Inline Pump
Introduction
At Vickers Fluid Systems of Eaton Aerospace, we know how important it is to listen. And listening to our customers has shown us that quality, reliability and performance are the standards by which a company’s products should be judged.

Since 1921, Vickers has gained vast experience in the design and manufacture of hydraulic pumps. This is the heritage passed on to our aerospace inline pumps. Refinement of the basic inline pumping concept has brought the industry a truly superior series of high performance pumps.

This bulletin is a technical description of the design and performance of the inline pump series. Also described is the variety of controls available.

Intended as a convenient reference for the system designer, the bulletin provides the necessary information to predict unit performance and to select the proper type of control.

Features and Benefits
Low Cost
Lower cost per horsepower than other pumps of comparable design.

Durable by Design
Generous bearing surfaces and reduced contact pressures result in a more durable pump.

More Horsepower
Lighter weight and higher speed capability provide a much higher horsepower-to-weight ratio.

Rapid Response
Step changes from peak demands to minimal flow can be accomplished within 50 milliseconds.

Smooth Operation
Low pressure pulsations minimize system disturbances and improve system life.

Economical Overhaul
Overhauls are economical because of the minimum number of parts and the simplified rotating group.

Reliability
Simplicity and conservative design parameters assure high reliability and the ability to tolerate off-design conditions.

Low Power Loss
Compact rotating group and small anti-friction bearing diameters result in minimum power loss.

Identification Code
Vickers Inline Series Pumps are identified by model numbers that indicate the displacement and design release number. (see diagram below.)
Basic Operations

Figure 1
The Vickers inline pump series is a family of positive-displacement, axial-position pumps designed to operate at either fixed or variable displacement. Figure 1 is a cross-section of the typical inline pump.

As the drive shaft rotates, it causes the positions to reciprocate within the cylinder block bores. The piston shoes are held against a bearing surface by compression force during the discharge stroke and by the shoe hold-down plate and retainer during the intake stroke. The bearing surface is held at an angle to the drive shaft axis of rotation by the yoke (Figure 2).

**Intake Stroke**

As each piston shoe follows the shoe bearing plate away from the valve plate, the piston is withdrawn from the cylinder block. During this intake stroke of the piston, fluid is supplied to its cylinder block bore through the valve plate inlet port.

**Discharge Stroke**

Further rotation of the drive shaft causes the piston shoe to follow the shoe bearing plate toward the valve plate. This is the discharge stroke of the piston and fluid is expelled from its cylinder bore through the outlet port of the valve plate.

Figure 3 illustrates the manner in which piston stroke is controlled by the yoke angle. Displacement variations that respond to pressure changes to vary the yoke angle are described on page 14.
Drive Shaft and Bearings
The drive shaft is a simple, single-piece design held in accurate alignment by two anti-friction bearings.

Bearing Size Minimized
The shaft is supported by radial bearings at each end. Since radial loads (due to position forces on the cylinder block) are distributed between the two widely spaced shaft bearings (front and rear, Figure 1), bearing size is minimized, reducing friction losses. This is especially important in high-speed applications where the fluid disturbance and power loss in submerged bearings increases appreciably with bearing rpm and time.

Self-Aligning Spline
The spline that drives the cylinder block is a major diameter fit, crowned slightly to provide cylinder block self-alignment.

Cylinder Block
Optimum hydraulic pressure balance between the cylinder block and the valve plate (Figure 4), ensures proper hold-down, minimizes internal friction and reduces torque losses.

No Support Bearing Needed
Radial loads resulting from piston reactions are carried to the drive shaft through the drive spline, eliminating the need for a support bearing on the cylinder block. Consequently, losses associated with such a large bearing are eliminated, in addition, this kind of load support provides optimum alignment conditions between the cylinder block, valve plate and drive shaft, since fewer mating surfaces are required to establish the proper geometric relationships of these components.

The effective center of the cylinder block spline is located near the rotation plane of the piston shoes to minimize movement action on the cylinder block.

Pistons and Shoes
Stress analysis methods combined with verification tests have been used to arrive at an optimum piston-cylinder block design. The Vickers design places special emphasis on shoe design, since this is a critical link in the efficiency, life and reliability of the in-line pump design.

Precise Pressure Balance
Precise pressure balance versus speed and load capability have been achieved without sacrificing efficiency and life. Thrust loads on the piston shoes are controlled by pressure balance to a point where the resultant loads can be adequately supported by the fluid film under their outer lands. (Figure 5).

Low Cylinder Wear
Since minimum piston engagement in the cylinder block bore is designed to be approximately two diameters, reaction forces between the pistons and cylinder block are minimized. This reduces bore wear so that internal leakage remains nearly constant with time.

Yoke
Extensive studies have been performed on the yoke to provide a design with an optimum deflection-to-weight relationship. The yoke pivot centerline has been positioned to allow optimum design of the actuating piston and control spring (Figure 6).

Shoe Bearing Plate
The use of a shoe bearing plate allows simplified yoke design, accurate lapping of the bearing surface and optimum material selection.

Piston Shoe Hold-Down Plate
During the inlet stroke, the piston/shoe subassembly requires force to pull it out of the cylinder block bore; this force is supplied by the hold-down plate. It is driven and guided by contact with the shoe necks, and is held in place axially by sliding contact with the hold-down plate retainer (Figure 7).

