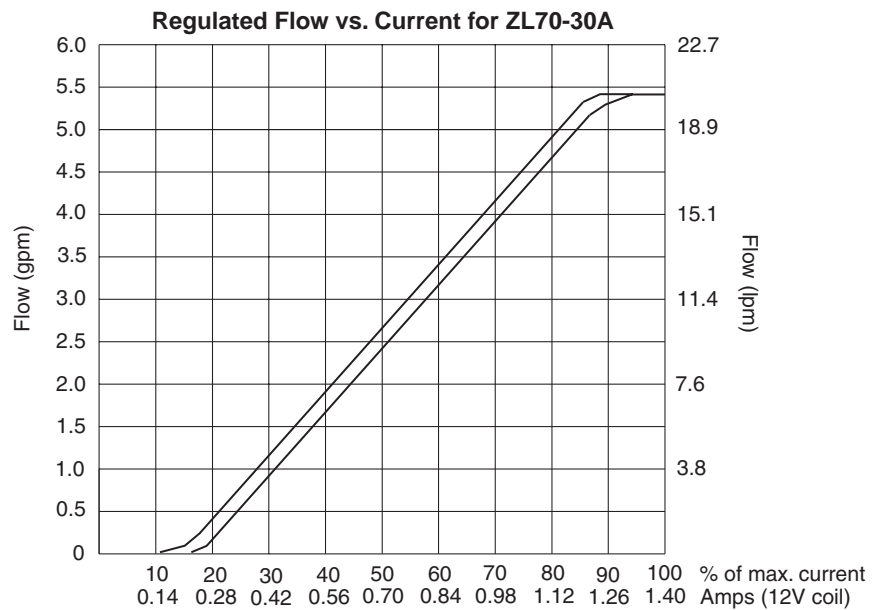




**Performance of ZL70-30**

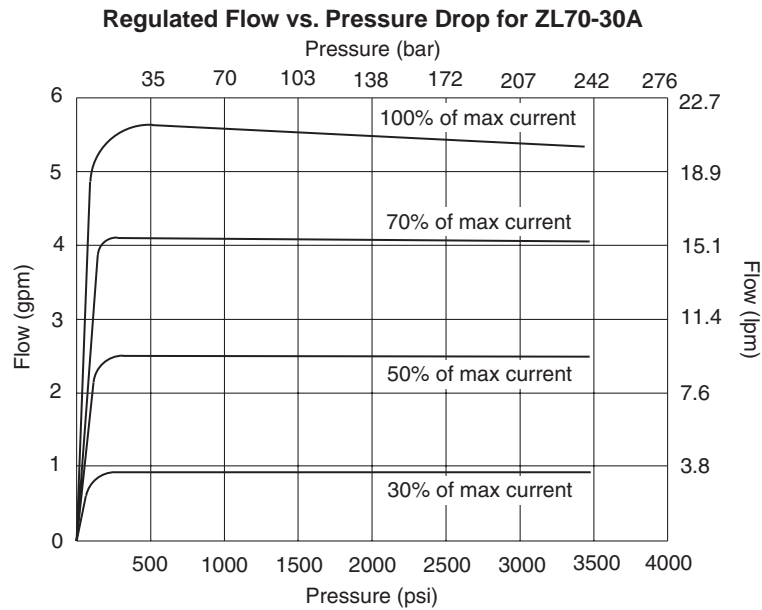
This section presents the two curves which describe the performance of the ZL70-30. These are the same type of graphs used to describe the performance of the restrictive type flow controls presented in the previous chapter. The first is the flow vs. current characteristic and the second is the pressure compensation characteristic.

The flow vs. current characteristic is shown below. While only one curve is shown on the graph, it actually represents the regulated flow regardless if port 2 or port 3 is the inlet port used. In other words, the regulated flow is symmetrical regardless of the inlet. The schematic below represents the hydraulic system used to test the valve for this characteristic. A relief valve was installed down stream of the regulated port and was set to open at 3000 psi while the inlet pressure was limited to 3500 psi. The PWM dither frequency of the controller which powered the ZL70-30 was set at 110 Hz.



As the graph on the previous page shows, the hysteresis is less than 6%. The flow range is 0 to 5.3 gpm. Saturation occurs at 90% of maximum current or 1260 mAmp for a 12V coil.

The next graph shows the compensation characteristic of the ZL70-30. The curves show that regardless of the change in pressure, the flow remains constant. This is true up to the curve which represents the flow at 70% of maximum current. Above this curve there is some change in flow as the load pressure increases. The flow decreases as the pressure increases or there is a droop in the characteristic. The compensation spring value represented by the bend in the curve is 190 psi. This curve and the previous curve show little difference in the characteristic with regard to direction of flow through the valve.

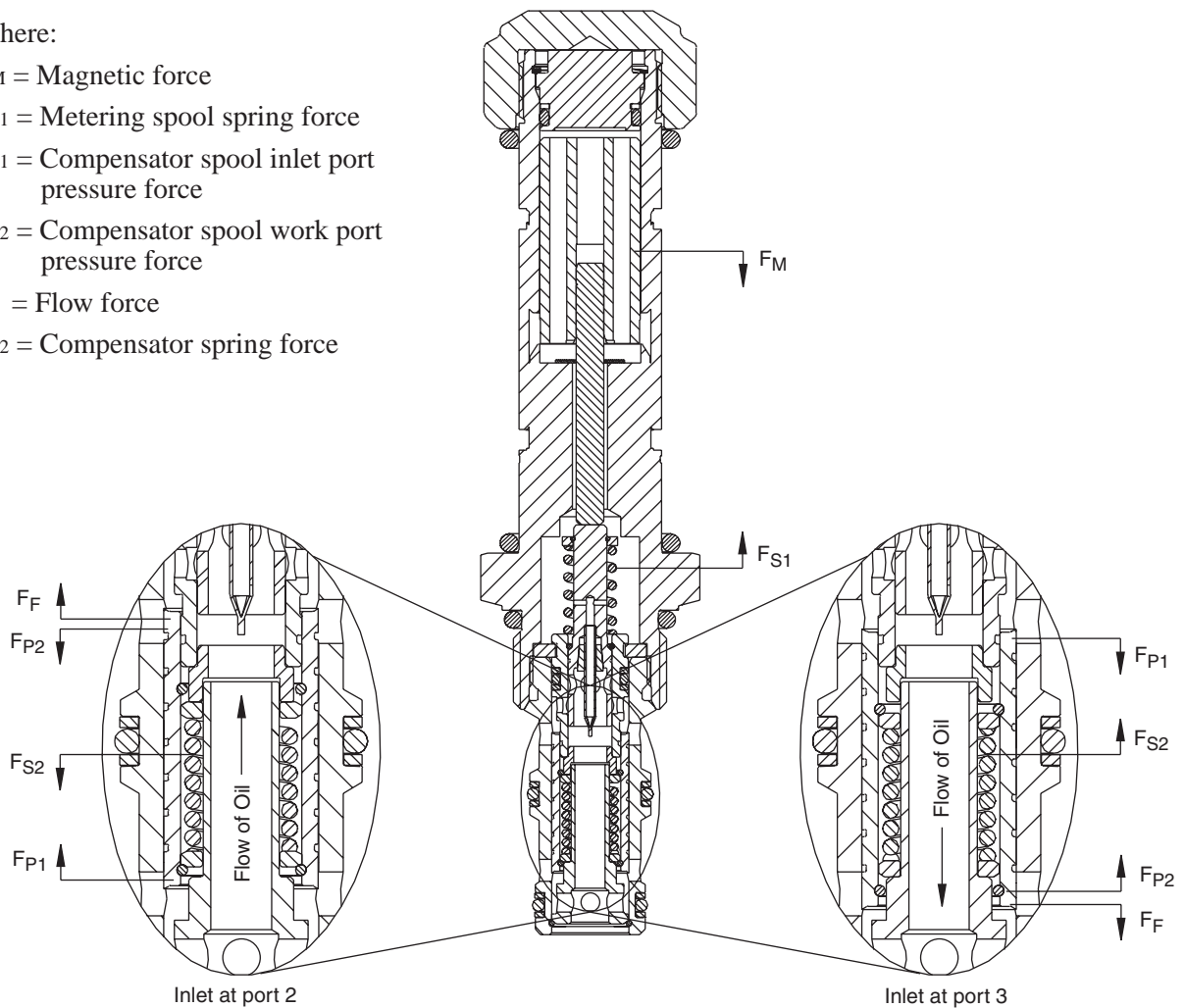


**Forces Acting on the ZL70-30**

The same forces that act on the PV70-30 also act on the components of the ZL70-30. These include the magnetic force, the metering spool spring force, compensating spool spring force and flow force. These are all indicated on the diagram below. All these forces act equally regardless of the direction of flow. The direction of the spring force and flow force change, depending on if the flow is from 2 to 3 or 3 to 2. This is shown in the two enlarged diagrams of the compensating spool below.

Where:

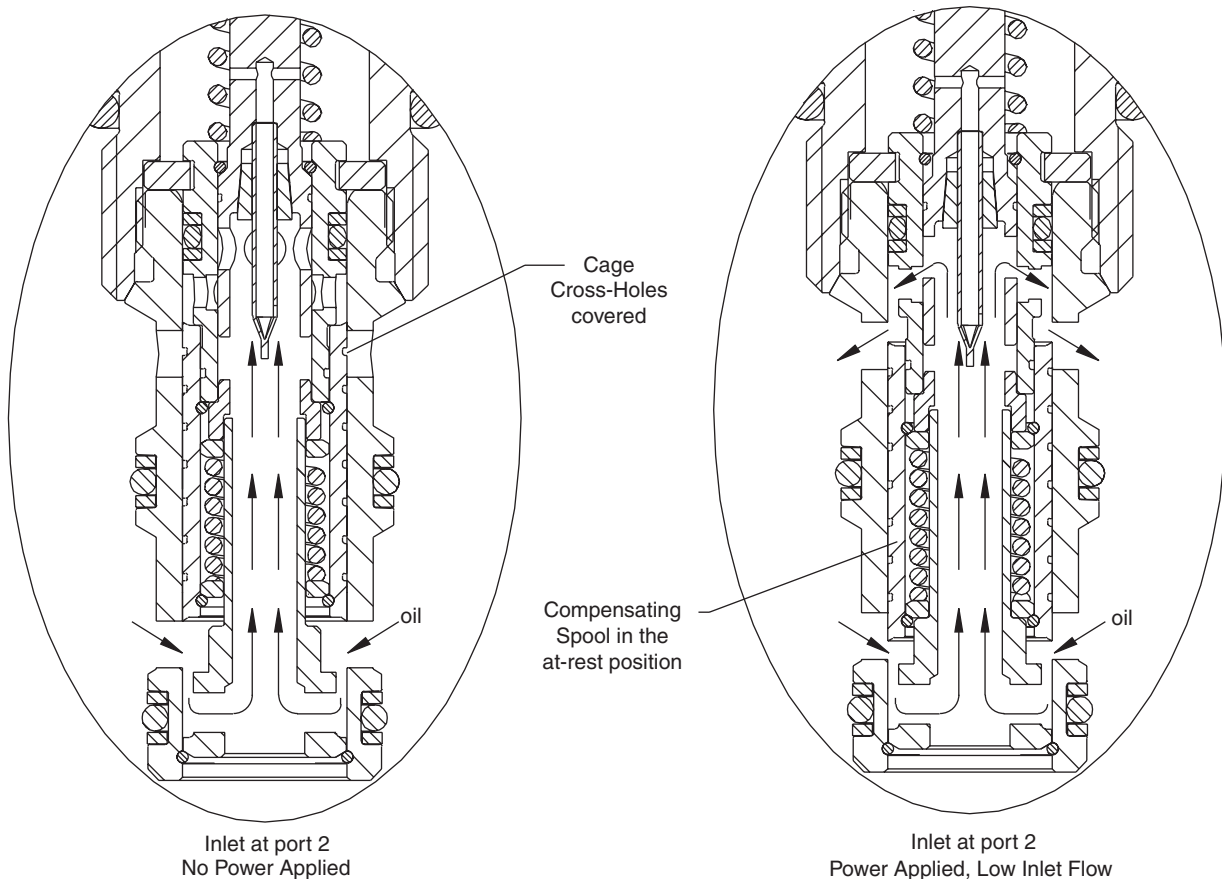
- $F_M$  = Magnetic force
- $F_{S1}$  = Metering spool spring force
- $F_{P1}$  = Compensator spool inlet port pressure force
- $F_{P2}$  = Compensator spool work port pressure force
- $F_F$  = Flow force
- $F_{S2}$  = Compensator spring force

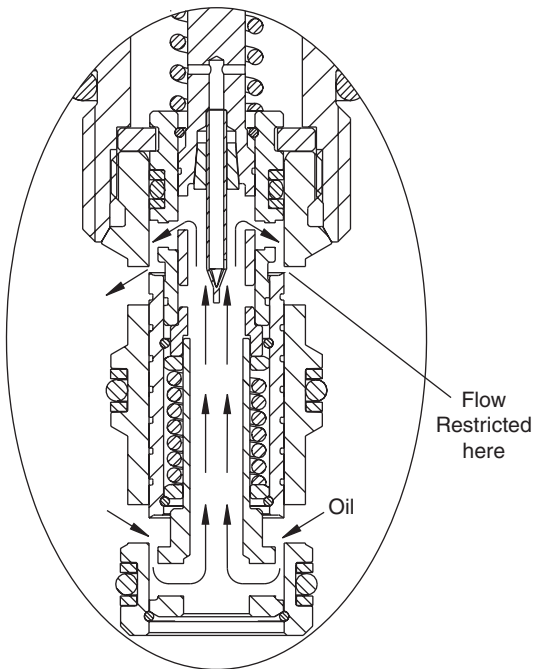


### Operation of ZL70-30

The diagrams that follow are used to describe the operation of the ZL70-30. The operation of the ZL70-30 is similar to that of the PV70-30. That is, power is applied to the coil, thus creating a magnetic force. The armature is attracted to the pole piece and pushes on the metering spool. Oil is then allowed to flow from port 2 to port 3 or port 3 to port 2. Port 1 is not connected externally from the valve to the hydraulic circuit.

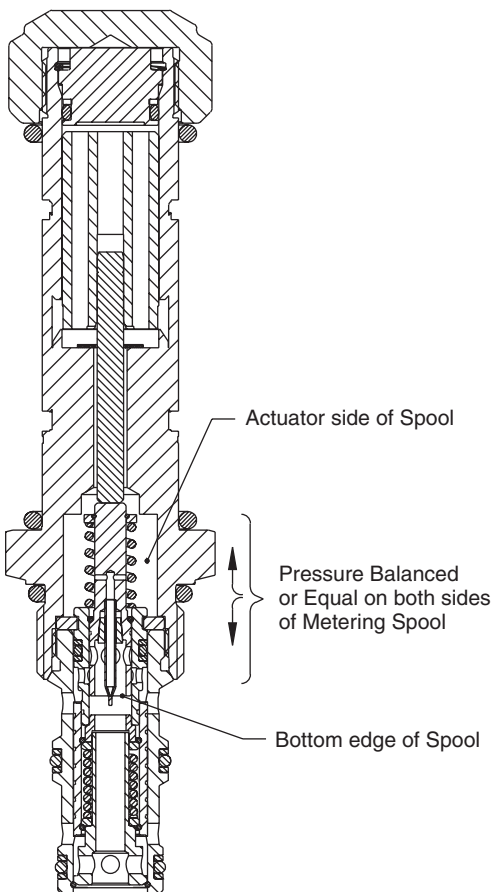
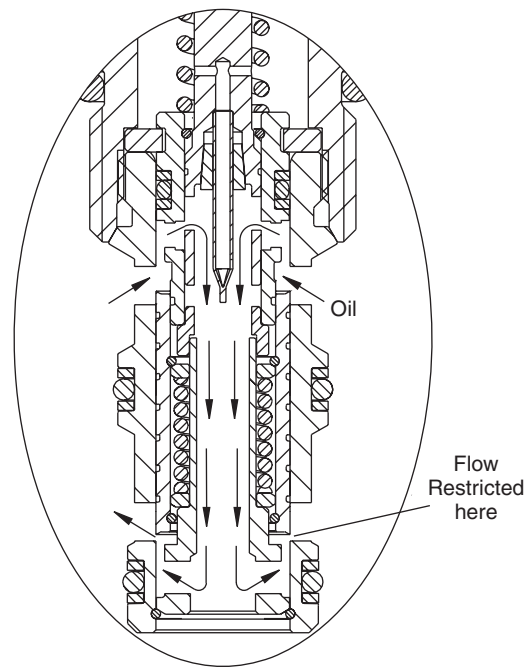
The first diagram to the left shows the valve when no current has been applied. Notice the metering spool is in the neutral or the de-energized position. Also, note that the compensating spool has completely covered the cage cross holes at port 3. This is because there is no pressure at port 3 to balance the inlet pressure at port 2. The next diagram to the right shows oil flowing from port 2 to port 3. Here the coil has been energized. Notice that the compensating spool has moved back to its initial at-rest position. As with the PV70-30, this is because the compensating spring force is greater than the force developed by the difference in pressure between ports 2 and 3. This would only occur if the inlet flow was actually lower than the desired regulated flow selected by the current applied to the ZL coil.





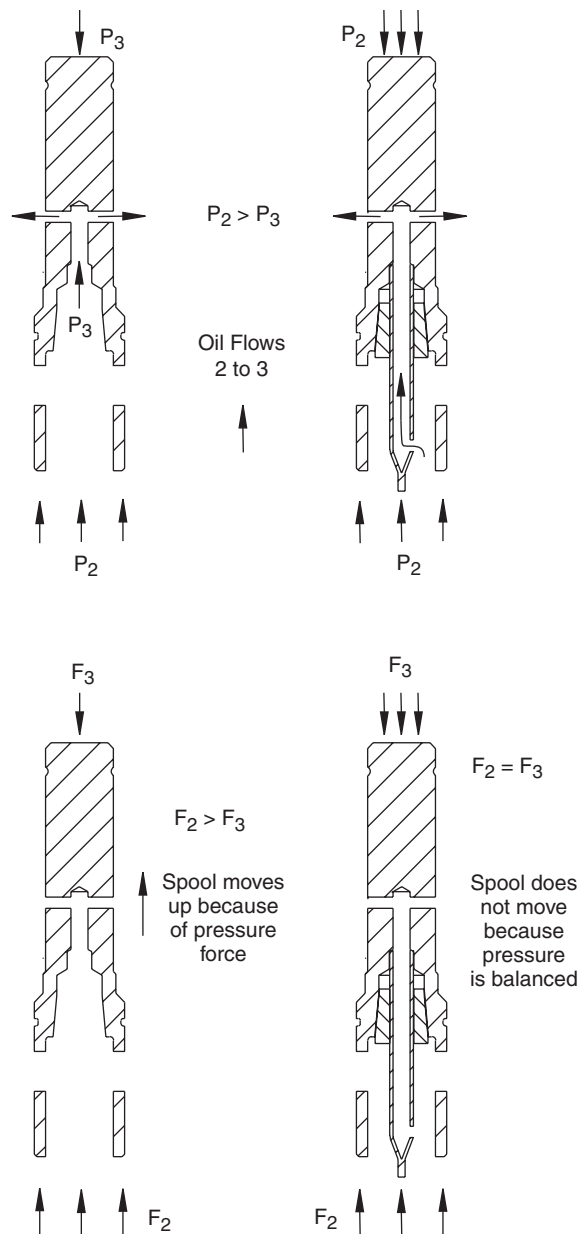
Typically, the flow available from the pump is greater than the desired regulated flow from the ZL70-30. When this is true, the compensating spool moves to restrict flow at port 3 as shown in the diagram to the left.

The diagram below shows the oil flowing from port 3 to port 2. As in the previous diagram, the compensating spool is metering the flow of oil but in this case, the flow is restricted at port 2 rather than port 3.

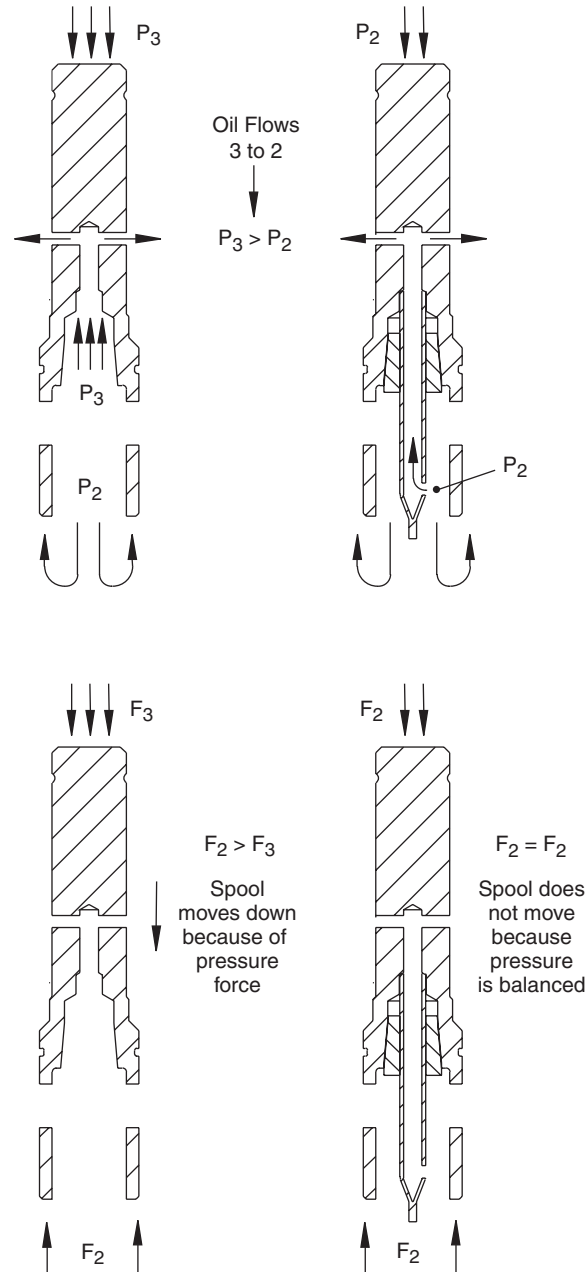


The performance (linearity, hysteresis, stability and symmetrical flow) of the ZL70-30 not only depend on the damping chamber and tight fit between the parts but also on the small tube inserted into the metering spool. This tube, known as a **pressure sense tube**, balances the pressure on either side of the metering spool. This means that the pressure inside the actuator portion of the valve is equal to the pressure at the bottom edge of the metering spool. These two areas of the valve are indicated in the drawing to the left.

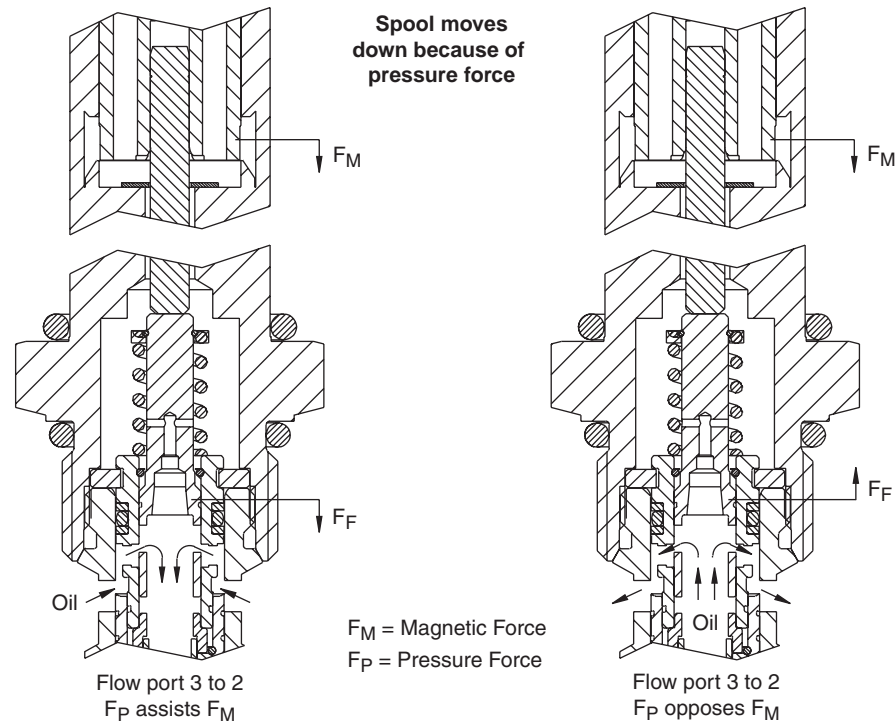
The following diagrams show how the pressure in the actuator would vary if the tube were not in place. In the two sets of diagrams following, the inlet is at port 2. Notice in both, that as the oil flows through the spool there is a decrease in pressure. In the diagram without the tube, the lower pressure is transmitted to the actuator side of the spool through the small hole in the center of the spool. In the assembly with the tube, the pressure at the bottom of the spool,  $P_2$ , is transmitted through the tube up to the actuator. Also shown below these pictures is a diagram of the forces acting on the spool. As the diagram to the left shows, the force acting on the bottom of the spool is greater than the force acting on the top of the spool. However, in the diagram to the right, the forces are balanced.



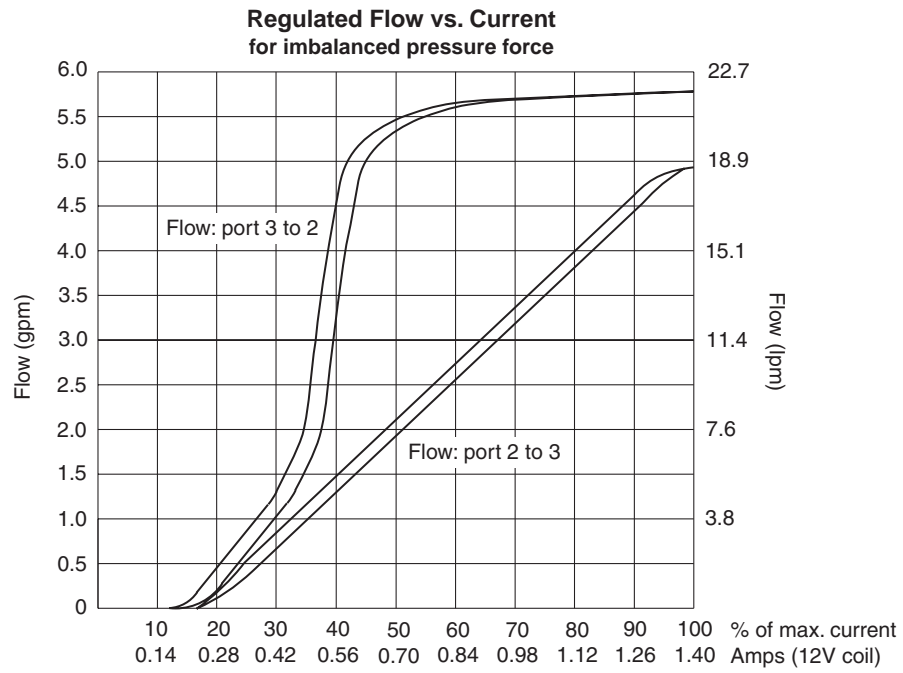
The next set of pictures depict what occurs when the direction of flow is reversed. The diagram to the left indicates that the pressure on the top of the spool is higher than the bottom and this causes the force on the top of the spool to be greater than the bottom. The picture to the right shows that the pressure and force are balanced because the tube transmits the pressure from the bottom of the spool to the top.



As shown on the previous page, if the tube was not present there would be a force imbalance acting on the metering spool. That is, there is a difference in the force on the actuator side of the spool versus the metering side. Not only would there be a force imbalance, but it also switches directions depending on the direction of the flow. If this were to occur when oil flows from port 3 to port 2, this pressure force would assist the magnetic force. This assistance in force is so great that the flow vs. current graph would not be a straight line, but would appear as a step instead. However, when the oil is flowing from port 2 to port 3, this force would work against the magnetic force. This is summarized in the two drawings below. This force imbalance would cause the flow vs. current characteristic to appear different based on which port was the inlet, as shown in the following graph.

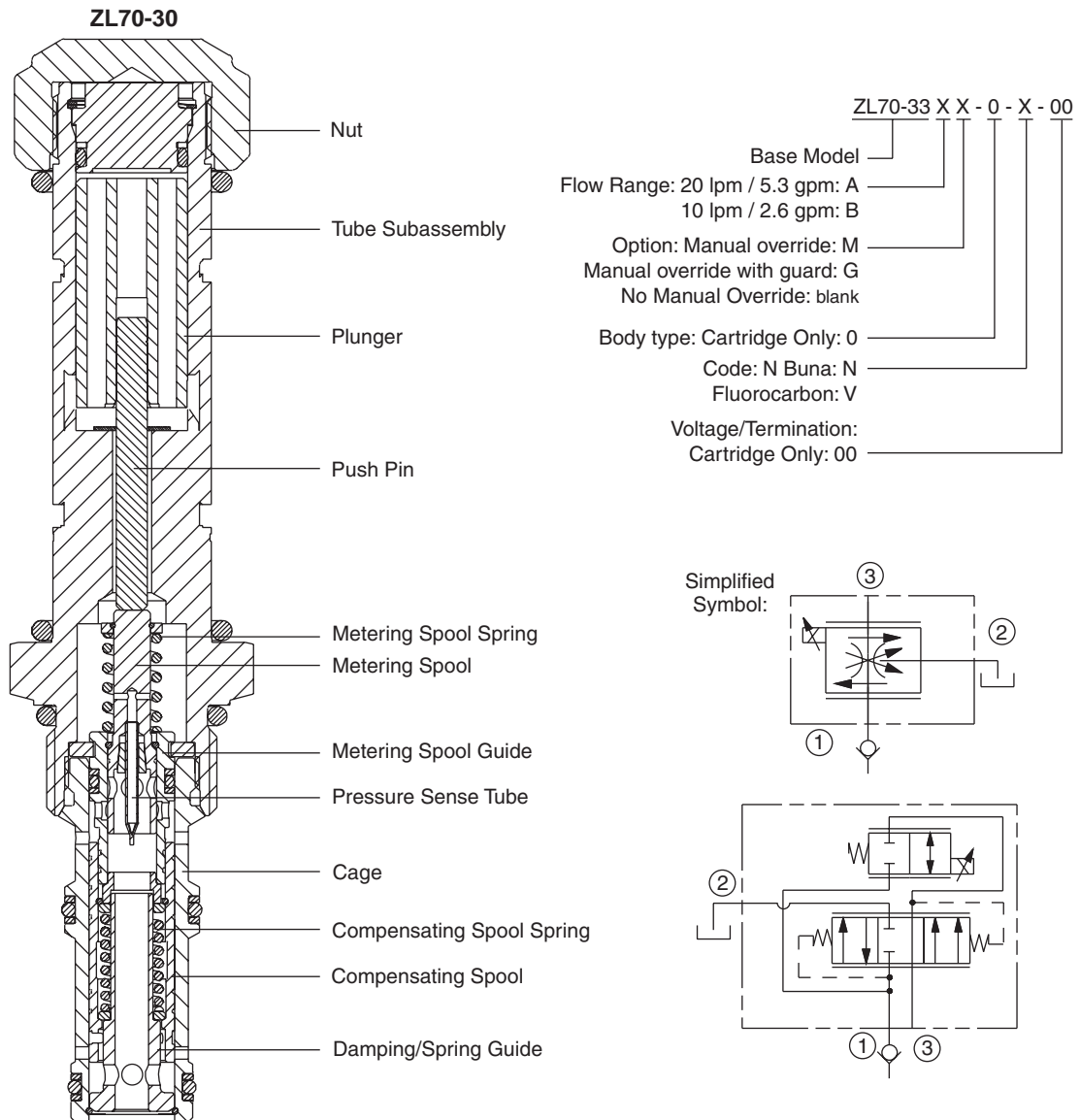






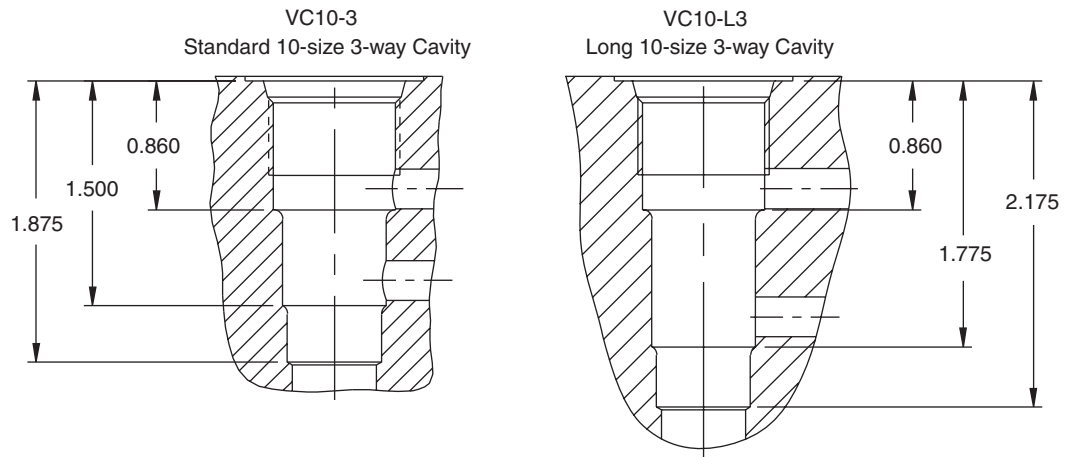
### ZL70-33

The ZL70-33 is the first of two valves specifically designed to proportionally control the lift and lowering of a single acting cylinder. It is a normally closed, internally compensated, bidirectional, proportional flow control. It is interesting to note that the ZL70-33 acts as a priority/bypass flow regulator when the inlet is at port 1. When the inlet is at port 3 and port 1 is blocked, the ZL70-33 acts as a restrictive type flow regulator. The cross section, symbol and model code are shown below. A section describing the performance follows as well as the operation of the valve itself. Also included is a section listing the forces acting on the valve. However, the detailed description of the forces can be found in other parts of this manual. The operation of the valve when used in a lift/lower circuit can be found in the applications chapter. In addition, the cross section of the cavity needed for the ZL70-33 and the ZL70-36 is shown on the following page.



