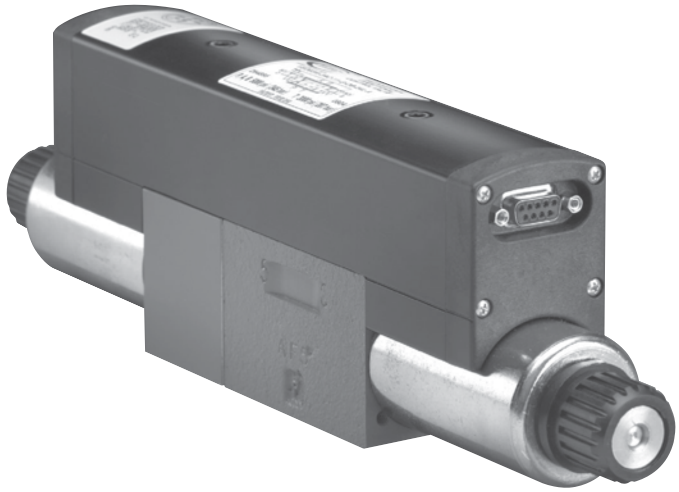
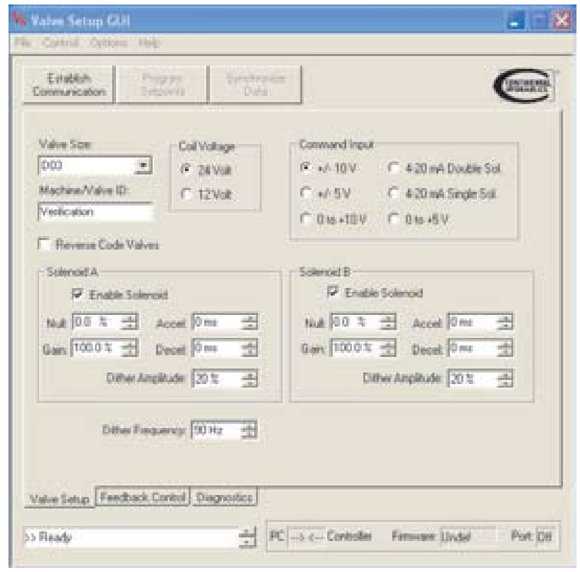
CONTINENTAL HYDRAULICS

# *NEW* VED03M
# PROPORTIONAL DIRECTIONAL CONTROL VALVES

SOLENOID ACTUATED, DIRECT OPERATING WITH ON-BOARD ELECTRONICS






## DESCRIPTION

NFPA D03/ISO 4401 Size 03 manifold mounted 4-way valves. These proportional directional flow control valves are direct operated, sliding spool and spring centered valves. They are used to control flow rate and direction. The valve features an on-board electronic amplifier and a Hall Effect spool position sensor for closed spool inner loop operation.

The on-board amplifier is a single or dual output amplifier designed to operate from a wide range of supply voltages. The supply voltage is efficiently converted to output current by using pulse width modulation (PWM). The output current is controlled to reduce the affects of temperature change on the valve solenoid. Each output is current limited to prevent overdriving the valve and to protect against short circuits with automatic fold back current limiting.

The on-board amplifier is packaged within a fully potted, anodized aluminum housing providing a minimum of IP65 environmental protection.

All set points are factory preset to catalog specifications. Changes to the programming are only accessible through the RS-232 9-pin connection using a Graphic User Interface (GUI) from a desktop, laptop or PDA. The GUI program is compatible with Windows 95, 98, 2000, NT and XP operating systems. PDA operating systems pocket PC 2000-2002 and CE.

Power and control connections are made using the 7/8 – 20UNEF 7-pin connector. (Shell size “14S” with a “A7” pin arrangement).

Command signals may be bipolar or unipolar voltage or current (4 - 20mA). The amplifier is designed to work with a position sensor for valve spool position feedback, providing maximum performance.

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## TYPICAL PERFORMANCE SPECIFICATIONS*

| | | | | |
|---|---|---|---|---|
| MOUNTING SURFACE | | | NFPA/T3.5.1M R1-1984 (D03) ANSI/B93.7M-1986 ISO/4401 SIZE 03 | |
| FLOW CAPACITY @ 145 psi (10 bar) (Full Loop Drop) | A/F*-26 Spools | Nominal | 7 gpm | 26 lpm |
| | A/F*-21 Spools | Nominal | 5.5 gpm | 21 lpm |
| | A/F*-16 Spools | Nominal | 4.2 gpm | 16 lpm |
| | A/F*-09 Spools | Nominal | 2.4 gpm | 9 lpm |
| | A/F*-03 Spools | Nominal | 0.8 gpm | 3 lpm |
| MAXIMUM OPERATING PRESSURE | | P, A, B Ports | 5000 psi | 345 bar |
| | | T Port | 3000 psi | 207 bar |
| TYPICAL RESPONSE TIME | Centered to 100% Spool Travel | | | 55 ms |
| | 100% Spool Travel Back to Center | | | 65 ms |
| HYSTERESIS | | Nominal w/Dither | <1% | |
| THRESHOLD | | Nominal w/Dither | <0.5% | |
| REPEATABILITY | | Nominal w/Dither | <0.5% | |
| OVERLAP | VED03M valves are factory preset to compensate for the effect of spool overlap. | | | |
| COIL DATA | Code 12L | Voltage | 12 VDC | |
| | | Current | 3.8 ohms (+/-10%) | |
| | | Wattage | 19 (@ 76°F./24°C.) | |
| | | Continuous Amps | 2.2 Max. | |
| | Code 24L | Voltage | 24 VDC | |
| | | Current | 15.2 ohms (+/-10%) | |
| | | Wattage | 19 (@ 76°F./24°C.) | |
| | | Continuous Amps | 1.1 Max. | |
| | | Insulation | Class H | |
| | | Duty Cycle | Continuous | |
| MOUNTING | | | Unrestricted (Horizontal Preferred) | |
| FLUID | Any hydraulic fluid compatible with Viton A or Buna N elastomers. | | | |
| | Fluid temperatures up to 150°F. (65°C.) will not appreciably affect valve performance, however, for safety, temperatures above 130°F. (54°C.) are not recommended. Minimum temperatures are determined by the maximum startup viscosity of 4000 SUS (863 Cs). Minimum viscosity is 30 SUS (0.3 Cs). | | | |
| | Fluid Cleanliness should be ISO 4406 Code 17/15/12 up to 3000 psi (315 bar); 15/13/11 for 3000 psi (315 bar) and above. | | | |

***NOTE:** Data taken with fluid temperature at 120°F. (49°C.) and viscosity at 100 SUS (20.6 Cs).

## PROGRAMMING REQUIREMENTS

| | PC | PDA |
|---|---|---|
| OPERATING SYSTEM | Win 95, 98, 2000, NT, XP | Win Pocket PC 2000-2002 Win CE |
| REQUIRED DISK SPACE | 1021kb | 320kb |
| COMMUNICATION PORT | Com1, or next available | Compact Flash Slot, Type II |
| COMMUNICATION CABLE | DB9/RS-232 male to female | Compact Flash to RS-232, Type I or Type II |