Shoe Hold-Down Plate Retainer
The retainer provides positive retention of the shoe hold-down plate during the intake stroke.

This retainer is secured to the yoke by screws to ensure optimum support of the shoe hold-down plate and minimum retainer loading in the areas where shoe lift forces (intake stroke) are highest. The retainer design and arrangement improve the high-speed capability of the pump.

Valve Plate
Valve plate kidney port slots have been designed to provide minimum power loss and pressure pulsation throughout delivery range. This is accomplished by designing the valve plate porting and yoke geometry...
so the piston chamber pressure is raised to system pressure before opening to the outlet port (precompression) and the piston chamber pressure is lowered to inlet pressure before opening to the inlet port (decompression).

**Decompression Phase**
During the decompression phase, the energy stored in the unswept volume of fluid at system pressure is returned to the system by motoring, rather than being lost by throttling to inlet pressure.

**Shaft Seal**
Vickers Aerospace Products has developed a shaft seal configuration especially designed for use in all aerospace pumps. Benefits of this design are longer life, greater reliability and lower overall pump costs. This face type shaft seal ([Figure 8](#)) is not a “package” design, but is a combination of simple elements. The elements most subject to wear can be repaired at overhaul and the sealing surface can be lapped, providing an inexpensive procedure for obtaining new seal performance.

**Rotating Sealing Element**
The simplified seal is made of high quality material such as bearing grade bronze or carbon. The major difference from other seals is that the sealing element rotates with the shaft while the heavier mating ring is stationary in the housing. The seal assembly is driven by two tabs on the retainer that engage with the pump drive shaft. Static sealing around the circumference of the drive shaft is accomplished with the elastomeric grommet, shown in [Figure 8](#), which is held in contact with the shaft by a garter spring. Dynamic sealing is effected by forcing the rotating element against the stationary mating ring. Constant force is provided by the wave washer; hydrostatic balance close to 100% assures that its operation is unaffected by a wide variation in case pressure.

**Spring Assures Contact**
Since the elastomeric grommet is held in contact with both the shaft and the sealing element by means of mechanical springs, the effectiveness of the seal does not depend on the elastic properties of the grommet. Efficient sealing is not disturbed by the effects of either age or temperature. Also, because of the low mass of the elements mounted on the shaft, the seal maintains full contact, even in environments with high vibration levels.

**External Sealing**
There are few external seals required in the standard design, and in newer designs none are subjected to high pressure. As a result, the pump has fewer potential external leakage paths. The structural integrity of the single piece housing has been proven by completing a 4-Life fatigue test (approx. 1,440,000 cycles). The design simplifies maintenance tasks, reduces pkg weight, and minimizes envelope requirements.

**Materials**
Materials are chosen for high strength, long life and optimum performance. Materials used in the standard design include bronze, and steel for the rotating group and cast aluminum for the housing and mounting flange. The valve block may be steel, titanium or aluminum. High performance pumps generally use wrought or forged housing and mounting flanges.
General Application Advantages
The hydraulic pump that supplies power to move the various loads of modern aircraft and defense vehicles is a critical component. Determining the best pump for the job involves three basic considerations:

- Reliability
- Performance
- Economy

**Reliability**
The safety of an airplane and its occupants, or the effectiveness of a missile, depends on the proper functioning of the hydraulic system. Realizing that product reliability is the most important factor to the consumer, Vickers sets this as the essential, guiding objective in all design, manufacturing and testing operations. This objective is further supported by the Product Support Department of Vickers Aerospace through field and overhaul work and statistical studies of pumps in use.

**Engineering Cooperation with Customers**
Close and active engineering cooperation with the customer is aimed at providing the proper hydraulic circuit design necessary to obtain the most reliable pump performance. The wide use of Vickers pumps in all kinds of military and commercial aircraft and the record these pumps have established over the past years are evidence of the emphasis Vickers places on reliability.

**Performance**
Certain standards and details of performance are required by hydraulic system designers. How well a pump meets or exceeds these specifications is obviously important. Vickers inline pump performance is defined in these five characteristics:

1) **Pressure Regulation**
   System pressure is automatically held within a given range for all flows from zero to full flow. The regulation range may be chosen to best fit a particular requirement and may be as small as 3% of related pressure. The regulation compensates for changes in load demand, temperature and variation in driving speed.

2) **Stability**
   Recovery from step loads and load perturbations is rapid, and pressure overshoots are not excessive when proper circuit design is employed.

3) **Temperature Range**
   Vickers inline pumps are designed to provide optimum performance in a Type II system as defined by MIL-H-5440G, -65 °F to +275 °F; (-54 °C to 135 °C) fluid temperature. Special modifications for operation at higher temperatures can be provided.

4) **Efficiency**
   Typical value of overall efficiency of Vickers inline pumps at rated operating conditions is 88%. Volumetric efficiency at rated operating conditions exceeds 96%. These efficiencies are maintained near these values for long operating durations.

5) **Weight**
   Vickers inline pumps have a high power-to-weight ratio. For example, the PV3-115 pump has a ratio of 4.5 hydraulic output horsepower per pound (continuous rating), that rises to 5.4 hp/lb for overspeed operation.

**Economy**
Economy involves original purchase price and operating costs. Operating costs in turn depend upon overhaul time, overhaul costs and reliability or unscheduled removal rate.

