POWER PUMP VALVE DYNAMICS— A STUDY OF THE VELOCITY AND PRESSURE DISTRIBUTION IN OUTWARD-FLOW BEVEL-FACE AND FLAT-FACE POWER PUMP VALVES

by
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ABSTRACT

The dynamics (movement) of the check valves in a reciprocating power pump determine whether the pump will operate quietly and smoothly, or noisily with pulsation and vibration. Although piping pulsation and vibration are, to a degree, functions of the system design, improper pump valve dynamics contribute a significant portion of the pulsation problems in many installations.

Many reciprocating power pumps, operating in industrial installations, have been found to be fitted with suction and discharge valve springs that are too weak. The weak springs do not close the valves soon enough, as the plunger reverses, causing the valves to be slammed onto their seats, creating a noisy pump, hydraulic shocks, and possible damage to the valves and/or seats. The hydraulic shocks can also cause damage to other pump components, the drive train, and to the system. Suction and discharge pipes may vibrate, piping and instruments may be damaged, and pump net positive suction head (NPSH) requirements may be high. To reduce the pulsations and vibrations, owners frequently resort to the installation of pulsation dampening equipment in the piping, when stronger valve springs may have been adequate to produce a quiet, smooth-running pump.

Starting with either the maximum valve lift, or the maximum closing impact velocity, simple equations of motion can approximate the displacement (lift), velocity, and acceleration of the valve, all based on crankshaft rotative speed. The maximum lift can then be used to calculate the NPSH required by the pump.

INTRODUCTION

Industrial power pumps are available in two basic configurations. Figure 1 shows a horizontal pump, with the typical vertical (axes) valves. Figure 2 shows a vertical pump, with the typical horizontal (axes) valves. These valves are simply check valves (nonreturn valves). Each valve is pushed open by the pumpage. It is pushed closed by the valve weight (if the axis is vertical), by the spring, and, if the spring is too weak, by the reverse flow of pumpage past the valve, which is caused by the reverse stroke of the plunger.

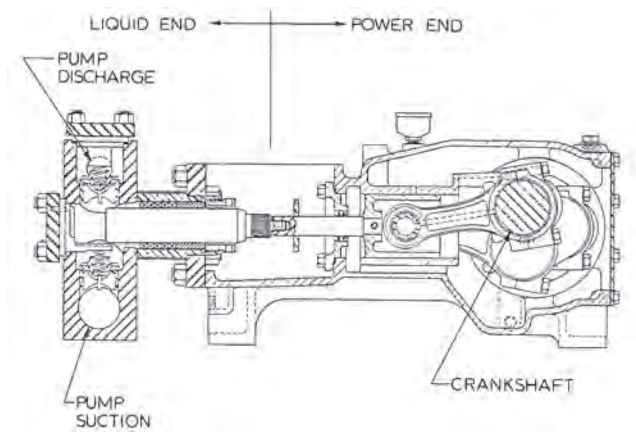

Figure 1. A Horizontal Triplex Power Pump Showing Vertical Wing-Guided Suction and Discharge Valves and Springs. (Courtesy Henshaw, 1987)

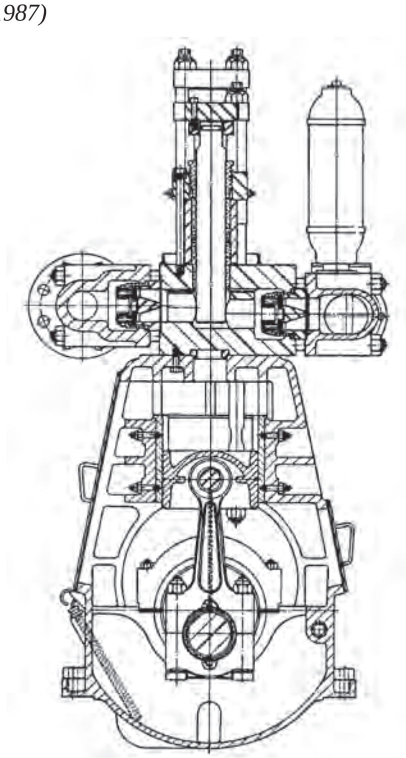

Figure 2. A Vertical Power Pump Showing Horizontal Disc-Type Suction and Discharge Valves and Springs. (Courtesy Henshaw, 1987)

The Need

The best valve and spring analyses known to the author, to date, are those developed by Worthington (1949 to 1953), although they do not always provide satisfactory results. They are largely based on approximations (Note: 1t is understandable that Worthington would use approximation shortcuts, because their equations were developed about 1950, a time when engineers used slide rules for their calculations. Iterations would have been very time-consuming. With the computer, one can quickly iterate for a more precise solution). They ignore the pressure drop in the valve seat (due to velocity), the pressure between the seating surfaces ("clinging" effect), acceleration and, to a degree, velocity. Other technical documents on pumps and compressors (Parry, 1989) (Singh and Madavan, 1987) (White, 1972) (Matsumura and Sugiyama, 1990) show that mathematical models can produce generally-acceptable agreement with test measurements, but they provide insufficient information to enable duplication of their work, and no equations to help evaluate or design power pump valves or springs.

This author became aware of the problems caused by weak valve springs as early as 1960, and in the intervening 40+ years, has searched for a reliable guide for the design and application of these springs. A weak spring allows the valve to close so late that it creates a hydraulic shock in the pump and system (it slams closed). It also results in the late opening of its mating valve, causing the mate to be "jerked," or shocked, open, creating an additional hydraulic shock. These pounding valves, and the resulting hydraulic shocks, cause excessive pump and system noise and vibration, and shorten the lives of pump and system components.