## ELECTRICAL/CONTROLSPECIFICATIONS

| | | | |
|---|---|---|---|
| POWER INPUT CONNECTION | | | 7/8-20UNEF Thread 14S-A7 |
| POWER INPUT (Typical) | 10 to 32 VDC* | @ 12 VDC | 3.4 amps |
| | | @ 24 VDC | 1.7 amps |
| POWER INPUT PROTECTION | No damage from reversed power leads or noise spikes. Board will not power if polarity is reversed. | | |
| ENABLE/ DISABLE | Close contact enable. To enable the outputs, power enable pin C to 9 - 32 VDC power source. This source may come from power pin A and may include a safety switch. | | |
| COMMAND INPUTS | | | +/-5 VDC<br>+/-10 VDC<br>0 to 5 VDC<br>0 to 10 VDC<br>4 - 20 mA |
| NULL ADJUSTMENT RANGE | | in 0.5% increments | 0 to 50% |
| GAIN ADJUSTMENT RANGE | | in 0.5% increments | 50 to 100% |
| RAMP RATE ADJUSTMENT | Limits the rate at which the valve opens or closes. Each solenoid has its own independent Accel and Decel adjustments. | | |
| | | in 5 mS increments | 0 to 30 seconds |
| POWER OUTPUT | Selectable<br>Independent outputs<br>Short circuit and overload protection<br>Open load detection<br>15kHz PWM high frequency output | | |
| DITHER FREQUENCY (Programmable) | | in 5 Hz increments | 30 to 360 Hz |
| DITHER AMPLITUDE (Programmable - % of I-max.) | | in 1% increments | 0 to 20% |
| TEMPERATURE RANGE | -40°F. to 185°F. (-40°C. to 86°C.) will not appreciably affect valve performance. However, for safety, temperatures above 130°F. (54°C.) are not recommended. | | |
| INTERNAL POSITION FEEDBACK | | | Factory calibrated |
| ENVIRONMENTAL PROTECTION | | | IP65 / NEMA 4 |

***NOTE:** For full valve shift, voltage must be at least the rated solenoid voltage.

**System Control Spools**

Selecting the correct spool is critical for best control in any application. A valve sized incorrectly can be the difference between correct consistent operation and poor overall control. Continental Hydraulics offers not only a wide variety of flow rates, but also offers a variety of metering functions. These metering functions are designed to match the load, actuator and circuit characteristics for the best possible control.

It is important to choose the correct spool for your application. Typically choose a spool that will pass the flow you need at approximately a 200 - 300 psid full loop pressure drop for best overall performance. Refer to the Flow Curve Charts. This pressure drop and/or back pressure provide system stiffness that is required for optimum control.

The metering characteristics of the spool will be based on the load characteristics an/or circuit design. Spool metering options available are combination metering, meter-in, meter-out, 2:1 ratio, 1.3:1 ratio and position control.

**Code "C" -** Combination metering spools meter fluid into and out of the actuator equally in either direction. Combination metering spools are highly recommended for motor circuits to provide both good acceleration and deceleration load control.

**Code "I" -** Meter-in spools meter fluid into the actuator. This style of metering should be used in circuits where the actuator is always working against a resisitive load or when a counterbalance valve is used to hold or keep the load from running away. This spool is not recommended for use to decelerate a load within the use of other devices.

**Code "O" -** Meter-out spools meter fluid out of the actuator. This style of metering is typically used in circuits where the load will create a run away condition.

**Code "PC" -** Position control spools are combination metering style spools, slightly under lapped in the center condition to provide better control at the null condition when used in closed loop cylinder positioning applications.

**Code "T" -** This 2:1 ratio metering spool is designed to give equal metering and excellent control over hydraulic cylinders that have a 2:1 bore to rod effective area ratio.

**Code "CY" -** This 1.3:1 ratio metering spool is designed to give the equivalent of an equal metering characteristic for most standard catalog bore and rod combination hydraulic cylinders giving better control than other styles of spools.

Combination, meter-in, 2:1 ratio and 1.3:1 ratio spools can easily be used with pressure compensators to provide proportional pressure compensated flow control.

# PROPORTIONAL DIRECTIONAL CONTROL VALVES

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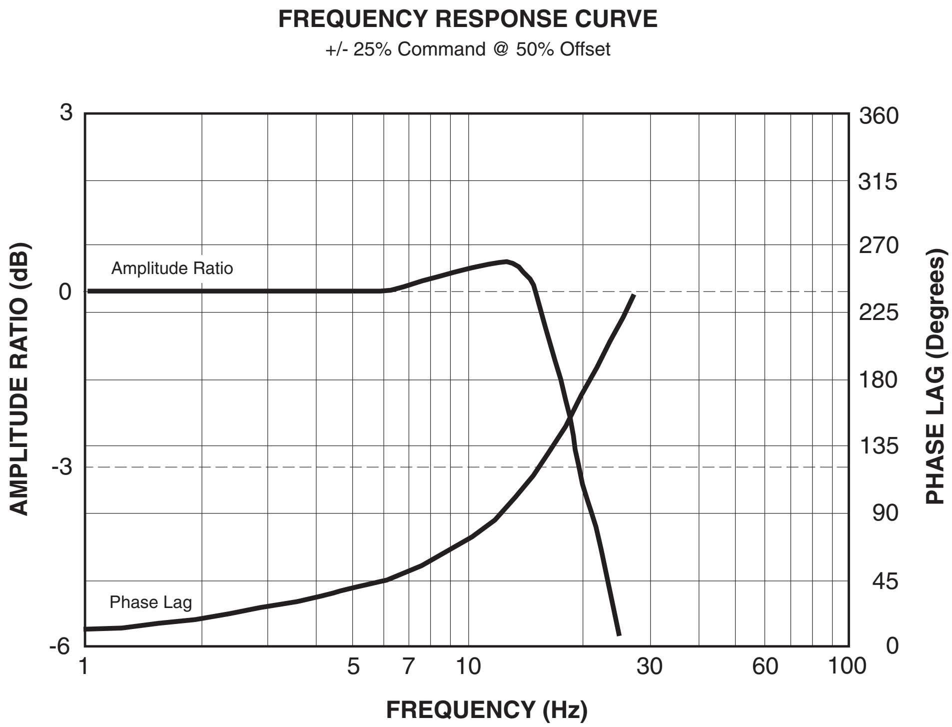
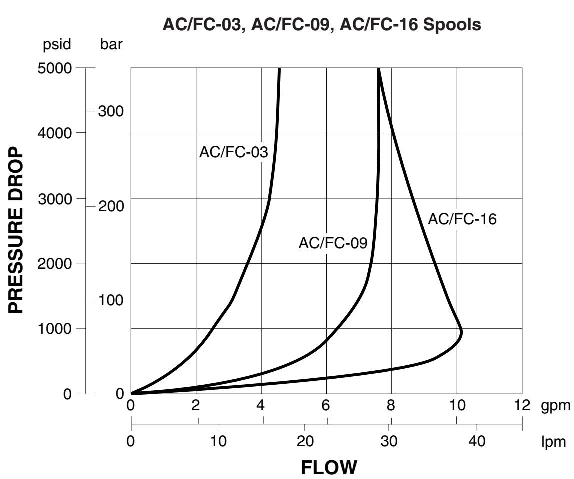
## FREQUENCY RESPONSE CURVES