**Life**
Vickers inline pumps have demonstrated long life capability both in qualification testing and in service.

Nearly all models have completed the qualification requirements of MIL-P-19692. The long time durability has been further demonstrated in a wide variety of applications and in other laboratory testing. One example is the extended endurance test of a PV3-115 pump, following qualification to MIL-P-19692. The pump was cycled between 10% and 90% flow at rated speed of 6000 rpm and at continuous Type II system temperatures. After 15,000 hours, the pump was still capable of meeting new pump performance requirements.

A pilot of performance versus time is shown in Figure 9.

**Service Cost**
The uncomplicated design of the Vickers inline pump provides a unit that has low overhaul cost. The man-hours required for disassembly, maintenance operations, reassembly and retest are minimal. Parts costs are low and a minimum of inexpensive overhaul tools are needed.

**Reliability**
The low wear rates, together with the added structural margin provided in the design, result in the ability of Vickers inline pumps to withstand frequent off design operating conditions of temperature, speed, pressure, etc., without noticeable damage or performance deterioration.

**Flexibility of Installation**
An additional consideration when choosing a hydraulic pump is flexibility of installation. Vickers inline pumps have standard mounting arrangements (QAD or bolt type) with a variety of inlet and outlet configurations available. The pressure control has been designed as an integral component, reducing space and weight. In addition, the pump design is ideally suited to manifold type inlet-outlet porting and the use of quick disconnect type mounting.

**Thru-Shaft Available**
For systems where mounting pad availability is at a minimum, Vickers inline pumps can be provided with a thru-shaft and special end cap. This feature allows additional accessories to be mounted to and driven by the Vickers pump, reducing engine mounting pad requirements. Custom designing provides the package configuration that best fits a particular application.

![Efficiency & Horsepower Versus Time](image.png)
Operational Characteristics
Life
The life of a hydraulic pump depends to a great extent upon the operating temperature, the drive speed, and the pressure and flow extremes to which it is subjected. In addition, cleanliness of the hydraulic fluid and sufficient inlet pressure are very important for assuring long life.

Normal Period of Operation
The normal period of pump operation between overhauls in aircraft application may range from 1000 to more than 15,000 hours of actual pump running time. For applications involving operation at fluid temperatures above 275° F (135° C), the overhaul period may be shorter.

Efficiency
Overhaul efficiency varies somewhat with drive speed and outlet pressure. This value usually exceeds 85% and may be more than 90% at rated pressure and moderate speeds. During idle, the pump provides only enough flow to satisfy its own internal leakage. In the case of the EDV pump, in which leakage is circulated at low pressure instead of rated pressure during idle periods, the power loss is further reduced.

Effects of Inlet Pressure and Temperature Extremes
Inlet pressure will adversely affect efficiency when it drops below the critical inlet pressure and causes cavitation. Extremes of temperature also will somewhat reduce efficiency. At high temperatures the fluid viscosity is lowered and leakage increases. Low temperature increases fluid viscosity and increases windage loss. However, the oil quickly warms up as the pump operates. The effect of temperature is minimal except at very high or very low extremes.

Pressure
Most Vickers pressure-compensated pumps are designed for 3000 psi (207 bar) maximum outlet pressure at zero flow and automatically limit the steady state pressure to this value. During transient periods, pressure surges may instantaneously exceed this by about one-third or less. Full flow maximum pressures are usually 2900 psi (200 bar). Higher pressure pumps, especially 4000 psi, 5000 psi and 8000 psi (276 bar, 345 bar and 552 bar) are being used in an increasing number of applications which are at the forefront of hydraulic technology. Dual range pumps provide two separate pressure ranges. (See Controls section, page 14, for description of compensators).

Temperature
Temperature limits for Type II systems per MIL-H-5440G continuous full-life operation are -65° F to +275° F; (-54° C to 135° C). Vickers standard inline units can be operated within these limits.

Driving Speeds
The driving speed recommended for a particular size pump is based on pump life considerations.

The maximum speed values shown represent an optimum compromise between long pump life (lower speeds) and maximum power output (higher speeds) and may be exceeded under certain conditions.
Transient Response

When a sudden load change causes system pressure to exceed the lowest value of the pump's pressure regulation range, or when a sudden load change occurs while system pressure is in the regulation range, the pump must rapidly adjust the output flow to meet the new load condition. This requires that the yoke be repositioned rapidly. The time elapsed during the pressure change is a measure of the dynamic response of the pump and its associated circuit and load, and depends upon the following factors:

1) The Speed at Which the Yoke Angle is Changed
This is a characteristic of the pump. For example, small yoke cylinder area and high control value gain contribute toward rapid yoke motion for both increasing and decreasing displacements.

2) Pump Driving Speed
Pump speed affects time response because the rate of flow is proportional not only to yoke angle, but also to pump driving speed. When pump speed is high, the output flow will be high and pressure build-up time will be short.

3) The Compliance of the Circuit
Circuit compliance is determined by the volume of oil under compression, the bulk modulus of the oil (varies with temperature and kind of fluid), elasticity of the lines and whether an accumulator is used. Small oil volume, high-bulk modulus, low-expansion lines and absence of an accumulator contribute to fast pump response. A given flow change will result in a rapid pressure change, causing the pump to regulate rapidly.

4) The Nature of the Load
The nature of the load influences how rapidly a change in load resistance or flow-control is reflected in a change in system pressure. Since changes in system pressure cause the pump to regulate, the nature of the load is a factor in the speed of pump response. In general, if the load has a high resonant frequency (low inertia, high spring rate, low damping), response of the pump to changes in load resistance or flow control valve settings will be faster than in the case of a low-frequency load system.