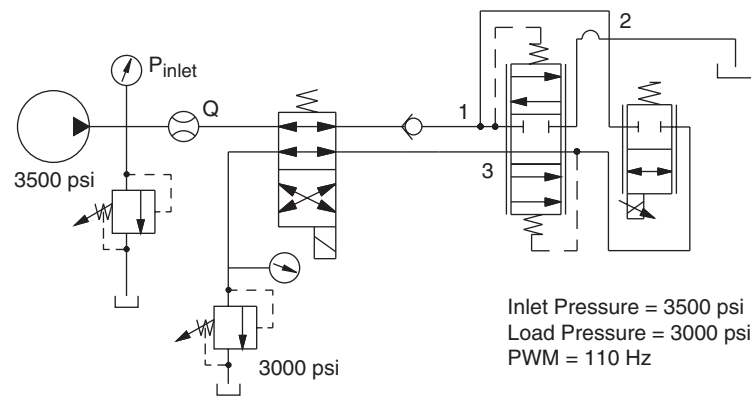
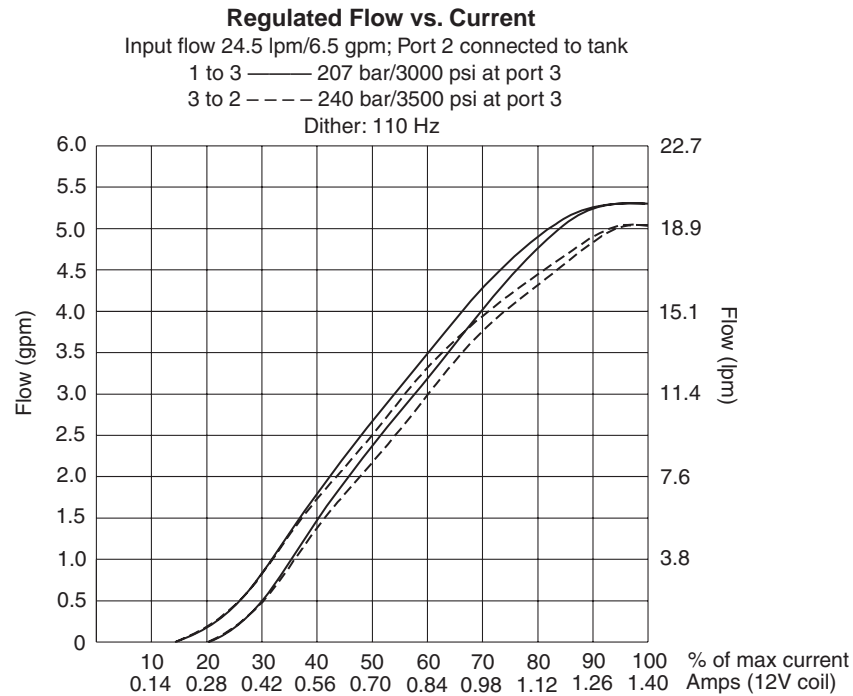
**Cavity for ZL70-33**

Shown below is a comparison of the cavity that must be used for the ZL70-33 and the ZL70-36. The diagram on the left is the standard HydraForce 10-size 3-way cavity. The diagram on the right is a long version of the 10-size 3-way cavity. The ZL70-33 and ZL70-36 must use this long cavity because they do not fit into the standard cavity. Further, the other types of PV and ZL valves must use the standard cavity as they will not function properly in the long cavity.



### Performance of ZL70-33

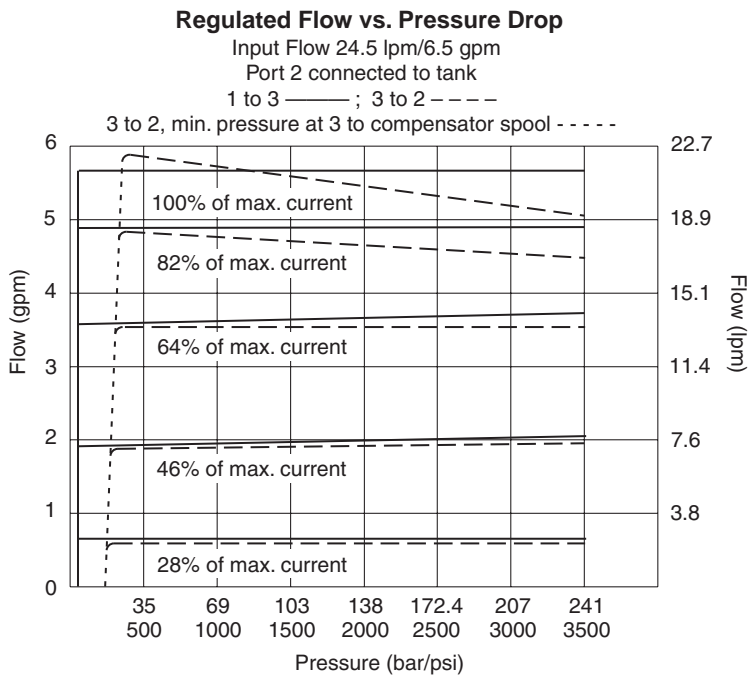
This section discusses the performance characteristics that describe the operating limits of the ZL70-33. These are the flow vs. current characteristic and the pressure compensation characteristic. The flow vs. current characteristic as well as the hydraulic schematic that represent the circuit in which the valve was tested are shown below.



Unlike the ZL70-30, there is a sufficient difference in the maximum regulated flow depending on whether port 1 or port 3 is used as an inlet, therefore two graphs are required. The hysteresis in both graphs is less than 6%. The threshold current is 20% or 0.28 amp for a 12V coil. The saturation current for both is 100% of the maximum operating current or 1400 mAmp for a 12V coil. The maximum regulated flow when the inlet is at port 1 is 5.3 gpm. When the inlet is port 3 the maximum regulated flow is limited to 5.1 gpm. See the graph on the previous page.

The graph below shows the compensation characteristic of the ZL70-33. Three line types are used to represent a different characteristic depending on the direction of flow. The curve on the left which joins all the compensation curves represents the compensation value of the valve. That is, the compensating spool begins to regulate at 75psi. The solid line represents the flow of oil from port 1 to port 3. The load pressure at port 3 was varied from a minimum setting of the relief valve to 3500 psi while the pressure and flow at port 3 were recorded on the graph. These lines show that regardless of the change in pressure, the ZL70-33 maintains a constant flow.

The dashed line represents the performance of the valve when oil flows from port 3 to 2. The next line to review is the one which starts at approximately 250 psi and runs almost parallel to the vertical axis. This line represents the minimum pressure required at port 3 to open the flow path at port 2 between the compensating spool and cage. The dashed lines which project horizontally from this one represent how the valve compensates for changes in load pressure with oil flowing from port 3 to 2.



### Forces Acting on ZL70-33

Detailed below are the forces acting on the ZL70-33 which are the same forces acting on the ZL70-30. These include the magnetic force, spring force and pressure forces. A detailed description of all these forces can be found in the previous sections of this manual.

Where:

$F_M$  = Magnetic force

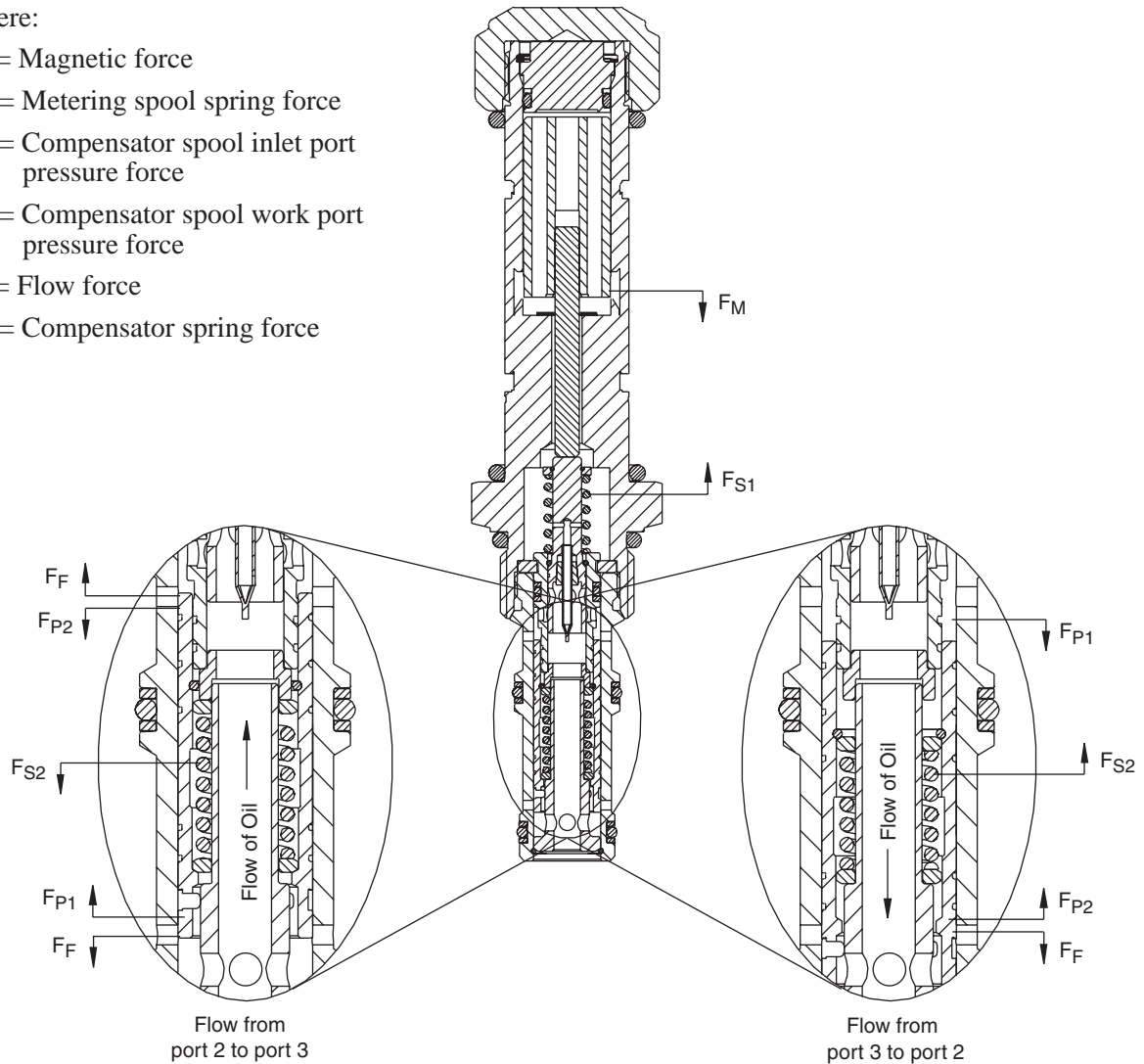
$F_{S1}$  = Metering spool spring force

$F_{P1}$  = Compensator spool inlet port pressure force

$F_{P2}$  = Compensator spool work port pressure force

$F_F$  = Flow force

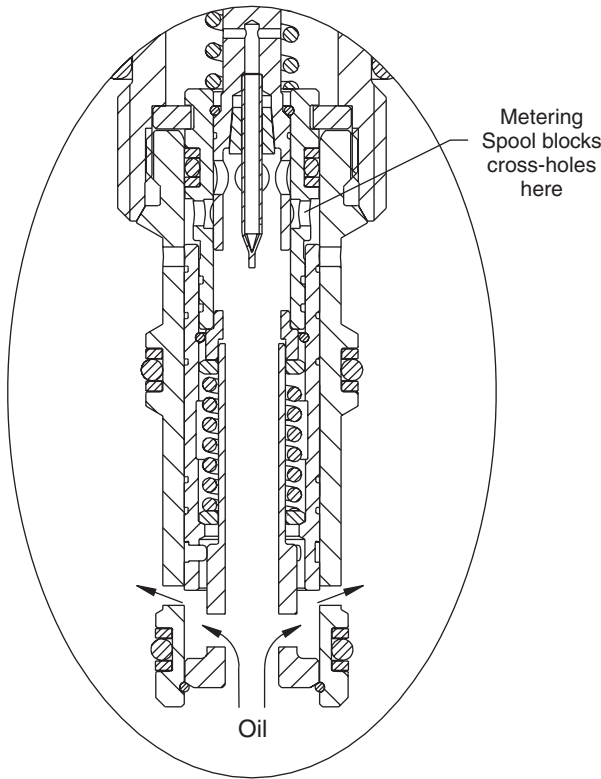
$F_{S2}$  = Compensator spring force



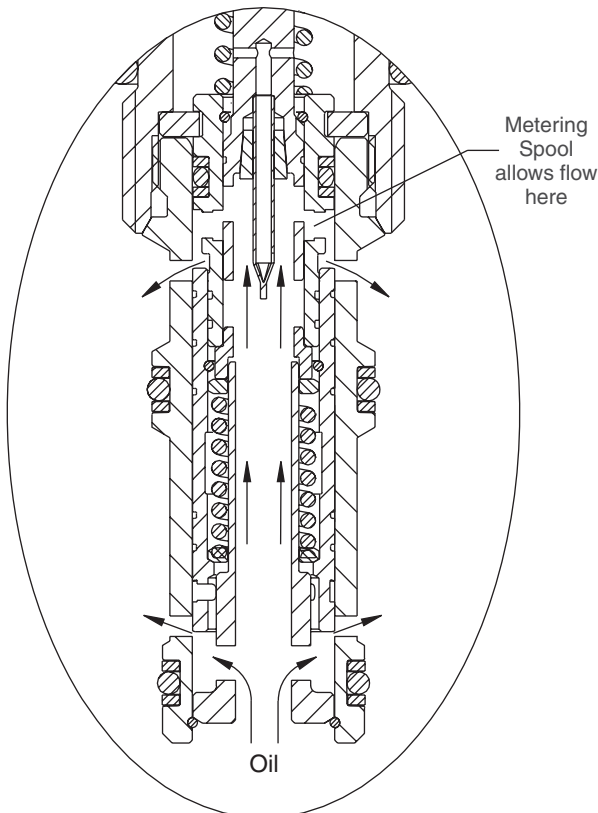
As with the PV70-30 there is a pressure sense tube in the metering spool to balance the forces acting on this spool, regardless of the direction of flow. If the tube was not present, the flow vs. current characteristic would differ between the two flow paths.

**Operation of ZL70-33**

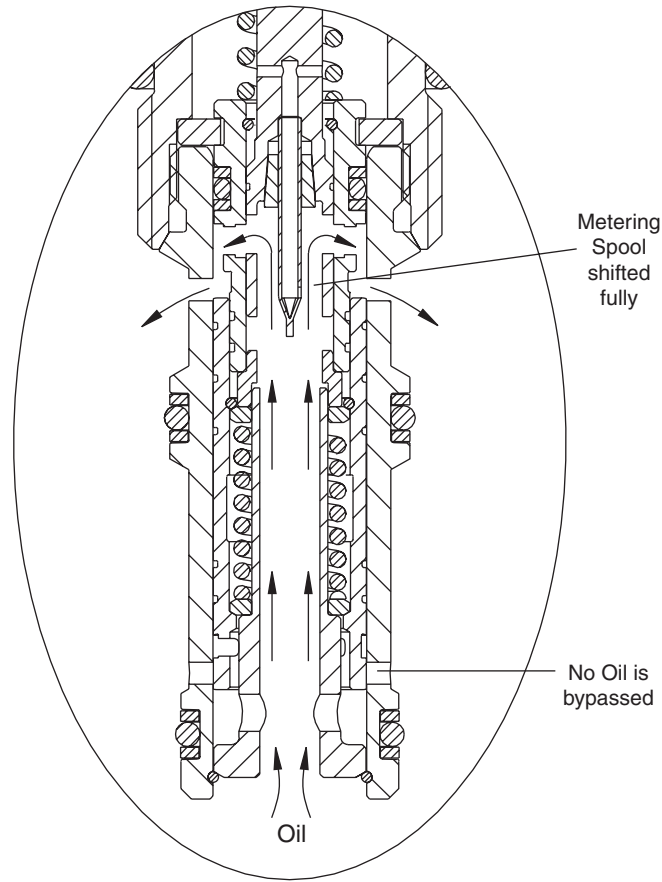
The following section details the operation of the ZL70-33. This includes what occurs as oil flows from port 1 through port 3 as well as the operation when port 3 acts as the inlet. In the first diagram at left, pump flow has been turned on but the ZL70-33 coil has not been energized. Notice that the cross holes of the metering spool guide remain blocked and the oil flows from port 1 to port 2. In this mode the valve is bypassing the pump flow.



In the diagram to the left, the coil has been energized and the metering spool has unblocked the cross holes of the guide. The compensator has also moved to allow oil to flow out of port 3. Notice that some of the oil is still flowing out of port 2 as well. In this mode the valve is acting as a priority/bypass valve.

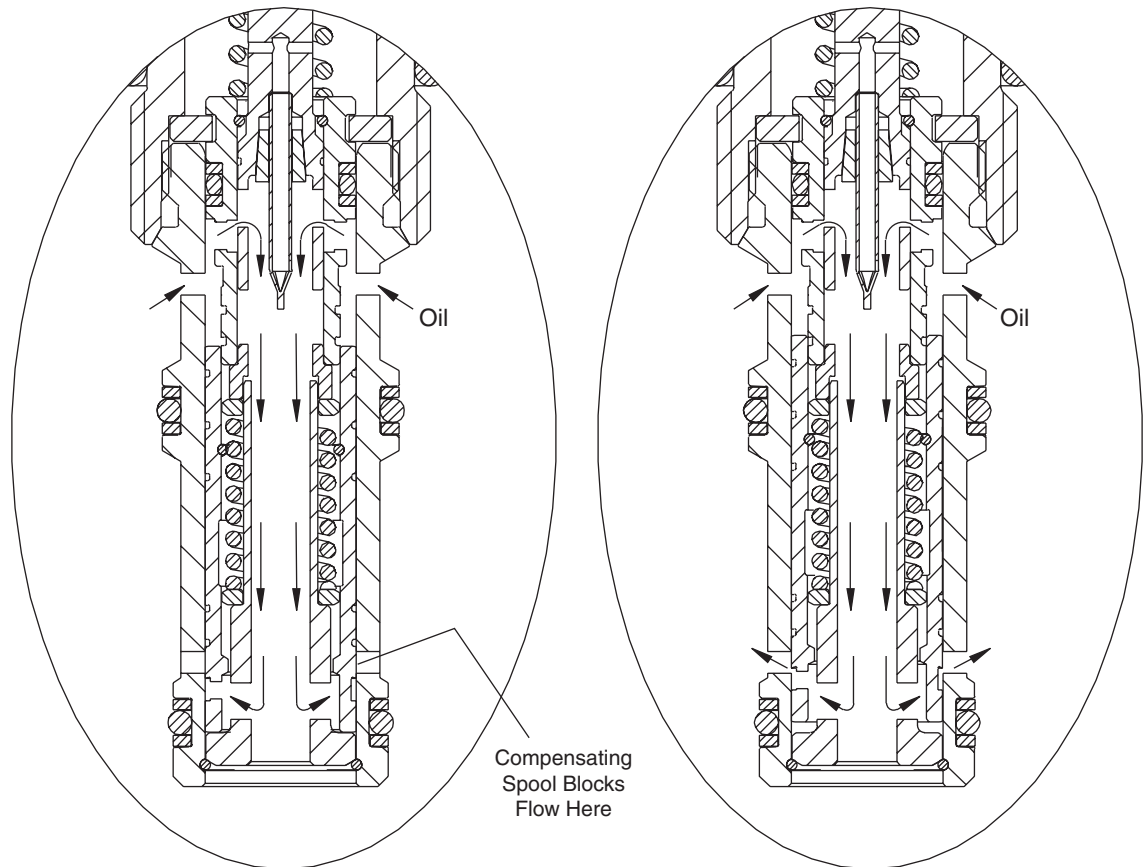


The diagram below shows the operation of the ZL70-33 with the inlet at port 1.  $I_{MAX}$  has been applied to the coil. This causes the metering spool to shift fully. The compensator has returned to the neutral position. This occurs because the inlet flow is less than or equal to the desired flow. The desired flow in this case is the flow at  $I_{MAX}$  or approximately 5.3 gpm. (This can be found on the flow vs. current characteristic graph given in the previous section.)



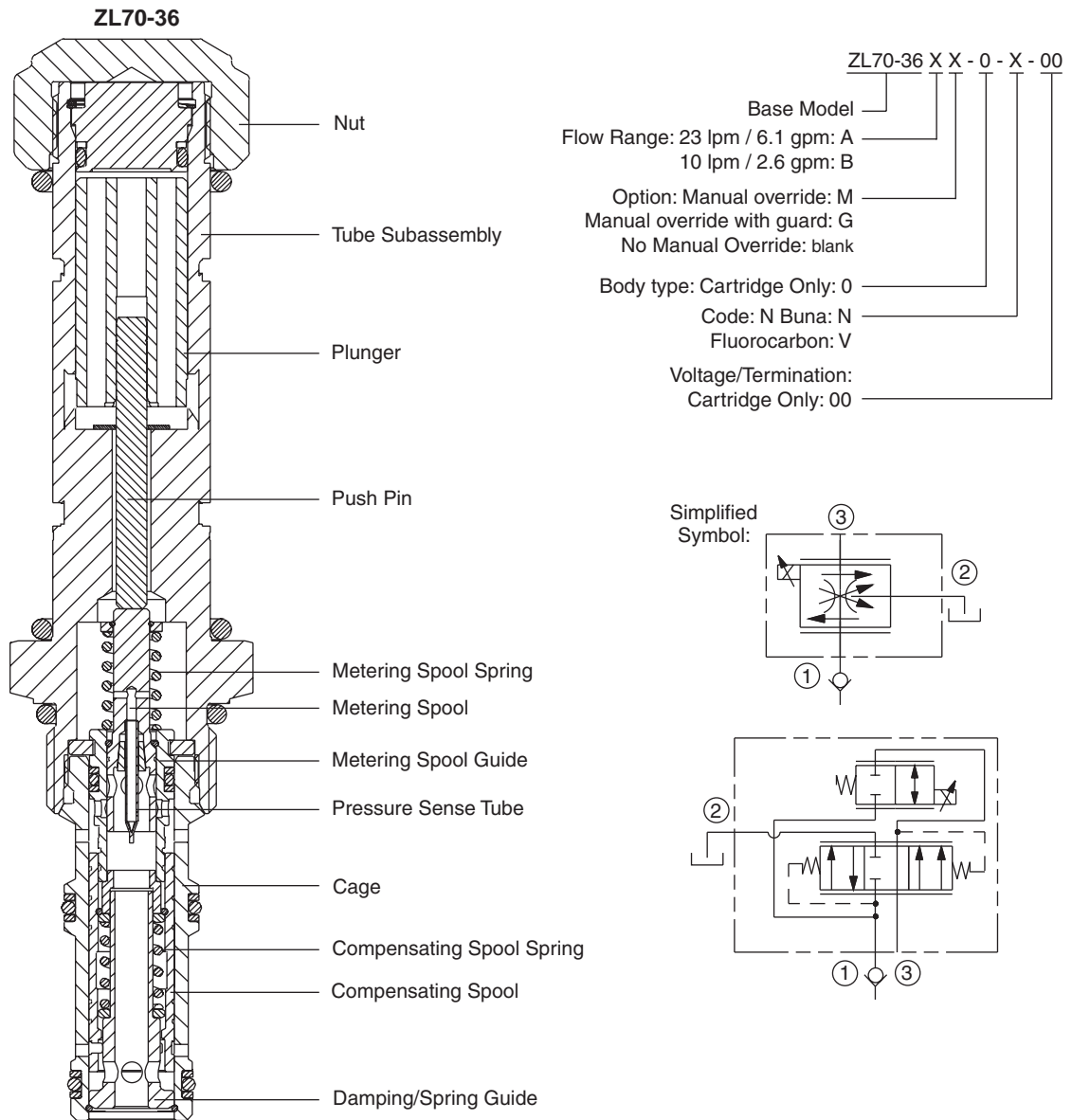


The following two drawings will be used to describe the operation of the ZL70-33 when oil is flowing from port 3 to port 2. No oil can flow from port 1 because there is a valve external to the ZL70-33 blocking the flow of oil. This will be explained further in the next chapter. In both drawings, current has been applied to the coil. The metering spool has moved to allow oil to flow through the center of the compensating spool. The difference between the pictures is the position of the compensating spool. In the picture to the left, the compensating spool is blocking the flow of oil. This shows that the differential pressure across the compensating spool has not exceeded 250 psi. 250 psi is the minimum pressure differential required for the ZL70-33 to regulate the flow of oil from port 3 to 2. This is shown on the pressure compensation characteristic curve. This was also depicted in the compensation characteristic shown in the performance of ZL70-33 section when the inlet is at port 3. In the drawing to the right, oil is flowing out of port 2. This occurs because the differential pressure across the compensator is greater than 250 psi.



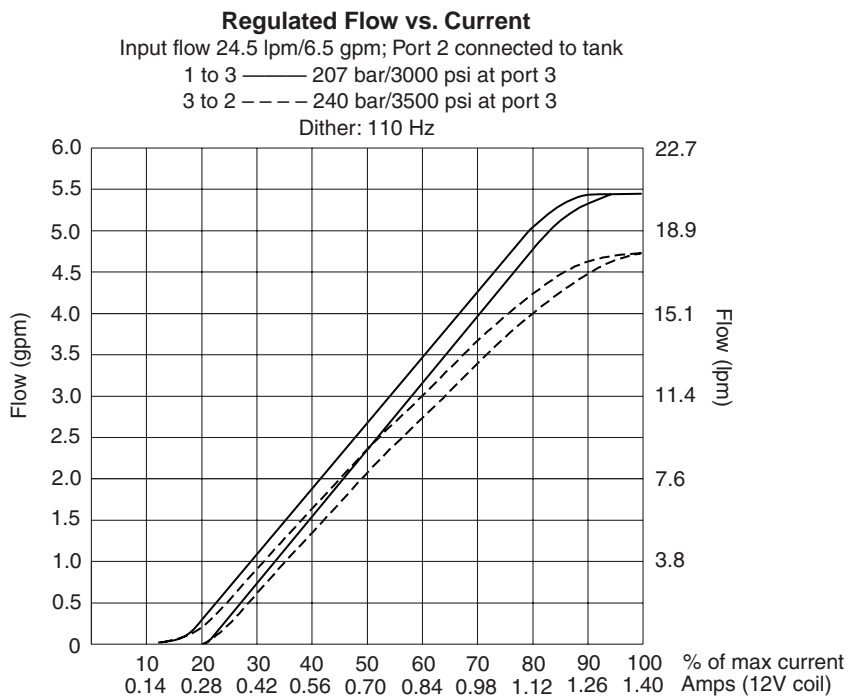
### ZL70-36

The cross section, schematic and model code of the ZL70-36 are shown below. This valve is a normally closed, bidirectional, internally compensated proportional flow control. Like the ZL70-33, this valve was specifically designed to control the movement of a single acting cylinder. The performance, forces and operation will be described in the following sections.

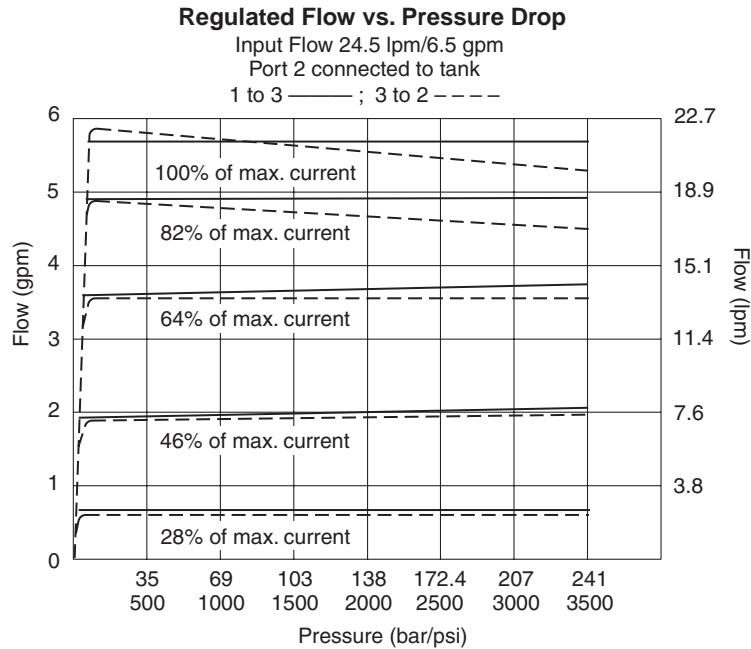


**Performance of ZL70-36**

Two graphs are shown in this section to depict the performance of the ZL70-36. The first graph shows the flow vs. current characteristic and the second shows the compensation characteristic. The schematic representing the hydraulic test can be found in the performance section of the ZL70-33. In both of the flow vs. current curves below, the hysteresis is less than 6%. The threshold current is 20% of  $I_{MAX}$  and the saturation when oil flows from port 1 to 3 or port 3 to 2 is 95% of  $I_{MAX}$  and 100% of  $I_{MAX}$  respectively. The maximum flow range is 0 to 5.5 gpm for the 1 to 3 direction but only 0 to 4.7 for the 3 to 2 direction.



The next graph represents the compensation characteristic for either oil flowing from port 1 to 3 or port 3 to 2. The line running almost parallel to the flow axis on the left can be used to determine the compensation value, which is 50 psi. Notice that unlike the ZL70-33, oil begins to flow immediately with the slightest change in pressure. This graph was made by setting the current to a specified level. Then the flow and load pressure were recorded while the inlet pressure was varied.



**Forces Acting on ZL70-36**

A cross section of the ZL70-36 with the forces labeled on the various components is shown below. These forces are the same as the ones listed on other proportional flow controls.

Where:

$F_M$  = Magnetic force

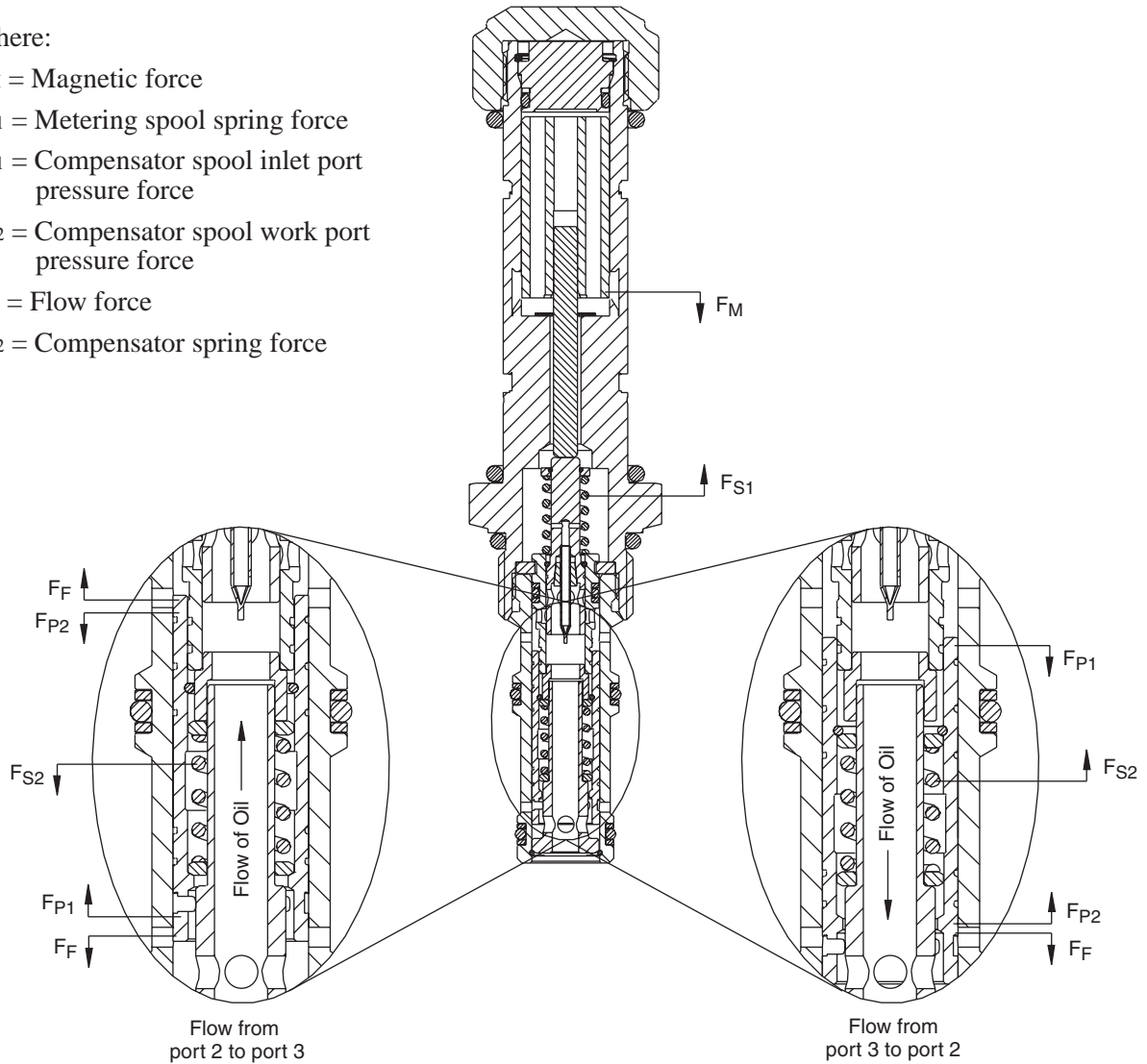
$F_{S1}$  = Metering spool spring force

$F_{P1}$  = Compensator spool inlet port pressure force

$F_{P2}$  = Compensator spool work port pressure force

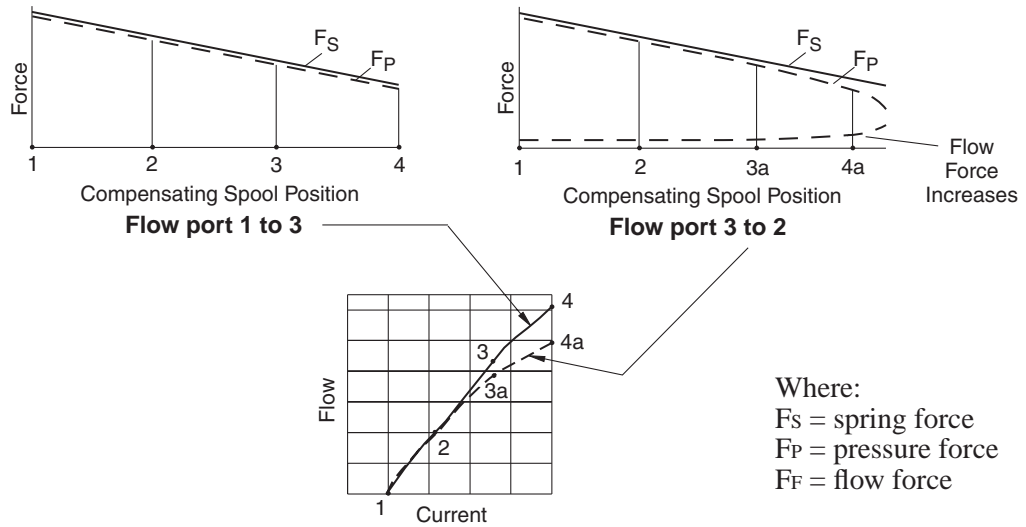
$F_F$  = Flow force

$F_{S2}$  = Compensator spring force



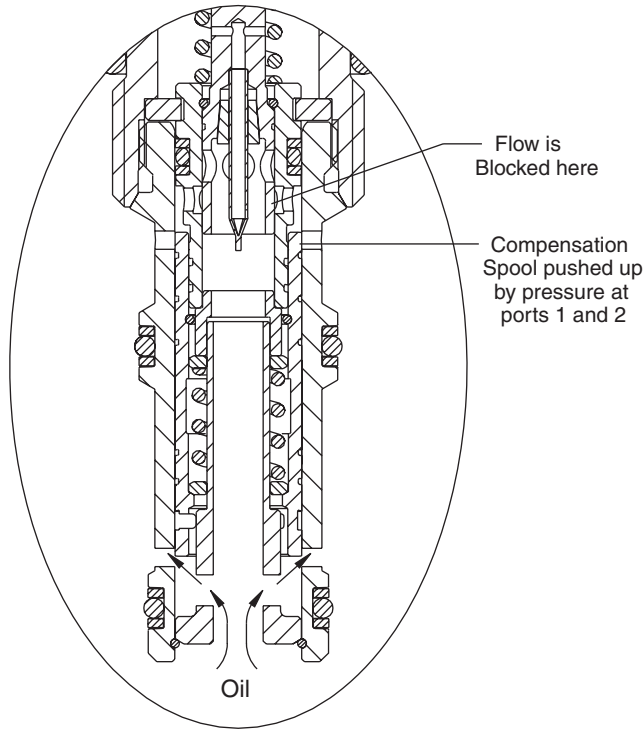
The flow force acting on the compensating spool when the oil flows from port 3 to 2 affects the performance more than when the oil flows from port 1 to 3. In fact, the flow force has more influence on the compensating spool in the ZL70-36 than the ZL70-33. This is because the compensating spool spring force in the ZL70-36 is less than that of the ZL70-33. The following graphs show how the flow force differs depending on the direction of flow and how it affects performance.