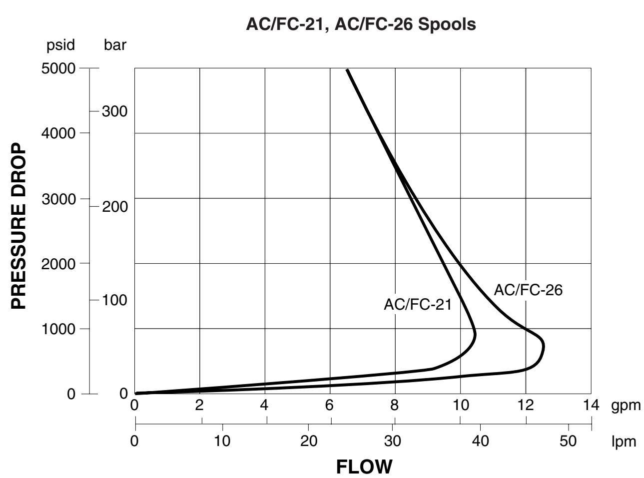
+/- 25% Command @ 50% Offset - Amplitude and Phase lag




## LIMITING POWER ENVELOPE CURVES

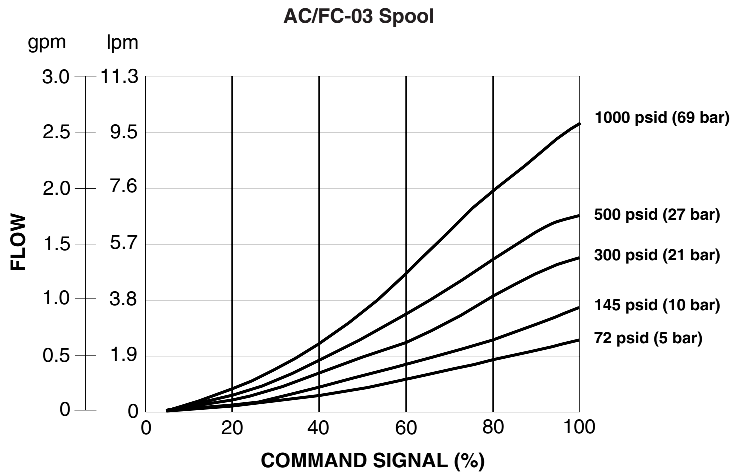
Full Loop @ 100% Command Signal

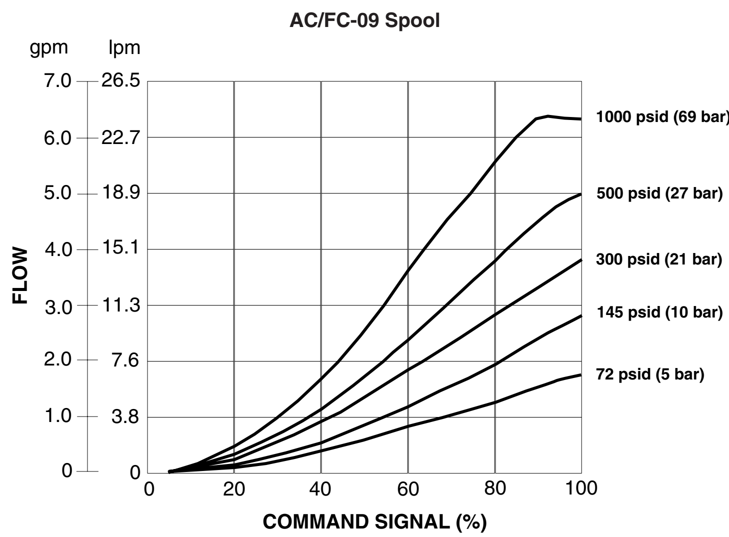


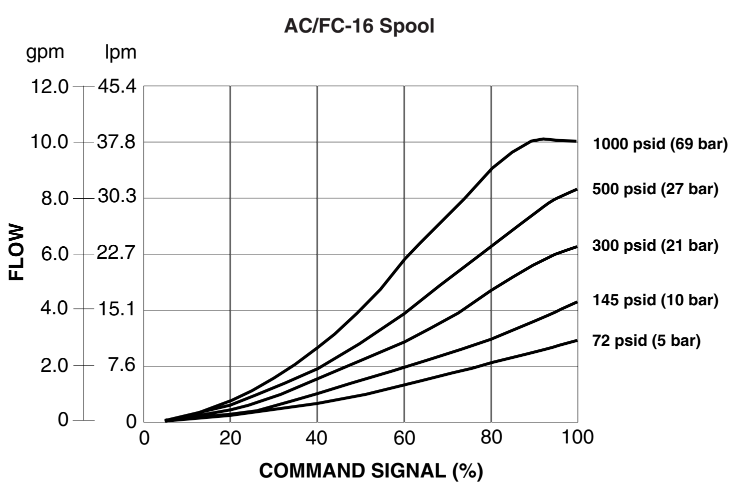



## FLOW CURVES AT CONSTANT PRESSURE DROP

P to A, B to T or P to B, A to T



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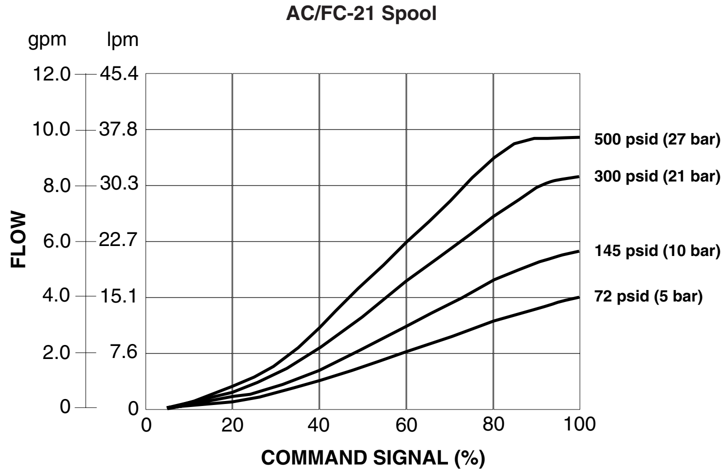
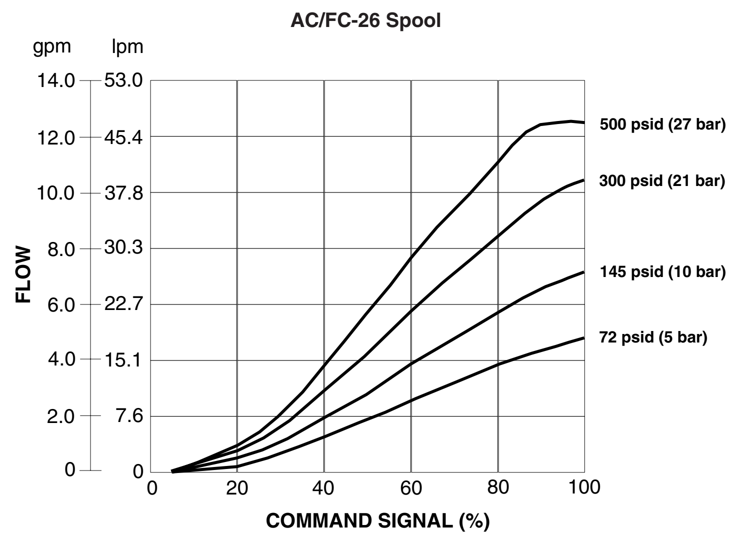
SOLENOID ACTUATED, DIRECT OPERATING WITH ON-BOARD ELECTRONICS