A late-closing discharge valve allows a momentary reverse flow through the valve seat, then, as it slams closed, causes a high-pressure shock on the discharge side of the pump, and a low-pressure spike in the pumping chamber. The suction valve is then "jerked" open, adding to the low-pressure pulse in the chamber and transmitting that pulse into the inlet manifold and pipe. A late-closing suction valve allows a momentary reverse flow through its seat, the valve being driven closed by the liquid pushed by the plunger, then the instantaneous stopping of the flow as the valve hits the seat creates shocks in the pump inlet and pumping chamber. The discharge valve is then driven from its seat, creating a hydraulic shock in the pumping chamber and sending that shock wave into the discharge system. Such valve action results in a noisy, vibrating pump and shaking pipes. It has also been found to break valves and valve seats, and can contribute to shorter life of piping components and gauges, as well as other pump components such as packing, gaskets, gears, crankshafts, and fluid cylinders.

This paper provides equations ("models") that allow the plotting of valve displacement (lift), velocity, and acceleration as functions of the crankshaft angle, provides equations that allow calculation of valve assembly pressure drops (that can be converted to NPSHR), and offers guidelines for the design, or selection, of springs that produce smooth-running, quiet power pumps.

FIELD EXPERIENCE

On various occasions, with pumps of various manufacturers, installation of "stronger" valve springs transformed a problematic situation into a smooth-running, satisfactory installation. ("Stronger," means, primarily, increasing the "preload" of the spring on the valve—not necessarily the stiffness of the spring.) Following are some examples of installations that involved the author:

1. Oil company, Goliad, Louisiana—A user reported that, when he increased the speed of his 4 inch (102 mm) stroke triplex, pumping butane, from 200 rpm to 300 rpm, the discharge pipe started shaking. He asked where he could buy a pulsation dampener. After installing stronger valve springs, a dampener was no longer required. Discharge pipe vibration was, again, low.

2. Oil company, California—A 5 inch (127 mm) stroke triplex was breaking (wing-guided) valves. The operator wanted stronger valves. Stronger springs stopped the breakage.

3. Wate-jetting company, Wixom, Michigan—A 3 inch (102 mm) stroke triplex, pumping water at about 5000 psi (34 mPa), was very noisy, did not achieve acceptable volumetric efficiency, and required excessive NPSH. Stronger springs quieted the pump, volumetirc efficiency increased to the expected value, and NPSHR dropped 50 percent.

4. Curacao, Caribbean—A pair of 5 × 7 ⅛ (127×181) 500 hp (370 kW) quintuplex pumps, running 217 rpm, in a desalination plant, were noisy, and the piping was vibrating excessively. The discharge pressure had a variation of ± 40 percent. Valve closing lags were measured at 21 degrees to 27 degrees of crank rotation. Much stronger springs, along with system modifications, resulted in satisfactory performance. Valve closing lag dropped to <5 degrees. Discharge pressure variations dropped to ± 4 percent.

5. Oil company, offshore Louisiana—A 3¼ × 7 (83×178) 500 hp (370 kW) vertical triplex pump running 233 rpm, with 5000 psig (34 mPa) discharge, on salt water, was reported to be noisy, had experienced numerous failures of the valves and seats, and vibrated excessively. Tests showed indications of cavitation, even though the inlet pressure was 200 psig (1400 kPa), and the pump had a large pulsation bottle at the inlet. Stronger valve springs quieted the pump. It ran so smoothly that a nickel was balanced, on edge, on the suction manifold.

6. Steel mill, Middletown, Ohio—Four 3⅞ × 5 (98×127) triplex pumps, operating in parallel, in a hydraulic system, at 300 rpm, were noisy, and pipes were shaking. Inline centrifugal boosters were vibrating excessively. The 1000 psi (7000 kPa) discharge pressure was pulsing ± 600 psi (4000 kPa). Spikes to 1000 psig (7000 kPa) were measured in the suction manifold. Stronger valve springs reduced discharge pressure variations to 100 psi (700 kPa). Noise and vibration dropped to acceptable levels. (The booster pumps were eliminated, and an individual suction line was run from the tank to each pump.)

7. Water-jetting company/chemical company, Illinois—A 1¾ × 4¼ (44×108) quintuplex pump, in a 5000 psi (34 mPa) water-jetting system, running 400 rpm, was breaking (disc) valves and seats. Increasing spring preload from 3 pounds (13 Newtons) to 18 pounds (80 Newtons) stopped the breakage.

Some of these pumps were quieted some by just stretching the valve springs, illustrating that the problem was as much preload as spring stiffness.

DEVELOPING THE MODEL

To satisfy the need for tools to better analyze power pump valves and springs, all forces acting axially on the valve (to push it open and closed) were quantified. A series of equations was derived, and a number of computer programs were written.

(These equations are for the more conventional outward-flow valve. A valve that has some, or all, flow going radially inward across inside-diameter sealing faces requires a different set of equations.)

Figure 3 is a cross-section of a flat-face, single-ported, outward-flow disc type valve. As shown in Figure 4, when the valve is pushed open by the pumpage, the pumpage impinges on the underside of the valve, flows radially outward between the valve and the seat, through the lift, or "escape," area, then past the outer diameter (OD) of the valve. Figure 5 illustrates the valve assembly areas used to quantify the forces on the disc. Figure 6 shows the components of a wing-guided valve, with Figure 7 showing the key dimensions of that assembly.