5) The Use of an Accumulator
Although the use of an accumulator decreases the speed of pump yoke response, the speed of load response is the important factor. With an accumulator in the system, flow will be supplied rapidly to the load when needed, thereby contributing to fast load response. Although the pump yoke repositions more slowly because of the accumulator, the load does not sense the reduced yoke response since the pump's function is temporarily taken over by the accumulator.

Minimum Accumulator Size Desirable
In order to keep the accumulator size at a minimum, the pump response should be as fast as possible. The size of the accumulator determines the time during which it can supply a given flow at a steady pressure. If a pump requires a relatively short time to adjust flow to a new rate, a small accumulator may be used or the accumulator may be eliminated.

Inlet Pressurization

When a pump is driven at high speed with insufficient pressure, cavitation may result due to the vacuum created during the intake stroke, and its sudden collapse. Also, when flow demand is suddenly increased, the inertia of the fluid in the inlet line can reduce the pressure at the pump inlet below the critical value and produce cavitation and water hammer. This has the tendency to tear particles of metal from the affected surfaces with resulting erosion. The amount of inlet pressurization required to prevent cavitation increases with the pump operating speed, as shown in the chart on page 59.

The chart shows recommended inlet pressure for long life when operated in MIL-H-5606 or MIL-H-83282 fluid. For operation with phosphate ester based fluids, inlet pressures higher than those shown are recommended. Short time operation below the recommended values can be accommodated. Inlet pressure limitations of specific installations should be discussed with the Vickers application engineer. Vickers pumps can be supplied with various control arrangements, each particularly advantageous for certain individual application requirements. The pressure regulation characteristics of a pump are determined by the type of control employed.
Types of Controls
Standard Types
The standard control types are the “flat cut-off” and “differential cut-off” compensators.

Flat Cut-Off Type
This control provides nearly constant pressure through the entire flow range (Figure 10) by limiting the system pressure increase to about 3% from full flow to zero flow. It has the advantages of nearly constant output pressure, high power output, minimum size and weight and fast response.

Description
A step-by-step description will give a better understanding of what happens in the control circuit of the flat cut-off type pump. For simplicity, an ideal pump is described (one that maintains a constant load pressure for all values of flow within the capacity of the pump, except when there is insufficient load to build up the pressure). Actually, as mentioned above, pressure rises about 3% as flow decreases from maximum to zero; this is caused by leakage from the control circuit and is important for pump stability.

Refer to Figure 11. Assume that initially there is no resistance to flow. This will give zero system pressure and maximum flow (yoke to maximum angle).

Increasing Load
As the load (resistance to flow) is increased, the pressure rises and flow remains maximum until the pressure reaches the pressure setting of the compensator valve spring. (A pressure setting or rated pressure of 3000 psi (207 bar) is assumed here; however the same description of operation also applies to pumps of higher rated pressures.) Therefore, 3000 psi (207 bar) system pressure is just sufficient to center the compensator valve spool.

Flow Proportional to Compensator Valve Opening
As further load increase causes the system pressure to exceed 3000 psi (207 bar), the compensator valve spool is moved downward (as shown in Figure 11) and flow from the Ps line is metered to the yoke actuating piston. This flow increases with compensator valve opening and therefore with system pressure increase above 3000 psi (207 bar).

Control Flow Integrated
The yoke actuating piston integrates the control flow; thus, the velocity of the piston (and yoke) is approximately proportional to the position of the compensator valve. The rate of system flow reduction varies with pressure above compensator pressure setting.

The yoke angle is reduced until the flow is just sufficient to give the set system pressure.

At this point the yoke is in its new position and the compensator valve spool is centered.

Decreased Load
If the load is decreased, the system pressure is temporarily reduced and the compensator valve spool will be displaced upward, opening the yoke actuating piston to case pressure.

The yoke actuating spring will cause the yoke angle to increase until the flow is just sufficient to again give the set system pressure. The yoke is in its new position and the compensator valve spool is centered.

Pump Leakage Flow Maintained
If the main line flow is completely blocked, the yoke will be moved to near center, with only enough displacement to provide the pump leakage flow. The system pressure will be maintained and the pump will provide flow when required.

Ideal Cause Was Assumed
In the preceding explanation, it was assumed that the pump has ideal regulation; that is, no increase in supply pressure as flow decreased from maximum to zero. As stated, this is not obtainable in practice because of leakage. Any leakage out of the control circuit requires an opening of the compensator valve to replace the leakage and maintain yoke position. An increase in supply pressure is required to produce the compensator valve displacement. Also, since a smaller yoke angle requires higher control pressure (compression of the yoke spring is greater), the leakage increases which, in turn, requires a greater compensator valve displacement and, therefore, a greater increase in supply pressure. A certain amount of control circuit leakage is desirable, since it limits the gain and helps to assure stable operation.
Factors Influencing Regulation
Other design constants influence the regulation, but leakage is the reason that any regulation range exists. Factors tending to make the static regulation curve more vertical (higher gain) include low spring rates (compensator valve and yoke), low leakage, large areas (compensator valve and yoke actuating piston) and high yoke centering force.