## FLOW CURVES AT CONSTANT PRESSURE DROP

P to A, B to T or P to B, A to T

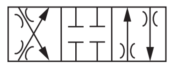
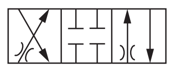
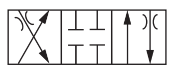
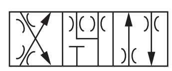
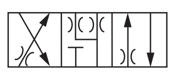
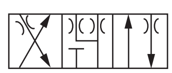


# PROPORTIONAL DIRECTIONAL CONTROL VALVES

SOLENOID ACTUATED, DIRECT OPERATING WITH ON-BOARD ELECTRONICS

## SPOOL and SPOOL FLOW RATES CODES

| SPOOL FUNCTION CODE** | DESCRIPTION | SYMBOL | FUNCTION |
|---|---|---|---|
| AC | METER IN METER OUT |  | |
| AI | METER IN |  | |
| AO | METER OUT |  | |
| FC | METER IN METER OUT |  | MOTION CONTROL |
| FI | METER IN |  | |
| FO | METER OUT |  | |
| PC | METER IN METER OUT | | POSITION CONTROL |
| CY | 1.3:1 Flow Ratio | | CYLINDER SPOOL |
| T | 2:1 Flow Ratio | | |

****NOTE:** Consult factory for spool availability.

| SPOOL FLOW RATE CODE** | NOMINAL FLOW* gpm (lpm) |
|---|---|
| 26 | 7.0 (26) |
| 21 | 5.5 (21) |
| 16 | 4.2 (16) |
| 09 | 2.4 (9) |
| 03 | 0.8 (3) |
| 31 | 8.2 (31) |
| 25 | 6.5 (25) |
| 20 | 5.2 (20) |
| 15 | 4.0 (15) |
| 05 | 1.3 (5) |
| 31 | 8.2 (31) |
| 25 | 6.5 (25) |
| 20 | 5.2 (20) |
| 15 | 4.0 (15) |
| 05 | 1.3 (5) |
| 26 | 7.0 (26) |
| 21 | 5.5 (21) |
| 16 | 4.2 (16) |
| 09 | 2.4 (9) |
| 03 | 0.8 (3) |
| 31 | 8.2 (31) |
| 25 | 6.5 (25) |
| 20 | 5.2 (20) |
| 15 | 4.0 (15) |
| 05 | 1.3 (5) |
| 31 | 8.2 (31) |
| 25 | 6.5 (25) |
| 20 | 5.2 (20) |
| 15 | 4.0 (15) |
| 16 | 4.2 (16) |
| 05 | 1.3 (5) |

***NOTE:** Flow at 145 psi (10.0 bar) pressure drop (full loop).
****NOTE:** Consult factory for spool availability.

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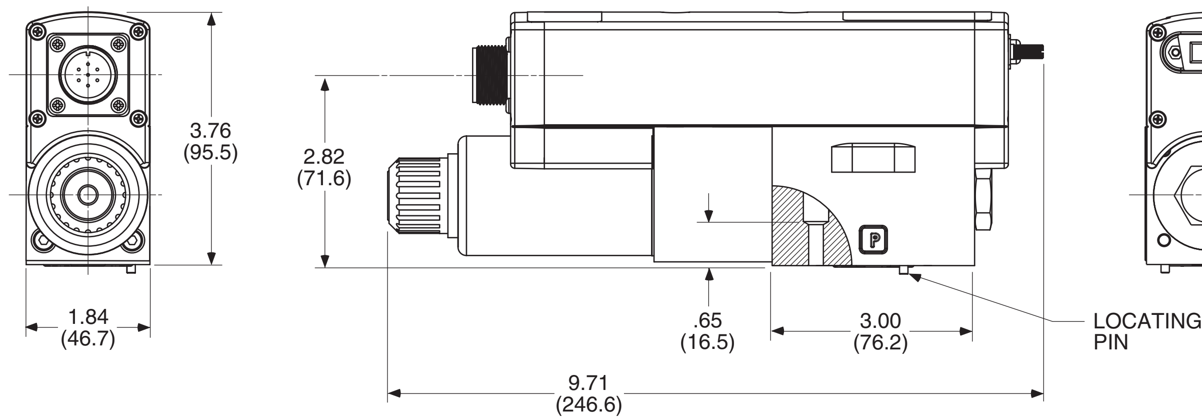
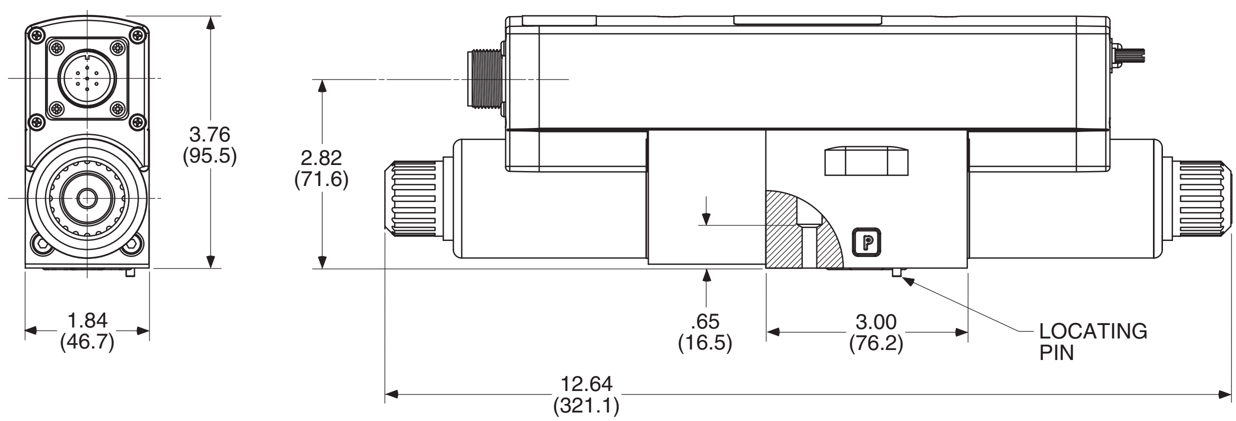
## DIMENSIONS