The first graph shows how the spring force changes to match the pressure force. The various points labeled on the graph represent the position of the compensating spool in relation to the flow achieved when the current is varied. These points are also shown on the flow vs. current graph in the middle. Notice that point 1 on the force graph represents the highest spring force, but it also represents the lowest flow. This is because the metering spool is closed so the pressure at port 2 pushes up on the compensating spool and compresses the compensating spring. A pictorial view of the compensating spool in this position can be found in the first operational diagram of the following section. As the metering spool moves to uncover more of the metering spool guide cross holes, the compensating spool moves accordingly. A constant differential pressure is then maintained across the metering spool. As the compensating spool moves, the spring force decreases. This is shown in the decreasing slope of the line on the graphs. Notice the flow force is not shown on this graph. This is because the two different flow forces cancel each other out when the oil flows from port 1 to port 3.

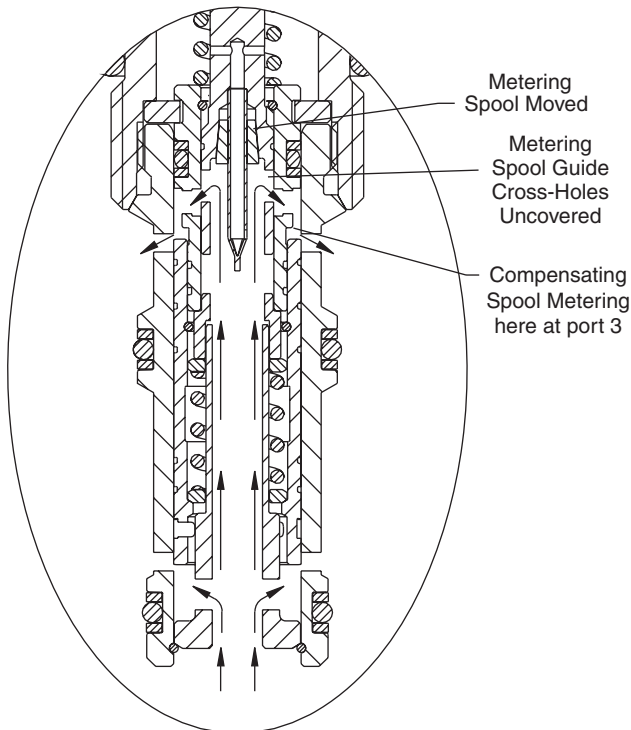


The second force vs. spool position graph shows the changes in force as oil flows from port 3 to port 2. In this graph the flow force is shown. This is because the flow forces do not completely cancel. The beginning portion of this graph is similar to the previous. The difference is in the second half. As more flow passes through the valve, the flow force increases. Eventually, the flow force is so great that the flow cannot increase further. This is because the compensating spool is beginning to wash shut.

**Operation of ZL70-36**

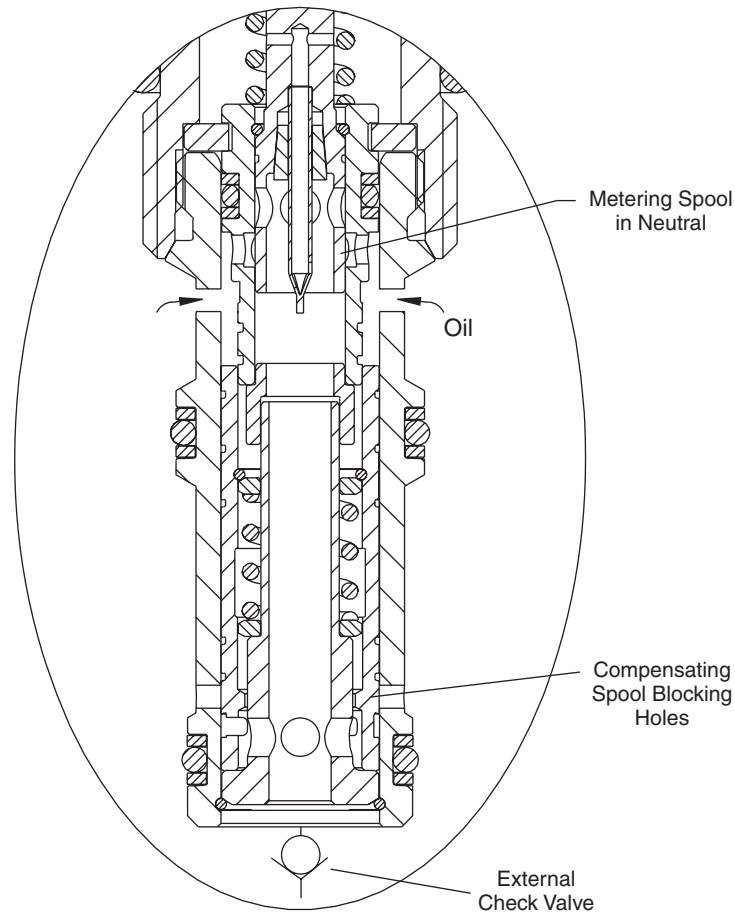


The following section describes the operation of the ZL70-36. The first two drawings show the operation when oil is flowing from port 1 to port 3. As with all the normally closed proportional valves, the flow of oil is blocked between the inlet and outlet until the threshold current is applied. The diagram to the left shows the condition where no power has been applied to the coil. Like the ZL70-33, all the oil is bypassed to tank.

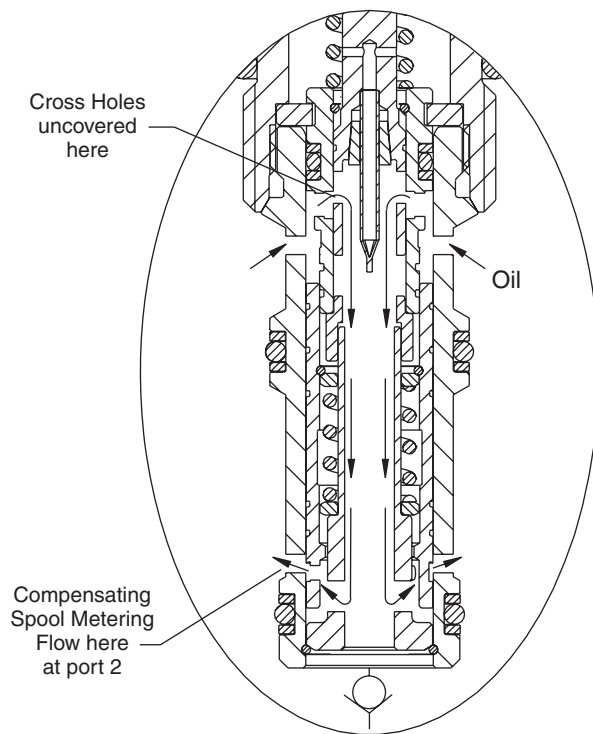


The next drawing shows the metering spool has moved from the neutral position because current has been applied to the coil. The oil is now flowing to both the work port and bypass port. The compensating spool is metering the flow at port 3.

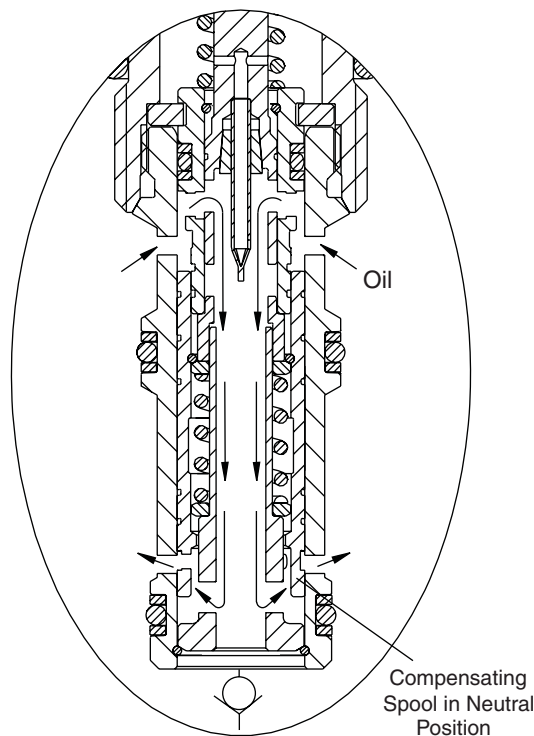
The next three diagrams show the operation of the ZL70-36 when oil is flowing from port 3 to port 2. The first diagram below shows that the metering spool is in the neutral position. The compensating spool has moved to completely cover the cage cross holes. This occurred because there is no pressure at port 2 to balance the pressure at port 3. Again, no oil is flowing because no power has been applied to the coil.







The diagram at left shows that the metering spool has moved and the cross-holes of the metering spool guide have been uncovered. Oil is now flowing from port 3 to port 2. The compensating spool is metering the flow of oil at port 2.



This last diagram shows that the oil is still flowing from port 3 to port 2. The difference between this picture and the one above is that the compensating spool is no longer metering flow at port 2. This is because the pressure difference between port 3 and port 2 does not exceed the pressure compensating spring force. However, unlike the ZL70-33, oil continues to flow. This is because the compensating spool is free to float in the ZL70-36. The compensating spool in the ZL70-33 is held in place by the compensating spring.

## Summary

In this chapter the following concepts were presented:

- The patent covering the invention of the bidirectional flow control.
- The performance and operation of the restrictive style, bidirectional ZL70-30.
- How the pressure sense tube improves the performance of the ZL70-30
- Why the flow force affects the maximum flow through the ZL70-36 more than the ZL70-30.
- The various compensating values for the ZL70 type valves.
- The operation of the ZL70-33 and the ZL70-36 and how the valves change from priority/bypass type to a restrictive type valve depending on the direction of flow.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. What is the basic premise of the patents covering the bidirectional flow control? \_\_\_\_\_
2. Why is the pressure sense tube used in the ZL70-30? \_\_\_\_\_
3. What are the differences in the forces acting on the compensating spool of the ZL70-30 when the flow changes direction? \_\_\_\_\_
4. What is the compensating pressure of the ZL70-30? \_\_\_\_\_
5. True or False. The ZL70-30 is a restrictive type flow control. \_\_\_\_\_
6. What is the minimum pressure required at port 3 of the ZL70-33 to move the compensating spool so that oil can flow? \_\_\_\_\_
7. What is the compensating value of the ZL70-33 when oil flows from port 1 to 3? \_\_\_\_\_
8. What is the compensating value of the ZL70-36 when oil flows from port 1 to 3 or from port 3 to 2? \_\_\_\_\_
9. What are some of the main differences between the performance of the ZL70-33 and the ZL70-36? \_\_\_\_\_
10. True or False. The ZL70-36 can either be a restrictive type flow control or a priority bypass type flow control, depending on the direction of flow. \_\_\_\_\_

**Notes:**

# Chapter 11: Poppet Style Flow Controls

## Objectives

- Learn about the poppet style proportional flow controls.
- Learn about the dual flow gain vs. current characteristic.
- Discover the similarities in the construction of a poppet flow control compared to a solenoid valve.
- Develop an understanding of how the poppet flow controls operate.

## Introduction

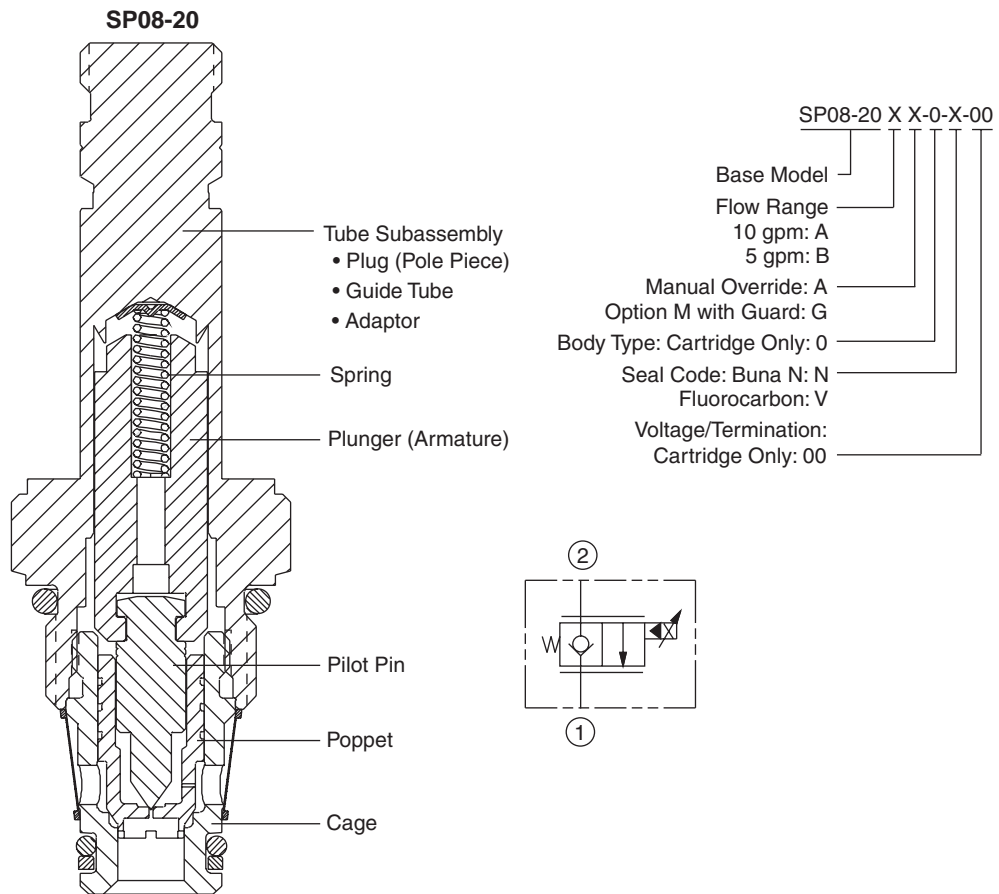
The pilot operated poppet style, proportional flow controls are introduced in this chapter. The methods used in the manufacturing of the components as well as the assembly are presented. As with other valves presented in this manual, the performance, operation and forces are presented.

# HYDRAFORCE

### SP08-20

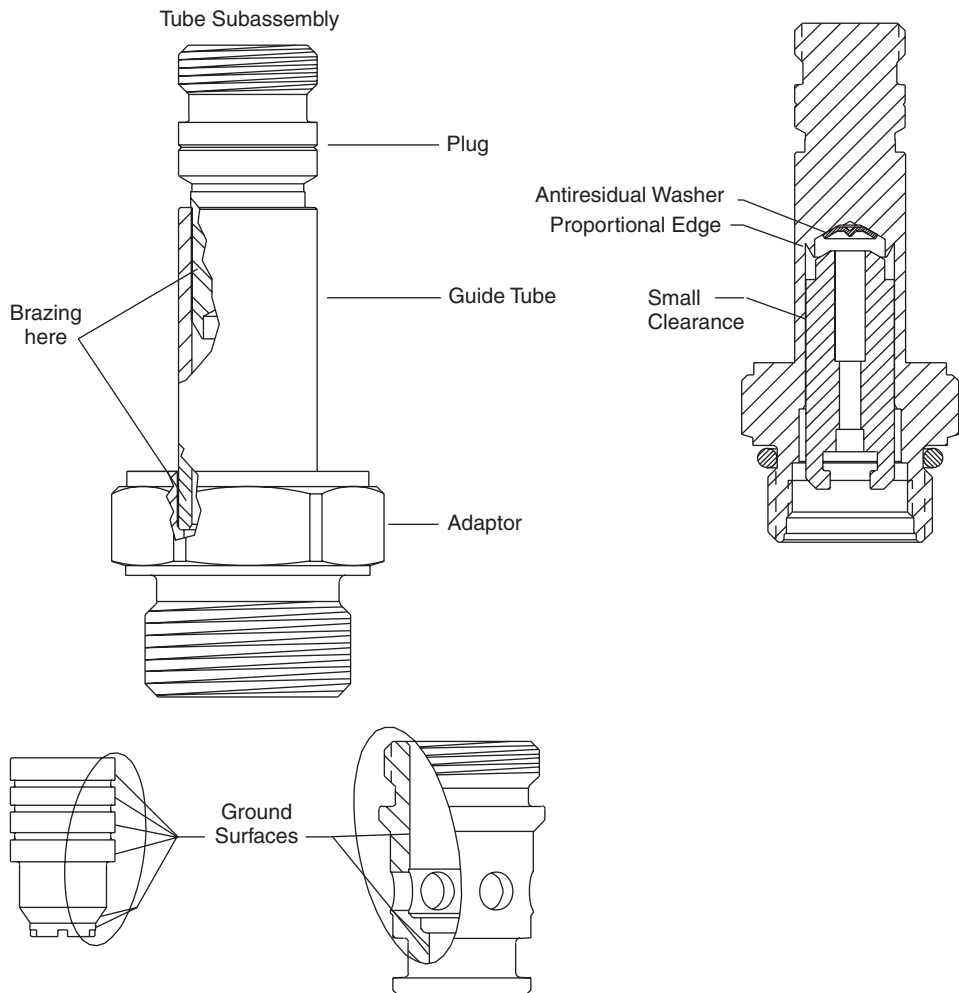
The SP08-20 is a pilot operated poppet style, proportional flow control valve. Because it is a poppet style, the leakage characteristic is low. Therefore, the valve is suitable for load holding applications. The coils used to operate this valve are the same as those used on the SV products. This includes the E series coils. A further discussion regarding the operation of the coils mated to the SP valves is provided in the operation of coils chapter.

A cross section of the valve, model code scheme and schematic of the SP08-20 are shown below. In the sections that follow, the construction and manufacturing methods of the components are presented, as well as the forces acting on the components, performance and operation of the valve.



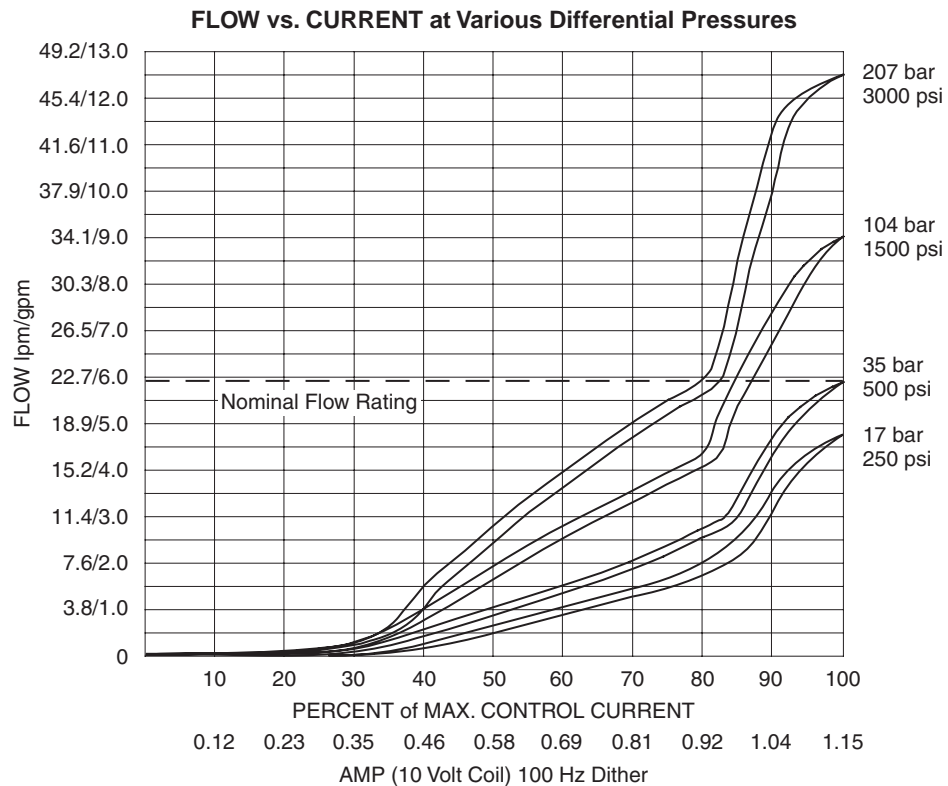
**Construction of SP08-20**

In general, it can be said that the construction of the SP08-20 is exactly like the SV08-20. Notice on the previous page that the valve is constructed of the same components as the SV08-20. The only component that is found in the SP that is not in the SV is the anti-residual washer. This keeps the plunger from magnetically latching to the pole piece. The valve is made up of a cage, poppet, pilot pin, tube subassembly and spring. However, this is where the similarities end. The tube subassembly, like the SV is made from three components and these are then brazed. The difference here is in the proportional edge on the pole piece face. This edge is necessary to provide the direct correlation between current and flow. Notice that unlike the other proportional flow control valves, this is a pull style actuator. The armature looks similar to the one in the SV, yet the clearance between the tube and plunger has been decreased to reduce hysteresis and improve stability. For this reason, the processing of the plunger includes a polishing operation as does the bore of the guide tube. Also, a special grinding process for the cage has been developed internally at HydraForce as well as one for the poppet. These parts are ground to assure low leakage and low hysteresis. Further, the parts are hardened to provide surfaces that are resistant to wear and contamination.



### Performance of SP08-20

This section describes the graphs used to illustrate the performance of the SP08-20. This includes the flow vs. current characteristic, the pressure drop vs. flow characteristic as well as a frequency response graph. The first graph below shows the flow vs. current performance. Notice that the curves below have a knee, or the slope changes at approximately 85% of  $I_{MAX}$ . This dual slope characteristic will be explained in the operation section.

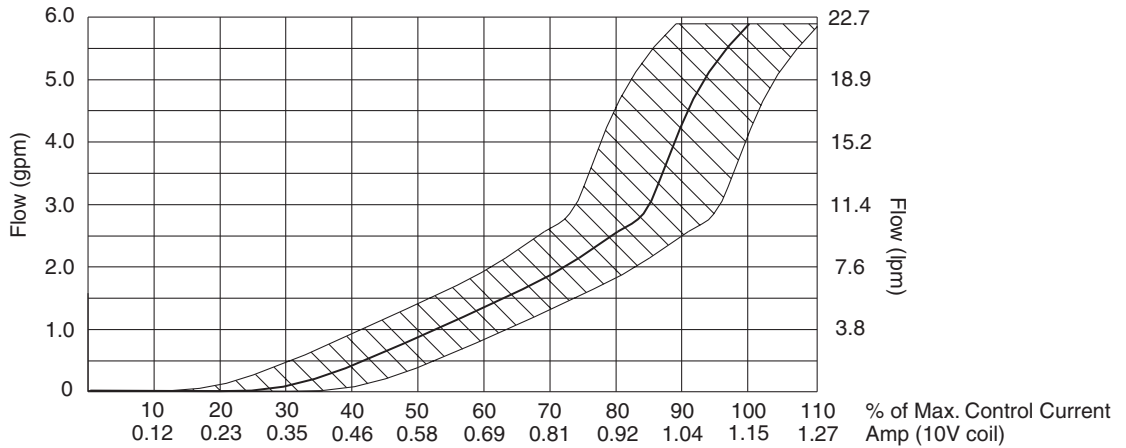


The graph above shows the operating range as well as the hysteresis. The operating range was determined at a differential pressure of 500psi across the SP08-20. This implies a maximum flow of 5.7 gpm. The hysteresis shown on the graph varies depending of the current level. If the current is below 85% of  $I_{MAX}$ , the hysteresis is less than 5%. If the current is greater than 85% of  $I_{MAX}$ , the hysteresis is less than 7%. The four curves on this graph represent various differential pressures across the SP08-20. The graph at 250 psi represents a graph that may describe the performance achieved with a compensator. The graphs at the higher differential pressure demonstrate the ability of the valve to be used in a system where there is no compensator. That is, the valve is able to meter flow regardless of the change in pressure.

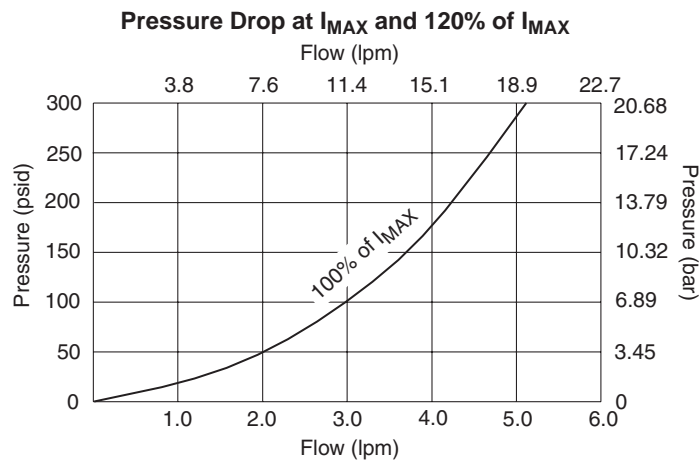
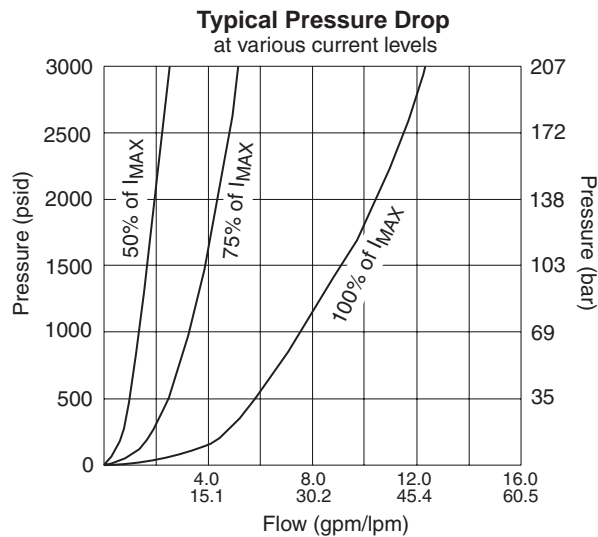


**Reproducibility of SP08-20**

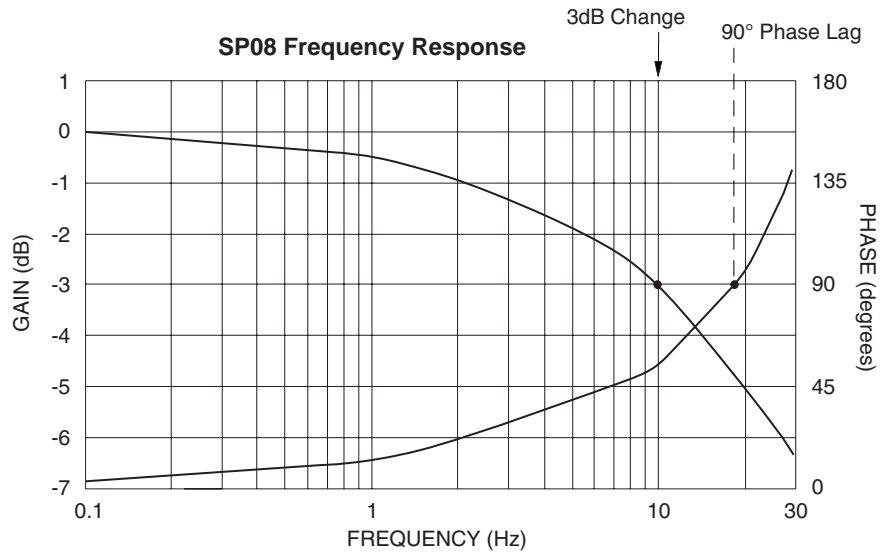
The following graph relates to the previous. The graph below shows the manufacturing limits for the flow vs. current characteristic at an inlet of 500 psi. Essentially, this graph shows the flow vs. current characteristic may vary as much as  $\pm 10\%$  from the nominal curve given on the previous graph. This valve to valve tolerance band can be expected on any of the SP products.



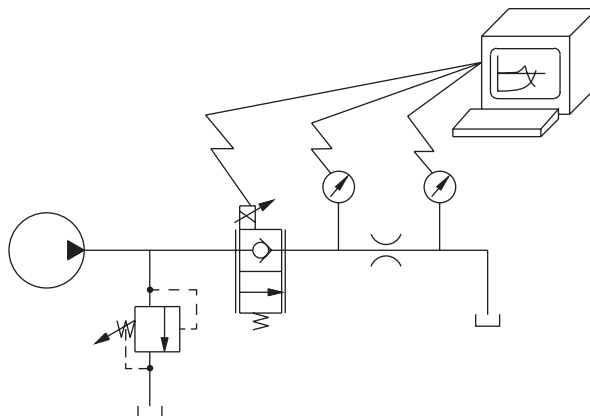
The following graphs describe the pressure drop across the valve when the current is set at various levels. The first graph shows the pressure when the current is set to 50% and 75% of  $I_{MAX}$  as well as when the current is at  $I_{MAX}$ . The second graph is an enlarged view of the first. It shows only the pressure when the current is set to  $I_{MAX}$  and 120% of  $I_{MAX}$ . This curve demonstrates that in low duty cycle applications, a lower pressure drop may be possible if the current can be increased above  $I_{MAX}$ .



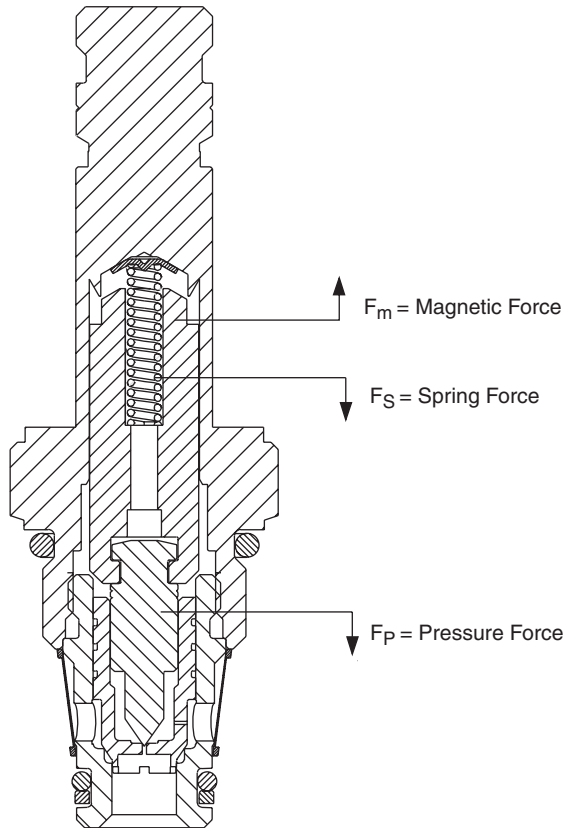
The last graph is the Bode plot or frequency response characteristic. The bandwidth or operating range is measured by either picking the frequency where the magnitude exceeds a 3dB change or where the output lags the input by 90°. The points on the graph below show that the bandwidth is either 10 Hz at a 3dB change or 18 Hz at a 90° phase lag.



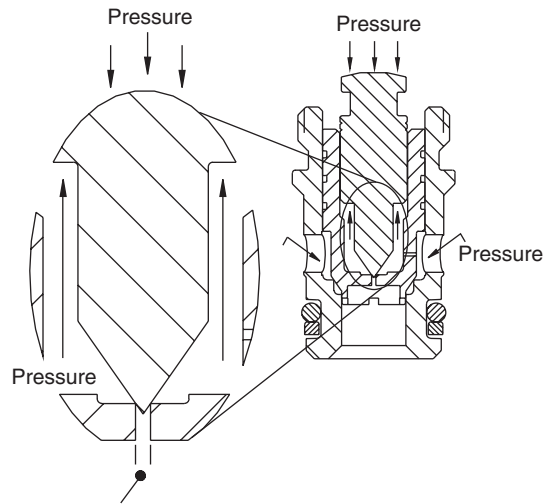
The test was done with the current set at the midpoint of the range and it was varied sinusoidally. The output measured was the pressure drop across a fixed orifice. The circuit is shown below. The frequency response bandwidth is greater than 5Hz for the SP product range.



### Forces Acting on SP08-20



There are three main forces acting on the components. These are the spring force, magnetic force and pressure force acting on the pilot pin. See the drawing to the left for the relative direction the forces act. The description of the spring force can be found in various other chapters. The pressure force acting on the pilot pin is summarized in the drawings below. The pressure is applied equally to all surfaces of the pilot pin, except for the area where the pilot pin seats. Since the pilot pin seals off this area, the pressure cannot be applied to this part of the pilot pin. Therefore, there is a force pushing on the pilot pin seat. (This force is further clarified in the HydraForce Solenoid Manual).

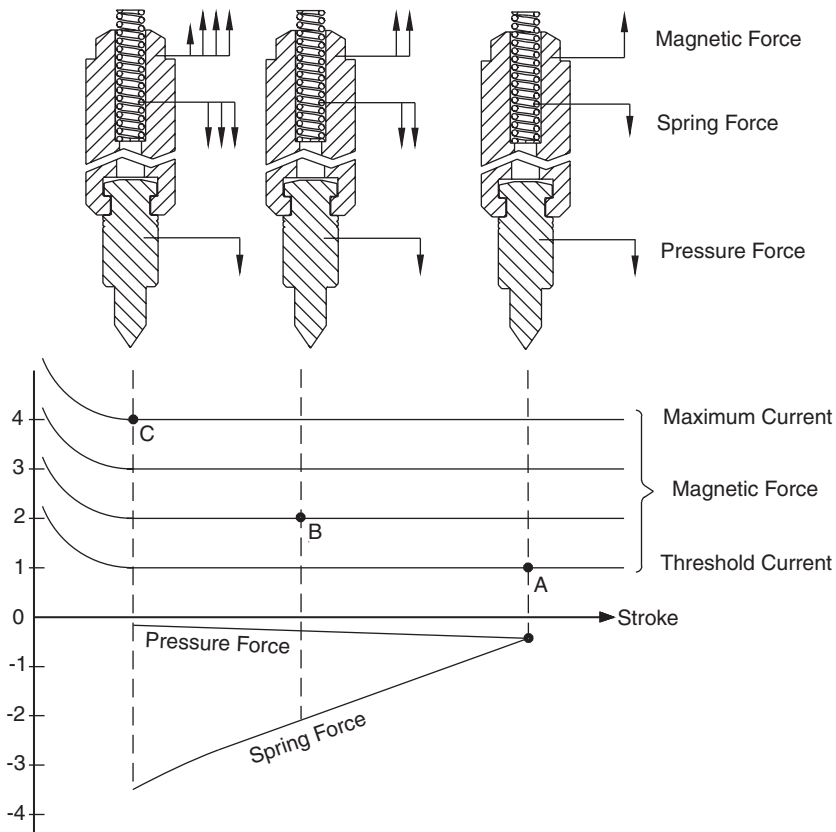


Area of Pilot Pin Seat

$$F_p = \text{Pressure Force} = \text{Pressure} \times \text{Area of Pilot Pin Seat}$$

The last force shown on the cross section is the magnetic force. The force vs. stroke characteristic for the armature is flat or horizontal. This is the characteristic for a linear solenoid actuator and can be seen on the graph below.