Differential Cut-Off Type
This type of control provides a somewhat greater proportional decrease in pressure as flow increases from zero to maximum (Figure 13). A typical pressure regulation range is 20% of rated pressure, e.g. 600 psi (41 bar) for a 3000 psi (207 bar) setting. Advantages include proper load division in systems employing two or more pumps in parallel, minimum transient pressure surges and a high degree of stability.

Description
The following step-by-step description explains how the differential type pump operates to automatically limit system pressure and to provide a proportional decrease in system pressure from the maximum to a given pressure, as flow increases from zero to maximum. The regulation range assumed is 2400 to 3000 psi (165 to 207 bar). The range can be altered to obtain optimum characteristics for a particular application.

System Pressure Less Than 2400 psi (165 bar)
Refer to Figure 12. The compensator valve spool will remain displaced upward at system pressure Ps less than 2400 psi (165 bar) due to the preload of the compensator valve spring.

With the compensator valve spool displaced upward, the yoke actuating piston is ported to case. Therefore, the yoke actuating spring holds the yoke at the maximum angle (maximum flow position).

System Pressure More Than 2400 psi (165 bar)
As system pressure (Ps) exceeds 2400 psi (165 bar) (due to increase in load), the upper metering orifice of the compensator valve opens and the bottom orifice closes.

Flow from the control pressure (Pc) line enters the yoke actuating piston, compressing the yoke spring and decreasing the yoke angle. As the spring is compressed, the control pressure, Pc, rises.

When the increase of Pc equals the amount by which Ps exceeds 2400 psi (165 bar), the valve will be centered to keep the yoke in its new position.

Flow Reduction Proportional to Ps above 2400 psi (165 bar)
The yoke position is proportional to yoke spring compression and to the increase in Pc above case pressure. Therefore, reduction of yoke angle and flow are proportional to the increase in Ps above 2400 psi (165 bar).

Decreasing System Pressure
When Ps is between 2400 and 3000 psi (165 to 207 bar), any decrease in Ps causes the compensator valve spool to be displaced upward and oil in the yoke actuating piston is ported to case, until Pc decreases enough to allow the spool to return to the centered position again. At this point the yoke remains in its increased-flow position. Therefore, the proportional relationship applies also to increase of flow with respect to decrease of pressure.

Summary
In summary, pump output flow is inversely proportional to system pressure excess above 2400 psi (165 bar), as the pressure varies within the regulation range 2400 to 3000 psi (165 to 207 bar). Flow is maximum for 2400 psi (165 bar) and less. Flow is zero for 3000 psi (207 bar), except for internal leakage flow to supply lubrication. System pressure cannot exceed 3000 psi (207 bar) (except for small transient surges during sudden load changes) and the pump will rapidly provide flow when required.
Electrically Depressurized Variable Pump with Blocking Valve

This control further reduces the power loss when the pump is on standby by reducing system pressure to a lower value – example, 600 to 1000 psi (41 to 69 bar) by means of an electrical signal.

**Description**

The solenoid valve (Figure 14) is normally de-energized. In this position the pump pressure compensator operates normally to provide full outlet pressure. The blocking valve is maintained in the open position (as shown) for outlet pressure above e.g. 400 psi (28 bar).

Energizing the solenoid will port outlet pressure to a depressurizing piston that moves the compensator valve spool down. This connects pump outlet pressure directly to the yoke actuating piston. Since the control pressure required to hold the yoke in zero stroke position is only that necessary to overcome the yoke springs, e.g. 1000 psi (69 bar) for a 3000 psi (207 bar) pump, the pump is depressurized to a 1000 psi (69 bar) level. This results in a lower input torque to the unit and a lower power loss.

High pressure is also ported to the spring chamber of the blocking valve when the solenoid is energized. This hydraulically balances the blocking valve piston and allows the spring to move the piston down to the closed position. The pump is thereby isolated from the system.

The EDV/blocking valve feature has three primary uses:

1. It enables individual pumps to be taken off the line during system checkout and facilitates troubleshooting system malfunctions.
2. It enables a disabled system to be shut down in flight to avoid additional system damage or to prevent loss of system fluid.
3. If the pump is depressurized during an engine start, the peak torque of the pump during startup is reduced. This is particularly important if the pump is to be driven by an auxiliary power unit or air turbine. It may be impossible to start the power unit or air turbine unless this feature (or a flow bypass) is incorporated.

**EDV Used with Either Type Cut-Off**

Although Figure 14 shows a flat cut-off compensator, the EDV feature also may be used with the differential type. It has no effect on the pump’s regulation characteristics except when the solenoid is energized. Since the driving speed of the pump is continuous and unaffected by depressurization, the outlet pressure builds up very rapidly when the solenoid circuit is opened.
EDV Feature without Blocking Valve
If the hydraulic system contains a check valve close to the pump outlet port and normal system pressure is maintained during periods of pump depressurization, the blocking valve may be eliminated. This results in savings of cost, envelope and weight.

Dual Range Control
This control provides two separate output pressure ranges (Figure 16). Selection is made by means of an external signal (electrical, mechanical or hydraulic). It permits the use of smaller hydraulic components in systems that require short, high-power demand periods. The dual range control also reduces friction and leakage losses during low-power demand periods. Extended pump life is still another benefit.

Preload Determines System Pressure
As previously explained, the preload on the pressure compensator spring determines the system pressure at which the pump begins regulating. Compressing the spring (increasing preload) causes the pump to regulate at a higher pressure, and conversely, relaxing the spring (decreasing preload) provides regulation at a lower pressure.