Dimensions shown in: Inches (millimeters)

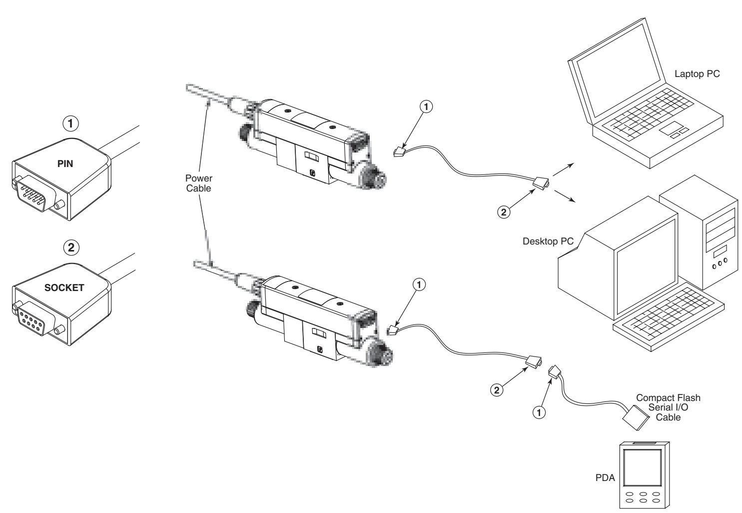
### Single Solenoid




### Double Solenoid




## CONNECTION to COMPUTER or PDA



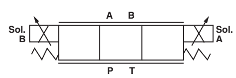
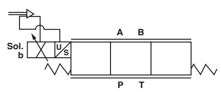
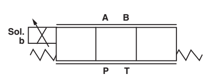
# PROPORTIONAL DIRECTIONAL CONTROL VALVES

SOLENOID ACTUATED, DIRECT OPERATING WITH ON-BOARD ELECTRONICS

## ORDERING INFORMATION

| | SELECT ONE | SELECT ONE | SELECT ONE | | SELECT IF REQ'D | SELECT IF REQ'D | SELECT ONE | | DESIGN LETTER |
|---|---|---|---|---|---|---|---|---|---|
| **VED03M –** | ☐ | ☐ **–** | ☐ **–** | **G –** | ☐ **–** | ☐ **–** | ☐ | **L –** | **A** |
| | BASIC VALVE | SPOOL FUNCTION | FLOW RATES | SEALS | MECHANICAL OPTIONS | ELECTRICAL OPTIONS | SOLENOID | SOLENOID MFG. | |

### BASIC VALVE

| CODE | DESCRIPTION | SYMBOL |
|---|---|---|
| 3 | DOUBLE OPERATOR 3 POSITION SPRING CENTERED |  w/OB |
| | |  Standard Valve |
| 5 | SINGLE OPERATOR 2 POSITION SPRING CENTERED |  w/OB |
| | |  Standard Valve |

### ELECTRICAL OPTIONS

| CODE | DESCRIPTION |
|---|---|
| D | DIN 43650 SOLENOID CONNECTORS |
| OB | ON-BOARD ELECTRONICS 7 PIN CONNECTION INTERNAL HALL LOOP |

### SOLENOID MFG.

| CODE | DESCRIPTION |
|---|---|
| L | LISK |

### SOLENOID

| CODE | VOLTAGE |
|---|---|
| 12 | 12 VDC |
| 24 | 24 VDC |

### MECHANICAL OPTIONS

| CODE | DESCRIPTION |
|---|---|
| OMIT | STANDARD DOUBLE SOLENOID AND SINGLE SOLENOID "A" PORT END |
| R | SINGLE SOLENOID "B" PORT END |

### SPOOL FUNCTION

| CODE |
|---|
| REFER TO CHART FOR SPOOL AVAILABILITY PAGE 7 |

### FLOW RATES

| CODE |
|---|
| REFER TO CHART FOR FLOW RATES PAGE 7 |

### SEALS

| CODE |
|---|
| VITON SEALS STANDARD |

TYPICAL ORDERING CODE:

**VED03M-3AC-26-G-OB-24L-A**

## TERMINOLOGY

**Backlash.** The free play between interacting mechanical parts. Occurs when motion is reversed.

**Compliance.** The springiness of an object. Amount of displacement per unit of force.

**Deadband.** The amount the spool must travel from the center condition to the point that flow starts. Caused by the overlap of the spool lands to the valve body lands.

**Dither.** Used to reduce the effects of friction of the spool to the body. A small amount of oscillating power added to the output power going to the valve coil. This signals rate and/or amplitude is adjustable so the effect will keep the spool in motion, but will not affect the output from the valve.

**Flow Gain.** Relationship of control flow to input current, typically expressed as GPM/ma.

**Frequency Response.** The measurement of how the output responds to an oscillating input signal of varying frequency with fixed amplitude. Measured in terms of decibels and/or phase lag. Decibels (dB) is given at the –3dB point (the point at which the output is approximately 70% of the commanded output). Phase lag is given at the –90° point. Phase shift as compared to the input signal (the output is at 100% shift when the command is at 0%).

**Gain (Current maximum).** Sets *maximum* amperage to the solenoid. Used to the maximum flow or pressure from the valve to the system. Do not adjust pass the maximum the system can supply or the system may not respond as desired if long ramp times are used.

**Hysteresis.** The difference in the input current to produce the same output when going from center to full shift and back to center. Typically measured at 50% signal in both directions.

**Inertia.** The property of an object that resists change in motion. The inertia of an object is dependent on the mass and shape. Simply put an object at rest tends to stay at rest; an object in motion tends to stay in motion.

**Internal Leakage.** There are two sources of internal leakage. The first is the leakage between the main body and spool and the second is pilot flow (some proportional and servo components require a small pilot flow through a hydraulic amplifier known as “quiescent flow”).

**Linearity.** The maximum deviation of the control flow from the best straight line of flow gain, expressed as a percent of rated current.

**Null Bias (Current minimum).** Sets *minimum* amperage to the solenoid. Used for deadband reduction, or will set a minimum flow or pressure. Always adjust null before adjusting gain pots.

**Pressure Compensator.** Devices used to create a consistent pressure differential between the inlet and the outlet of an orifice. The most commonly used in electro-hydraulic circuits are the *restrictive* and *by-pass* types. Care must be taken when applying these components since they naturally will have inconsistent pressure drops at various flow rates through a given valve. However, they will improve the system performance when widely changing loads are seen.

**Pressure Drop**. In order to have flow, there must be differential in pressure between two points. It will also require some amount of force (pressure) to push the fluid through an orifice. A pressure drop, unlike in standard hydraulic systems, is a “good thing” and is required. Pressure drops creates stiffness in the system, stiffness = controllability. Although pressure drop results in wasted energy through heat, it is the cost of getting in control.

**Pressure Gain.** A measure of the change in control port pressures as the input current is varied about the zero flow point.