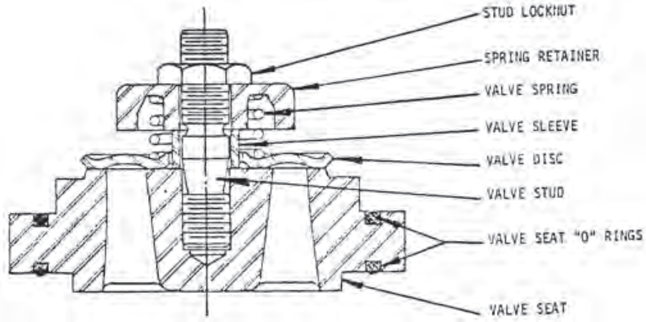

Figure 3. A Center-Guided Outward-Flow Flat-Face Disc-Type Valve Assembly. (Courtesy Henshaw, 1987)

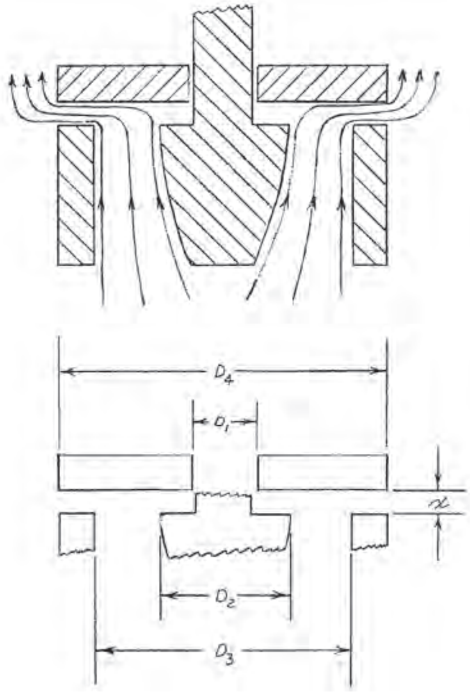

Figure 4. The Liquid Velocity Pattern and Pertinent Dimensions of a Center-Guided Outward-Flow Flat-Face Disc-Type Valve Assembly.

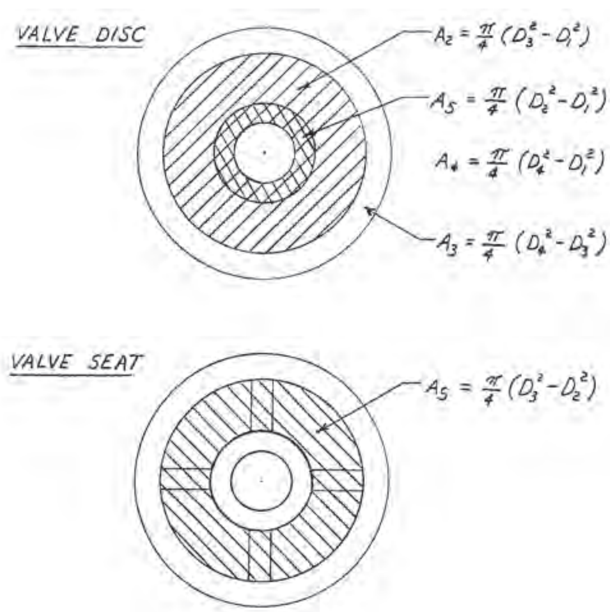

Figure 5. The Pertinent Areas of an Outward-Flow Valve Assembly.

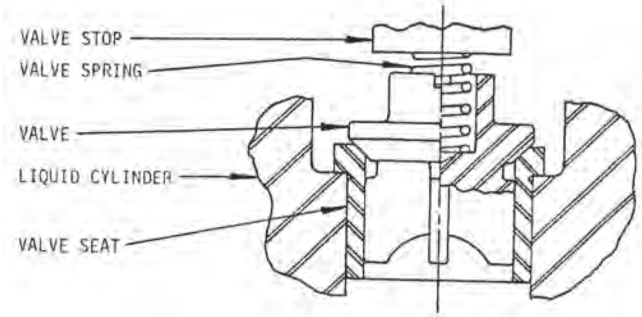

Figure 6. An Outward-Flow Bevel-Face Wing-Guided Valve Assembly. (Courtesy Henshaw, 1987)

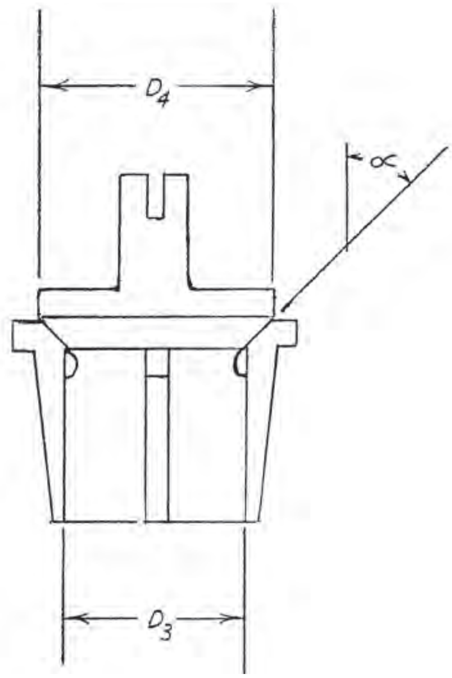

Figure 7. The Pertinent Dimensions of an Outward-Flow Bevel-Face Wing-Guided Valve and Seat.