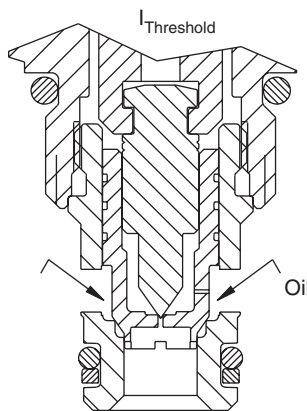
The three forces are summarized on the graph below. The magnetic force is represented by four lines on the plus side of the graph. Four lines are shown to indicate that the magnetic force is a function of the current applied. The line closest to the x-axis is the magnetic force developed when the threshold current is applied. The line furthest from the x-axis represents the magnetic force when the maximum current is applied. The pressure force and spring force are shown on the negative side of the graph. The fact that one force is positive and the others are negative is a matter of convention simply to show the forces act opposite to each other.



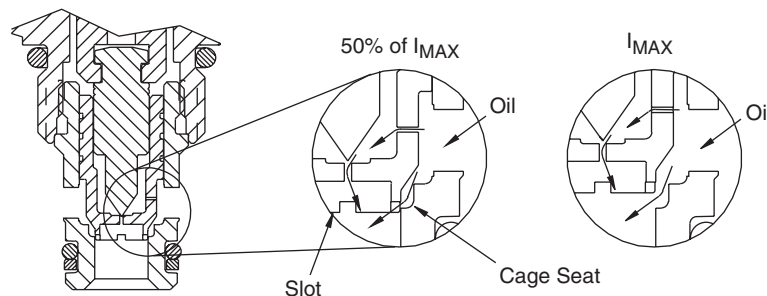
The graph above illustrates that the forces are balanced throughout the stroke of the armature. Also, the pictures of the spring, pilot pin and plunger show the forces being applied to the individual components. For example, at point A, the magnetic force developed by the threshold current is equally matched by the pressure and installed spring force are 0.5 while the magnetic force is 1.0. Using the force balance equation  $F_M = F_P + F_s$  along with these numbers from the graph, shows that the forces balance or  $1 = 0.5 + 0.5$ . If the current is increased, the valve opens. This is because as the current increases, the magnetic force increases proportionally. The magnetic force causes the armature to move, compressing the spring, and the forces remain balanced whether at point A, B or C or anywhere in between.

### Operation of SP08-20

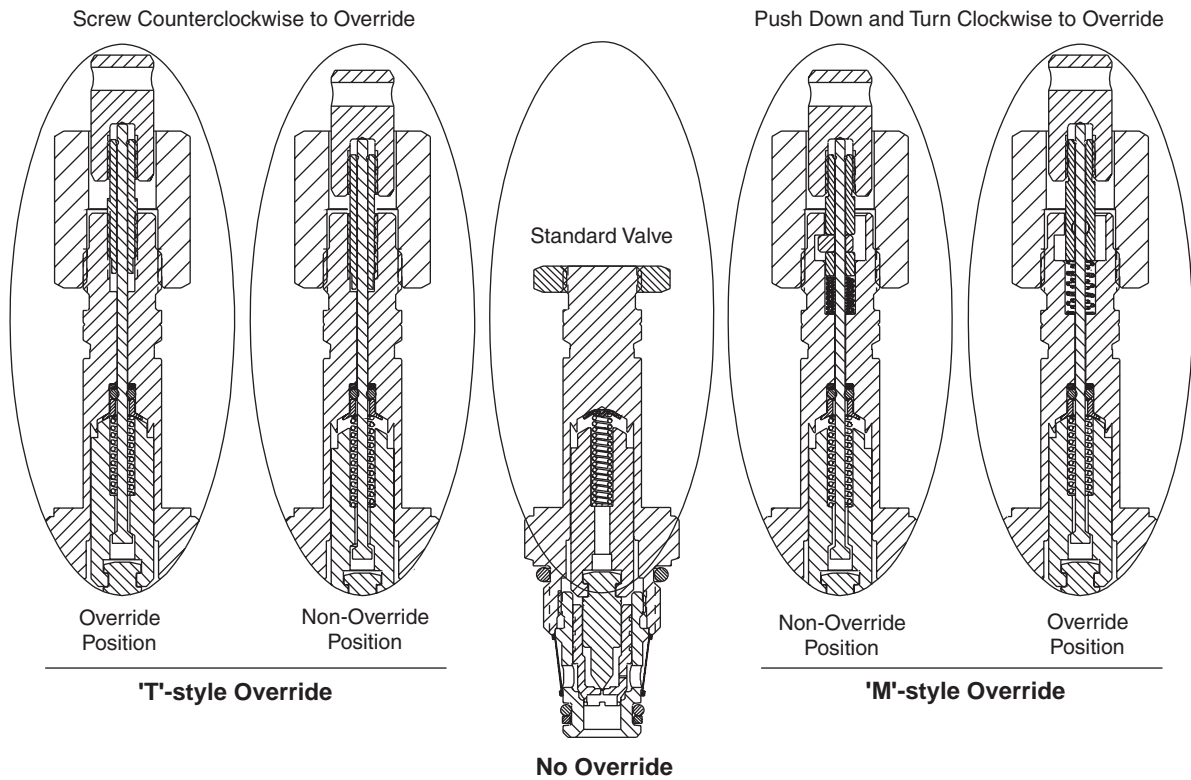
The following three diagrams depict the operation of the SP08-20 when various levels of current are applied. As with any proportional valve, when current is applied, a magnetic force is created by the solenoid. The armature or plunger is then attracted to the pole piece. In the case of the SP08-20, the pole piece is the plug of the tube subassembly. As the plunger moves toward the plug, it begins to compress the spring. Before this occurs however, the magnetic force must overcome the pressure force acting on the pilot pin. The first diagram below shows that oil is blocked from flowing between ports 2 and 1. This picture represents the valve operation before the threshold current is reached.



The diagrams below show the pilot pin and poppet are no longer seated. The picture to the left represents the performance when approximately 50% of  $I_{MAX}$  has been applied. The picture to the right represents the position of the poppet at  $I_{MAX}$  applied to the coil. Notice that the poppet is further from the seat than the one on the left. Both of these pictures represent how the opening of the valve is dependent on the current applied. The position of the poppet in these pictures illustrates how the knee in the flow vs. current graph is formed. In the middle picture the slots in the poppet (known as the *crown of the poppet*) are still below the edge of the seat. This crown feature provides for a small change in flow as the current changes. Once the crown is out of the seat area, the metering gain characteristic is dependent on a combination of the cone of the poppet and the crown. This is a much larger gain, hence the shape of the flow vs. current graph.



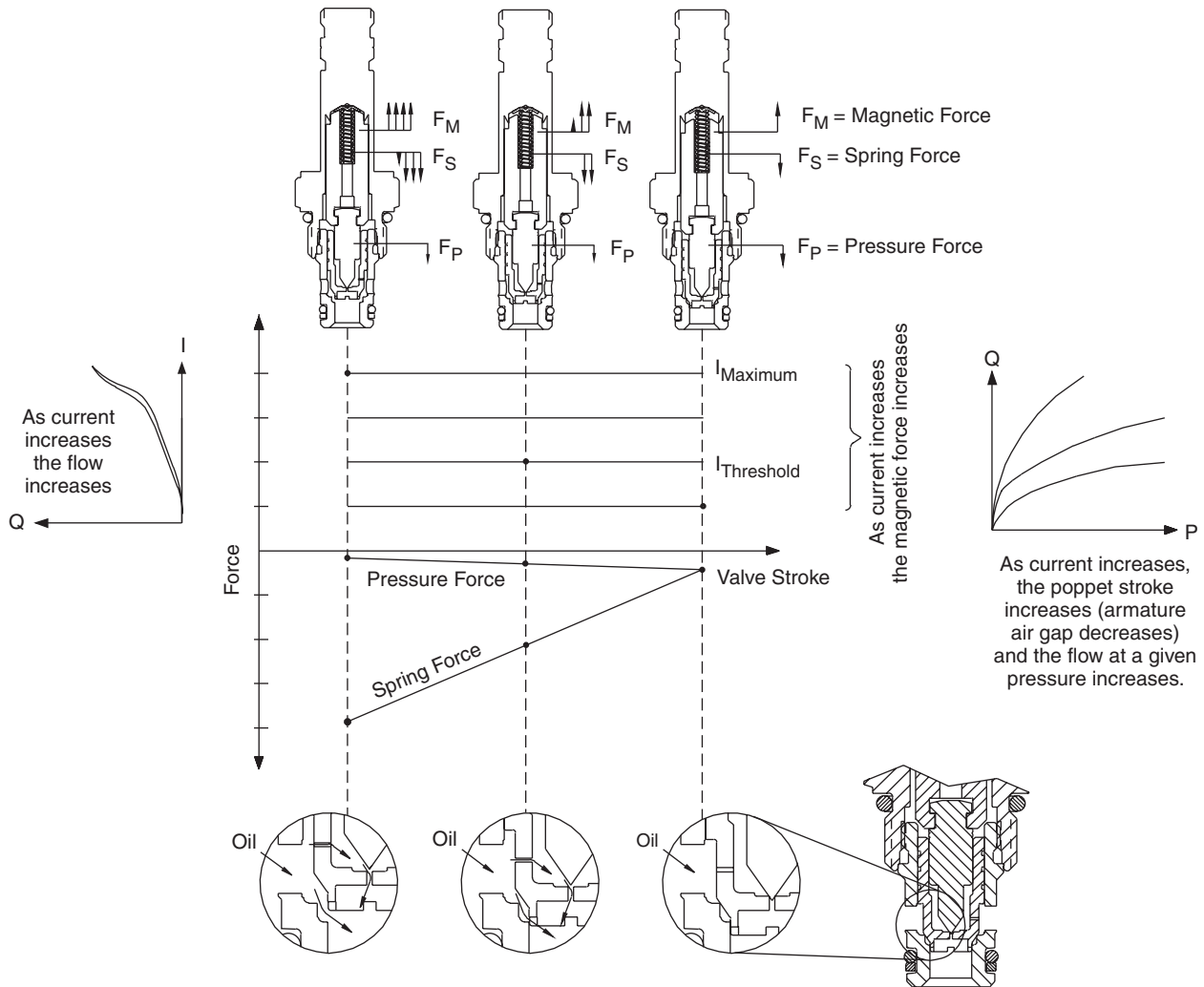
**Manual Override for SP08-20**



The diagram above shows the two styles of standard overrides available for the SP08-20. These are the 'M' and the 'T' type overrides. The one on the right is a detent style. To operate this type, the user pushes down on the override and turns it clockwise until it hits a stop. The user can then release it and the springs move the plunger to the override position. In the case of the 'T' style override, shown on the left, the user simply turns the override counterclockwise. This turning action gradually moves the valve components into an override position until a positive stop is achieved. The 'M' style gives two discrete positions like an on/off valve and the 'T' style gives infinite positions like a proportional valve. It is important to note that there is no change in performance in regards to the flow vs. current characteristic for these two overrides. This is not true of the 'J' style override (not shown). To operate this override, the operator simply pulls on the knob. The flow achieved at a given current when the valve is constructed with this type of override is less than the standard valve, because of a decrease in magnetic force.

### Tying It All Together

The diagram below is a summary of how the forces interact to influence the performance of the valve.





The diagram to the left ties together the performance, operation and forces that were presented in previous sections. The graph in the middle summarizes the forces acting on the components. These forces are further clarified by the small pictures of the valves above the graph. Moving right to left as the current increases, the magnetic force increases and the air gap decreases. This results in an increase in spring force. The pictures below the graph show the valve in operation. Again, from right to left as more current is applied, the poppet opens further. This relationship between flow and an increase in current is depicted by the flow vs. current graph to the left of the force graph. This graph was turned on its side to show the relation between flow and force. The last graph on this page to the far right shows the flow relation between pressure drop and how current influences this characteristic. Each line starting from the one closest to the pressure axis represents an increase in current. This graph shows as the current increases more flow passes through the valve at the same current. Another way of saying this is that the pressure drop through the valve decreases as current increases.

## Summary

In this chapter the following concepts were presented:

- Discovered what a two way, proportional, poppet flow control is.
- Learned about the similarities between the SP and the SV valves.
- Understand why special manufacturing processes were developed for the SP valve.
- Brief descriptions of the performance curves were presented.
- A graphical and pictorial description of the forces acting on the components was discussed.
- The operation of the manual overrides was presented.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. List some similarities between the SV08-20 and the SP08-20. \_\_\_\_\_
2. True or False. The SP08-20 uses a push style actuator like the other proportional flow control valves. \_\_\_\_\_
3. Describe the three significant forces acting on the components of the SP08-20. \_\_\_\_\_
4. When are the forces of the SP08-20 not balanced? \_\_\_\_\_
5. What special processes were developed for the SP products? Why was this done? \_\_\_\_\_
6. Why is a pressure drop curve where the current is equal to 120% of  $I_{MAX}$  described in the performance section? \_\_\_\_\_
7. True or False. The performance of the SP08-20 remains unchanged when any of the manual override selections are selected. \_\_\_\_\_
8. True or False. The SP08-20 uses the same coil as the EHPR08-33. \_\_\_\_\_

**Notes:**

# Chapter 12: Proportional Directional Control

## Objectives

- Learn about the spool type proportional directional control valves available from HydraForce.
- Discover the similarities and differences between the proportional directional controls and the on/off directional controls.
- Understand the construction of the proportional directional controls.
- Learn about the metering characteristics of the proportional directional control valves.
- Learn about proportional directional control valves with an integral load sense port.
- Understand the performance of the “L” option spool valves.

## Introduction

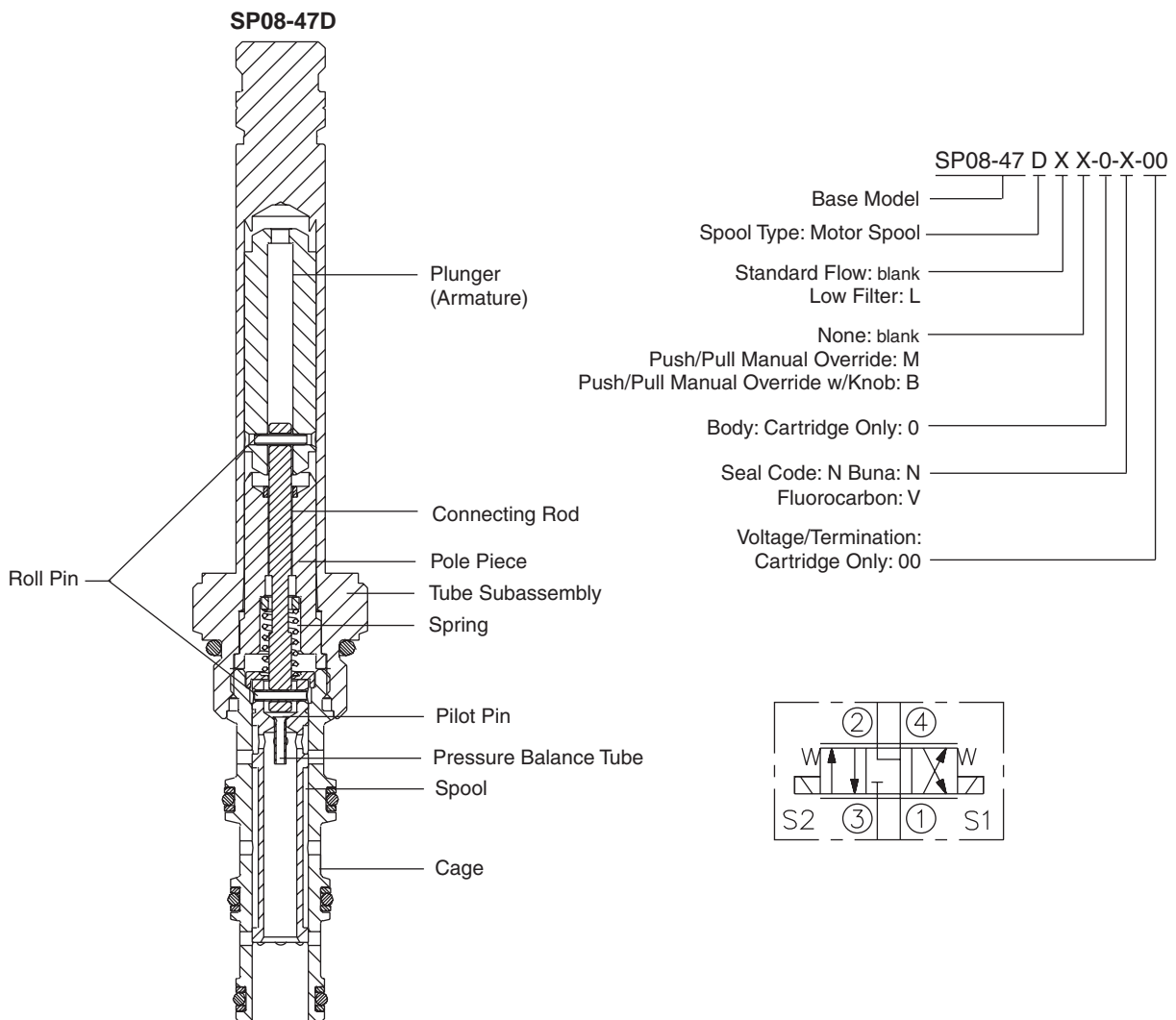
In this chapter the proportional spool type directional control valves will be presented. This includes both two position, four-way and three position, four-way valves. Also, a three position valve unique to HydraForce with a load sense port built-in will be described. The performance, operation and forces acting on the components will be presented in detail. Beside the typical pressure drop and flow vs. current graphs, a pressure compensation curve will be presented. This should give a further understanding of the performance and operation of these valves.

# HYDRAFORCE

### SP08-47D

The SV08-47D is a 3-position, 4-way proportional directional valve. Like the SV08 and SV10-47, this valve uses two coils acting independently of each other to actuate the valve to the secondary positions. The construction is similar to that of on/off style valves. However, they differ significantly in the shape of the armature and pole piece face. Two types of spool configurations are offered in the SPXX-47 series of products, the C spool (all ports blocked) and the D spool (motor spool, work parts connected to tank and inlet blocked). Within the 08 size product there is also an L version (low flow) of each type. The L version has a lower maximum control flow and it also has a lesser amount of droop in the flow vs. pressure compensator characteristic as compared to the standard version. This will be explained further in the performance section.

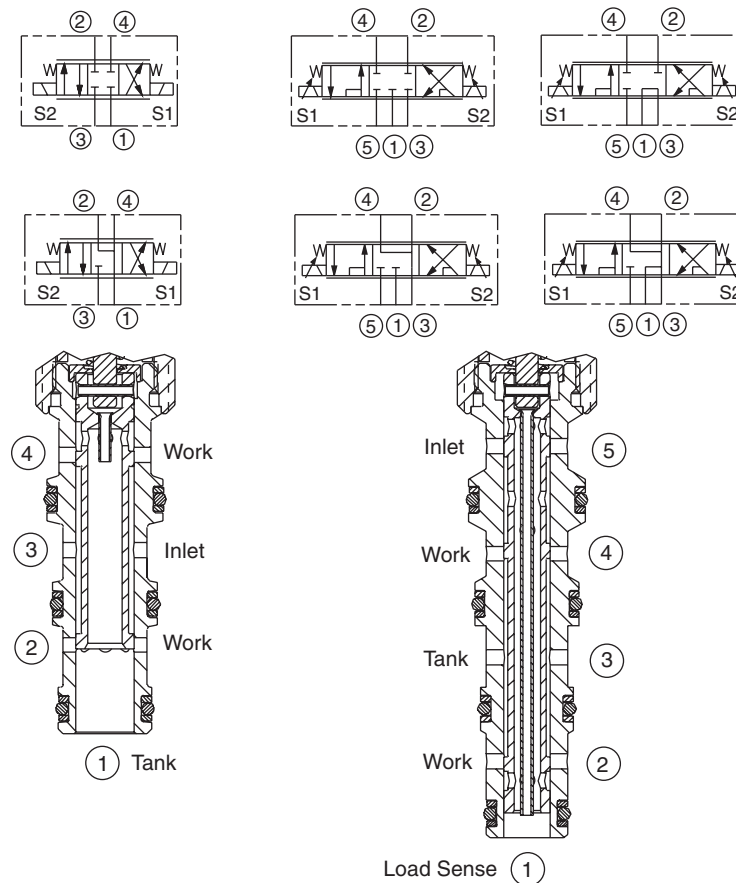
Shown below is the cross section, model code and symbol. In the following sections the performance, forces acting on the valve and operation of this valve are discussed. Also included in this section is a brief description of the SP10-57 and 58 series.



**Comparison of SP10-47, SP10-57 & SP10-58**

The diagram below shows the cross section of the SP10-47D and the SP10-57D. Also shown are the symbols that represent the various products available in this series of valves and includes the 58 style. While the model code denotes that this is a five ported valve, the hydraulic function is similar to the 47 product in that there is an inlet port, tank port and two work ports. The fifth port is intended to be used to feed a load sense line.

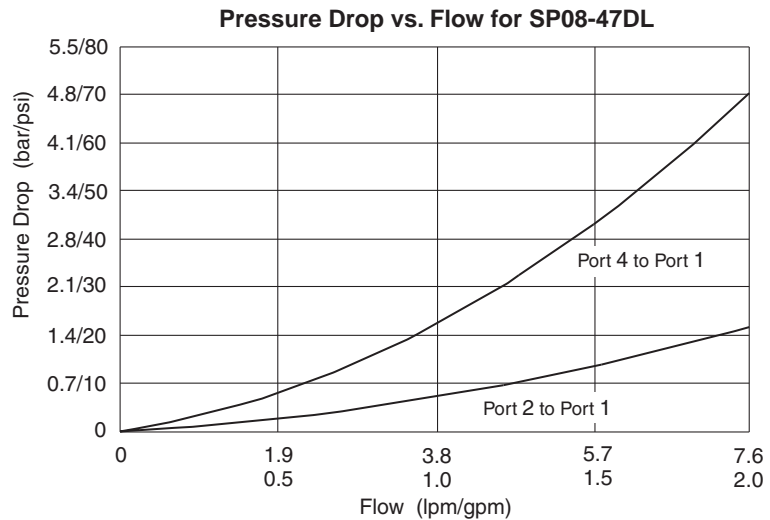
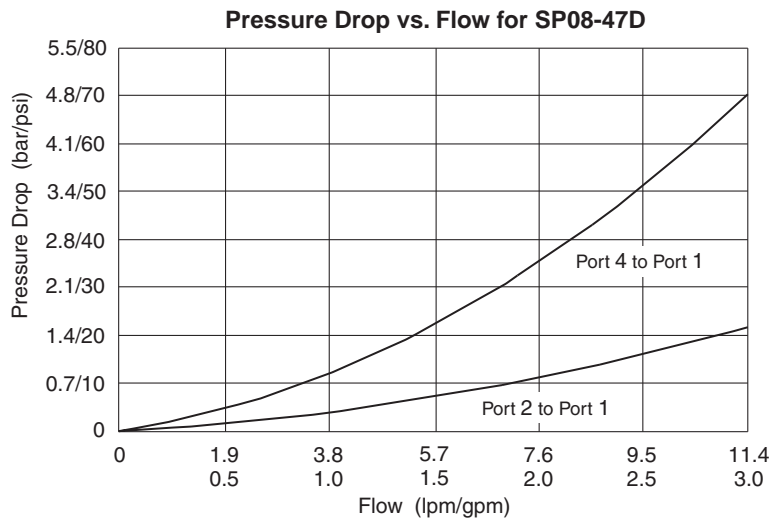
The port logic is different for the 57 and 58 compared to the 47 style symbol. Port 5 is the inlet port on the 57 and 58 where port 3 is the inlet port for the 47. Also, port 3 is the tank port for the 57 and 58 while port 1 is tank in the 47. Because of these differences, the manifold designer cannot simply stretch the block to add the load sense functionality. Instead, if the design of a block that originally used a 47 style valve is migrated to one that uses a 57 style, the manifold design will need to change significantly.



Notice that neither the ‘A’ spool (tandem center) or ‘B’ spool (all ports connected in neutral) are shown. The reason is that these types of flow paths would cause a possible unsafe condition. Development of these spools showed that during transition, the tank port would be connected to one of the work ports prior to the opposite work port being connected to pressure. This could result in an unexpected movement of the actuator which could result in a load dropping when an operator would not expect it.

### Performance of SP08-47D

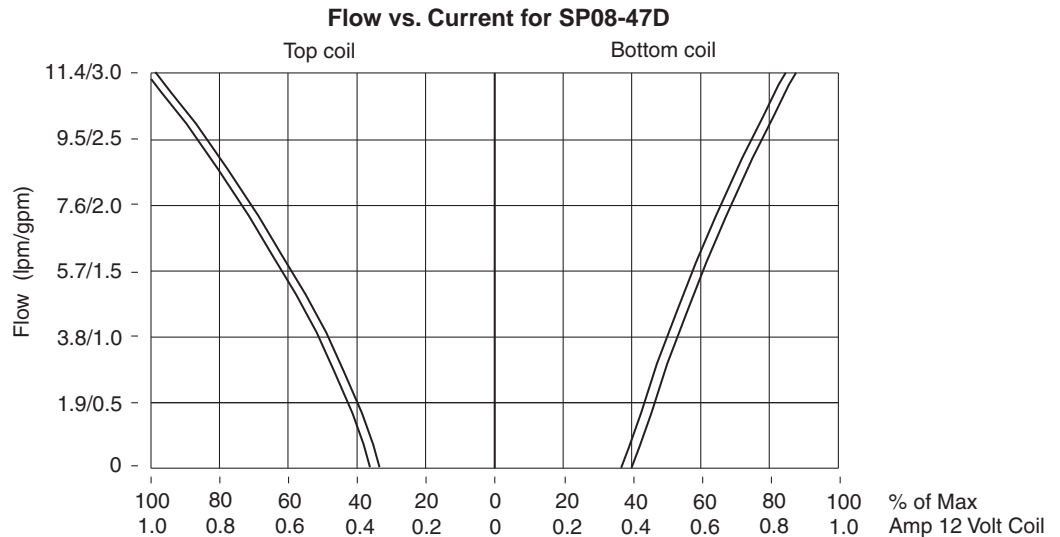
In the following section, the graphs that describe the performance of the SP08-47D and the SP08-47DL are presented. The performance of the standard and low flow spool are included in this section. These performance characteristics include pressure drop, flow vs current and the pressure compensation characteristic. The two graphs below represent the pressure drop of the SP08-47D and SP08-47DL.



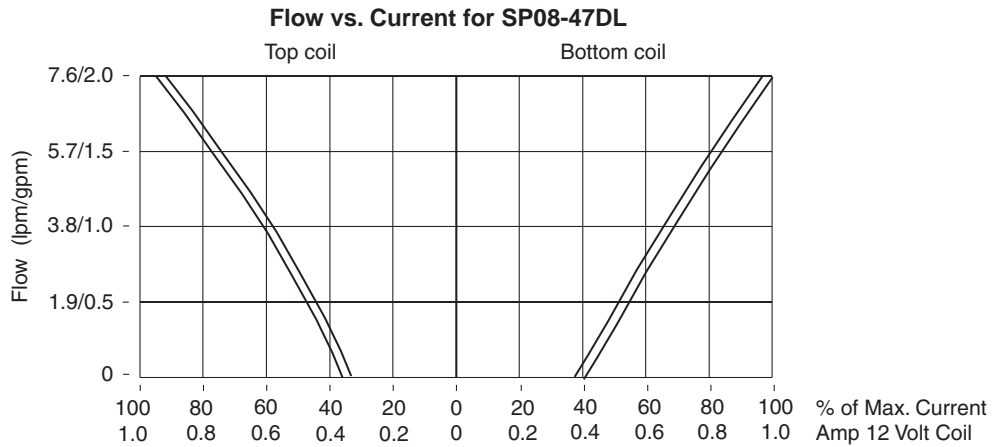


The pressure drop shown in the previous two graphs was recorded with I<sub>MAX</sub> applied to the coil. Notice that the pressure drop from port 2 to 1 is lower than that of 4 to 1. If this valve or the C spool valves are connected to a double-acting cylinder, the blind side should be connected to port 2 to give the lowest overall effect on system pressure. The curves show that the pressure drop through the standard spool is the same as the low flow spool at the rated flow for each spool type. That is the control orifice of the L spool is smaller at I<sub>MAX</sub> as compared to the standard. Thus at equal pressure drop, the flow will differ.

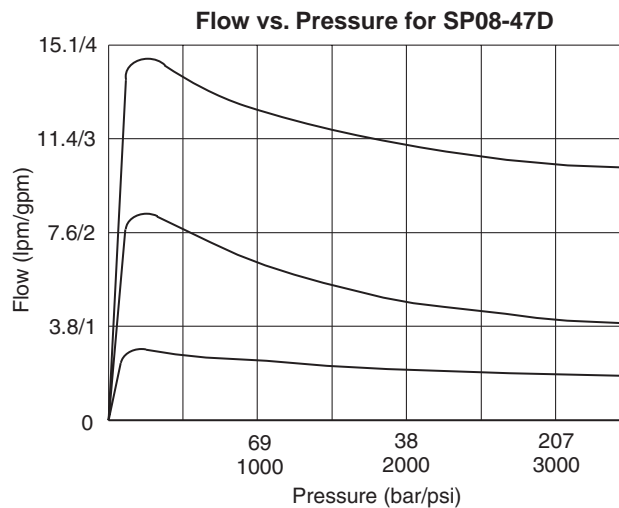
The next graph used to describe the operating characteristic of these valves is the flow vs. current characteristic. The maximum flow is based on an inlet pressure at 500 psi. For the standard spool type the flow range is 0 to 3.0 gpm and the flow through the L type at this same pressure is 0 to 2.0 gpm. The graph also shows that hysteresis is less than 7% for both spool styles.

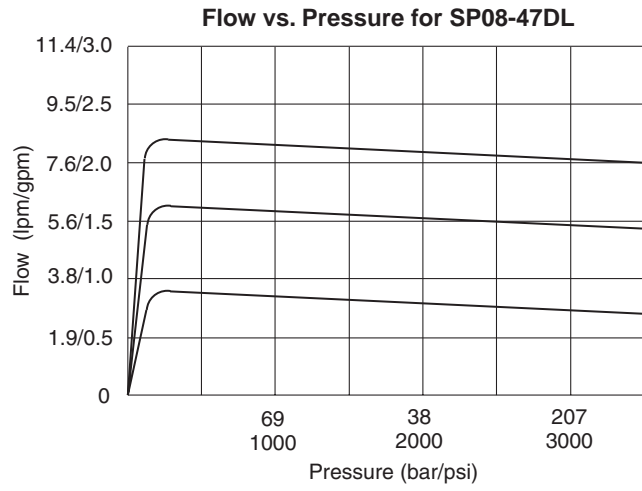


Notice that the bottom coil or S2 curve is steeper than the top coil curve. This will be explained in the operation section.



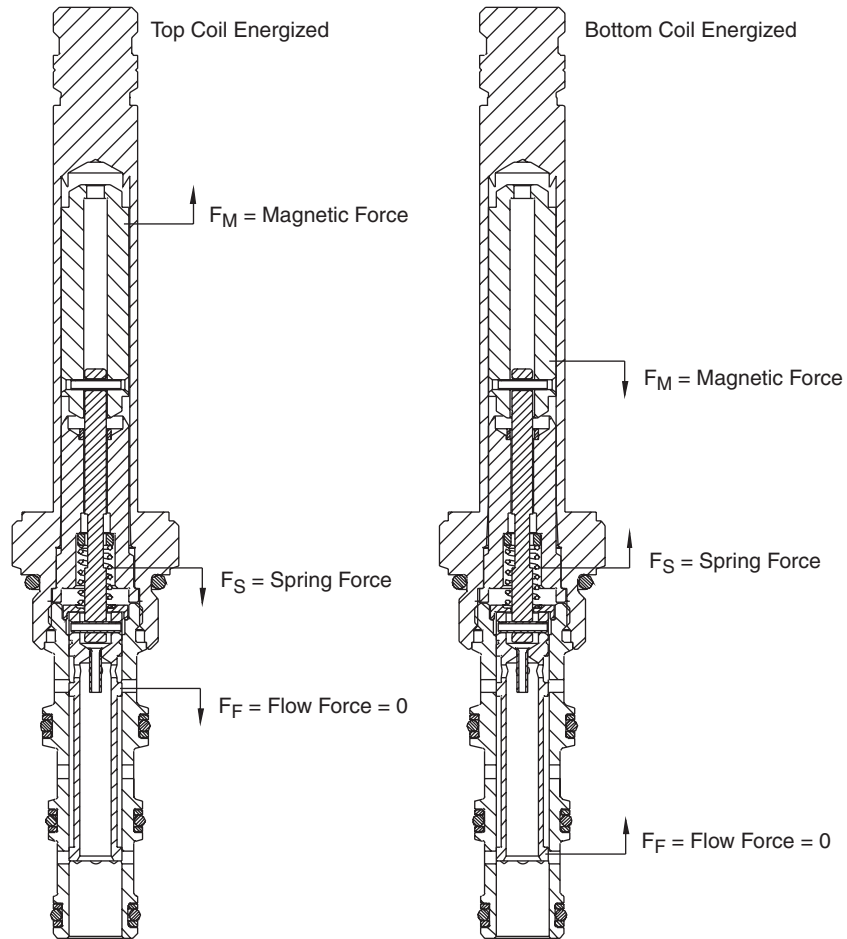
The last two graphs show the change in regulated flow as the pressure differential across the valve varies. The first graph shown below is the pressure compensation curve of the SP08-47D. The graph shows that as the pressure drop across the inlet, through the work ports increases, the flow decreases. (Note: Each curve represents a different current level which is held fixed throughout the test.) This change in flow or droop for the standard spool (SP08-47D) is 35%. The change in flow for the SP08-47DL is only 18% which can be seen on the next page.