**Pulse Width Modulation (PWM).** An effective method of controlling electrical power without creating heat. PWM is the amount or percent of time that power is ON for one cycle. If power is on for 25% and off for 75% of a cycle of a 12 volt supply, the average amount would be 3 volts. The frequency must be significantly higher than the valve response.

**Ramp Accel.** Limits the rate an *increasing command* can open or increase the valve output.

**Ramp Decel.** Limits the rate a *decreasing command* can close or decrease the valve output.

**Repeatability.** The ability to repeatedly return to the same output for the same input from the same direction.

**Resolution.** The smallest amount of input that results in a change in output.

**Step Response.** The amount of time it takes a spool to shift for a stepped input signal.

**Symmetry.** The degree of equality between the flow gain of one direction and that of the reversed direction.

**Threshold.** The minimum change in reverse output with the reversal of input signal. Percent of command change required to show a change in output.

## CONCEPTS

**How Does a Direct Acting Proportional Flow Control Valve Work**

There are three components or items that are required. First, a spool and body assembly that are designed to gradually open or close a flow path (metering notches or orifice) as it is moved along its distance of travel. The second component is a proportional solenoid and spring arrangement that will position the spool to a point, where the electromotive force developed by the current applied to the coil is balanced by the resistive force pushing back from the spring. The third component that makes the proportional valve work is the amplifier card. An amplifier card is the component that will convert a low power electrical command signal into a higher power controlled current to run the solenoid.

**System and Valve Sizing**

In order to correctly "size" a valve for a given application, you MUST know the application. Start with the load itself and ask the following questions:

- What is the load? Is the load an over running (flywheel) type, restrictive type (always pushing back) or both in the movement?
- How many Hz is required to do the work?
- What amount of time is available to make the move?
- How fast is the step response of the valve being considered?
- What are the pump flow and pump response time?
- How accurately must the load maintain (plus or minus what % of force, rpm, position)?
- What force is required to move the load in the time allotted? *Force required to move the load = load mass + force due to acceleration + load friction + external force + seal friction*

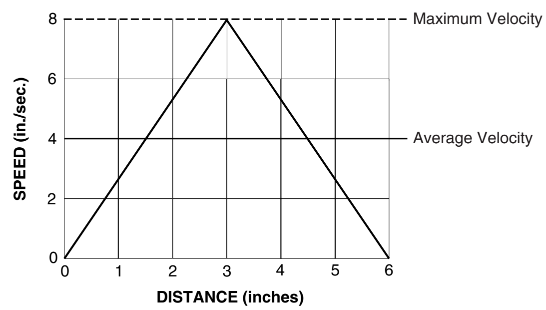
**Example:** To determine the correct system components, review this typical cylinder application that helps to show the relationships of time, mass, cylinder size and flow rates.

- Cylinder has a 2 inch (50.8 mm) bore, 1 inch (25.4 mm) rod and 6 inch (152.4 mm) stroke.
- Load is 5000 lbs. (2268 kg) in the extend mode; 1000 lbs.(453.6 kg) in the retract mode.
- Cycle time is 1.5 seconds in both the extend and retract mode.
- The load is a vertical elevator type lift where a second operation will remove the load off the platen.
- A smooth acceleration and deceleration is desired on the extend stroke.

Based on the above information, it will be assumed that a counterbalance valve will be used to prevent the cylinder from free falling during the retract mode. A uniform acceleration and deceleration motion profile will be used.

**Typical Formula and Calculations Required**

1. Cylinder area for extend for a 2 inch bore = (2 x 2) x .7854 = 3.14 sq. in.
2. Cylinder effective area for retract a 1 inch rod = 3.14 - [(1 x 1) x .7854] = 2.36 sq. in.
3. Average (extend mode) velocity for a uniform acceleration and deceleration means that one-half the time of the extend cycle will be acceleration and half deceleration or .750 seconds. Therefore, the maximum velocity = distance/time or 6 inches/.750 seconds = 8 in./sec.




4. System peak flow can now be calculated using the peak velocity of the cylinder and the extend area. Peak flow in gpm = (Vm x area x 60)/231 or (8 x 3.14 x 60)/231 = 6.53 gpm peak flow.
5. Acceleration is then velocity maximum/time or 8 in. per sec/.750 sec. = 10.67 in./sec.$^2$.
6. The force of acceleration = load mass x acceleration (mass is weight/gravity) or (5000/386.4) x 10.67 = 138 pounds.
7. Total force pressure = force of acceleration + load/extend area = (138 + 5000)/3.14 = 1640 psi.
8. System pressure = total force pressure required at the cylinder + valve pressure drop (see performance curve of valve being used) + line loss + seal friction = 1640 + 195 (estimated) + 100 + 165 = 2100 psi estimated system pressure.

To show how time, flow or cylinder size affect each other in the above example:

- Change the 1.5 second time to 1 second results in 9.8 gpm and an additional 50 psi.
- Drop the peak flow available to 3.5 gpm results in an increase of pressure at the cylinder as the load will need to be accelerated to its peak speed quicker. This may result in damage to the product being moved if the g forces are too great.
- Change the cylinder size will change both the flow rates and pressures required.

**Spool Selection**

Selecting the correct spool is critical for best control in any application. A valve sized incorrectly can be the difference between correct consistent operation and poor overall control. Continental Hydraulics offers not only a wide variety of flow rates, but also offers a variety of metering functions. These metering functions are designed to match the load, actuator and circuit characteristics for the best possible control.

When selecting a spool for given application, it is recommended that the spool's rated flow be as closely matched to the maximum flow required. Size the spool for approximately a 200 - 300 pressure drop across the valve (see flow curve charts) for best overall performance. This pressure drop and/or back pressure provide system stiffness that is required for optimum control.

The metering characteristics of the spool will be based on the load characteristics an/or circuit design. Spool metering options available are combination metering, meter-in, meter-out, 2:1 ratio, 1.3:1 ratio and position control.

The flow control range of a valve should be kept within a 20:1 ratio for the best results. Do not expect to have one valve control 0.5 gpm and 50 gpm accurately at either condition.

Due to external factors like high oil or ambient temperature, coil power losses may effect the maximum output from the valve used. In situations where high temperatures may come into play, size the valve so the maximum flow is achieved prior to the maximum rated current of the coil is used.

**Pressure Drop and Flow Relationships**

It must be understood that *all proportional and servo control valves are orifices*. With that stated, the relationship between pressure drop (the difference between the inlet pressure and the outlet pressure) is expressed by general formula for flow through an orifice: $Q = K(A)\sqrt{\Delta P}$

Q = Flow
K = Orifice Constant
A = Area
$\Delta$P = Pressure Drop

An example of this would be a valve rated for passing 10 gpm (37.8 lpm) at a pressure drop of 100 psi (6.9 bar) will pass about 14 gpm (53.0 lpm) at a pressure drop of 200 psi (13.8 bar). As you see if the load required pressure changes or drops by 100 psi (6.9 bar), the flow will automatically change by 40%. Care must be taken to watch how the load being controlled may change. If wide swings are possible, other components may be required to compensate for the effect.

**Open Loop Control Systems**

The system responds to a command input signal to vary the output accordingly, but there will not be any corrections made to the output based on what is happening at the load.