ALL FORCES

The forces acting on the valve (disc) are as follows (convention: up is positive for forces, lift, velocity, and acceleration):

- F_P = Sum of all forces resulting from static pressures.
- F_i = Force from inertia of pumpage impinging on bottom of disc. (It is assumed that the pumpage on top of the disc moves in unison with the disc.)
- F_S = Force on disc from valve spring
- W_2 = Weight of disc if valve axis is vertical. If axis is horizontal, $W_2 = 0$.

PRESSURE FORCES

The sum of all static-pressure forces is:

$$F_P = P_2A_2 + P_3A_3 - P_4A_4 \tag{1}$$

The friction loss through the valve seat is negligible, so P_2 is equal to P_1 less the velocity head of the pumpage flowing upward though the seat, as follows:

$$P_2 = P_1 - \rho V_S^2/2 \tag{2}$$

$$P_2A_2 = (P_1 - \rho V_S^2/2)A_2 = P_1A_2 - (\rho V_S^2/2)A_2 \tag{3}$$

The static pressure in the lift area, P_3, is equal to P_4 plus the loss at the exit of the disc (taken as $\rho V_{ex}{}^2/2$), less the velocity head of the pumpage in the lift flow area. The effective velocity head is approximated using the velocity at the entrance to the lift area.

$$P_3 = P_4 + \rho V_{ex}{}^2/2 - \rho V_{en}{}^2/2 = P_4 + (\rho/2)(V_{ex}{}^2 - V_{en}{}^2) \tag{4}$$

$$\begin{aligned} P_3A_3 &= [P_4 - (\rho/2)(V_{en}{}^2 - V_{ex}{}^2)]A_3 \\ &= P_4\,A_3 - (\rho/2)\,A_3\,(V_{en}{}^2 - V_{ex}{}^2) \\ &= P_4\,(\pi/4)(D_4{}^2 - D_3{}^2) - (\rho/2)\,A_3\,(V_{en}{}^2 - V_{ex}{}^2) \\ &= P_4\,(\pi/4)D_4{}^2 - P_4\,(\pi/4)D_3{}^2 - (\rho/2)\,A_3\,(V_{en}{}^2 - V_{ex}{}^2) \end{aligned} \tag{5}$$

The static pressure force on the top of the disc is P_4A_4.

$$P_4A_4 = P_4(\pi/4)(D_4{}^2 - D_1{}^2) \tag{6}$$

The total of all static pressure forces is therefore:

$$\begin{aligned} F_P = {} & P_1A_2 - (\rho V_S{}^2/2)A_2 + P_4\,(\pi/4)D_4{}^2 - P_4\,(\pi/4)D_3{}^2 \\ & - (\rho/2)\,A_3\,(V_{en}{}^2 - V_{ex}{}^2) - P_4(\pi/4)\,D_4{}^2 + P_4(\pi/4)\,D_1{}^2 \end{aligned} \tag{7}$$

The third and seventh terms cancel each other. Collecting remaining like terms and substituting A_2 for the D_3 and D_1 terms produces the following:

$$F_P = (P_1 - \rho V_S{}^2/2 - P_4)A_2 - (\rho/2)\,(V_{en}{}^2 - V_{ex}{}^2)\,A_3 \tag{8}$$

The first term is the differential static pressure acting on A_2 pushing the valve open, and the second term is the differential static pressure acting on A_3 pushing the valve closed.

The Clinging Force

This second term is herein defined as the "clinging" force. It pushes the valve toward the seat (down) when there is flow radially outward between the valve and seat (through the lift area). (A wider seating surface creates a larger clinging force, thereby requiring a higher differential pressure to kick the valve fully open.) (This clinging tendency can be demonstrated with a spool and piece of paper. The paper will cling to the bottom of the spool when one blows down into the spool.) The clinging force can be further analyzed as follows:

$$\begin{aligned} F_C &= (\rho/2)\,(V_{en}{}^2 - V_{ex}{}^2)\,A_3 = (\rho/2)\,[(Q_e/A_{en})^2 - (Q_e/A_{ex})^2]\,A_3 \\ &= (\rho\,Q_e{}^2/2)[(1/\pi D_3 x\sin\alpha)^2 - (1/\pi D_4 x\sin\alpha)^2](\pi/4)(D_4{}^2 - D_3{}^2) \\ &= (\rho\,Q_e{}^2/8\pi(x\sin\alpha)^2)[(1/D_3)^2 - (1/D_4)^2](D_4{}^2 - D_3{}^2) \\ &= (\rho/8\pi)(Q_e/x\sin\alpha)^2[(D_4/D_3)^2 + (D_3/D_4)^2 - 2] \end{aligned} \tag{9}$$

(As D_4 approaches D_3 [as the seating surface width approaches zero], F_C approaches zero.)

To simplify calculations, the clinging coefficient is herein defined as:

$$K_C = [(D_4/D_3)^2 + (D_3/D_4)^2 - 2]\,/8\pi)(\sin\alpha)^2 \tag{10}$$

(K_C is dimensionless and is established solely by the dimensions of the valve assembly. α is the angle the seating surface makes with the axis of the valve. With a flat-face valve, as in Figure 3, α = 90 degrees, and $\sin\alpha = 1$.) So that:

$$F_C = K_C\,\rho\,(Q_e/x)^2 \tag{11}$$

The equation for the total static-pressure forces can then be written as:

$$F_P = (P_1 - P_4)A_2 - \rho V_S{}^2\,A_2/2 - K_C\,\rho\,(Q_e/x)^2 \tag{12}$$

($P_1 - P_4$) is the total (stagnation) differential pressure across the valve assembly. For a suction valve (with no significant restriction upstream or downstream of the valve), $P_1 - P_4$, at its peak value, is the NPSH required by the pump.