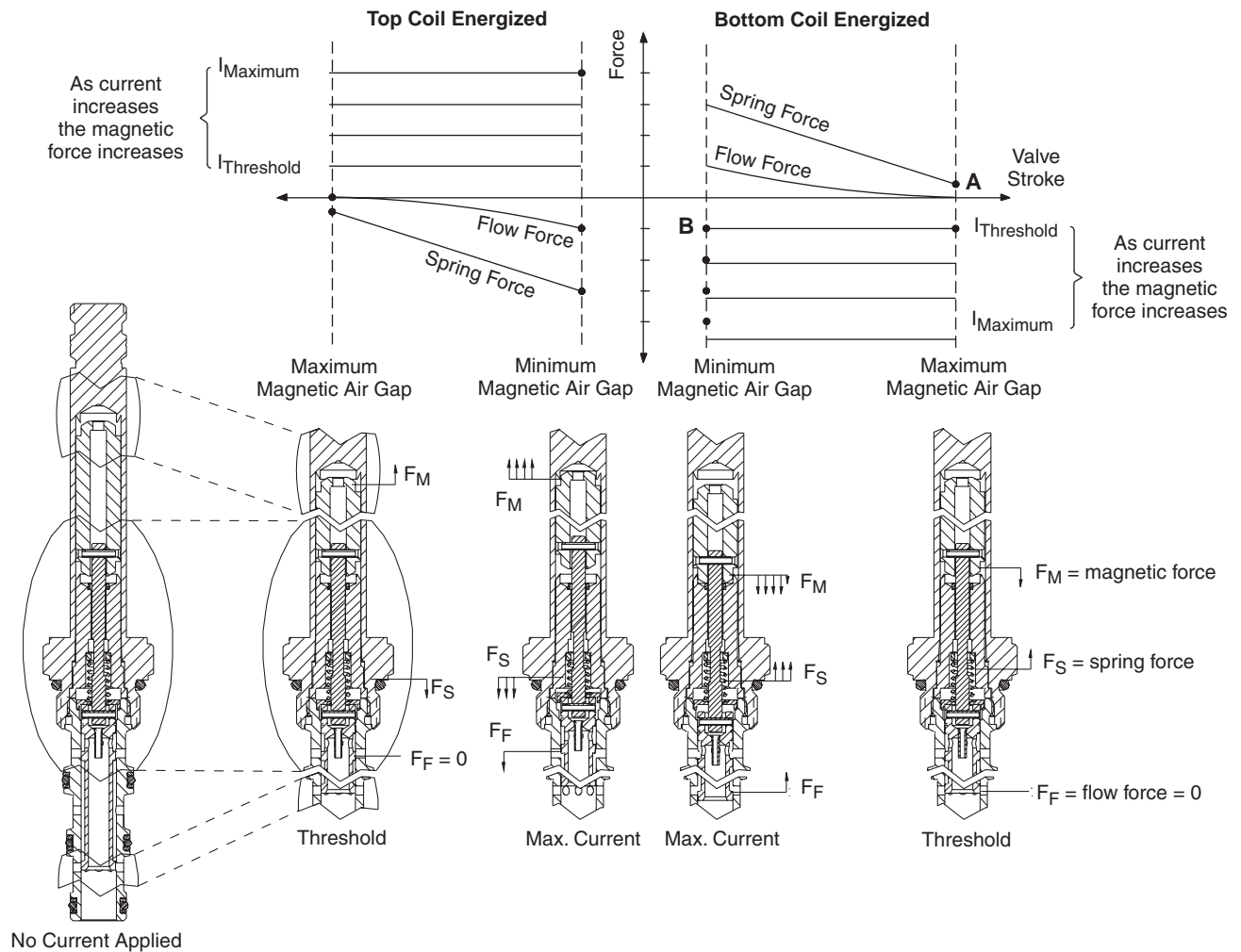
### Forces Acting on SP08-47D

The diagram below shows the valve with an indication of the dominant forces. Below this is a graph showing the magnetic force, flow force and spring force, similar to that of the SP08-41.

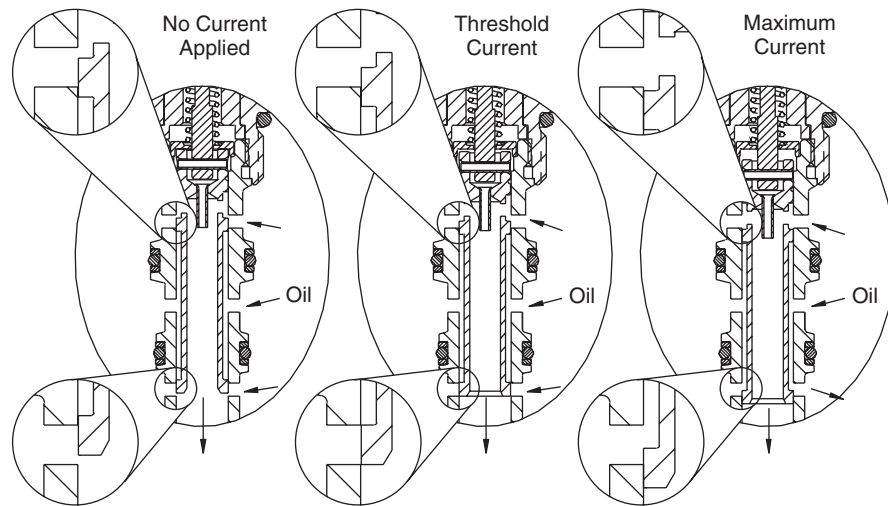


**Operation of SP08-47D**

Shown below are several pictures of the SP08-47D when the bottom or S2 coil is energized. The description of the operation when the top coil is energized is the same except port 4 is the work port and port 2 is connected to the tank. The first picture, below left, shows the valve in the neutral state. The work ports are connected to the tank and inlet is blocked. Going from left to right, the next picture shows the position of the spool when the threshold current is applied to the coil. Notice that oil from port 4 is blocked while oil can flow from port 3 to 2. This type of spool configuration is considered a meter in type spool. Finally, the expanded views to the right show the spool when 50% of the maximum current is applied and when  $I_{MAX}$  is applied.



Notice in the graphs on the previous page, that the magnetic force in the bottom reaches a higher value than the top coil. The magnetic flux is stronger for the bottom coil because the adaptor is able to concentrate the flux better than the coil nut. This is why the flow vs. current for one coil versus the other differs. The spring force for either the S1 or S2 direction is the same but the magnetic force is different. Since the magnetic force of the bottom coil is greater, it is able to overcome the spring force at a lower level. That is why the S2 or bottom coil curve is steeper.



## Summary

In this chapter the following concepts were presented:

- The difference between the port logic of the SP10-47s, 57s and 58s.
- How the flow force affects the pressure compensation characteristic of the proportional spool valves.
- The difference in performance between the SP08-47D and SP08-47DL.
- The definition of meter-in and meter-out spools.
- Why the flow vs. current characteristic curve differs between the S1 and S2 coil.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. Why is there a difference between the Flow vs. Current characteristics of the S1 coils as compared to the S2 coil? \_\_\_\_\_
  
2. True or False. The tank port for the SP10-47D, SP10-57D and SP10-58D is port 1. \_\_\_\_\_
  
3. True or False. The maximum flow that is controlled by the SP08-47D is the same as the SP08-47DL. \_\_\_\_\_



# Chapter 13: Application of Proportional Flow Control

## Objectives

The objectives for this chapter are as follows:

- Review the benefits of compensation.
- Learn about the cost to add compensation to a hydraulic circuit.
- Learn about internal vs. external compensation.
- Understand how proportional flow control valves regulate the movement of cylinders in a fork lift.
- Learn about the use of a proportional flow control to regulate the speed of hydraulic motors.
- Learn about the use of the SP12-20 in a lift/lower circuit.
- Understand how the SP08-47 can improve the performance of the hydraulic system.

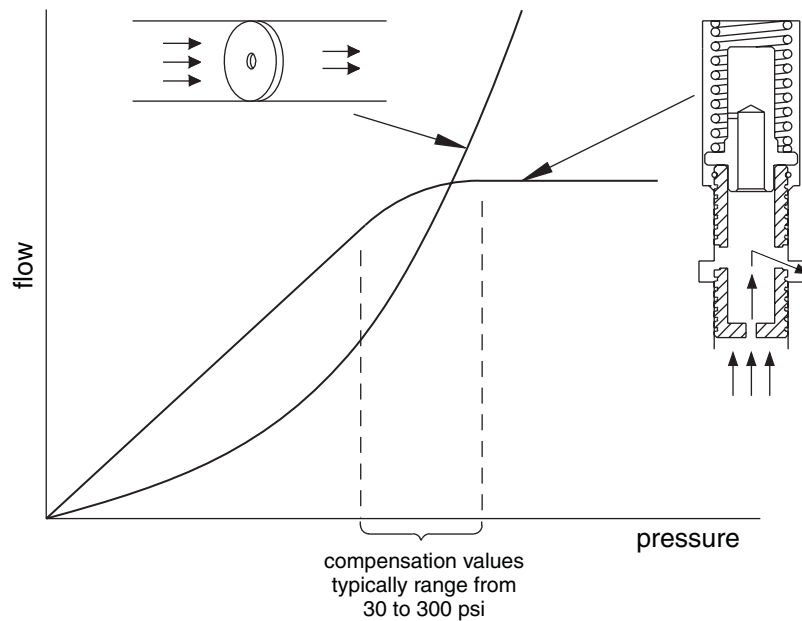
## Introduction

Several applications using proportional flow controls will be introduced in this chapter. The costs and benefits of compensation are given first, and the various compensating values of the HydraForce flow controls are tabulated. Following this is the proportional control of cylinders in a forklift, the control of a motor in a salt spreader and motors driving a track vehicle.

## Pressure Compensation

### Benefits and Costs

A spool and spring which make up the compensation function of a flow control were presented in the previous chapters. The compensation function assures the desired flow is maintained regardless of the changes in load pressure. Also, the performance and operation of a fixed orifice and pressure compensated orifice were discussed in Chapter 7. The flow of the fixed orifice varies when the pressure required to move the load changes. In other words, as the pressure drop changes, [recall  $Q = 31 A (DP)^{1/2}$ ] the flow changes. With the compensator, the spool moves in response to the pressure and spring in order to maintain a constant flow. The graph below summarizes the benefit of the pressure compensated flow control. Assume the hole in both the fixed orifice and the compensator are of a diameter such that the regulated flow is the same at the pressure drop equal to the compensation value. The graph shows how the flow through the fixed orifice continually changes as the pressure changes while the flow through the compensator remains unchanged once the compensation value is reached.



The benefit of having consistent flow regardless of the load pressure, translates into consistent machine behavior. This means productivity improves because the operator is able to predict the machine motion and speed. The improvement comes at the cost of an additional valve that is slightly more expensive than the fixed orifice. There is one other cost associated with a compensator which is the addition of pressure drop. By adding another valve or function to the system, the pressure drop increases. The increase in pressure drop means that the hydraulic system requires more energy. This energy is measured in horsepower (hp) or kilowatts (kw). Since the pressure drop across a compensator does not perform any work (i.e. doesn't move a cylinder or motor), this is considered to be a loss. A simple equation is used to determine how much energy is lost due to the compensator regulating flow. This equation is:

$$\text{hp} = \text{flow (gpm)} \times \text{pressure (psi)} \div 1714$$

To convert to kilowatts, simply multiply the result by 0.75. Let's look at an example of what the power would be for pressure compensation to occur when 6 gpm flows through the PV70-30. This is found by multiplying 6 gpm by 165 psi and dividing by 1714. The loss would be 0.58 hp.

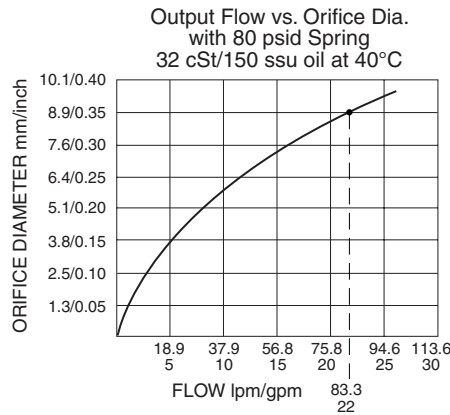
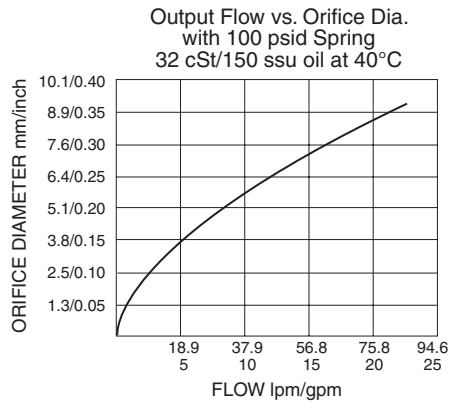
This horsepower loss is the reason why the lightest compensator spring should be selected. This is one of the factors which the system designer can control. By decreasing the compensating spring value, the horsepower loss decreases accordingly. For example, if the spring value was 100 psi then the horsepower loss would be 0.35 hp ( $0.35 \text{ hp} = 6 \text{ gpm} \times 100 \text{ psi} \div 1714$ ). The flow could be reduced as long as the actuator size can be varied accordingly. If this level of horsepower loss is unacceptable while the machine is idling, a 2-way solenoid valve can be installed between the proportional valve and the pump to dump the oil to tank when the hydraulic power is not required. Further, the horsepower loss across the compensator generates heat. The horsepower loss and the heat generated when a dump valve is used is less than a system which does not have a dump valve. This may allow the size of the cooler to be decreased as well as the size of the reservoir.

## Compensation Values

The pressure compensation values at which the spring force balances the pressure drop force across the compensating spool are tabulated below. The first table is for the external compensators used with the PV70/72-33/35.

<b>EC10-30</b> (priority style)	40 psi (2.8 bar) 80 psi (5.5 bar) 160 psi (11.0 bar)	<b>EC16-32</b> (restrictive style)	80 psi (5.5 bar) 160 psi (11.0 bar)
<b>EC12-30</b> (restrictive style)	100 psi (6.9 bar) 160 psi (11.0 bar)	<b>EC12-34</b> (restrictive style)	100 psi (6.9 bar) 160 psi (11.0 bar)
<b>EC10-40</b> (priority/bypass style)	40 psi (2.8 bar) 80 psi (5.5 bar) 160 psi (11.0 bar)	<b>EC16-32</b> (restrictive style)	250 psi (13.8 bar)
<b>EC12-40</b> (priority/bypass style)	60 psi (4.1 bar) 75 psi (5.2 bar) 100 psi (6.9 bar) 160 psi (11.0 bar)	<b>EC10-42</b> (priority/bypass style)	80 psi (5.5 bar) 150 psi (10.3 bar)
<b>EC16-40</b> (priority/bypass style)	40 psi (2.8 bar) 80 psi (5.5 bar) 160 psi (11.0 bar)	<b>EC16-42</b> (priority/bypass style)	80 psi (5.5 bar) 150 psi (10.3 bar)
<b>EC10-32</b> (restrictive style)	80 psi (5.5 bar) 150 psi (10.3 bar)	<b>EC12-42</b> (priority/bypass style)	80 psi (5.5 bar) 160 psi (11.0 bar)
<b>EC12-32</b> (restrictive style)	80 psi (5.5 bar) 160 psi (11.0 bar)	<b>EC42-M42</b> (priority/bypass style)	80 psi (5.5 bar) 150 psi (10.3 bar)

The use of the EC16-40 was not previously described. It could be used with the PV72-33 for various application reasons. One reason may be that the flow discharged through the bypass exceeds the rating of the EC12-40. Another reason may be to use a lower compensating spring value in an EC16-40 compared to the EC12-40 and yet meet the same maximum regulated flow. The two graphs on the following page are used to describe how a lighter spring in the EC16-40 can meet the same flow as an EC12-40 with a higher compensator spring value. These graphs show the value of the regulated flow for a given orifice size for either the EC12-40 or the EC16-40. Notice that the graph for the EC12-40 represents the performance when the compensation value is 100 psi while the compensation value for the EC16-40 is 80 psi. The reason that this occurs is because the flow force acting on the 16 size compensator is lower than that of the 12 size EC. Since the flow force is greater, some of the force from the 100 psi bias spring is needed to keep the spool from washing shut.



By comparing a specific orifice size, it is apparent that the EC16-40 is able to regulate the same amount of flow as the EC12-40, even though the compensating value is lower. For example, the flow regulated with an orifice diameter of 0.35 inches is 22 gpm for either compensator.

The table below summarizes the compensating values of the internally compensated proportional flow controls. The values listed can be used to determine the energy loss of the hydraulic system by using the equation previously given in this section.

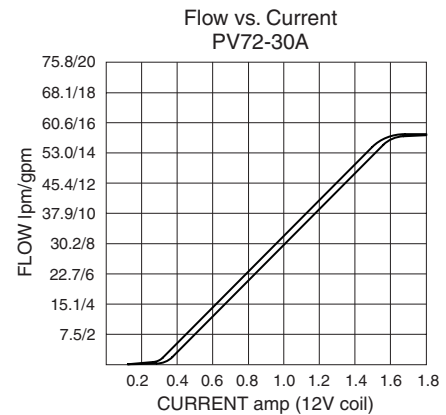
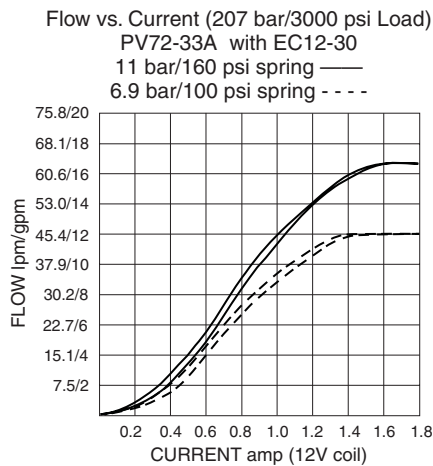
PV08-30	125 psi	8.6 bar
PV08-31	125 psi	8.6 bar
PV70-30	125 psi	8.6 bar
PV70-31	125 psi	8.6 bar
ZL70-30	190 psi	13.1 bar
ZL70-31	190 psi	13.1 bar
ZL70-33	75 psi	5.2 bar
ZL70-36	50 psi	3.5 bar
PV72-20	245 psi	16.9 bar
PV72-21	245 psi	16.9 bar
PV72-30	165 psi	11.4 bar
PV72-31	165 psi	11.4 bar
ZL72-30	230 psi	15.9 bar
ZL72-31	230 psi	15.9 bar
ZL72-33	150 psi	10.3 bar
ZL72-36	145 psi	10.0 bar
ZL76-30	205 psi	14.1 bar
PV76-31	205 psi	14.1 bar
ZL76-30	285 psi	19.7 bar
ZL76-33	100 psi	6.9 bar
ZL76-36	145 psi	10 bar

**Internal vs. External**

Shown in previous chapters are the internally and externally compensated flow control valves. There are inherent benefits to using each. The obvious reason for the internal version is that only one cavity is required to provide this function. This minimizes the space in the manifold as well as eliminates the cross drilling between the two cavities. This reduction in machining and manifold space translates into cost savings.

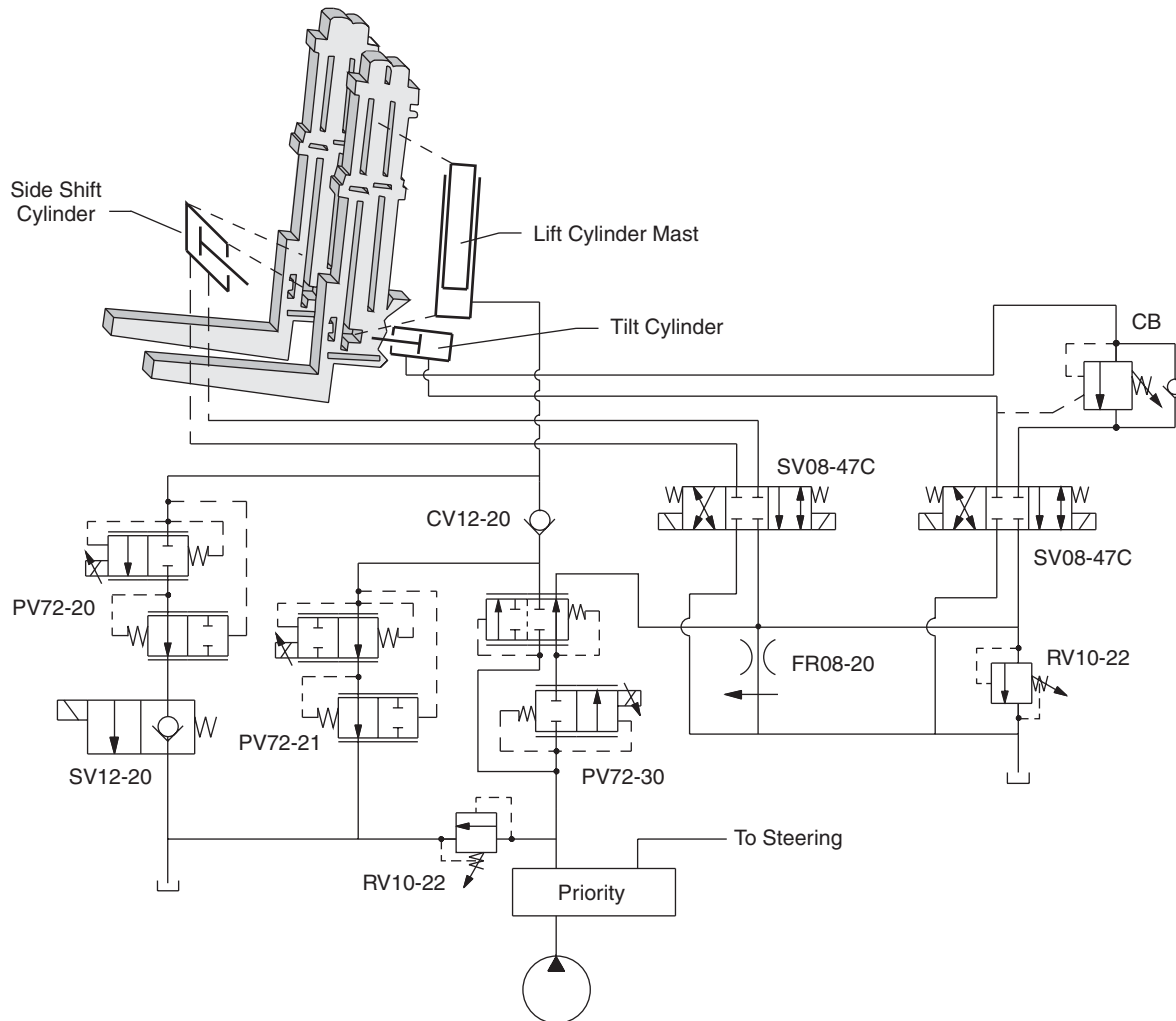
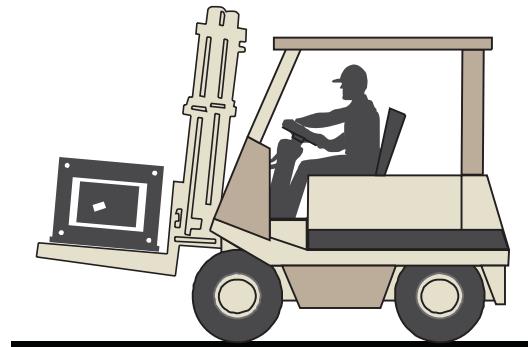
There are several reasons why externally compensated flow controls may be selected over internally compensated ones. One is that there are several bias spring values or pressure compensation values to choose from. This allows for tuning the system to give the desired maximum flow rate while using the complete resolution of the proportional valve. As an example, compare the two graphs below for the PFR72-33 (externally compensated) and the PV72-30 (internally compensated). The maximum control flow of the PV72-30 is 15 gpm. Assume that the desired maximum control flow of a given application is only 12 gpm. The operator would only be able to use 85% of the maximum control current. Since the threshold is at 20% of the maximum control current, the range of control current which results in a change in flow, is only 65% of  $I_{MAX}$ . In contrast, instead of reducing the resolution, the compensation value of the PFR72-33 could be 100 psi and still reach the desired maximum flow with 93% of  $I_{MAX}$ . A further benefit is that the horsepower loss is reduced by going from a 165psi compensation value in the PV72-30 to the 100 psi spring in the PFR72-33.

One other benefit to using the externally compensated flow control is the ability to modify the product where instability in the system exists. Since the construction of the externally compensated valves is relatively simple, it is easy to modify. This includes additional metering holes, modifications of spring rates or the addition of orifices in the sense line. While HydraForce has applied the standard product to hundreds of applications, there have been some where the product needed to be modified to improve overall system performance.



## PV and ZL Valves used in Fork Lifts

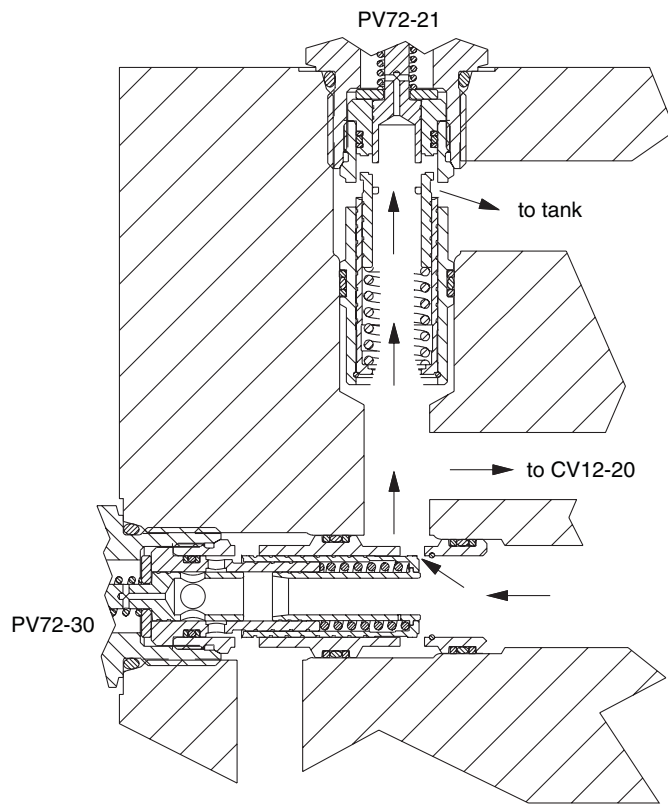
In this section, two different examples of proportional control of a fork lift mast will be shown. The first involves the use of two internally compensated PVs and the second involves the use of a ZL70-36. Both have practical applications in industry and several other solutions or circuits can be designed for this application.



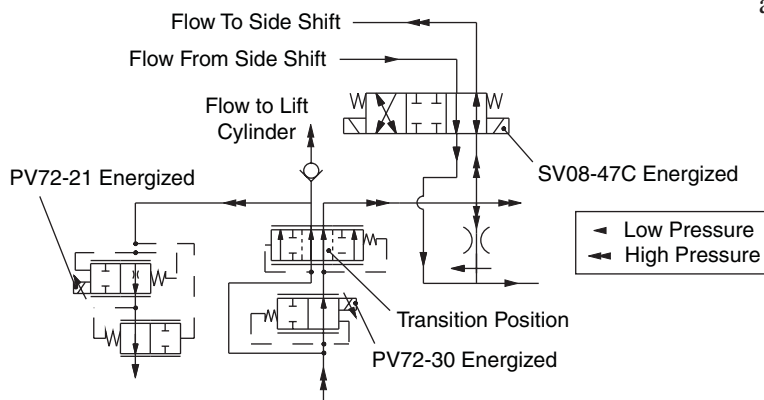


The circuit on the previous page proportionally controls the fork lift mast, the tilting of the fork lift mast and shifting the forks from side to side. A fourth function which can extend the forks from the frame, known as the reach function, is not included.

The circuit operates as follows: while the vehicle is operational, but none of the functions are selected, oil flows from the pump through the bypass port of the PV72-30, continuing through the PV72-21 until returning to tank. This is seen in the diagram below.

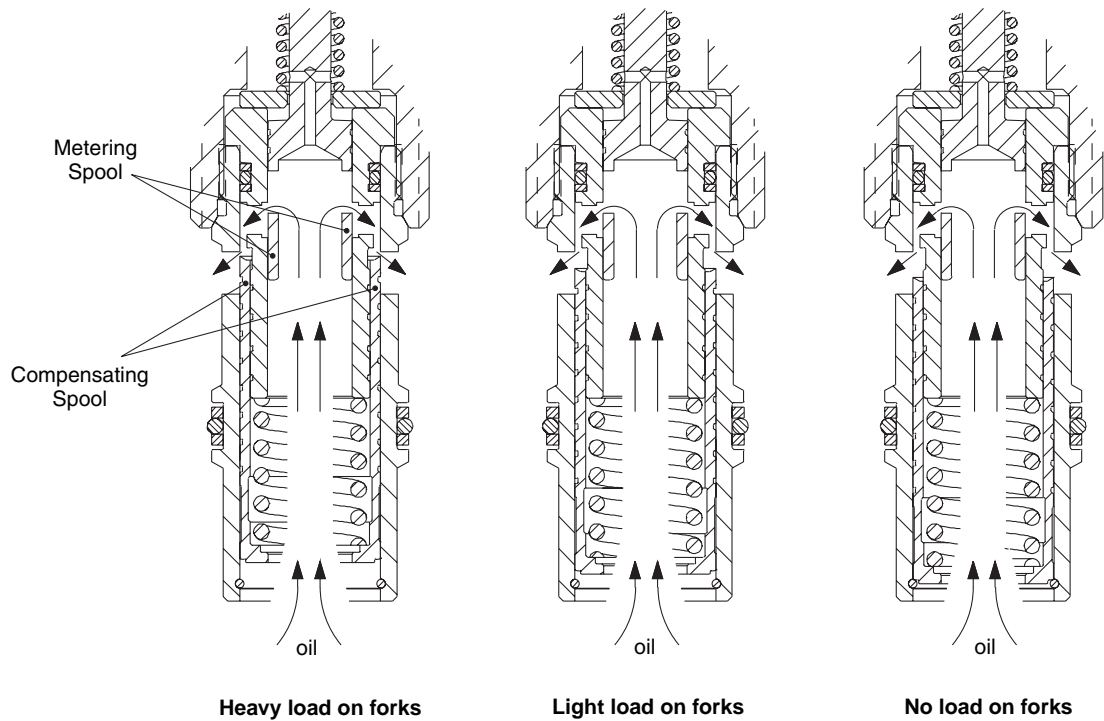


If the operator wants the forks to be raised, power to the PV72-21 is applied. As more current is applied to the PV72-21, less flow is diverted to tank and instead, oil flows to the lift cylinder. While oil is flowing to the mast cylinder, the tilt, or side shift function could be selected as well. In order for these functions to operate, the solenoid valve of the desired function is energized and then power to the PV72-30 can be applied. Since the cylinders controlling these functions are smaller than the lift cylinder, the change in flow of oil to the lift cylinder is negligible. That is, while the operator is watching the load go up, he will see no change in speed as he simultaneously selects the tilt or side shift function. When these functions are required to operate, the current may be limited to 50% of  $I_{MAX}$  or less. More current than this would cause these functions to operate too quickly. In fact, the PV72-30 was not selected for the maximum flow it could regulate, rather, the amount of flow it could bypass to the lift portion of the circuit. The portion of the circuit schematic shown here (lower left) shows that the tilt function has been selected while the PV72-30 and PV72-21 are powered.

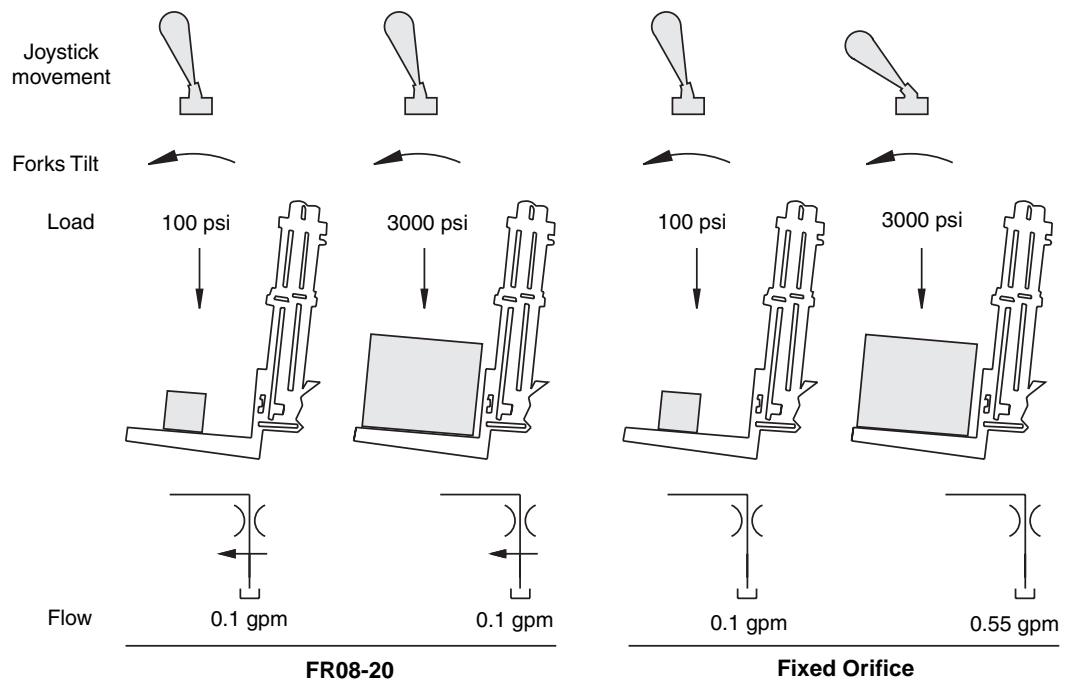


If the operator wants to lower the forks, the PV72-21 is de-energized and the SV12-20 is energized. Current can then be applied to the PV72-20 to vary the lowering speed of the forks. Since the PV72-20 is internally compensated, the lowering speed is not dependent on the load sitting on the forks. Flow from the mast cylinder will discharge at the same rate, regardless of the weight. Also, the compensating spool allows oil to flow, unrestricted, from the mast cylinder to tank even if there is no load on the forks. The three pictures below clarify these points.

**Metering Spool of the PV72-20 remains in the same position while the position of the Compensating Spool varies**

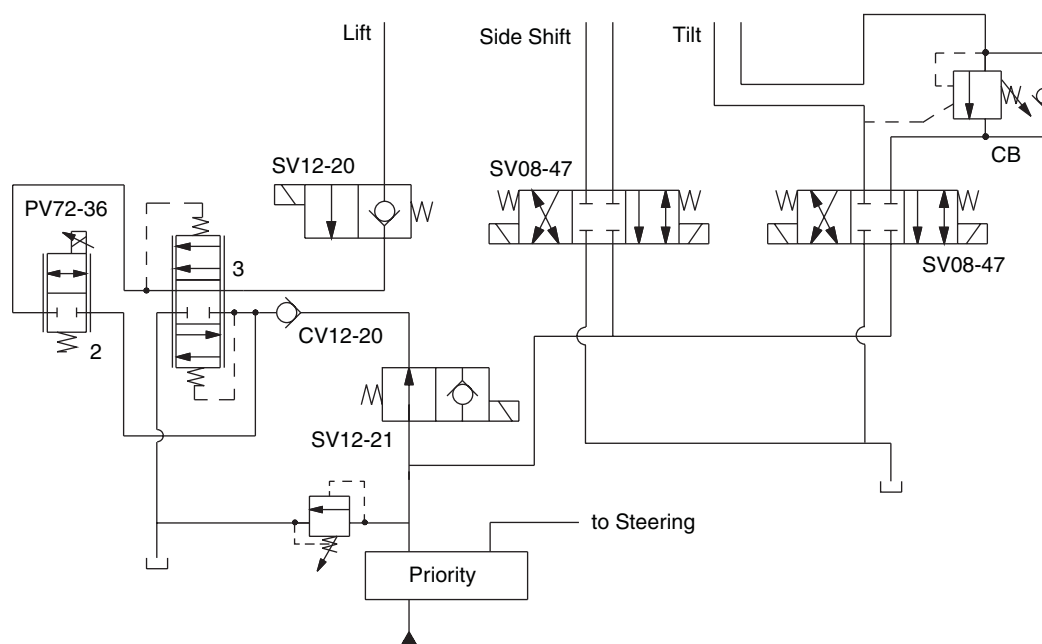


Notice that an FR08-20 is used in this circuit. This bleed orifice is required only when the PV72-30 is de-energized. It is needed to drain trapped pressure which would otherwise hold the compensation spool in the neutral position. The FR08-20 was selected instead of a simple orifice because the amount of oil that bleeds to tank will not vary when the load pressure of the tilt or side shift varies. While FR08-20 does cost more than a simple orifice, the oil draining to the tank will be consistently low. This will improve the energy efficiency of the hydraulic system. The machine operation will also feel consistent to the operator. The change in flow for the simple fixed orifice is compared against the consistent flow of the FR08-20 in the diagrams below.