**Closed Loop Control Systems**

The system responds to the command input as in the open loop system, but the output will be corrected via a comparison of the command input signal to a feedback signal coming from a source at the load.

An example of open and closed loops would be a car going from a flat surface to an incline without adjusting the gas pedal (command input source). This is open loop. In closed loop, adding cruise control (feedback input signal) will adjust the system output closing the loop for the desired control. A true closed loop control system will sense the system output and automatically correct any difference between the desired system reaction and the actual system reaction.

# PROPORTIONAL DIRECTIONAL CONTROL VALVES

SOLENOID ACTUATED, DIRECT OPERATING WITH ON-BOARD ELECTRONICS

Closed loop systems for hydraulic applications can be defined by three methods. Each required a certain type of logic to achieve the best performance.

- Position Control
- Velocity Control
- Force or Pressure Control

The basic concept of **position control** is to move to a point and stop. This requires a logic system that will in essence have a command source (analog) of one polarity and value, and a feedback source that will be the same in value but opposite in polarity. Once in position, the two signals will cancel each other providing the control valve a zero or off command, and the valve will close. Digital systems will typically be commanded to a position of X pulses, and once the system has counted out the correct number of pulses, it will send an off signal to the control valve.

The concept of **velocity control** is to set an actuator speed and hold it constant. Unlike position control where the valve must close to hold position when the loop error (position error) is zero, the valve in velocity control must hold open when the loop error (velocity error) is zero to maintain desired velocity. The error, between the feedback and command, is summed with the command resulting in an open valve. When the velocity error goes to zero, the output to the valve holds steady. Any further errors in velocity will adjust output up or down to correct the loop.

**Force or pressure control** is similar to velocity control. In pressure control, the valve must remain energized (open) when the loop error (pressure error) is zero to maintain desired pressure. The error, between the feedback and command, is summed with the command resulting in an energized valve. When the pressure error goes to zero, the output to the valve holds steady. Any further errors in pressure will adjust output up or down to correct the loop.

These systems will also require other mathematical calculations to help gain speed and accuracy. These calculations are done in the “P I D” loop closure part of the control circuit.

- P Proportional
- I Integration
- D Derivative

“PID” example: You need to move your vehicle from point A to point B, down the road with several curves using only ”P” and “I”. On the first run, drive the course using only the rear view mirror. You will not travel as fast (Proportional) towards the destination as by the time you see the road has curved (Phase Lag), you will need to correct your course. As you move along (Time), the curve will cause you to turn the wheel more as you note that you are further off target (your brain multiples the error by the time involved (Integration). At some point you will have over compensated and you will go off target with the opposite error, and the process will start over again. You solve this by slowing down (Proportional) so the effectiveness of the corrections (Integration) are more substantial; or add a side view (a little Derivative) so the corrections can be made quicker; or add a forward view (a lot of Derivative) that will anticipate the corrections based on your eyes seeing the change allowing to start the corrections quicker as they come up.

- **Proportional Term** - (moderate frequencies). As the proportional term is increased, the effectiveness of Integration is lowered and the effectiveness of Derivative will come into effect later.
- **Integraton Term** - (low frequecies, adds phase lag). The primary benefit is the reduction of steady state error.
- **Derivative Term** - (high frequecies, adds phase lead). Helps improve responsiveness and stability.

**Adjusting “PID”**

The adjustment procedure is to reduce the “I” and”D” term values to minimum so the “P” term value can be set with little or no effect from the “I” and “D” terms. Increase the “P” term until the system instability occurs. Set the “P” term about 30% less than that point. Next, raise the “I” term until the system is about to go unstable, then increase “D” term to improve system stability. Repeat increasing “I” and then “D” as needed.

**Rule of Thumb** - Always select a feedback device that measures what you want to measure! An example of this would be to use a load cell on the cylinder rather than a pressure transducer. A pressure transducer is a device that does not take into account seal friction, mechanical friction, etc. that will be subtracted from the actual force that is being exerted.

**Formulas and Reference Material**

- Current is measured in Amps (A)
- Voltage is measured in Volts (V)
- Resistance is measured in Ohms (O)
- Inductance is measured in Henries (H)
- Capacitance is measured in Farads (F)

Ohm's Law: Voltage = Current x Resistance for DC
$V = I \times R$

Power: Power (Watts) = Current x Voltage
$W = I \times V$

Flow through an orifice: $Q = K (A) \sqrt{\Delta P}$ (see Pressure Drop and Flow Relationships)

Force due to Acceleration: The force to overcome the combination of several load and inertia components can become a large factor in high speed applications. The following information and mathematical formulas will be required to calculate the overall requirements.Force required = load mass + acceleration + external force + seal friction.

- Load mass (in pounds) can be total weight or a percentage (%) of total weight as dictated by angle of incline and/or coefficient of friction.
- Acceleration Force = load mass x acceleration
  - Load Mass = [mass/386.4 (gravity)]
  - Acceleration = [Max. velocity (in./sec.)/time to move (sec.)].
- External Force: Any changes made to the load due to external sources (example would be an addition or subtraction of weight due to a box coming on or off a conveyor).
- Seal Friction: Use 10% of maximum force.

## FLUID POWER FORMULAS

### Basic Formulas

| Formula for: | Word Formula: | Letter Formula: |
|---|---|---|
| Fluid Pressure (in pounds per square inch) | Flow Rate = $\frac{\text{Force (pounds)}}{\text{Unit Area (sq. inches)}}$ | $P = \frac{F}{A}$ or psid = $\frac{F}{A}$ |
| Fluid Flow Rate (in gallons per minute) | Flow Rate = $\frac{\text{Volume (gallons)}}{\text{Unit Time (minute)}}$ | $Q = \frac{V}{T}$ |
| Fluid Power (in horsepower) | Horsepower = $\frac{\text{Pressure x Flow}}{1714}$ | $HP = \frac{P (Q)}{1714}$ |
| Velocity through Piping (in feet per second) | Velocity = $\frac{\text{0.3208 x Flow Rate thru I.D. (gpm)}}{\text{Internal Area (sq. Inches)}}$ | $V = \frac{.3208 (Q)}{A}$ |
| Compressibility of Oil (in additional required oil to reach pressure) | Volume = $\frac{\text{Press. x Volume Oil Under Press.}}{\text{250,000 (approx.)}}$ | $VA = \frac{P (V)}{250{,}000}$ (See Note 1 Below) |
| Compressibility of a Fluid | Compressiblity = $\frac{1}{\text{Bulk Modulus of Fluid}}$ | $CB = \frac{1}{BM}$ (See Note 2 Below) |
| Specific Gravity of a Fluid | Specific Gravity = $\frac{\text{Weight of 1 Cu. Ft. of Fluid}}{\text{Weight of 1 Cu. Ft. of Water}}$ | $SG = \frac{W}{64.4283}$ |
| Viscosity to Centistokes (SUS to Cs) (For 32 SUS to 100 SUS) | Centistokes = 0.2253 x SUS - $\frac{194.4}{\text{SUS}}$ | $Cs = 0.2253 (SUS) - \frac{194.4}{SUS}$ |
| (For 100 SUS to 240 SUS) | Centistokes = 0.2193 x SUS - $\frac{134.6}{\text{SUS}}$ | $Cs = 0.2193 (SUS) - \frac{134.6}{SUS}$ |
| (For 240 SUS and greater) | Centistokes = $\frac{\text{SUS}}{4.635}$ | $Cs = \frac{SUS}{4.635}$ |