It is necessary to convert Q_e, the flow rate through the lift area, to plunger volume displacement rate, Q_2, and valve velocity, V_v. The flow rate through the lift area is less than plunger volume displacement rate by the volume displacement rate of the valve.

$$Q_e = Q_2 - A_4V_4 \tag{13}$$

Then:

$$F_P = (P_1 - P_4)A_2 - \rho(Q_2/A_S)^2\,A_2/2 - K_C\,\rho\,(Q_2 - A_4V_V)^2/x^2 \tag{14}$$

IMPULSE FORCE

The inertial, or impulse, force of the pumpage impinging on the bottom of the valve (disc) can be calculated from the momentum equation. Assuming the velocity change to be the difference between the seat (fluid) velocity and valve (disc) velocity:

$$F_i = \dot{m}\Delta V = (\dot{W}/g)(V_S - V_V) = Q_2\,W_3(Q_2/A_S - V_V)/g = \rho Q_2(Q_2/A_S - V_V) \tag{15}$$

Wright (1958) provided equations that revealed the impulse force for the wing-guided bevel-seat valve to be only 30 percent of this calculated value, yet was 130 percent of this value for the single-ported flat-face valve. Herein is introduced the impulse coefficient, K_i, to correct the calculated force to the measured force, to be used as follows:

$$F_i = K_i\,\rho Q_2(Q_2/A_S - V_V) \tag{16}$$

TOTAL OF FORCES ON VALVE (DISC)

The total of all forces on the valve (disc) is:

$$F_T = F_P + F_i - F_S - W_2 \tag{17}$$

$$\begin{aligned} F_T = {} & (P_1 - P_4)A_2 - \rho(Q_2/A_S)^2\,A_2/2 - (K_C\rho/x^2)(Q_2 - A_4V_V)^2 \\ & + K_i\,\rho Q_2(Q_2/A_S - V_V) - F_0 - Rx - W_2 \end{aligned} \tag{18}$$

From Newton's Second Law one knows that the net (total) force on the valve will produce an acceleration, such that:

$$F_T = ma = W_1a/g \tag{19}$$

The above equation can then be written:

$$\begin{aligned} W_1a/g = {} & (P_1 - P_4)A_2 - \rho(Q_2/A_S)^2\,A_2/2 - (K_C\rho/x^2)(Q_2 - A_4V_V)^2 \\ & + K_i\,\rho Q_2(Q_2/A_S - V_V) - F_0 - Rx - W_2 \end{aligned} \tag{20}$$

We now have the acceleration, velocity, and displacement (lift) of the valve (disc) all combined in one equation. It must be satisfied at all times that the valve is open. The instantaneous displacement of the plunger, Q_2, is the driving "force." It appears four places, all in numerators. Valve linear displacement, x, appears twice, once in a numerator and once in a denominator. Valve velocity, V_V, appears twice in numerators. These valve dynamics, driven by the plunger, determine the pressure drop across the assembly, $P_1 - P_4$.

To enable a solution to this equation, the pressure drop, $P_1 - P_4$, must be replaced with velocity and displacement terms common with the other terms in the equation.

CONSIDERING THE VALVE ASSEMBLY AS AN ORIFICE

The valve assembly can be considered an orifice. The pressure drop across the assembly is almost all a friction loss across the seating surfaces (through the lift area). This is the approach used by Worthington (1949 to 1953) and Matsumura and Sugiyama (1990) for the valve at its maximum lift (when stationary) and is extended here to the complete lift cycle. Calculations indicate that the inertia of the pumpage through the seat and lift area are negligible, and friction loss of the pumpage through the seat is negligible, so these effects are not accounted for in this evaluation.

The orifice equation is $V = c(2gh)^{0.5}$ where c = orifice coefficient and h = head (energy) drop across the orifice. The pressure drop is therefore:

$$\Delta P = W_3h = W_3(V/c)^2/2g = \rho(V/c)^2/2 \tag{21}$$

For the conventional valve, with pumpage flow radially outward, the effective velocity is taken as that at the entrance to the lift area (where velocity is maximum). Therefore:

$$\Delta P = \rho(V_{en}/c)^2/2 = \rho(Q_e/cA_{en})^2/2 = \rho[(Q_2 - A_4V_V)/c\pi D_3 x\sin\alpha]^2/2 \qquad (22)$$

This can now be substituted into equation (20), for $P_1 - P_4$:

$$W_1a/g = \rho A_2[(Q_2 - A_4V_V)/c\pi D_3 x\sin\alpha]^2/2 - \rho(Q_2/A_S)^2 A_2/2 - (K_c\rho/x^2)(Q_2 - A_4V_V)^2 + K_i\rho Q_2(Q_2/A_S - V_V) - F_o - Rx - W_2 \qquad (23)$$

In Equation (23) one sees that Q_2 now appears five places, all in numerators. Valve linear displacement, x, appears three times, once in a numerator and twice in denominators. Valve velocity, V_V, appears three times, in numerators.

INCORPORATING THE EQUATIONS OF MOTION

To calculate the velocity, V_V, and displacement, x, at any time, requires knowing the preceding values of acceleration, velocity, and displacement. The solution has been to divide the approximately 180 degrees (of crank rotation) that the valve is open into small enough increments that the acceleration can be considered as being constant during that increment. The following equations of motion can therefore be applied to assist in establishing values for a, V, and x:

$$V = V_1 + at \qquad (24)$$

$$a = (V - V_1)t \qquad (25)$$

$$x = x_1 + V_1t + 0.5at^2 = x_1 + V_1t + 0.5(V - V_1)t = x_1 + 0.5(V + V_1)t \qquad (26)$$

The time increment, t, can be converted to crankshaft angle as follows:

$$t = 30\Delta\theta/\pi N \qquad (27)$$

if θ is in radians, or:

$$t = \Delta\theta/6N \qquad (28)$$

if θ is in degrees.