No Pressure at “E”
As seen in Figure 15, the compensator spring is in the extended position (low preload) when there is no pilot pressure at “E”. This means that a relatively low system pressure is required to overcome the spring preload force and the pump will regulate at its lower range.

Pressure at “E”
When pilot pressure acts at “E”, the lower end of the spring is forced upward until a stop is reached. The position of this stop determines the amount of additional spring preload and thus determines the system pressure at which regulation in the higher range occurs. The pressure at “E” must be sufficient to hold the piston firmly against the stop. This pressure can be provided from an external source, or it can be supplied from the pump output through a solenoid-controlled valve.
Constant Horsepower Control

This control limits power output to a given value for a given pump speed and maintains it at a nearly constant level within a given flow range (Figure 18). It has the advantage of keeping pump power source requirements at a minimum.

The purpose of the constant horsepower is to limit the maximum pump power by beginning flow reduction at a given intermediate system pressure (point A) and to cause the pump to reach near rated pressure at a given intermediate flow (point B).

Yoke Spring Preloaded

Refer to Figure 17, System pressure, Ps acts at all times on area “H”. The yoke spring is preloaded with a force equal to area “H” multiplied by the pressure at which it is desired to have flow reduction begin (refer to point A in Figure 18). As system pressure reaches this value, the preload is overcome and further pressure increase will produce a proportional decrease in flow. The rate of the spring and the area “H” will determine the slope from “A” to “B” (Figure 18).

Provisions For Special Requirements

1) Integral Boost Stage

If your application requires the pump to operate at unusually low inlet pressure, a boost stage can be added. This is a centrifugal pump that permits operation at inlet pressure well below atmospheric. A typical application is an aircraft main engine pump with a minimum inlet requirement of five psia.

The impeller adds to the length and weight of the pump, but results in overall system weight reduction and increased reliability in applications where inlet pressure may be quite low.

2) Startup Bypass Valve

A simple, reliable, low-cost valve may be incorporated in the pump valve block to bypass flow and minimize torque during startup. This is sometimes important to reduce cranking torque of an engine or startup current of a motorpump. The flow is bypassed to inlet, at very low pressure, until a given flow is reached; the bypass valve then closes and remains closed until system pressure is reduced to near zero, at which time it opens and is ready for the next start. Operation is completely automatic and self-contained. There are no electrical connections and torque during startup is even less than that of the EDV feature in the depressurized mode.

3) Pressure Pulsation

A pulsation damping chamber (attenuator) can be incorporated in the valve block to provide reduction of the outlet pressure ripple. This adds to the envelope and weight of the pump, but is justified for applications where lower pressure pulsation levels are essential and the required compressibility is not provided by the load circuit.
**Servo Control**

Servo Control of variable displacement pumps is defined as a control where the output is proportional to the input signal. The output may be pressure or flow. The control valve usually is an electrohydraulic valve but could be direct drive, fluid, pneumatic or other type valve.

There are two methods of servo signal control for variable pumps and motors. The first is used where only a variable pressure setting is desired. This provides a variable delivery component with a variable pressure setting determined by the control signal.

The second method of servo control is to vary the displacement of the pump to provide flow proportional to the control signal. This type of control is generally used with a feedback network to control velocity or position of an actuator, or speed of a rotating device like a constant speed generator.

**Intelligent Control™**

Intelligent Control is a trademark for a Vickers controller that can be used with variable pumps or motors. This controller can furnish all the characteristics of the controls mentioned in this brochure plus additional control capability such as position control. In addition, the Intelligent Control has more accuracy and flexibility. One control function can be commanded with a single control module. In addition to various control functions, Vickers Intelligent Control has the capability to provide diagnostics, built in test, health monitoring and integrity testing. The pressure and power control mode characteristics in Figure 19 show the precise control that is available with the Intelligent Control. The control can also be programmed to limit transient cavitation, reduce starting torque requirements and control multiple components. The Intelligent Control has been designed to be central to a hydraulic energy management system that will minimize system heat rejection and reduce the total system weight.

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**Figure 18**

![Diagram](image18.png)

**Figure 19**

![Diagram](image19.png)
Application Performance and Installation Data
3000 psi (207 bar)
3000 psi (207 bar) has been the most commonly used system pressure for aerospace applications. Vickers has created variable displacement pumps to meet the full range of 3000 psi (207 bar) power applications. Virtually all Vickers pumps have been qualified and sufficient testing has been performed to generate the typical performance curves shown in this section. The table to the right presents the basic characteristics for Vickers 3000 psi (207 bar) pumps.

4000 psi (276 bar) and Higher
Higher system pressures have been selected for military applications and have been in use for several years. These applications normally require additional Vickers engineering involvement.