**Note 1:** Use 0.3208333 for greater accuracy. **Note 2:** Approximately .5 % per 1000 psig.

### Pump Formulas

| Formula for: | Word Formula: | Letter Formula: |
|---|---|---|
| Pump Outlet Flow (in gallons per minute) | Flow = $\frac{\text{rpm x Pump Displacement (cu. in./rev)}}{231}$ | $Q = \frac{n (d)}{231}$ |
| Pump Input Power (in horsepower required) | Horsepower = $\frac{\text{Flow Rate (gpm) x Press. (psi)}}{\text{1714 x Efficiency (Overall)}}$ | $HPIN = \frac{Q (P)}{1714 (Eff.)}$ |
| Pump Efficiency (overall in percent) | Efficiency Overall (%) = $\frac{\text{Output Horsepower}}{\text{Input Horsepower}}$ x 100 | $EFF._{OV} = \frac{HP_{OUT}}{HP_{IN}} \times 100$ |
| | Efficiency (%) = Volumetric Eff. x Mechanical Eff. | $EFF._{OV} = Eff._{VOL} \times Eff._{MECH}$ |
| Pump Efficiency volumetric in percent) | Vol. Eff. (%) = $\frac{\text{Actual Flow Rate Output (gpm)}}{\text{Theoretical Flow Rate Output (gpm)}}$ | $EFF._{VOL} = \frac{Q_{ACT}}{Q_{THEO}} \times 100$ |
| Pump Efficiency (mechanical in percent) | Mech. Eff. (%) = $\frac{\text{Theoretical Torque to Drive}}{\text{Actual Torque to Drive}}$ x 100 | $EFF._{MECH} = \frac{T_{THEO}}{T_{ACT}} \times 100$ |
| Pump Life ($B_{10}$ bearing life) | Bearing Life =Rated Hrs. x $\left(\frac{\text{Rated RPM}}{\text{New RPM}} \times \frac{\text{Rated PSI}}{\text{New PSI}}\right)^3$ | $B_{10}$ = Rated Hrs x $\left(\frac{RPM_R}{RPM_N} \times \frac{PSI_R}{PSI_N}\right)^3$ |

## FLUID POWER FORMULAS (Continued...)

### Thermal Formulas

One **British Thermal Unit (BTU)** is the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit. One horsepower = 2545 BTU/hr.

| Formula for: | Word Formula: | Letter Formula: |
|---|---|---|
| Reservoir Cooling Capacity (based on adequate air circulation) | Heat (BTU/hr.) = 2 x Temperature Difference Between Reservoir Walls & Air (F°) x Area of Reservoir (sq. in.) | BTU/hr. = 2.0 x $\Delta$ F° x A |
| Heat in Hydraulic Oil (approx.) (due to system inefficiency (SG = 0.89 - 0.92) | Heat (BTU/hr.) = Flow Rate (gpm) x 210 x Temperature Difference (F°) | BTU/hr. = Q x 210 x $\Delta$ F° |
| Heat in Fresh Water (approx.) | Heat (BTU/hr.) = Flow Rate (gpm) x 500 x Temperature Difference (F°) | BTU/hr. = Q x 500 x $\Delta$ F° |
| Heat in Hydraulic System Due to Unused Flow/Pressure | Heat (BTU/hr.) = Flow Rate (gpm) x 1.485 x Pressure Drop (psig) | BTU/hr. = Q x 1.485 x psig |

### Actuator Formulas

| Formula for: | Word Formula: | Letter Formula: |
|---|---|---|
| Cylinder Area (in square inches) | Area = $\pi$ x Radius$^2$ (inches) | A = 3.14 ($r^2$) |
| | Area = $\frac{\pi}{4}$ x Diameter$^2$ (inches) | A = $\frac{3.14}{4}$ ($D^2$) or A = 0.785 ($D^2$) |
| Cylinder Force (in pounds, push or pull) | Force = Pressure (psig) x Net Area (sq. in.) | F = psig x A or F = P (A) |
| Cylinder Velocity or Speed (in feet per second) | Velocity = $\frac{\text{231 x Flow Rate (gpm)}}{\text{12 x 60 x Net Area (sq. in.)}}$ | V = $\frac{231 (Q)}{720 (A)}$ V = $\frac{0.3208 (Q)}{A}$ (See Note 1 Below) |
| Cylinder Volume Capacity (in gallons of fluid) | Volume = $\frac{\pi \text{ x Radius}^2 \text{ (inches) x Stroke (inches)}}{231}$ | V = $\frac{3.14 (S)}{231}$ S = length of stroke |
| | Volume = $\frac{\text{Net Area (sq.in.) x Stroke (inches)}}{231}$ | V = $\frac{A (S)}{231}$ S = length of stroke |
| Cylinder Flow Rate (in gallons per minute) | Flow Rate = $\frac{\text{12 x 60 x Vel. (ft./sec.) x Net Area (sq. in.)}}{231}$ | Q = $\frac{720 (V)(A)}{231}$ Q = 3.117 (V)(A) (See Note 2 Below) |
| Fluid Motor Torque (in inch-pounds) | Torque = $\frac{\text{Pressure (psig) x Motor Displacement (cu. in.)}}{2\pi}$ | T = $\frac{psig (d)}{2 (\pi)}$ T = $\frac{P (d)}{2 (3.14)}$ |
| | Torque = $\frac{\text{Horsepower x 63025}}{\text{rpm}}$ | T = $\frac{63025 (HP)}{n}$ |
| | Torque = $\frac{\text{Flow rate (gpm) x Pressure (psig) x 36.77}}{\text{rpm}}$ | T = $\frac{Q (psig) (36.77}{n}$ T = $\frac{Q (P) (36.77)}{n}$ (See Note 3 Below) |
| Fluid Motor Torque/100 psig (in inch-pounds) | Torque/100 psig = $\frac{\text{Motor Displacement (cu. in./rev.)}}{0.0628}$ | T/100 psig = $\frac{d}{0.0628}$ |
| Fluid Motor Speed (in revolutions per minute) | Speed = $\frac{\text{231 x Flow Rate (gpm)}}{\text{Motor Displacement (cu. in./rev.)}}$ | n = $\frac{231 (Q)}{d}$ |
| Fluid Motor Power (in horsepower output) | Horsepower = $\frac{\text{Torque Output (inch-pounds)}}{63025}$ | HP = $\frac{T (n)}{63025}$ |

**Note 1:** Use 0.3208333 for greater accuracy. **Note 2:** Use 3.116883117 for greater accuracy. **Note 3:** Use 36.77071 for greater accuracy.