These equations were incorporated into a number of programs that produced plots of x, V, and a as functions of crankshaft rotative angle.

THE COMPUTER PROGRAMS

The programs calculate and plot displacement, velocity, and acceleration of the valve from the time it opens until it closes. The opening lag and the angle increment are some of the inputs into the program. The program iterates at each increment, using the previous displacement, velocity, and acceleration, and the current plunger flow rate, to calculate the current values for, x, V, and a.

CALCULATING THE VALVE LIFT AT 90 DEGREES CRANK ANGLE (L_{90})

To calculate the valve lift at 90 degrees crank angle, Equation (23) can be simplified, since valve velocity, V = 0. The flow rate through the valve becomes Q_1, and x becomes L_{90}. L_{90} can be calculated as follows:

$$L_{90} = (A/B)^{0.5} \qquad (29)$$

Where:

$$A = \rho Q_1^2[A_2/2(c\pi D_3\sin\alpha)^2 - K_C] \qquad (30)$$

$$B = W_1a/g + \rho Q_1^2(A_2/2A_S - K_i)/A_S + F_o + R\,L_{90} + W_2 \qquad (31)$$

Since L_{90} appears on both sides of the equation, iteration is required for a solution. (A trial value is plugged into the right side of the equation for a first trial solution.)

USING A PURE SINE CURVE FOR PLUNGER VELOCITY

Because the plunger (or piston) is driven by a crankshaft-connecting rod-crosshead system (a slider-crank mechanism), the velocity of the plunger is a slightly distorted sine wave. If the L/R ratio (connecting rod length ÷ radius of crank throw) is 5 (typical for a power pump), the velocity reaches a peak at 79 degrees on the suction stroke (11 degrees before midstroke) (for a single-acting plunger pump), and 101 degrees on the discharge stroke (11 degrees after midstroke). Since a pure sine curve is the average of the suction and discharge stroke velocities, the programs, which are based on pure sine curves, can be used for either a suction valve or a discharge valve. The pure sine curve has a peak value that is below the distorted sine wave curve less than 2 percent, so is felt to be adequate. If desired, the equations can be revised to include the more precise distorted sine wave plunger velocities.

TESTING THE MODEL

The program was tested against the tabulated test results from a 4 inch(102 mm) stroke horizontal pump at 150 rpm and 450 rpm as reported by Parry (1989). The paper does not give all valve dimensions, so some dimensions were approximated. Results from the 2.75-inch (70 mm) diameter plunger were used.

Figure 8 shows the plot produced by the program at 150 rpm with the "heavy" valve (springs). For c = 0.6 (from tests by the author), the program calculated a maximum lift of 0.17 inch (4.3 mm) (compared to the 0.19 inch [4.8 mm] test result tabulated in the paper), a 0.01-inch (0.3 mm) lift at 180 degrees, and a 5 degree closing lag (compared to the tabulated 0 degrees, which is thought to be impossible. The mating suction valve was tabulated as having a closing lag of 5 degrees.). Changing c to 0.85, as used by Worthington (1949 to 1953), reduces the calculated maximum lift to 0.13 inches (3.3 mm), 0.06 inches (1.5 mm) less than the reported 0.19 inches (4.8 mm). It therefore appears that a c of 0.6 is more appropriate for a disc valve.

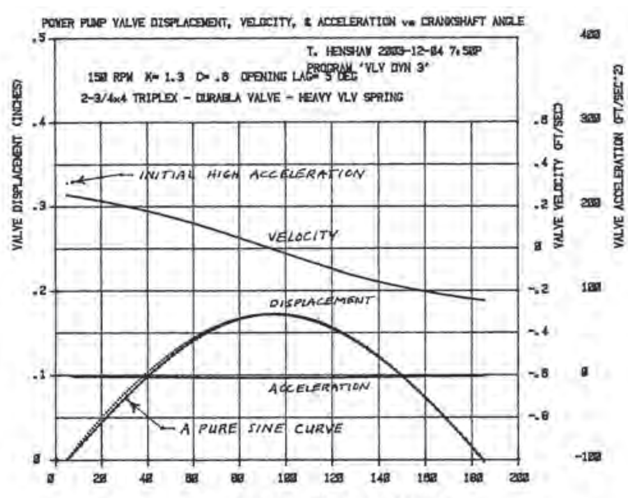

Figure 8. A Calculated Plot of Valve Dynamics of a Disc Valve in the 2.75×4 Power Pump.

For 450 rpm, the program showed that, had the lift not been limited (to 0.25 inch [6.4 mm]), the valve would have lifted to 0.55 inch (14 mm) (about three times a reasonable value for that speed), would have been 0.1 inch (2.5 mm) off the seat at the end of the stroke, and would have hit the seat with a velocity of 2 ft/sec (0.8 m/s). The program calculated a closing lag of 13 degrees, although the measured lag was reported as 28 degrees (which resulted in a reported 4.7 percent backflow). The difference is possibly because the program assumed that the spring was pushing on the valve from the fully lifted 0.55 inches (14 mm), whereas the valve tested, being held against a stop at the 0.25 inch (6.4 mm) lift, was not allowed to begin accelerating toward the seat until about 160 degrees of crank rotation. Displacement, velocity, and acceleration were all off the graph for a portion of the stroke.