### Basic Model Characteristics of 3000, 4000 and 5000 psi Pumps

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
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</thead>
<tbody>
<tr>
<td>PV3-003</td>
<td>0.030</td>
<td>0.5</td>
<td>18,000</td>
<td>22,500</td>
<td>2.38</td>
<td>9.00</td>
<td>1.7</td>
<td>0.8</td>
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<tr>
<td>PV3-006</td>
<td>0.061</td>
<td>1.0</td>
<td>15,000</td>
<td>18,750</td>
<td>3.96</td>
<td>15.00</td>
<td>2.4</td>
<td>1.1</td>
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<tr>
<td>PV3-008</td>
<td>0.08</td>
<td>1.31</td>
<td>13,500</td>
<td>16,800</td>
<td>4.68</td>
<td>17.72</td>
<td>3.4</td>
<td>1.6</td>
</tr>
<tr>
<td>PV3-011</td>
<td>0.11</td>
<td>1.803</td>
<td>12,500</td>
<td>15,600</td>
<td>5.95</td>
<td>22.53</td>
<td>3.7</td>
<td>1.8</td>
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<tr>
<td>PV3-019</td>
<td>0.192</td>
<td>3.15</td>
<td>12,100</td>
<td>15,100</td>
<td>10.07</td>
<td>38.12</td>
<td>3.7</td>
<td>1.7</td>
</tr>
<tr>
<td>PV3-022</td>
<td>0.22</td>
<td>3.605</td>
<td>10,000</td>
<td>12,500</td>
<td>9.52</td>
<td>36.05</td>
<td>4.6</td>
<td>2.1</td>
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<tr>
<td>PV3-032</td>
<td>0.32</td>
<td>5.244</td>
<td>9,000</td>
<td>11,250</td>
<td>12.47</td>
<td>47.19</td>
<td>6.0</td>
<td>2.7</td>
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<tr>
<td>PV3-044</td>
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<td>7.210</td>
<td>8,000</td>
<td>10,000</td>
<td>15.24</td>
<td>57.68</td>
<td>7.1</td>
<td>3.2</td>
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<tr>
<td>PV3-049</td>
<td>0.488</td>
<td>8.0</td>
<td>8,800</td>
<td>11,100</td>
<td>18.60</td>
<td>70.40</td>
<td>6.4</td>
<td>2.9</td>
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<tr>
<td>PV3-056</td>
<td>0.56</td>
<td>9.177</td>
<td>8,200</td>
<td>10,250</td>
<td>19.88</td>
<td>75.25</td>
<td>7.1</td>
<td>3.2</td>
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<tr>
<td>PV3-075</td>
<td>0.75</td>
<td>12.29</td>
<td>7,000</td>
<td>8,750</td>
<td>22.73</td>
<td>86.03</td>
<td>8.9</td>
<td>4.0</td>
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<tr>
<td>PV3-115</td>
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<td>18.85</td>
<td>6,600</td>
<td>8,250</td>
<td>32.86</td>
<td>124.4</td>
<td>11.5</td>
<td>5.2</td>
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<td>PV3-150</td>
<td>1.50</td>
<td>24.58</td>
<td>6,000</td>
<td>7,500</td>
<td>38.96</td>
<td>147.5</td>
<td>15.0</td>
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<tr>
<td>PV3-205</td>
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<td>5,900</td>
<td>7,400</td>
<td>45.97</td>
<td>174.0</td>
<td>19.8</td>
<td>9.0</td>
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<td>PV3-240</td>
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<td>39.33</td>
<td>5,300</td>
<td>6,600</td>
<td>55.06</td>
<td>208.4</td>
<td>22.5</td>
<td>10.2</td>
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<td>PV3-300</td>
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<td>49.16</td>
<td>5,000</td>
<td>6,250</td>
<td>64.94</td>
<td>245.8</td>
<td>28.0</td>
<td>12.7</td>
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<tr>
<td>PV3-375</td>
<td>3.75</td>
<td>61.45</td>
<td>4,800</td>
<td>6,000</td>
<td>77.92</td>
<td>295.0</td>
<td>34.5</td>
<td>15.6</td>
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<td>PV3-400</td>
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<td>65.55</td>
<td>4,400</td>
<td>5,500</td>
<td>76.19</td>
<td>288.4</td>
<td>33.5</td>
<td>15.3</td>
</tr>
<tr>
<td>PV3-488</td>
<td>4.9</td>
<td>80.30</td>
<td>4,100</td>
<td>5,125</td>
<td>86.97</td>
<td>329.2</td>
<td>46.3</td>
<td>21.0</td>
</tr>
</tbody>
</table>

**American Standard**

1) Flow \[ Q = \frac{\text{in}^3/\text{rev} \times \text{rpm}}{231} \] (gpm)

2) Torque \[ T = \frac{\text{in}^3/\text{rev} \times \text{psid}}{2\pi} \] (lb-in)

3) Power (Shaft) \[ \text{hp} = \frac{\text{Torque} \times \text{rpm}}{63,025} \] (hp)

3a) Power (hydraulic) \[ \text{hp} = \frac{\text{gpm} \times \text{psid}}{1714} \] (hp)

**Metric Units**

1) Flow \[ Q = \frac{\text{mL/rev} \times \text{rpm}}{1000} \] (L/min)

2) Torque \[ T = \frac{\text{mL/rev} \times \text{bar} \times \pi}{20\pi} \] (N•m)

3) Power (Shaft) \[ W = \frac{\pi \times \text{Torque} \times \text{rpm}}{30} \] (watt)

3a) Power (hydraulic) \[ W = \frac{5/3 \times \text{L/min} \times \text{bar} \times \pi}{5/3} \] (watt)

KW = \[ \frac{\text{L/sec} \times \text{MN/m}^2}{1000} \] (kilowatt)

* KPa/100 or 10 MN/m² may be used in place of bar

Note: The above equations give theoretical values: that is, 100% efficiency is assumed. Actual pump outlet flow is less by the amount of pump internal leakage, and input torque to the pump is greater by the amount of torque loss.
Pump Efficiencies at Normal Recommended Speeds. 3000, 4000 and 5000 psi Pressures.