Notice that the joystick, which controls the tilt function, is in the same position in the first two pictures to the left. This joystick controls the current applied to the PV72-30 and the power applied to the SV08-47C controlling the tilt function. Also notice that the amount of oil flowing through the FR08-20 is the same, regardless of the size of the load. In the two pictures to the right, the joystick is in two different positions and the flow is different. There is more flow exiting the orifice because the load is greater. Recall that flow through a fixed orifice depends on the load. The joystick is moved further so that more current can be applied to the PV72-30. This will allow the same amount of flow to be diverted to the tilt portion of the circuit as in the three diagrams to the left. The tilt cylinder can then move at the same speed as the cylinders in the other three pictures.

In the next circuit, the ZL72-36 is the only proportional valve required to control the lift cylinder. It replaces the function of both the PV72-21 and PV72-20 in the previous circuit. The SV12-20 is still required as a load holding valve. The operation of the ZL72-36 is similar to the operation of the PV72-21 when the forks are being raised. Initially, all the flow is diverted to tank. As current to the ZL72-36 is increased, more oil flows to the mast cylinder and less is diverted to tank. As the operator requires the speed of the forks to increase, more power is applied to the ZL72-36.



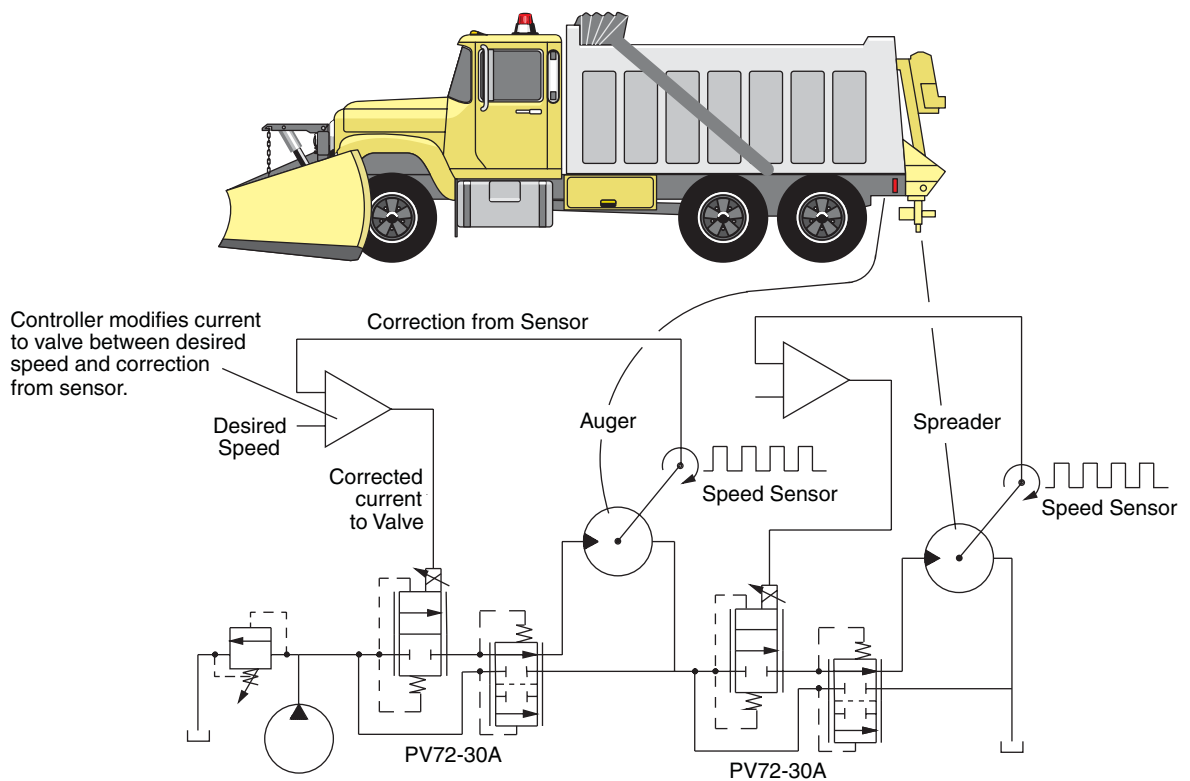
During lowering, the SV12-21 is energized to block the pump flow from influencing the lowering of the mast cylinder. This is only required when the other functions are selected, otherwise, the pump may be turned off when the forks are being lowered. The SV12-20 also must be energized and then the PV72-36 can be energized. Oil then flows from port 3 to port 2 of the ZL72-36. While the forks are being lowered, either the side shift or tilt function can be selected. Notice in the example above, the tilt and side shift are controlled through on/off solenoid valves and not proportionally.

During raising or lowering of the forks, the flow through the ZL72-36 is constant for a given current level applied to the coil, regardless of the load the fork lift is moving. Like the PV72-20, oil flows freely through the valve even when there is no load on the forks.

To summarize, the use of the internally compensated proportional valves allow for electrohydraulic lift/lower control of a fork lift mast. Further, when the operator desires a certain speed, that speed can be selected, regardless of the load the fork lift is moving.

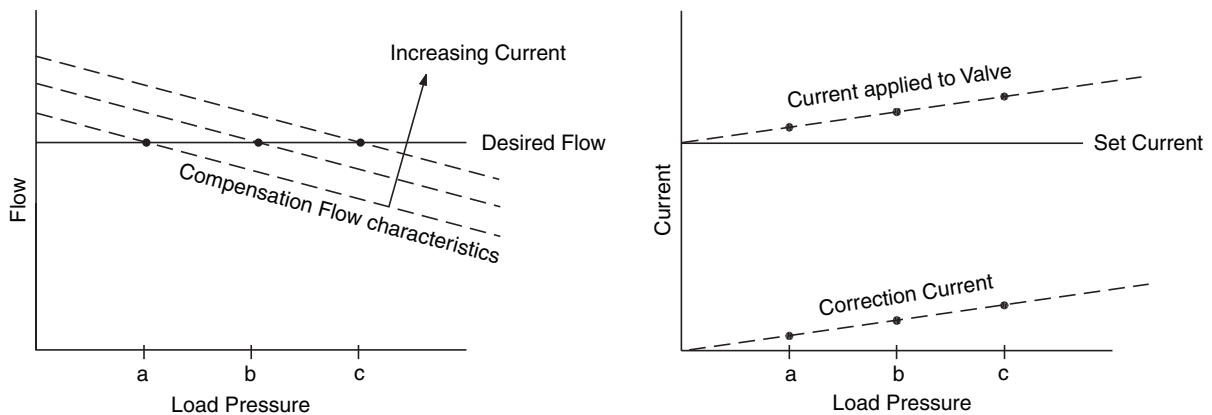
### PV72-30 in a Salt Spreader

An application for areas of the world which see snow, is a salt or grit spreader. These are mounted on the back of snow plows. A picture of one is shown below, along with an hydraulic circuit. This circuit is made up of two proportional flow controls and two motors in series. The motors drive the spreader and auger functions. The spreader disperses the salt or grit onto the roadway, while the auger, a corkscrew or helical shaped device, feeds the salt or grit to the spreader.



Compensated flow controls (PV70-30) are used in this circuit for two reasons. The primary reason is to assure constant speed, regardless of the change in pressure. The compensation feature internal to the valves is the mechanical feedback maintaining constant flow to the motors. There is also an electronic feedback loop in the form of a speed sensor to assure constant speed is maintained. This electronic feedback corrects for the droop in the compensation curve.

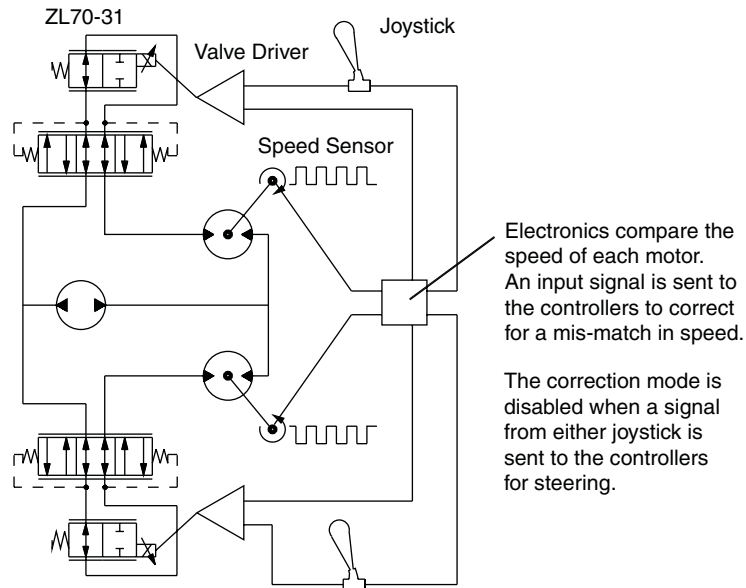
Below is a graph which shows the desired flow against changes in load pressure. Also shown on the graph is the compensation characteristic of the flow control. Three compensation curves are shown to represent how the current to the PV is changed to correct for the droop. The second graph shows the change in current compared to the load pressure. The graph shows how the set current from the operator remains unchanged regardless of the change in load pressure. However, the current applied to the valve increases because of the correction current determined by the speed sensor loop. The letters a, b and c shown on the two graphs are related to each other through the flow vs. current characteristic for the PV70-30. That is, in order for the desired flow to be maintained, the current must increase from point a to b and then to c as the load pressure varies. The change in current is actually relatively small to compensate for the droop. The graphs are exaggerated to illustrate this point.



The second reason why compensated flow controls are used, is to act as back up to the speed sensor. That is, while there are speed sensors to assure the actual speed of the spreader or auger meet the desired speeds, these sensors often fail. The sensors fail because of the highly corrosive salt environment. When the sensor fails, the operator needs the ability to set the auger and spreader speed to a 'limp' home mode. The limp home mode means that the spreader and auger speed could be set to a fixed speed that may spread less than the required salt or grit for the given conditions. This is better than not being able to spread any salt or grit. If the flow controls were not compensated, the speed of the auger may increase, causing the amount of salt sent to the spreader to increase as well. This increased amount of salt at the spreader would cause the system pressure to rise. The spreader will continue to spin as long as the load requirement does not exceed the system relief setting. Once the load pressure is the same as the system relief setting, oil begins to flow across the relief valve. Eventually, all the oil may flow through the relief valve causing the motors to stall.

## ZL70-31 and Parallel Motors

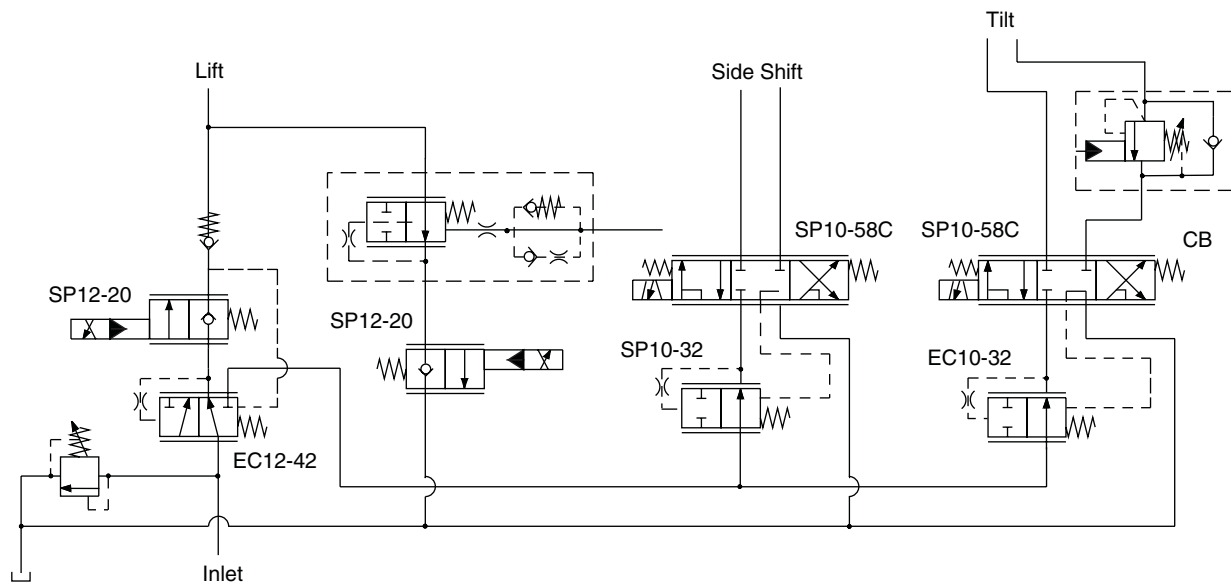
The following application is that of two ZL70-31 valves controlling the flow through two motors in parallel. Such a circuit may be used on a slow moving track vehicle.



The circuit operation is based on a variable pump controlling the forward and reverse speed and two ZL70-31 valves controlling the steering. This type of steering system is known as *steer-by-wire*. This term is used because there is no mechanical link between the operator and the steering system. The input from the driver is through movement of a joystick which outputs an electrical signal to the steering valves. The speed sensors continually monitor the motor speed. This information is used to assure straight line driving. When the vehicle is going straight (no input to the steering joysticks) the speed of the two motors are compared against one another. If there is a difference in speed, a correction factor is fed to the controller. The controller will then supply current to one of the ZL70-31s to restrict flow at the respective motor. This will reduce the speed of that motor which will cause the speed between the two motors to become equal. The use of the ZL70-31 valves allows for the system to be run in reverse when the flow from the pump is reversed, yet only one proportional valve is required to control the speed of each motor. When the operator requires the vehicle to turn, she would move either the left or right joystick for the desired direction. Power is then applied to the appropriate ZL70-31 to restrict flow to the respective motor. The vehicle will turn because there is a relative difference between the speed of the two motors.

## Proportional Directional Control Application of SP Valves

In the previous section where the fork lift operation was described, we showed how to use the PV product in a lift lower circuit and to vary the speed to auxiliary functions. In this section the use of the SP poppet product will be shown in the same application. The advantage for using the SP poppet product is the load holding capability. The disadvantages are; increased hysteresis and an increase in the variability of the valve to valve reproducibility. Both of these disadvantages can be minimized through various software schemes. Shown below is one possible solution for using an SP12-20 for lift along with an EC12-42 which gives priority to the lift function. The SP12-20 can also be used to control the lowering of the mast when used with the EC12-34. A compensator is used with the SP12-20s to assure consistent performance regardless of load. It is not needed to limit the pressure differential across the SP.



Also, shown in the schematic above are two SP08-58Cs. The proportional directional controls replaced the on/off valves shown in the previous section. These valves allow for precise control of the speed of these functions. This in turn allows for smooth movement of the load as well as enhancing the operator's ability to stop the load at a desired position. While this circuit gives improved control over the on/off circuit there may be an undesired interaction between the functions if they are used simultaneously. This is because of the pressure droop characteristic for these valves. Recall from the previous chapter, as the pressure differential increases, the flow decreases. The same occurs in this circuit. If the pressure required in the adjoining cylinder is greater then the flow will vary. A load sense shuttle along with an in-line compensator could correct this problem. Going a step further, the SP10-58C and 58D could be used along with the compensator. In order to justify this, the cost of the cavity for the LS04-30 is saved. The cost of LS04 essentially is added into the cost of the SP10-57.



## Summary

In this chapter the following concepts were presented:

- The costs and benefits of internal vs external compensator.
- A table listing the compensation values for the internally compensated flow controls.
- How to use proportional flow control valves in a forklift circuit.
- The use of proportional flow controls to accurately control the speed of a hydraulic cylinder.
- How electronic closed loop feedback can improve system performance.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. Why would a PV72-30 be selected to only regulate 5 gpm instead of a PV70-30? \_\_\_\_\_
  
2. What is the benefit of compensation? \_\_\_\_\_
  
3. If the PV72-21 is regulating 12 gpm, how much horsepower is required while the valve is compensating for pressure changes? \_\_\_\_\_
  
4. What are the pros and cons of using a SP12-20 in a lift/lower circuit? \_\_\_\_\_
  
5. What would an SP08-57 be used instead of an SP08-47? \_\_\_\_\_
  
6. Why was an FR08-20 be used in the first fork lift circuit? \_\_\_\_\_

# Chapter 14: Application of Coils

## Objectives

The objectives for this chapter are as follows:

- Learn how the system voltage affects the performance of proportional valves.
- Understand how voltage drop across a controller may affect the current available for a coil.
- Learn how duty cycle coupled with temperature influence the current draw of a proportional coil.

## Introduction

In this chapter, we will learn how the interdependencies of voltage, temperature, duty cycle and controller voltage drop influence the current available for a coil. Included in this chapter are graphs to reference the system voltage required at various ambient temperatures in order for a coil to draw maximum current.

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## Applying Coils

In proportional valves, performance depends on the current in the coil. Coil current is a function of the applied voltage and the resistance of the coil. Increasing voltage will raise the current level while increasing resistance will decrease the current level. In most mobile equipment electrical systems, the applied voltage is not controlled, but varies around the nominal battery voltage. In the case of battery operated vehicles, this voltage decreases continually until the next charging cycle. The resistance of the coil changes because of outside temperature and the power applied. The rate at which the coil can dissipate the power in the form of heat affects the resistance. As the temperature of the winding increases, the resistance of the coil also increases. This increase in resistance causes a decrease in the current within the coil and can lower the output of a proportional valve. This is true unless a closed loop current driver compensates for the change in coil resistance. However, even with the controller, the coil resistance may reach a point where the power supply is unable to provide sufficient power.

In order to maintain maximum flow or pressure at high ambient temperatures, it is important to know the actual applied voltage to the coil, including any voltage drop across the controller. Generally, on engine driven equipment, where alternator voltage is several volts above battery voltage, a coil rated at nominal voltage works well. On battery operated equipment a coil rated at several volts below nominal voltage may work best.

## Coils in Operation

To ensure optimal performance and cross-functionality, HydraForce determined the maximum current that could be applied to the coil under varying conditions. This was done by reviewing the voltage range available in mobile equipment, the amount of time the coil is powered, ambient temperature and the typical voltage drop across an electronic controller. Each of these factors must be considered simultaneously when looking at an application. These parameters are each discussed in the following sections. Further, an example is given for the pressure control coils.

### System Voltage

This section discusses the relation between the system voltage range and the current available for the coil. All other parameters considered (i.e. temperature, duty cycle etc.) are related back to voltage. That is, without sufficient voltage the coil cannot draw the current needed when the temperature is hot or the coil is on continually. A typical system voltage range specified by customers is +/-15% of nominal voltage. For a 12 volt system this is 10.2V to 13.8V. The following table shows the current available from various coils with 10.2V applied, and at room temperature. Compare the required current levels with the actual current. Notice that a considerable safety margin is built into each coil. For example, the required current draw for the EHPR08-33 to reach maximum control pressure is 1.2 Amps. However, the current draw at 10.2V is 2.0 Amps ( $10.2V / 5.1 \text{ Ohms} = 2.0 \text{ Amp}$  at 20°C). Comparing the 2.0 Amp to the 1.2 Amp shows that the system voltage could be much lower.

<b>Coil Size</b>	<b>Resistance 12V Coil</b>	<b>Current Draw* at 10.2V</b>	<b>Required Current for Pressure Control Valves</b>
01	5.1 Ohms	2.00 Amp	1.20 Amp
08	9.8 Ohms	1.04 Amp	0.65 Amp
10	7.2 Ohms	1.42 Amp	1.10 Amp
70	5.0 Ohms	2.04 Amp	1.50 Amp

\* Current Draw is calculated at 20°C

### Controller Requirements

Next, consider the effect a controller might have on the voltage available to drive the coil. Some controllers may require up to 1.5V to power the electronics. This voltage, or overhead (above the valve requirement) supplies power to create the dither, stabilized voltage, ramping and other features in controllers. The power draw of the electronics reduces the voltage available at the coil, or, in other words, another 1.5V of the nominal voltage is subtracted off. This means that if the operating system is 12V and the voltage range is + 15%, then the low end of the voltage range, 10.2V, is further reduced to 8.7V. The following table is a revised list, comparing the required current draw to the current draw available with the 8.7V.

<b>Coil Size</b>	<b>Resistance 12V Coil</b>	<b>Current Draw* at 10.2V</b>	<b>Required Current for Pressure Control Valves</b>
01	5.1 Ohms	1.71 Amp	1.20 Amp
08	9.8 Ohms	0.89 Amp	0.65 Amp
10	7.2 Ohms	1.21 Amp	1.10 Amp
70	5.0 Ohms	1.74 Amp	1.50 Amp

\* Current Draw is calculated at 20°C

Again, comparing the current draw to the required values shows that there is still a safety margin. Many OEM customers design their own electronics for their applications. If this is the case, the power is not subtracted directly from the power applied to the coil. These customers are usually able to supply the full voltage range to the valve.

### Ambient Temperature

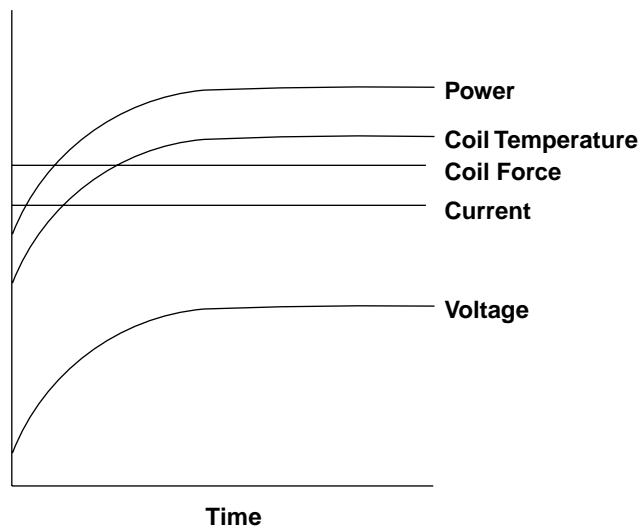
While ambient temperature does not reduce the voltage available, it does cause the coil resistance to change. The following table lists the room temperature resistance and the resistance at 40°C. The current is also tabulated at 8.7V (allowing for low voltage and voltage drop across the controller).

<b>Coil Size</b>	<b>Resistance at 20°C 12V Coil</b>	<b>Resistance at 40°C</b>	<b>Current Draw* at 8.7V</b>
01	5.1 Ohms	5.5 Ohms	1.58 Amp
08	9.8 Ohms	10.6 Ohms	0.82 Amp
10	7.2 Ohms	7.8 Ohms	1.115 Amp
70	5.0 Ohms	5.4 Ohms	1.611 Amp

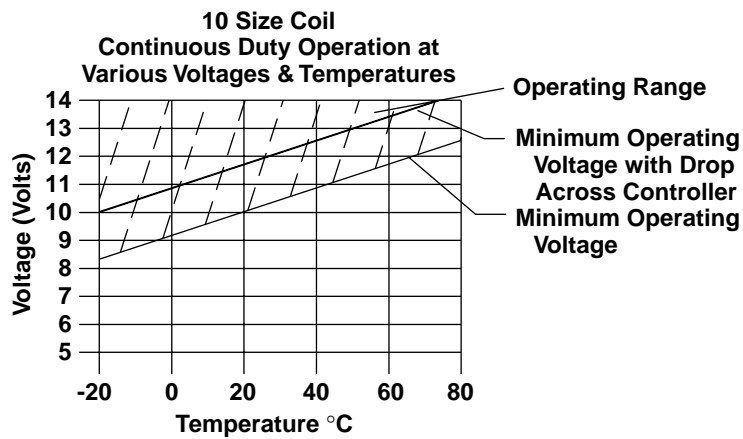
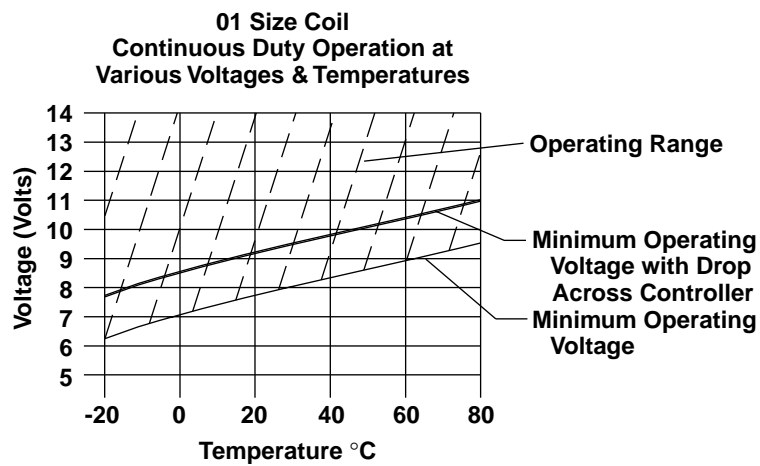
\* Current Draw is calculated at 20°C

### Continuous Duty

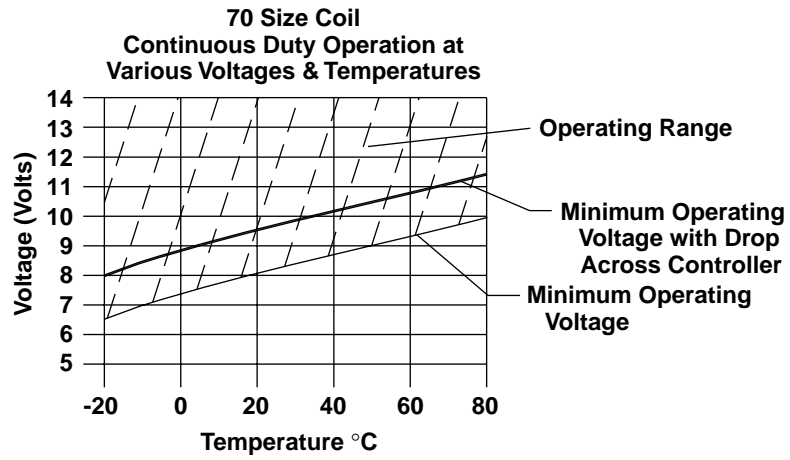
This section will examine the effect of powering the coil continuously at  $I_{MAX}$ . When  $I_{MAX}$  is applied to a coil for more than one and a half hours, the coil resistance stabilizes. Under this condition, the duty cycle of the valve is continuous. The term duty cycle in this section refers to the amount of time an operator commands the valve to work at a given current level. The graph below shows  $I_{MAX}$  applied continuously. Also shown on the graph is the voltage, power and coil temperature. The voltage charted on the graphs on the following page is the value at which it stabilizes. The graph shows the current remaining flat. Recall the magnetic force is directly proportional to the current, so this also remains constant. In order for these two parameters to remain constant, others must vary. As power is applied, the coil heats up and the resistance increases. Ohm's Law dictates that the voltage and power must increase in order for the current to remain constant.



The charts below summarize how the various parameters affect the voltage required to operate the proportional pressure control valves. The graphs show the interrelation between temperature, current, voltage and voltage drop across a coil for continuous duty operation. Two lines are shown. One represents the voltage required with overhead for a controller and the second is if the controller is powered independently. Any voltage above these lines is sufficient to supply the current necessary to operate the valve continuously.

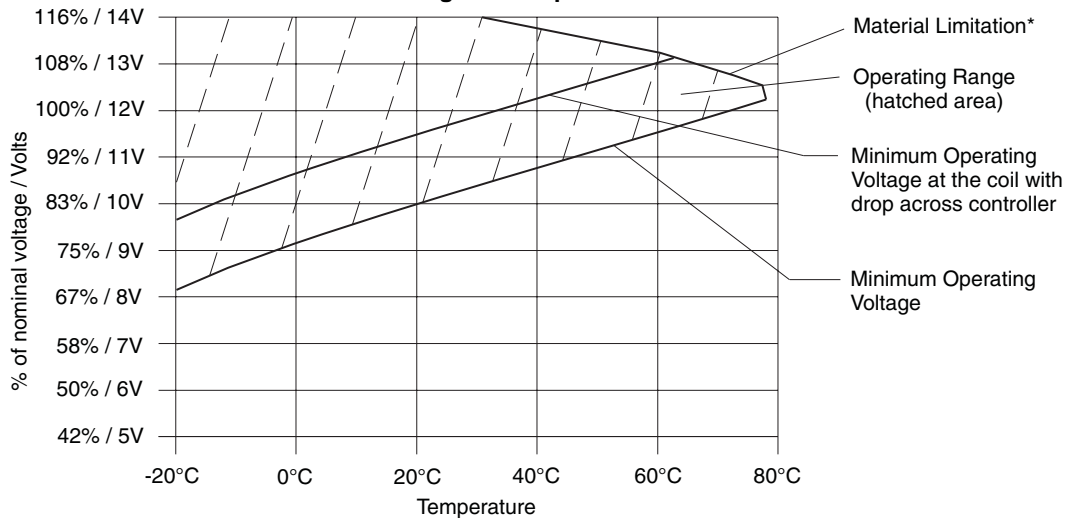






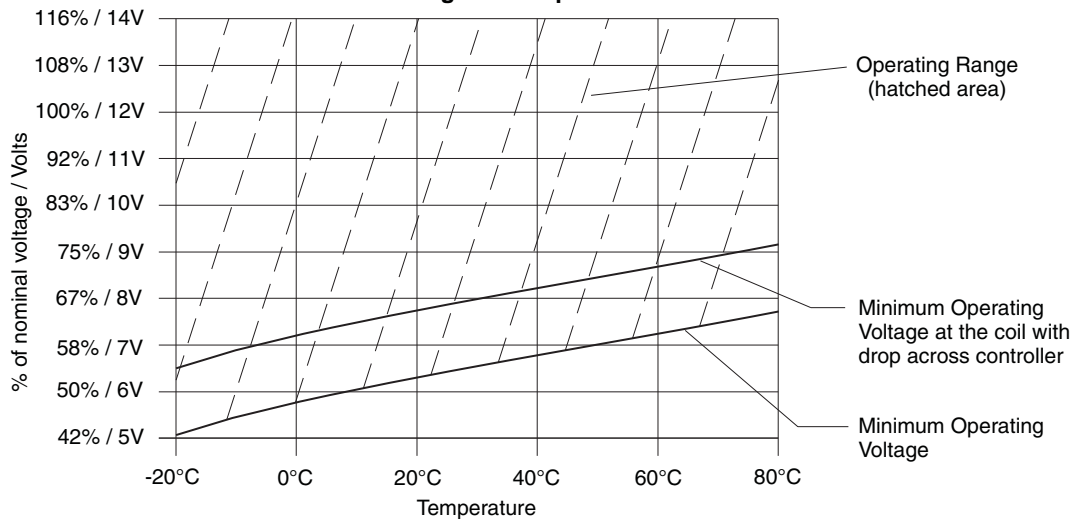
The following charts describe the operating conditions that affect the maximum current draw of the proportional flow control coil. The maximum current required for the 70 and 72 series valves ranges from 1250 to 1500 mA for a 12V coil. The 76 series valves requires an  $I_{MAX}$  of 1600mA. Further, the 08 series flow control valves require 1400 mA when using a 12V 01 size (EHPR) coil. As noted, these current values are for 12V system voltage.

**01 Size Coil, Continuous Duty, at 1400 mAmp  
at various voltages & temperatures**

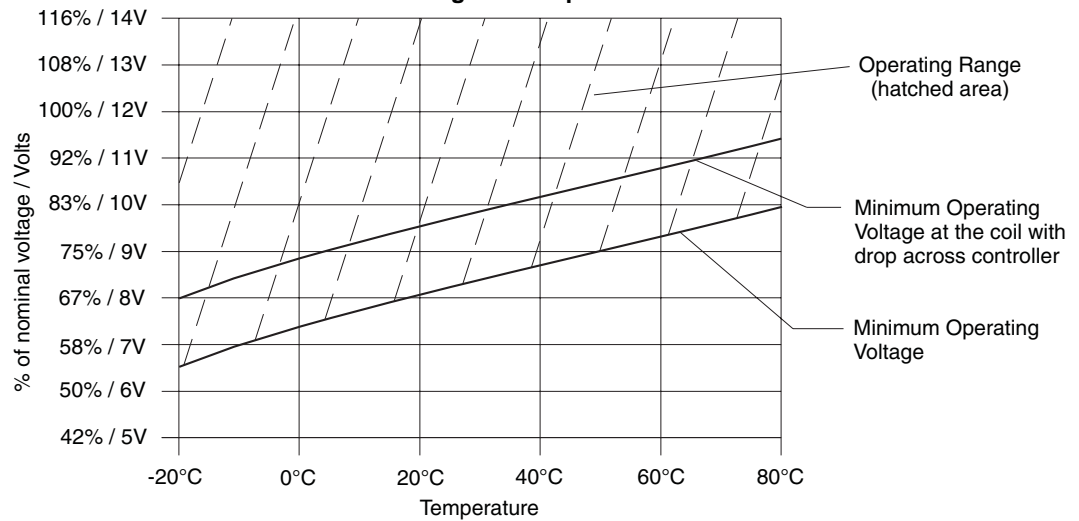


\* Material Limitation: The temperature rating of the plastic and magnet wire insulation is 180°C. The graph depicts an operating range that exceeds this temperature. It is not recommended to operate the coil under these conditions.