To see how a stronger spring would have improved performance, the spring preload was arbitrarily increased by a factor of 10 (77.5 lbs (345 Newtons), instead of the reported 7.75 lbs (34.5 Newtons). The program reported a more reasonable maximum lift of 0.19 inch (4.8 mm), a 180 degrees lift of 0.013 inch (0.3 mm), and a closing velocity of 0.71 ft/sec (0.22 m/s). Spring stiffness was then increased by a factor of 10 (to 1940 [28,300] instead of 194 lb/ft [2830 N/m]), leaving the preload at 77.5 lbs (345 Newtons). The maximum lift dropped to 0.16 inch (4.1 mm), the 180 degree lift dropped to 0.012 inch (0.03 mm), and the closing velocity dropped to. 0.70 ft/sec (0.21 m/s). Although the stiffer spring reduced the maximum lift about 16 percent, the 180 degree lift dropped only about 8 percent, and the closing velocity dropped only about 1 percent. In both cases the valve closing lag was 5 degrees. The spring preload was then changed back to the reported 7.75 pounds (34.5 Newtons), leaving the stiffness at 1940 Ib/ft (28,300 N/m). The L_{90} lift increased to 0.25 inch (6.4 mm), the L_{180} lift increased to 0.047 inch (1.2 mm), the closing velocity increased to 1.47 ft/sec (0.45 m/s), and the closing lag increased to 9 degrees. This illustrates that spring *preload* has a greater impact on valve dynamics than spring *stiffness*, supporting the field experience reported earlier.

The tested "heavy" (spring) was suitable for 150 rpm, but unsuitable for 450 rpm. For 450 rpm, it needed to be stronger by a factor of at least 10.

The program was also used to analyze a wing-guided discharge valve in a 0.66×4.25 (17×108) water-jetting pump running 510 rpm with a discharge pressure of 40,000 psig (275 mPa). Figure 9 shows the lift, velocity, and acceleration if the valve were to open at 10 degrees of crank rotation. It shows "oscillations" in velocity and acceleration at the beginning of the lift, although the lift settles into a sine curve quickly.

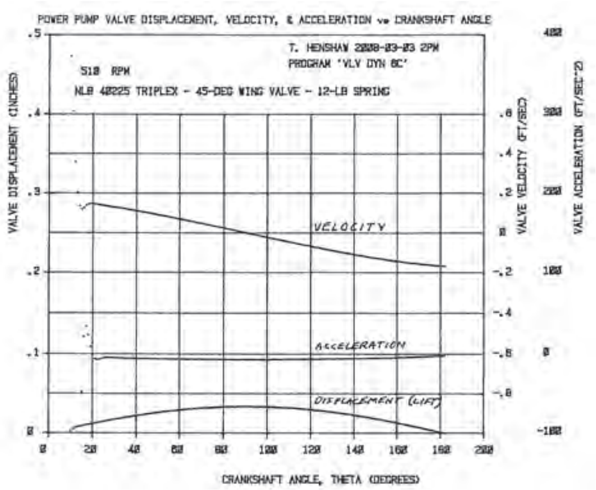

Figure 9. A Calculated Plot of Valve Dynamics of a Wing-Guided Discharge Valve in a 0.66×4.25 Power Pump with a Valve Opening Lag of 10 Degrees.

But because cool water shrinks 10 percent when compressed isentropically from atmospheric pressure to 40,000 psig (275 mPa), the discharge valve is not kicked open until about 50 degrees of crank rotation, as shown in Figure 10. The plot shows that the valve overshoots the sine curve, to a maximum lift of 0.036 inch (0.91 mm) at 55 degrees of crank rotation (only 0.004 inch [0.10 mm] more than the measured wear marks on the assembly), then undershoots slightly before settling to a sine curve. The velocity and acceleration take longer to settle to their respective trigonometric curves.

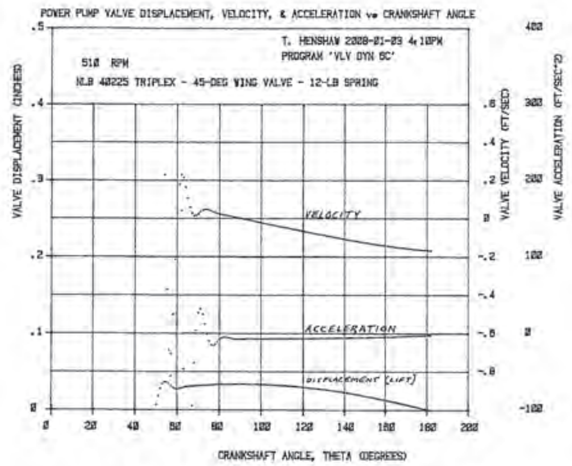

Figure 10. A Calculated Plot of Valve Dynamics of a Wing-Guided Discharge Valve in a 0.66×4.25 Power Pump with a Valve Opening Lag of 50 Degrees.

(It may be noted that the spring on this valve is considerably stronger than required by the proposed 72/N maximum lift of Equation [35].)

CURVE SHAPES

The shape of the displacement (lift) curve, for a properly-sprung valve, is shown by the model to be very near a sine curve (which is also shown by Wright [1955], Worthington [1949 to 1953], and Matsumura and Sugiyama [1990]), displaced by the angle of the closing lag. The velocity is therefore a cosine curve (confirmed by the model), and the acceleration is a minus sine curve (also confirmed by the model), except for momentary high values at the beginning of the lift (which can be caused by late opening), and possibly at closing. The greater the opening lag, the greater the initial high values, and the longer the duration of the departure from the sine and cosine curves. The greater the closing lag, also the greater the departure from the pure trigonometric curves.