<table>
<thead>
<tr>
<th>Pressure (psi)</th>
<th>Volumetric Efficiency</th>
<th>Torque Efficiency</th>
<th>Overall Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>3000 psi</td>
<td>0.96</td>
<td>0.92</td>
<td>0.885</td>
</tr>
<tr>
<td>4000 psi</td>
<td>0.95</td>
<td>0.94</td>
<td>0.89</td>
</tr>
<tr>
<td>5000 psi</td>
<td>0.93</td>
<td>0.925</td>
<td>0.86</td>
</tr>
</tbody>
</table>

Performance Calculations

The following efficiencies are given as a guide for obtaining preliminary flow, torque and power performance values for rated speed and pressure. Since flow and torque losses also vary with fluid temperature, type of fluid and other operating conditions, we recommend that a Vickers application engineer be consulted before finalizing the design parameters. This is especially true if outlet pressure or pump driving speed for the application is different from rated pressure (3000 psi) and normal speeds listed on pages 11 and 21. The curves on pages 24 through 56 give performance values for most of the models at three different speeds.

A generalized curve of recommended inlet pressures for various pump speeds is included on page 59.

FOR EXAMPLE

for a PV3-075 pump, using typical efficiencies:

Flow
To obtain the value of pump outlet flow, calculate the theoretical value using pump equation number 1) from page 21, and multiply that flow by the applicable volumetric efficiency. Subtract outlet flow from theoretical flow to obtain flow loss.

\[
Q_{\text{theo}} = \frac{75 \text{ cu in/rev}}{231} \times 7000 \text{ rpm} = 22.7 \text{ gpm}
\]

\[
Q_{\text{outlet}} = (22.7 \text{ gpm}) \times 0.96 = 21.8 \text{ gpm}
\]

\[
Q_{\text{loss}} = 22.7 - 21.8 = 0.9 \text{ gpm}
\]

(Multiplying by .96 Volumetric Efficiency)

Torque
To obtain the value of pump input torque, calculate the theoretical value using pump equation number 2) from page 21, divide that torque efficiency. Subtract theoretical torque from input torque to obtain torque loss.

\[
T_{\text{theo}} = \frac{358 \text{ lb•in}}{2\pi} \times 3000 \text{ psid} = 389 \text{ lb•in}
\]

\[
T_{\text{input}} = 389 \text{ lb•in} \div 0.92 = 339 \text{ lb•in}
\]

\[
T_{\text{loss}} = 389 - 339 = 31 \text{ lb•in}
\]

(Dividing by .92 Torque Efficiency)

Shaft Input Power
Shaft input power is calculated by multiplying input torque (obtained above) by shaft speed, and dividing by 63,025 (see pump equation number 3 from page 21).

\[
HP_{\text{input}} = \frac{389 \text{ lb•in} \times 7000 \text{ rpm}}{63,025} = 43.2 \text{ HP}
\]

(389 lb•in is Tinput calculated above)

Hydraulic Output Power
Hydraulic output power is calculated by multiplying outlet flow (obtained above) by differential pressure, and dividing by 1714 (see pump equation 3a from page 21).

\[
HP_{\text{output}} = \frac{21.8 \text{ gpm} \times 3000 \text{ psid}}{1714} = 38.2 \text{ HP}
\]

(21.8 gpm is Qoutlet calculated above)

Power Loss
Power loss is obtained by subtracting the value of hydraulic output power form shaft input power. Power loss can also be obtained by using either of the following equations:

\[
\text{Power Loss} = \text{Power Input} \times (1 - \text{overall efficiency})
\]

\[
\text{Power Loss} = \frac{\text{Power Output}}{\text{overall efficiency}} \times (1 - 1)
\]

\[
HP_{\text{loss}} = 43.2 - 38.2 = 5.0 \text{ HP}
\]

(also, \( HP_{\text{loss}} = (43.2) (1 - .885) = 5.0 \text{ HP} \))

(and, \( HP_{\text{loss}} = (38.2) \left( \frac{1}{0.885} \right) - 1 = 5.9 \text{ HP} \))
Typical Performance Data and Installation Drawings
PV3-056

inches (mm)

39
Metric Conversion of Hydraulic Units

- Pressure:
  - \( \text{no. of bar} = 0.06895 \times \text{no. of psi} \)
  - \( \text{no. of kPa} = 6.895 \times \text{no. of psi} \)
  - \( \text{no. of psi} = 0.145 \times \text{no. of kPa} \)

- Torque:
  - \( \text{no. of N\cdot m} = 0.1130 \times \text{no. of lb-in} \)
  - \( \text{no. of lb-in} = 8.851 \times \text{no. of N\cdot m} \)

- Flow:
  - \( \text{no. of L/min} = 3.785 \times \text{no. of gpm} \)
  - \( \text{no. of gpm} = 0.2692 \times \text{no. of L/min} \)

- Volume:
  - \( \text{no. of mL (cc)} = 16.39 \times \text{no. of in}^3 \)
  - \( \text{no. of in}^3 = 0.06102 \times \text{no. of mL} \)
Inlet Pressures

![Inlet Pressures Graph](image-url)
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  - Fax: (610) 583-3985

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