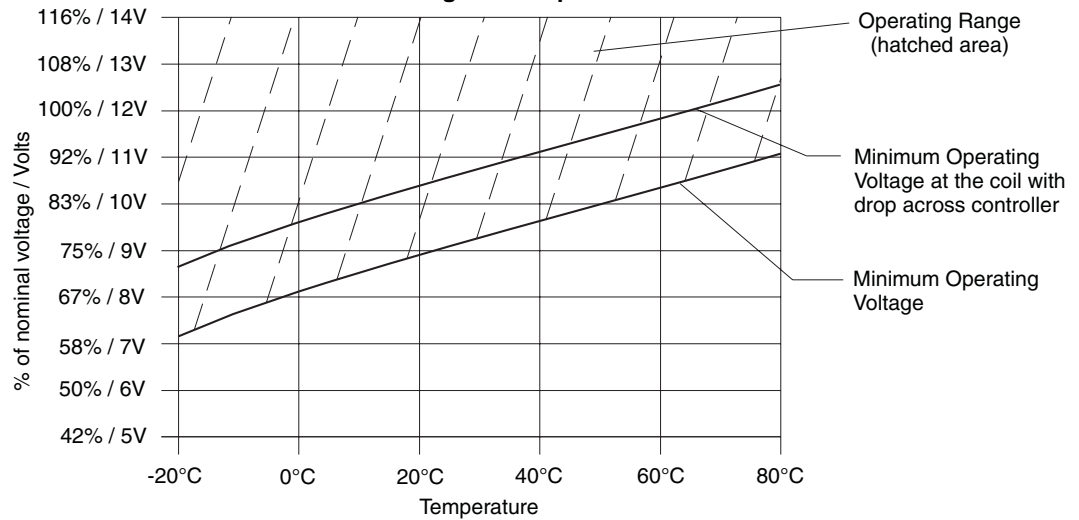
**70 Size Coil, Continuous Duty, at 1250 mAmp  
at various voltages & temperatures**



**70 Size Coil, Continuous Duty, at 1500 mAmp  
at various voltages & temperatures**



**70 Size Coil, Continuous Duty, at 1600 mAmp  
at various voltages & temperatures**



The coils used on the pressure controls and flow controls are typically the HydraForce standard 12v and 24v coils. The charts on the previous pages show that these products can be used continuously at room temperature or higher in many cases even in low voltage conditions.

When HydraForce developed the SP product, the rating system for continuous duty shifted from  $I_{\text{MAXIMUM}}$  to a value termed  $I_{\text{AVERAGE}}$ . In general, it is expected that in actual application, the current applied to the SP valve will vary. Sometimes the current applied may be close to maximum, while at other times it may be close to the threshold current. Therefore, the increase in coil resistance resulting from the power applied will typically stabilize around a nominal or average value. This stabilized average current value is defined as:

$$I_{\text{AVERAGE}} = (I_{\text{THRESHOLD}} + I_{\text{MAXIMUM}}) \div 2.$$

The graphs illustrate the operating range of HydraForce standard coils on the SP valves. The graphs show the voltage required to continuously maintain average current. The voltage supplies sufficient power to reach maximum current on an intermittent basis. Since it is recommended to use the SP valve with a closed-loop current controller, a voltage drop of 1.5V across the controller has been taken into consideration in these graphs.

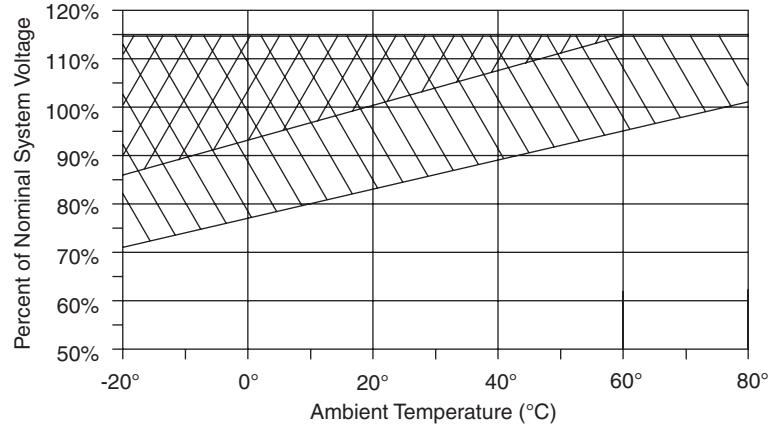
For example, the graph for the 08 size 10VDC coil shows that at an ambient temperature of 20°C, maximum current is available with only 83% of nominal system voltage. If ambient temperature rises to 80°C, maximum output is achieved only if 102% of nominal voltage is available to the coil. However, with the 12VDC coil, 102% of nominal voltage is required at 20°C. Notice that the voltage required at 80°C is above the maximum 115% of nominal voltage line. This indicates that the 12VDC coil is not suitable for this ambient condition regardless of the system voltage available.

**08 Size Coil**

Percent of system voltage required to maintain average current ( $I_{ave}$ ) at various ambient temperatures

10 or 20 VDC Coil Operating Range: \\\

12 or 24 VDC Coil Operating Range: ///

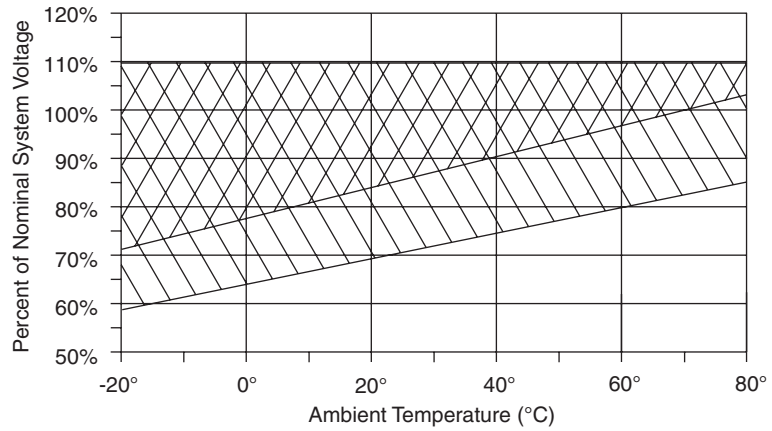


**10 Size Coil**

Percent of system voltage required to maintain average current ( $I_{ave}$ ) at various ambient temperatures

10 or 20 VDC Coil Operating Range: \\\

12 or 24 VDC Coil Operating Range: ///



## Summary

In this chapter the following concepts were presented:

- The affect of voltage temperature and operating time on the current draw available for the proportional valve.
- How the pressure control valves and the flow control valves are rated for continuous duty at  $I_{MAX}$ .
- The term  $I_{AVE}$  was introduced to describe the duty cycle under which the SP product was rated.

## Review Questions

Use the following review questions as a measure of your understanding of the chapter material. Answers are provided in the appendix.

1. What is the maximum ambient operating temperature of the 01 coil when used in a continuous duty application with the PV08-30? Why is there a maximum temperature limitation? \_\_\_\_\_
2. What parameters are considered when applying a coil in a continuous duty application? \_\_\_\_\_
3. What is the minimum operating voltage required for a 12V coil to maintain  $I_{MAX}$  at the coil of the PV76-30? (Assume 40°C ambient temperature and 1.5V drop across a controller. \_\_\_\_\_
4. What coil voltage is recommend if the voltage system relies on a battery as the primary power source? If an SP valve is being used in a system? \_\_\_\_\_
5. Describe what  $I_{AVERAGE}$  is and how it influences the performance of the SP products. \_\_\_\_\_
6. If an eletrical system is driven from an engine is there a greater concern for the coils being able to draw sufficient current? \_\_\_\_\_
7. What is the voltage drop across a typical control? \_\_\_\_\_
8. Define continuous-duty use of a coil. \_\_\_\_\_

**Notes:**



# Appendix A: Motion Control Theory

## Objectives

The objectives for this appendix are as follows:

- Learn about valve step response.
- Show examples and terminology related to valve and system damping.
- Gain an understanding of bode plots.
- Define open and closed loop systems and show examples of each.

## Introduction

Hydraulic valves control the distance, speed and acceleration at which an actuator moves and the force it applies to an object. The science or discipline that describes the interaction between the valves and actuator is known as *Control Theory*. In this chapter we will give an introduction of the terms used in control theory. Also, several examples will be provided to clarify these terms.

# HYDRAFORCE

## Motion Control

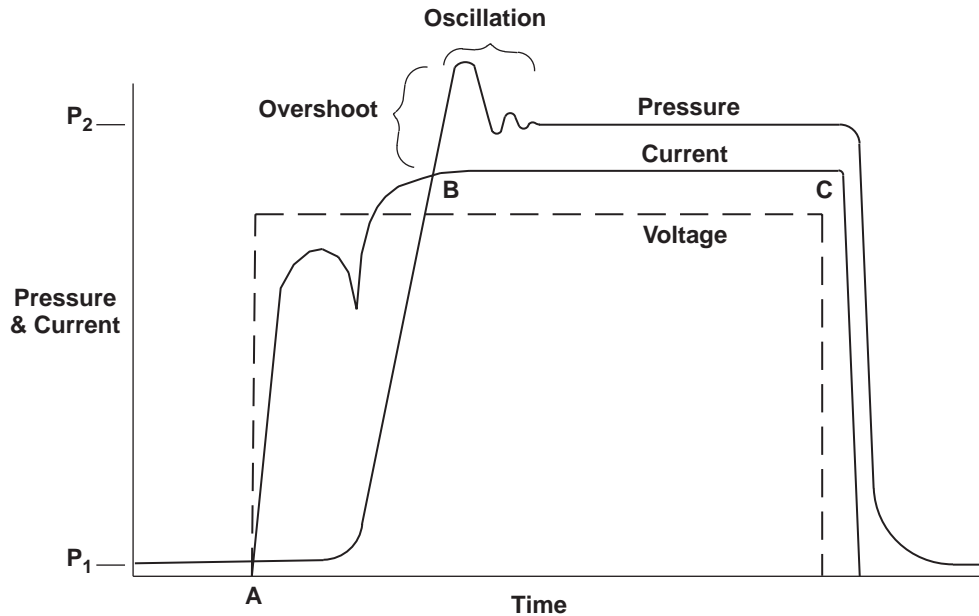
Motion Control Theory describes the movement of an object, such as a crane or bulldozer. Control theory describes how fast the object travels, its position, how quickly it can accelerate or decelerate and the force that it may exert on another object.

One of the main functions of hydraulics is to assist in repetitive motion. Basic hydraulic components such as solenoid valves, needle valves and relief valves can control the direction, speed, and force of a hydraulic cylinder which is doing repetitive motion. However, the motion of the object, such as a loader bucket controlled by a cylinder, may appear jerky, sluggish or move too quickly, depending on the operator. These valves do not adapt to the operator, or to the load. They operate in on/off or two distinct modes, with no settings in between. Proportional valves can operate at the extreme on/off position or anywhere in between. They allow the hydraulic system to adapt to the changing environment. For example, if the vehicle is traveling too fast, the current to the flow control can be changed to reduce the speed. This change in current could be controlled by the operator varying the position of the joystick. It may also occur without any operator input with sensing devices built in to the system. An example of this type of device is the cruise control mechanism on an automobile.

Several concepts which are part of motion control theory will be covered in the following sections. These include, step response, the damping force, frequency response, open loop control and closed loop control. The first three topics give an indication of how quickly the valve will respond to the changes required to control the motion of the system smoothly. The remaining two topics describe the two basic methods used in controlling a system.

### Step Response

Response time is the time it takes for the valve to shift from one state to the next. HydraForce typically records values which are based on the valve going from Off to the maximum current and back to Off again. The graph on the following page shows a typical response graph for a pressure control valve such as the TS38-20.



The first curve to be discussed is the voltage. It is turned on at point A and increased immediately to a constant level. Some time later, it is turned off, as indicated by point C, where it immediately falls to zero.

The current trace goes from zero at point A to  $I_{MAX}$  at point B. While the current is increasing between these two points, it momentarily dips while the plunger is moving, then continues to increase when the plunger stops moving. The term “plunger” refers to the HydraForce name for the solenoid armature. This is the part of the valve that becomes magnetic when current is applied to the coil.

When the plunger becomes magnetic, it begins to move and apply force to the parts of the valve which control the hydraulic function. In this example, pressure begins to increase towards  $P_2$ , the maximum control pressure. In some valves, the pressure continues to increase past  $P_2$ . The difference between  $P_2$  and where the pressure finally stops increasing is known as *overshoot*.

The graph then shows the pressure decreasing and increasing a few times until it finally flattens out at  $P_2$ . This increase and decrease is known as *oscillation*. The valve is said to be oscillating when the output characteristic, like pressure, is increasing or decreasing around some value such as  $P_2$ .

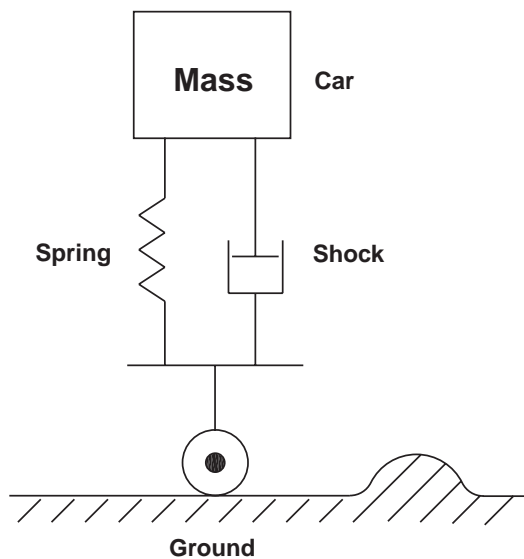
When the current is finally turned off, it immediately decreases. The pressure drops very quickly but tends to round off or decrease at a slower rate when the pressure gets near  $P_1$ . The reason for this slow decrease in pressure is due to the damping characteristic of the parts inside the valve.

Note: As with hysteresis, the overshoot and oscillation may be reduced with the use of PWM and dither.

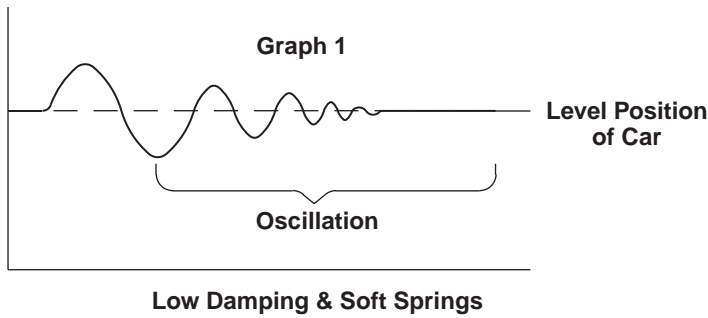
## Damping

The pressure curve on the previous page shows a force known as the *damping force* which causes the pressure to round off at the end. It is also responsible for the *oscillation dissipating* or *dying out*. When the current to the coil is turned On or Off, there is a change in energy. The parts inside the valve respond by moving rapidly from one position to the next. Damping is required to dissipate the sudden change in energy. When a force such as the magnetic force of the armature is applied to the parts, they begin to move. The parts begin to move with a given amount of speed, to a certain position within the valve, in order to give the desired flow or pressure. Since there are no brakes inside the valve, as in a car, the parts rely on damping to slow them when the desired position is reached. The following example illustrates this concept.

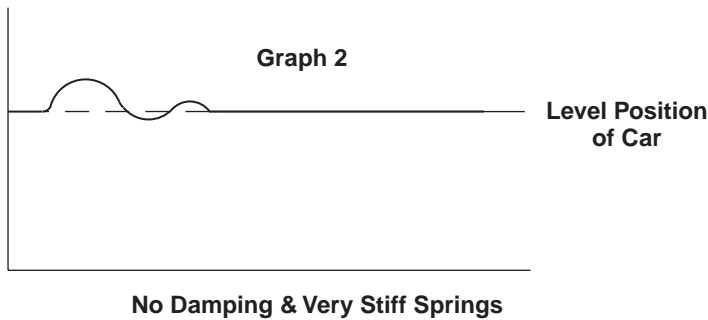
The shocks or struts in your car dampen the movement of the car when it hits a bump. The simple diagram below represents the car on its springs and shocks. The four springs and shocks on a car are represented here by one spring and one shock.



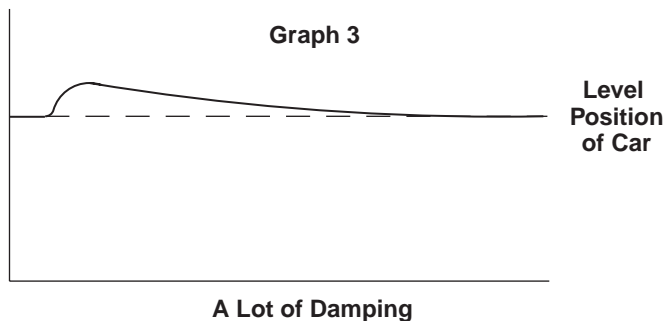
Imagine driving along a road and coming up to a speed bump. The car goes up and down over the bump and then several more times depending on the stiffness of the shocks. The following graphs show the position of the car relative to the ground.



Graph 1 to the left, shows a typical situation for the majority of passenger cars. The car goes up and over the speed bump. When the body of the car clears the bump, it dips down below the level position, and bounces back up and down again. The car may bounce up and down several times before leveling off.

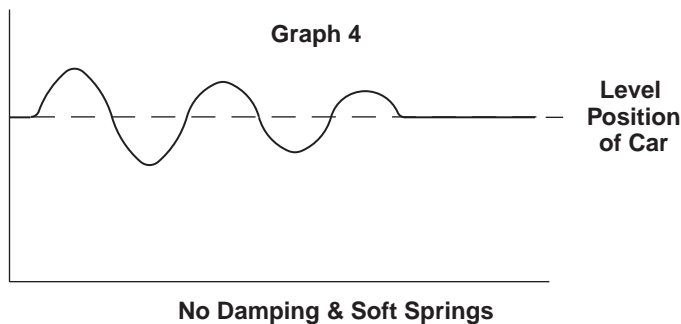


Graph 2 shows the car going up and over the speed bump and then back to its level position. This is typical of a pickup truck or sports car with very stiff springs or an extreme case such as a tractor which has no suspension.



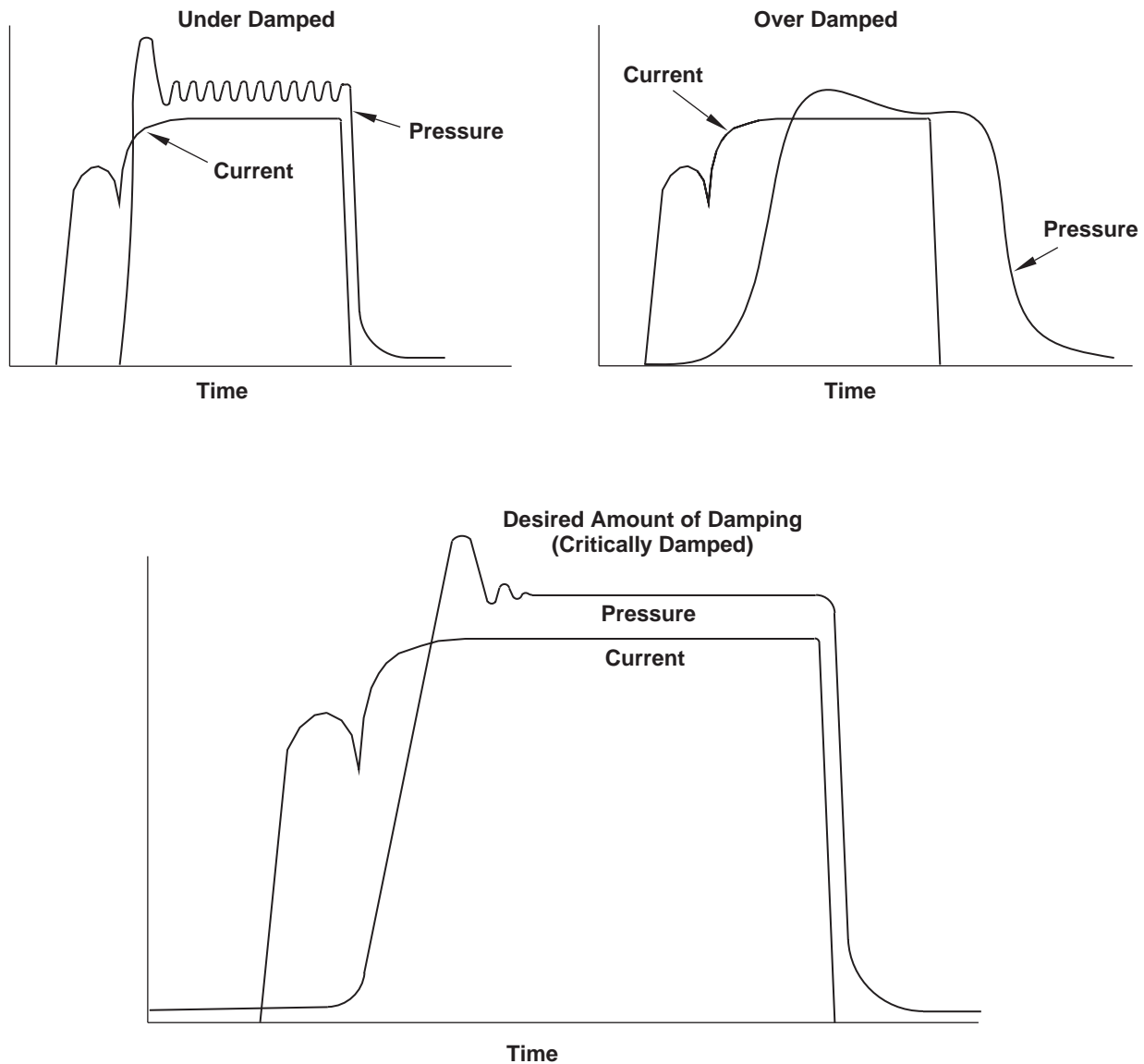
Graph 3 shows that once the car gets over the top of the speed bump it still takes some time before getting back to the level position. This is not a real situation for a car, it is given merely as an example. In this example we would consider the car “over-damped.”

The performance with an old car with worn out shocks is shown in the Graph 4. The body of the car continues going up and down after the car goes over the bump. In this case, we would consider the car “under-damped.”



These examples relate to valves because there is some amount of damping in a proportional valve. If there is not enough damping, the parts of the valve would bounce up and down, as shown in Graph 4. If there is too much damping, it may take a long time for the parts of the valve to reach the required position, as shown in Graph 3.

The following three graphs show the performance of a valve which is under damped, over damped and one that has some overshoot and oscillation. The third graph shows the desired performance, despite the pressure graph not following the current perfectly. This small amount of overshoot and oscillation shows that the valve has some damping.

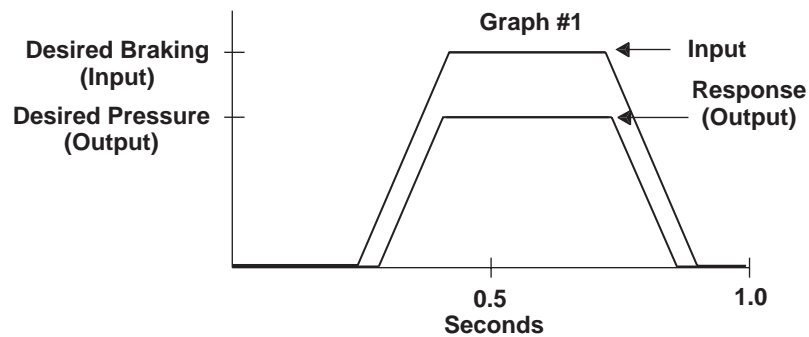
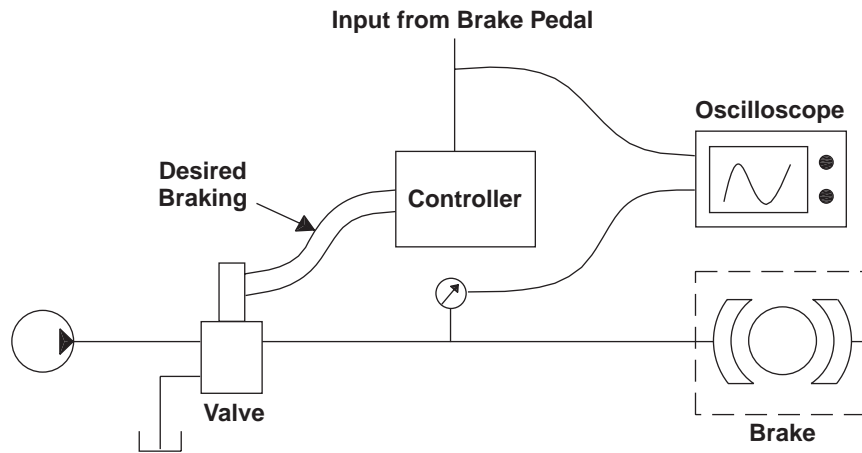


Damping in a valve can be accomplished by two methods. In the first method, the parts are very close together and the clearance or fit is small. This takes advantage of the viscous friction force. Another method of damping is by metering the flow into and out of a chamber known as the damping chamber.

### Frequency Response

Frequency response deals with dynamic behavior or the transient state of a mechanical system such a valve. In the following sections a fictitious example is given to introduce the topic. After this, the method used to actually measure the frequency response will be described, as well as related definitions.

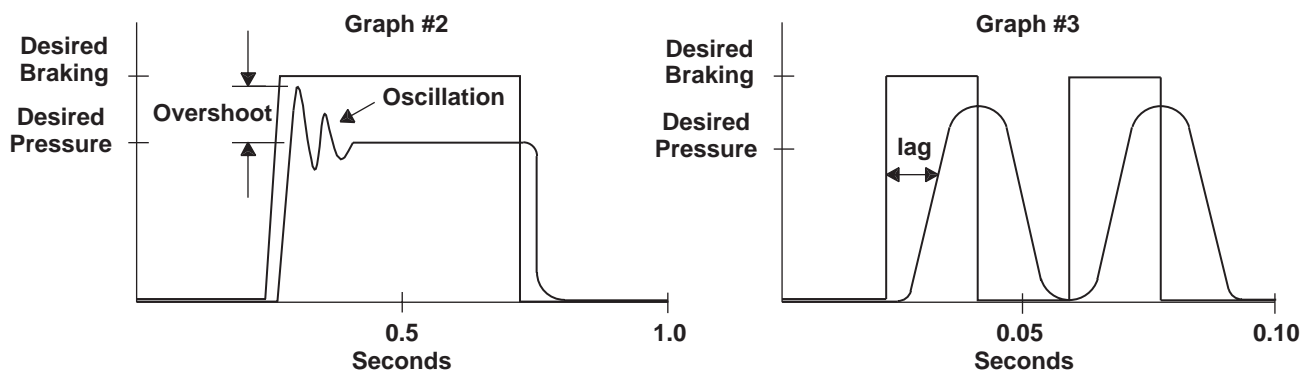
The following is a fictitious or simplified example of the brake system of a car. The brake system shown below, as well as the graphs, will be used to demonstrate some basic concepts of frequency response. In the example, we will look at how quickly the valve can react to the input current or the system disturbance. In short, this is the frequency response of the valve. By knowing the frequency response of the valve, it can be determined if the valve will stay in sync or in phase with the input. If the valve is not in phase, the frequency response indicates if the output is higher or lower than desired. Further, the frequency response indicates how fast the input can change before the output is delayed or lags behind the input.



Graph #1 above, shows that the pressure closely follows the desired brake input and reaches the correct pressure for that amount of braking. In other words, the slopes of the two graph lines are parallel and the output or desired brake pressure starts building almost as soon as the input or brake is applied. In terms of frequency response, the output from the valve is in phase with the input and the pressure reaches the desired amplitude or magnitude.

In Graph #2, the desired braking or input, is faster. This is noticeable from the steeper rise or slope of the desired braking. The valve stays in phase, but some amount of overshoot exists. In addition, there is a small oscillation in the pressure, until it flattens out to the desired brake pressure.

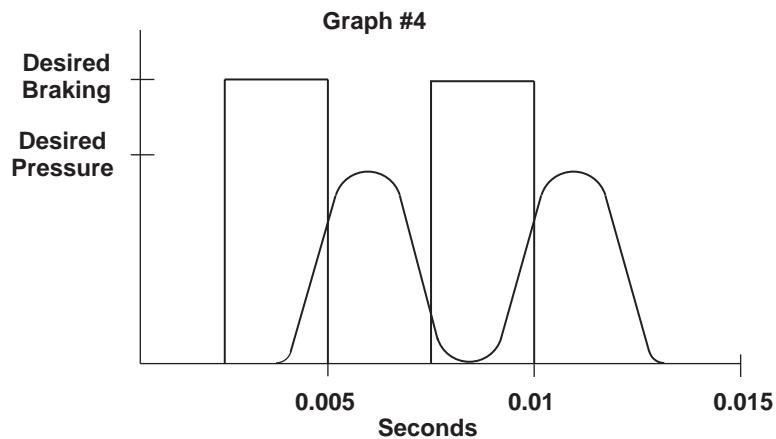
The third graph illustrates the controller trying to pulse the brake, as if in a panic situation. (Notice that the time on the graph is 0.1 second, rather than 1.0 second on graphs 1 and 2). In this case, the pressure is not able to follow the desired braking signal. It is out of phase or lagging behind the input. If the valve had a higher frequency response, the pressure would closely match the input, as in the first graph. In addition, the magnitude of pressure would not overshoot the required level.



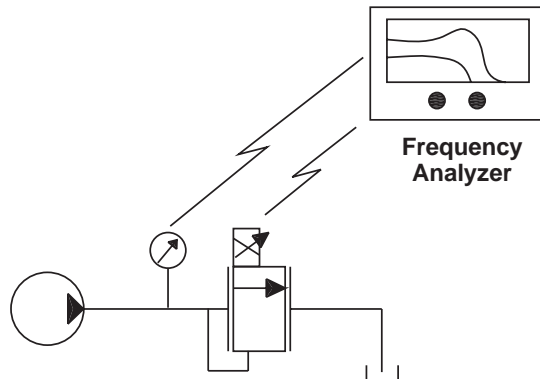
The pressure is slow to build (or lags behind the input) because the valve is slow. This occurs because mechanical and viscous friction may cause the parts to stick inside the valve. In other words, the valve may be over-damped. Once the valve begins to regulate the pressure, it causes the pressure to exceed the desired level. The pressure overshoots because of the momentum (mass multiplied by the speed) of the parts and the momentum of the oil. In other words, the parts cannot stop instantly once the desired pressure is reached.

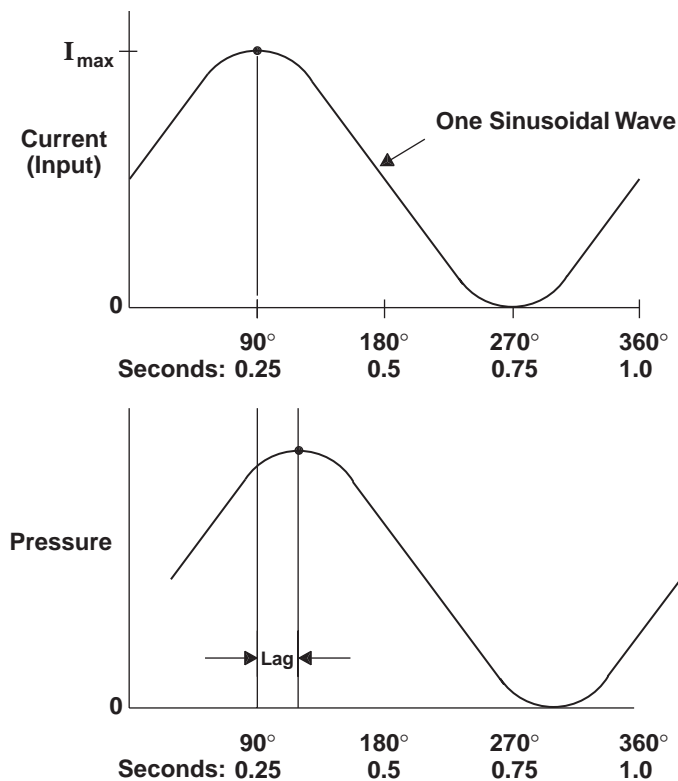


Graph #4 shows the controller trying to pulse the brake 200 times faster than in Graphs 1 or 2. Notice in Graph 4 below, that the total amount of time is 0.01 seconds rather than the 1.0 seconds shown in the first two graphs. Two pulses occur in 0.01 seconds on the fourth graph vs. one pulse in the first graph. Each pulse takes only 0.005 seconds to go on and off. Dividing 1 second by .005 seconds gives the value of 200 times faster than the pulse shown in the first graph. In this case, the valve is even more out of phase (compared with Graph 3) with the desired braking, from the controller. The pressure continues to grow, despite the current having been turned off. This occurs because of the dynamics and inertia of the parts inside the valve. Again, in this example, the valve is unable to regulate the pressure to the desired pressure level, because the current was turned off before the valve could build pressure to the desired level.



As mentioned in the example above, the frequency response is described by the amplitude of the output and how closely the output follows the input or lags behind the input. The frequency response for a valve can be found experimentally. This is done using a device known as a frequency analyzer. One type of frequency analyzer sends out a sinusoidal (or *sine* for short) signal as the output, in the form of current. It then collects information from the system (typically pressure). A block diagram of this type of test is shown below.





The graph above shows the current oscillating in a sinusoidal wave form between maximum current and zero. The graph also shows that the sine wave was applied to the coil in one second. Another way of looking at it, is the rate at which the current varies between on and off occurs at one cycle per second or one hertz. A *hertz* is abbreviated as Hz and is a unit of measure, indicating the number of cycles per second. This means that if the sine wave were repeated many times, the current would still go on and off once every second.

The second graph shows a similar sine wave, with one small difference. The peak and valley of this sine are shifted to the right. In frequency analysis terminology, this shift is known as a *phase lag* between the input and the output. The angular unit of measure ( $^\circ$ ) degree, is used to describe the amount the output sine wave lags behind the input sine wave.

In addition to generating current in sine waves at a rate of 1 Hz, the frequency analyzer can generate sine waves at a faster rate or higher frequencies. The output for valves is typically varied or *swept* from 1 Hz to 100 Hz. The frequency analyzer would then compare the output sine wave (pressure) to the input sine wave (current) for each frequency to determine the lag at each frequency.

Another term used in frequency analysis is *magnitude*. The magnitude is a measure of the difference in the amplitude or value that the pressure reaches at some sinusoidal input frequency, as compared to the pressure value if the current was left on at  $I_{max}$  continuously. This comparison is made using a unit of measure known as a decibel or dB for short. The decibel was originally developed for measuring the intensity of sound and has been adopted in frequency analysis.

The definition of a decibel for use in measuring the response of a valve:

$$\text{Decibel (dB)} = (20 \log |O_n / O_0|)$$

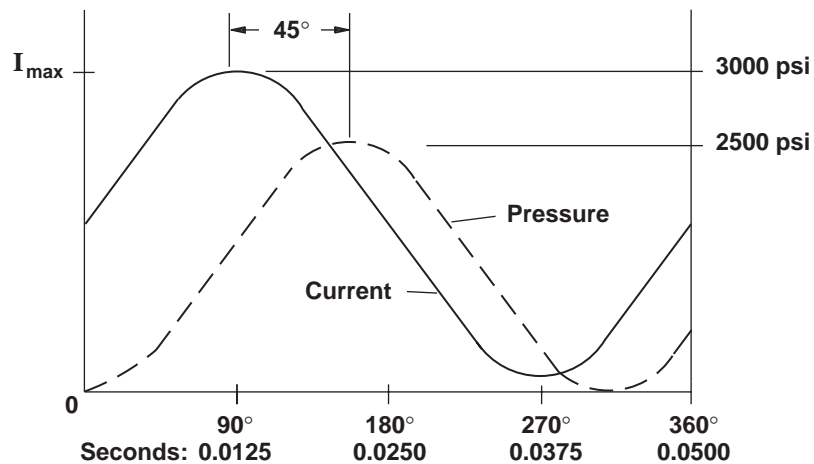
Where:

$O_0$  = output pressure with continuous current (steady state) applied to the coil.

$O_n$  = output pressure when the current is varied at a sinusoidal frequency.

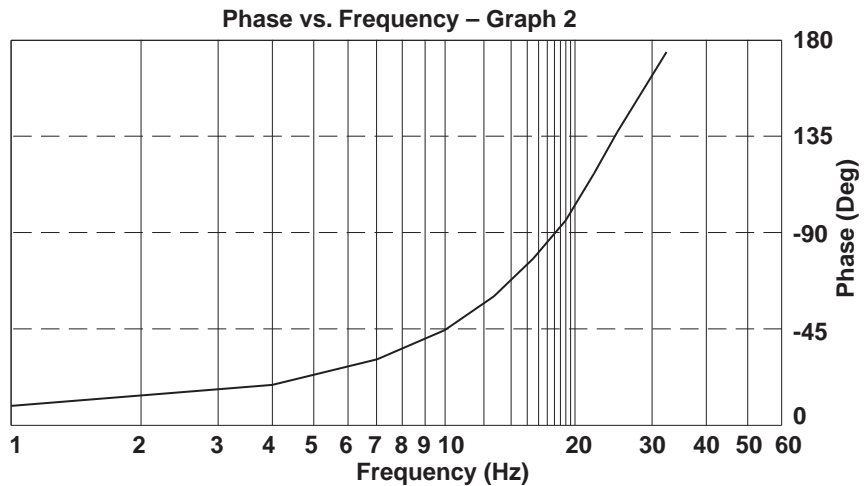
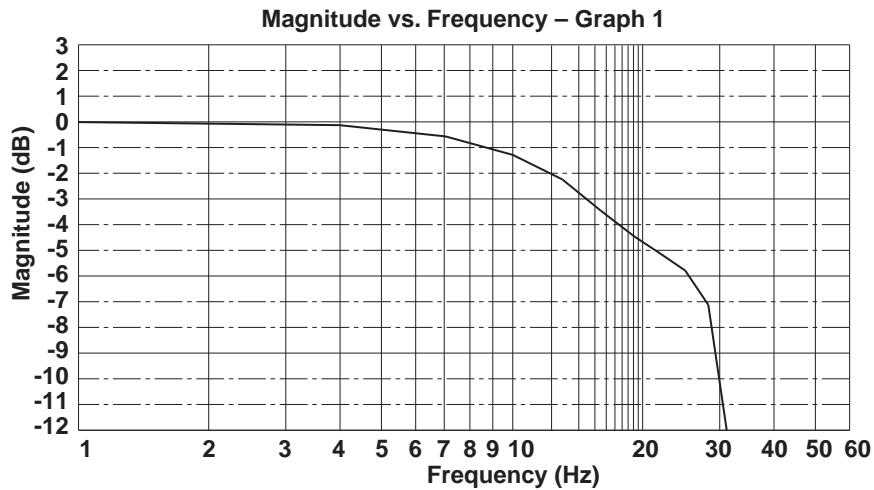
For example, let's assume that the steady state pressure for the TS38-20 at  $I_{max}$  is 3000 psi. If the output or pressure with an input current of 1 Hz is measured to be 3000 psi, then the magnitude is 0 dB ( $20 \log |3000 / 3000| = 0$ ).

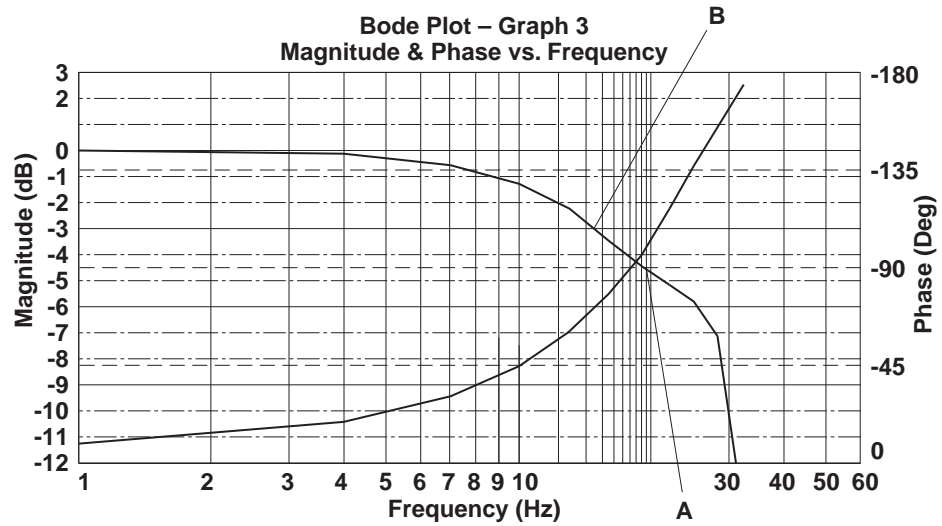
However, if the frequency of maximum current turning On and Off increases to 20 Hz, the pressure may only reach 2500 psi. The magnitude then is 2.0 dB. Further, the output may lag the input by  $45^\circ$ . The output compared to the 20 Hz input is shown in the graph below.



Numerous values for the magnitude and phase lag can be plotted against the frequency, at which each was measured. This type of plot is known as a Bode plot. Below is a table of various magnitudes and phase lags listed against the frequency at which they were recorded. These two values are plotted on a linear scale and the frequency is plotted on a logarithmic scale. The two graphs are shown individually, below, however, they are typically overlaid on one another as shown in the third graph. The third graph is known as a Bode Plot.

Frequency (Hz)	Magnitude (dB)	Phase (Degree)
1	0	-8
10	-1.25	-45
11	-2.00	-56
12	-2.75	-66
13	-3.25	-70
14	-3.40	-78
15	-3.75	-82
16	-4.20	-90
20	-4.65	-102
30	-10.0	-158





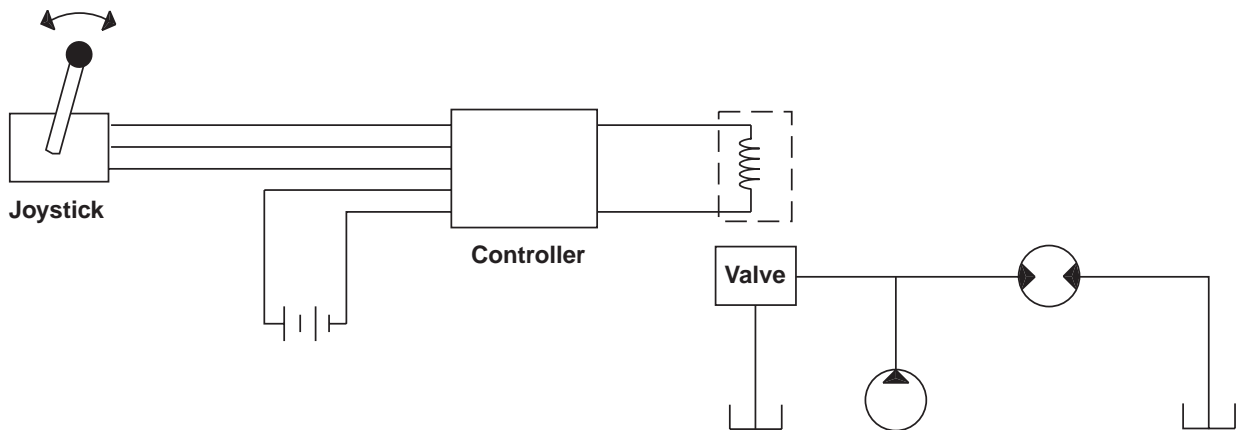
The Bode Plot is used to determine the behavior or stability of the hydraulic and/or mechanical system in which the valve is used. The frequencies at which the valve can operate and maintain the system stability is defined by the maximum frequency known as the **bandwidth**. The minimum frequency is zero Hz or **steady state**.

The bandwidth can be defined using two different points on the graph. The value typically reported in catalog literature is either the frequency at which the phase lags by 90° (Pt. A), or the magnitude decreases by 3 dB (Pt. B).

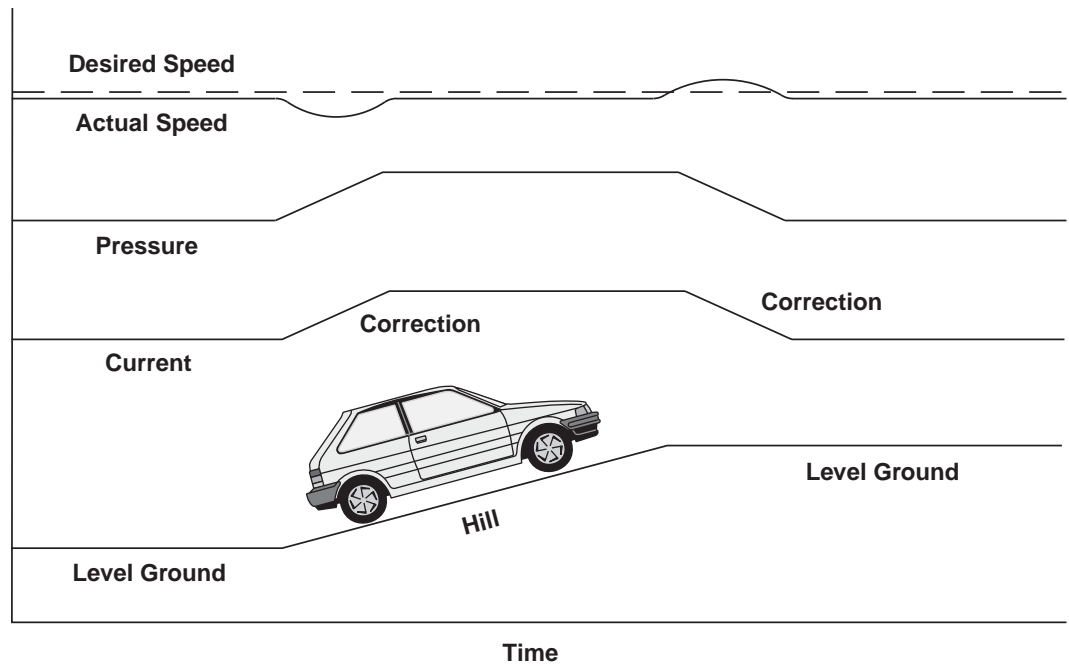
### Open Loop Control

Open loop control is associated with electrical and hydraulic systems. Essentially, it is a system where an operator gives a command and the system tries to obey that command. However, events may occur which cannot be compensated by that original command. When this happens, the operator must make a correction or choose to do nothing. This type of system is best illustrated with an example. The system below is used to regulate the speed of a car. The further the operator moves the joystick, the faster the car goes. If the car encounters a hill, it will slow down if the driver does not compensate and correct the speed.

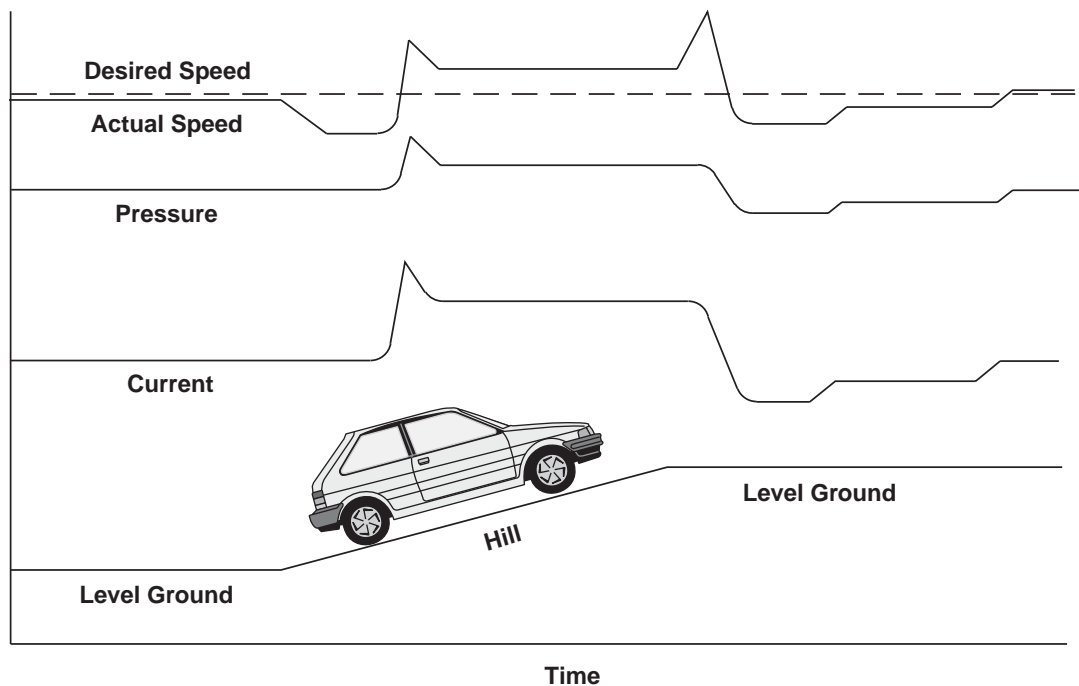
As mentioned above, without feedback, two scenarios for the speed of the vehicle exist. The first possibility is that the operator may not even attempt to adjust for the change in terrain, and the second scenario could be that she attempts to adjust but cannot get the speed back to exactly what it was. Both of these scenarios are shown in the following graphics.



The following graph shows that the current applied to the valve never changes. As a result, the speed decreases as the car goes up the hill. The speed of the vehicle returns to the desired speed once it is back on level ground.

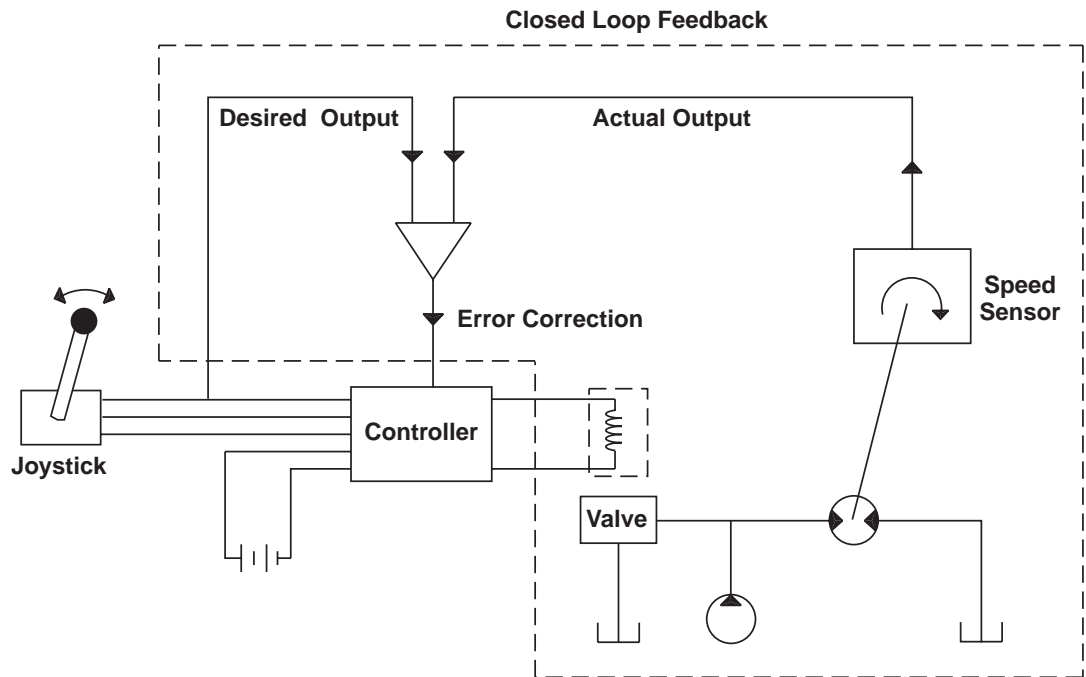


The next graph shows the operator trying to use the speedometer as her feedback device. The operator watches the speedometer while changing the input. The input from the joystick can be varied to correct the output (pressure and speed). In the graph below, the operator is trying to maintain the speed by adjusting the current, but cannot get back to the original or desired speed.



## Closed Loop Control

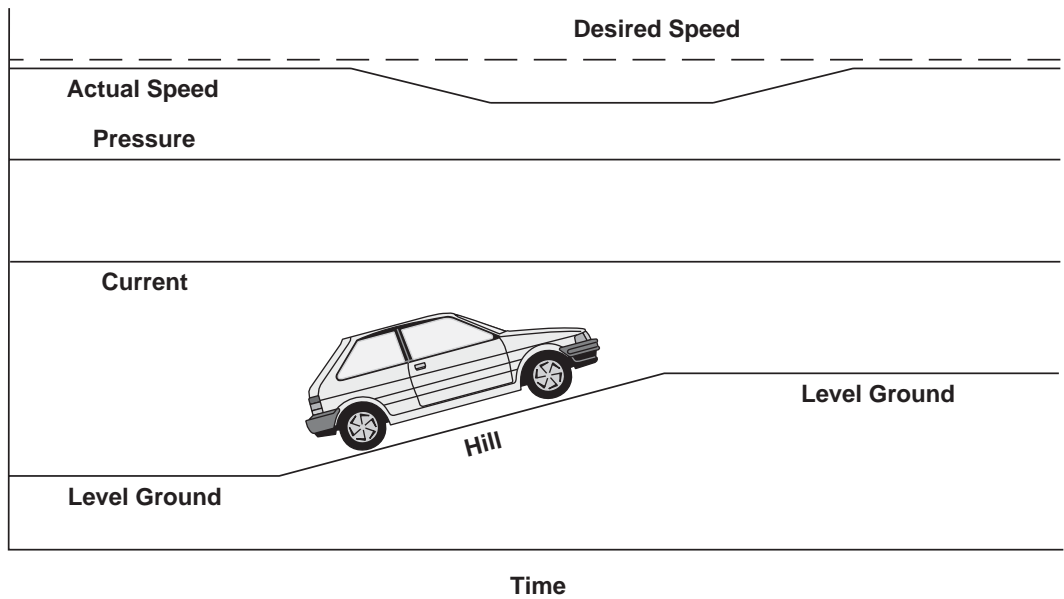
A closed loop control system is one which uses information (feedback) from the hydraulic or mechanical system to correct the output and ensure the desired input is met. Again, the concept is best described by using an example. The following block diagram is an example of a system using closed loop control.



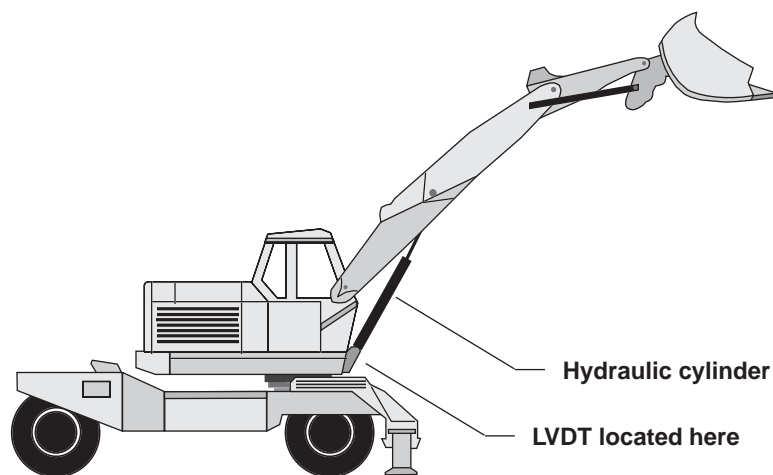
The system shown in the block diagram above works as follows: The operator moves the joystick to obtain a desired speed. A current from the controller is applied to the valve to regulate the pressure, to meet that speed. The speed sensor measures the speed and actual speed is compared against the desired speed. If the vehicle is going up hill, the actual speed may not meet the desired speed. At that point, a correction from the speed sensor is fed into the controller, to increase the current to the coil which will increase the pressure and speed. This method of comparing the actual output to the desired input is known as feedback.

The following example and graph show how the current corrects the pressure and speed when the vehicle begins climbing a hill. The graph shows that initially the speed does decrease, but after a short time, the current applied to the valve increases, bringing the speed back up to the desired speed. As the vehicle continues to climb the hill, the current stays at a higher constant level because the slope or grade of the hill does not vary. When the vehicle reaches the top of the hill, and the road is level again, the speed is higher than desired because the current has not yet been adjusted. Once the current adjusts, the speed returns to the desired speed.





Various sensors or devices can be used for feedback. One such device, as previously mentioned, is a speed sensor. The cruise control on a car has a speed sensor. Another sensor used in hydraulic systems is a pressure transducer. This is used in conjunction with a pressure control valve to compare actual pressure to desired pressure. A position sensor, such as an LVDT (linear variable displacement transformer) is another example of a sensor, as is the Hall Effect sensor. The position sensor can measure the position of a component, relative to a fixed position. For example, the distance a cylinder rod travels, may relate to the position of a bucket on an excavator. By maintaining the position of the bucket, the excavator could dig consistently at a desired depth.





# Appendix B: Quiz Answers

## Introduction

This appendix provides the answers to the quizzes at the end of each chapter.



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## Chapter 2: Quiz Answers

1. A potentiometer may be used to directly regulate the voltage applied to a coil to save money.
2. Smooth DC is available from batteries.
3. The acronym PWM stands **P**ulse **W**idth **M**odulation.
4. PWM is the rapid turning on and off a voltage source at a fixed frequency. By varying the on time or duty, the current applied to the device can be controlled.
5. The dither duty cycle is typically fixed at 50%.
6. Threshold is commonly called  $I_{min}$ . Maximum current is known as  $I_{max}$  or saturation current.
7. True. The maximum rated output for the valve is achieved at  $I_{max}$ .
8. The ramp function controls the rate of change in current output relative to a predetermined time. This function is used so that the output controlled by the proportional valve increases or decreases smoothly even if the input changes suddenly.
9. Dependent ramps are ones where the increasing or ramp while the valve is turning on, is equal to the decreasing ramp that controls the current to the valve while it is turning off. Independent ramps allow the on rate and off rate to be set separately.
10. False, current driven implies that the current is regulated by a closed loop current electronic controller.
11. The advantage of using current is that as a coil heats up the force generated remains the same if the current remains unchanged. When a coil heats up the resistance increases. If the voltage remains the same the current will decrease in accordance with Ohms law. With an electronic controller the voltage applied to the coil is regulated by the fact that the current is continually measured. As the coil heats up the control allows a higher amount of voltage to be applied to the coil. This is done by increasing the on-time of the PWM source.
12. The potentiometer is an input device. The amount of resistance is directly related to the current applied to the valve coil.
13. One example of where potentiometer could be found is in a joystick.

### Chapter 3: Quiz Answers

1. Pressure rise is a measure of how much the pressure increases above the set pressure.
2. Pressure droop is dependent on the flow the valve. Pressure droop is caused by an increase in flow force. In general the flow force is such that it causes the valve spool to close off the metering holes.
3. False, pressure compensation is a term used with a device that can maintain constant flow regardless of the change in pressure.
4. The pressure control range is identified by matching up the threshold hold current with the initial change in pressure for the low end of the range. The high end of the range is related to the pressure achieved at the maximum allowable current or  $I_{max}$ .
5. Hysteresis is essentially the valve components resisting to move when commanded. It is the difference in output as the input current is increased and decreased. Hysteresis is typically measured at one point.
6. Magnetic force increases as the current increases.
7. In order to minimize hysteresis dither is superimposed on to smooth DC signal or a low frequency PWM is used to control the current.
8. Hysteresis is the result of mechanical friction, viscous friction or damping and coercive magnetic force (the resistance for change in strength of the magnetic field).
9. Pressure range is bounded by  $I_{min}$  (where there is an initial change in flow) and  $I_{max}$ . The rate of change in pressure may or may not decrease when the current reaches  $I_{max}$ .  $I_{max}$  for a pressure control is typically based on coil limitation.

## Chapter 4: Quiz Answers

1. It is important to assure that the manual override screw is turned out fully so that the spring force set by the manual override does not become additive to the force developed by the coil.
2. The actuator used in the TS38-20 is known as a force motor actuator. The force versus stroke characteristic of such an actuator is nonlinear. If the stroke was larger than 0.005 the force developed by the coil and therefore the pressure controlled would not be linearly proportional to the current.
3. Spring force is a linear function. As the spring is compressed more the force increases in direct proportion.
4. A linear actuator is used to control the movement of the EHPR spool.
5. The force of the EHPR will remain constant as the air gap changes as long as the current remains constant.
6. The hysteresis of the TS38-20 is less than 3% when a PWM or dither frequency of approximately 200 Hz is applied to the coil.
7. The pressure range of the TS38-21 depends on the setting selected. The graph in the manual shows that the pressure range is from 3000 psi to 100 psi. This is the 'A' setting of the valve. The other two settings are the 'B' and 'C' setting which have a maximum setting of 2000 psi and 1000 psi respectively.
8. The typical application for the TS38-21 is to control the pilot pressure applied a larger spool. The larger spool typically controls the pressure in a cooling fan drive system.
9. The pressure droop in the EHPR08-33 is caused by an increase in flow force as the flow through the valve increases.
10. The amount of that the EHPR08-33 is rated for is 1 gpm. Most hydraulic circuits use a higher flow rate. The EHPR is used to control the pilot pressure applied to a larger directional that is sized such that it can control a flow much greater than 1 gpm.

## Chapter 5: Quiz Answers

1. The TS10-27 is a two stage pilot operated valve. When no current is applied the valve remains closed until the pressure equals that of the valve setting. Once the pressure setting is reached the valve opens and oil is relieved to tank. When current is applied the pressure setting is reduced in proportion to the current applied.
2. The pilot section of a valve is the portion of the valve that controls the pressure above the main stage spool.
3. The main stage is the portion of the valve that controls the system pressure and flow.
4. False, the TS10-26 uses a force motor style actuator.
5. Three types of proportional pressure reducing relieving valves available from HydraForce includes the TS98-30, TS10-36 as well as the EHPR08-33. Others that are available in the catalog include TS98-T31, EHPR98-T33, TS98-T34 and the TS12-36.
6. False, the hysteresis of the TS10-27 decreases when low frequency PWM (approximately 200 Hz) is applied. This is true for most proportional valves.
7. The threshold current required in the TS98-30 is a result of the spring acting against the spool.
8. Like many of the pressure controls there are three pressure ranges available.
9. True, the TS98-30 does have a damping chamber. The reason for the chamber is to damp out oscillations that may occur in the load or at the inlet pressure. Another reason is to assure that the spool does not react too quickly to changes in the pilot section of the valve.
10. Pressure reducing/relieving valves are typically closed in transition in order to assure the valve remains stable as the transition from one mode to the other occurs.

## Chapter 6: Quiz Answers

1. One reason to use a proportional pressure control is to smoothly or gently engage the clutch.
2. The TS10-27 is used in fan drives systems because if the electrical system fails the valve will be set to maximum pressure, thus allowing maximum fan speed. In this type of failure situation it is better to have the maximum fan speed or the maximum amount cooling instead of the opposite.
3. The EHPR is typically used to pilot large directional control spools such as the PE series or ones found in stack valves.
4. Two pressure reducing valves along with a feedback device could control the position of a cylinder. One pressure reducing valve would be placed at each cylinder port. Each valve would be energized to move the load in one direction or the other. One valve would be energized to a higher current than the other to make up for the differential area between the rod side and blind side of the cylinder.



## Chapter 7: Quiz Answers

1. **1.55** gpm Flow rate = **31** (constant, based on oil viscosity of 32 cSt)  
x  
**0.005** (orifice area in square inches)  
x  
**10** (square root of pressure drop of 100 psi)
2. The disadvantage of a fixed orifice is the flow varies when the pressure drop across the orifice varies.
3. One typical application is to meter or control flow into a cylinder.
4. The viscosity of the oil used for performance testing at HydraForce is 32 cSt.
5. True. This is why the term “droop” is associated with pressure compensated flow controls.
6. There are two active metering orifices in a priority/bypass compensator. The first is to regulate flow to the priority port and the second is to regulate flow to the bypass port.
7. Saturation occurs when there is no output, regardless of the change in input.
8. Flow range is bounded by  $I_{min}$  (where there is an initial change in flow) and  $I_{max}$  (where the change in flow stops regardless of a change in current).

## Chapter 8: Quiz Answers

1. The dominate forces acting on the components of the PV70-33 are the spring force and magnetic force. If the pressure drop across the PV70-33 becomes too high then the flow force may become greater than the spring force or magnetic force. Other forces that influence response and hysteresis would be the magnetic hysteresis, mechanical friction and viscous damping or friction.
2. The PV70-33 must be used with an external compensator because the pressure drop across the valve must be limited so that it does not wash shut from flow forces.
3. The differential across the PV70-33 and PV70-35 must not exceed 300 psid.
4. The compensation value is a measure of pressure drop across the fixed orifice that is sufficient to cause the active spool the compensator to begin to regulate or limit flow.
5. The override in the PV product is a screw style override. This allows the metering spool to be opened as desired in a similar that he current controls the opening of the valve.
6. The maximum flow rating of the PV70-33 is determined by the number of holes in the cage and the amount that these holes can be uncovered when the spool moves.
7. One reason to use the EC12-40 with the PV70-33 instead of the EC10-40 is that the EC12-40 can handle a higher bypass flow.
8. The main difference between the PV70-33 and the PV70-35 is that the first is normally closed and the second is normally open.
9. When the load pressure suddenly changes in a circuit where a PFR70-33 is regulating the flow the flow will remain unchanged.

## **Chapter 9: Quiz Answers**

1. A PV72-20 is a normally closed restrictive style proportional flow control.
2. Port 2 of the PV70-30 needs to be blocked in order to make it a restrictive style flow control.
3. When the load pressure suddenly changes a compensating spool will move in response to assure that the pressure drop across the metering orifice remains the same. This assures the flow rate remains the same.
4. The difference between the PV70-30 and the PV70-31 is that the PV70-30 is a normally closed flow control and the PV70-31 is normally open.
5. There are typically two spools in the internally compensated flow controls.
6. True, the magnetic force is balanced by the spring force.
7. The damping is present in many valves to assure smooth and quiet operation.

## Chapter 10: Quiz Answers

1. The patent for the ZL products is the invention of the bidirectional proportional flow control.
2. The pressure sense tube is in the ZL70-30 to assure that the pressure in the armature area is the same as that just inside the metering spool.
3. The difference in the forces acting the compensating spool when the flow changes direction is simply a change in direction in relation to the change in flow. For example the pressure force noted as the *inlet port pressure force* acts in the same direction as the flow.
4. The pressure compensation pressure can be found on the regulated flow versus pressure drop graph. The compensation pressure appears to be approximately 250 psi. This characteristic is found where the knee of the curve occurs.
5. True, the ZL70-30 is a restrictive style flow control.
6. The minimum pressure required for flow to pass from port 3 to 2 of the PV70-33 is 250 psi.
7. The pressure compensation value of the ZL70-33 is 75 psi.
8. The pressure compensation value of the ZL70-36 is 50 psi.
9. The main difference between the ZL70-33 and the ZL70-36 is that the ZL70-36 allows oil to flow from port 3 to 2 when the load pressure is very low.
10. True, the ZL70-36 acts as a priority/bypass flow control when oil is flowing into port 1. When oil flows into port 3 the valve acts as a restrictive style flow control.

## Chapter 11: Quiz Answers

1. Some of the similarities between the SP08-20 and SV08-20 include the basic construction of the tube subassembly, which includes a similar adaptor and guide tube. The plug appears the same on the outside but differs internally because of the shape of the magnetic pole face. Other similar components are the cage, poppet and pilot pin.
2. False, the SP08-20 does not use a push style armature it uses a pull style armature.
3. The three significant forces acting on the components of the SP08-20 include the spring force, magnetic force and pressure force acting on the pilot pin.
4. The forces on the SP08-20 components are always balanced when the valve is operating. The pressure force may initially be higher when the valve is shut complete.
5. The special processes that were developed for the SP08-20 include the dual diameter grinding of the cage and the polishing of the armature. Both processes were developed to reduce mechanical friction.
6. This curve was included to show that the pressure drop through the valve may be decreased when current above  $I_{max}$  is available.
7. False, the performance will degrade when a J or Y style override is selected.
8. False, the SP08-20 uses the same coil as the SV08-20.

## Chapter 12: Quiz Answers

1. The slope of the flow versus current curves differs between the S1 and S2 coil of the SP08-47's because the magnetic force in the S2 direction is greater because the magnetic flux path is relatively stronger.
2. False, the tank ports differ. The tank port is at port 1 for the SP08-47 valves and at port 3 for the SP08-58 valves.
3. False. The maximum flow controlled by the SP08-47DL is less than that controlled by the SP08-47D.

## Chapter 13: Quiz Answers

1. The PV72-30 has a lower compensating value. Therefore, the horsepower loss will be less.
2. A pressure compensator is used maintain a constant pressure drop across an orifice. The benefit of such a combination is that the same across the orifice is maintained regardless of load pressure.
3. The value of the compensating spring in the PV72-21 is 2.45psi. The horsepower is equal to the flow (12gpm) multiplied by the pressure (2.45psi) divided by 1714 or:  
  
$$(2.45 \text{ psi pressure} \times 12 \text{ gpm flow rate}) \div 1714 = 1.72 \text{ hp}$$
4. On the plus side the SP12-20 is a load holding valve. Because of this the cylinder mast will not drift. The draw back to using an SP12-20 in such a circuit is that the hysteresis is greater than that of a PV and the reproducibility band is greater as well.
5. The SP08-57 has an additional port as compared to the SP08-47. This extra port is used to provide load sense pressure to a compensator. This type of valve in conjunction with an external pressure compensator, such as the EC10-32, will provide enhanced control when compared to the SP08-47 without a compensator.
6. The FR08-20 is in the system to bleed-off pressure from the regulated port. If this is not done, pressure on the regulated port would be trapped. This pressure would eventually cause the internal compensator spool of the PV to shift and close-off flow to the bypass port.

## Chapter 14: Quiz Answers

1. The maximum operating temperature for the 01 coil is 75°C when used with the PV08 products. The limitation is a maximum value to assure the material does not fail.
2. The parameters that should be considered to assure a proportional valve coil is properly applied is the available voltage, the maximum ambient temperature, the duty cycle and the voltage drop across the controller.
3. The minimum voltage required to assure that 1600 mA is available to drive a PV76 is 11.1V.
4. If the electrical system is battery driven, a 10V coil is recommended for a 12V system, and a 20V coil is recommended for a 24V system.
5. I-average can be found by adding the threshold current to the maximum current and dividing by 2. It is assumed that typically the SP valves will be used at a current higher than this value and lower than this value. Because the current is always changing when the valve is used continuously, the current level heating the coil is this average value. The valves are designed so that even when the coil heats at this average value, the resistance is low enough for the coil to draw the maximum current when the voltage given on the graph is available at a given temperature.
6. If the electrical system is driven by an alternator connector to an engine, there is less concern if the coil can draw sufficient current.
7. The voltage drop across many of the HydraForce controllers is 1.5V.
8. Continuous-duty operation is typically defined as applying the maximum current for a period of time that exceeds 30 minutes.



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