Close approximations of the equations of motion of a properly-sprung power pump valve are therefore as follows:

$$x = L_{90} \sin(\theta + \theta_{LC}) = L_{90} \sin \omega(t + t_{LC}) \tag{32}$$

$$V_V = L_{90}\, \omega \cos \omega(t + t_{LC}) = L_{90}\, \omega \cos(\theta + \theta_{LC}) \tag{33}$$

$$a = - L_{90}\, \omega^2 \sin \omega(t + t_{LC}) = - L_{90}\, \omega^2 \sin(\theta + \theta_{LC}) \tag{34}$$

Because valve velocity is a (slightly distorted) cosine curve, it has maximum values at 0 degrees and 180 degrees. At near 90 degrees (near midstroke) velocity is zero. Because acceleration is a (slightly distorted, negative) sine curve, it is zero at 0 degrees and 180 degrees, and has its peak (negative) value near 90 degrees (midstroke). So, while the plunger has its peak velocity near midstroke, the valve has its peak displacement (lift) and acceleration near midstroke, and the valve velocity is simultaneously zero. While the plunger has its maximum acceleration at 0 degrees and 180 degrees, the valve has its maximum velocities at (near) those times.

MAXIMUM ACCEPTABLE LIFT AT 180 DEGREES (L_{180})

To obtain smooth operation and acceptable volumetric efficiency, Worthington (1949-1953) said that the lift of the valve, when the plunger reaches the end of its stroke, should be limited to a value of $D_4/200$ (0.5 percent of the valve OD), which means that a larger valve would be allowed to have a higher end-of-stroke lift (L_{180}) and higher impact velocity, when the valve closes against the seat. It seems more reasonable that the allowable seating (impact) velocity would be independent of valve size, and would require

L_{180} to be an inverse function of pump rotative speed, which is confirmed by Thornton (1976).

MAXIMUM ACCEPTABLE VALVE CLOSING LAG (θ_L)

Instead of using end-of-stroke lift, L_{180}, to determine if the spring is strong enough, can one use the amount (degrees) of valve closing lag θ_L?

If one assumes that the backflow through the valve, caused by the reversal of the plunger prior to the valve closing, is equal to the volume displacement of the plunger during that increment of shaft rotation (a seemingly reasonable assumption—and confirmed by Collier [1983a]), and if one knows the amount of valve lag, one can calculate the fraction of the stroke lost to valve backflow ("slip").

The fraction of the stroke lost to backflow = $f_L = L_{SL}/L_S$ = slip per valve = 0.5 (1-cos θ_L) where θ_L = angle of valve closing lag.

For example, a 10 degree lag would produce a slip of 0.0076 or 0.76 percent. If both suction and discharge valves lagged 10 degrees, the total slip would be about 1.5 percent, a low number. The 28 degrees lag reported by Parry (1989) for both valves, at 450 rpm, would calculate a per-valve slip of 5.85 percent; for both valves, 11.7 percent, which is excessive. The paper reported a slip of 4.7 percent for the discharge valve and 6.9 percent for the suction valve, a total of 11.6 percent. Although the individual values are different, the total is within 0.1 percentage point—less than a 1 percent difference.

(A 14 degree lag on both valves would produce a calculated total slip of 3 percent, a value that seems to produce acceptable pump performance, at least on lower-pressure units.)

VALVE LIFT AT θ = 90 DEGREES (L_{90}) VERSUS SPEED

Is there an optimum 90 degree valve lift that is a function of speed? Although it seems that the end-of-stroke lift, L_{180}, or the angle of valve closing lag, θ_L, or the valve closing velocity, V_C, is more important than maximum lift, it would be helpful to be able to establish an approximate maximum desirable lift, L_{90}. That number would provide the basis for simplified calculations of velocity and acceleration, and would simplify spring design.

Wright (1955) said: "As a general rule one might say that permissible lift of a valve is inversely proportional to the speed of the pump," but failed to pursue this concept to establish the logical relationships that follow. This principle is illustrated in Figure 11.

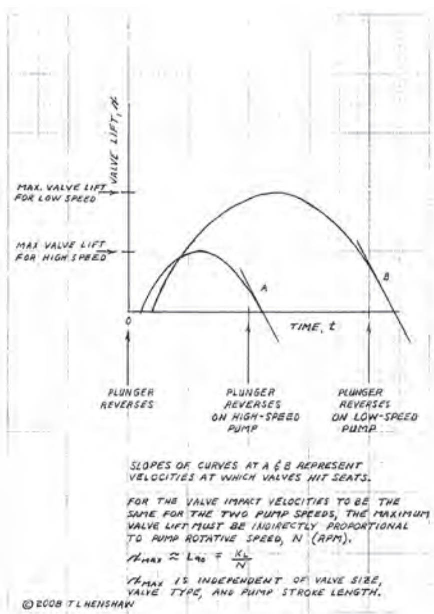

Figure 11. Illustration of the Effect of Crankshaft Rotative Speed (RPM) on Maximum Valve Lift to Obtain the Same Valve Impact Velocity.

Worster (1954), writing on reciprocating compressors, said that while acceptable valve "durability" was obtained with a lift of 0.200 inch (5 mm) at 300 rpm, it was necessary to reduce the lift to 0.040 inch (1 mm) for operation at 1800 rpm. If one uses this last point for reference, and let the N exponent be one, the equation for the approximate 90 degrees lift becomes 72/N inches (6/N ft) (1800/N mm). For 300 rpm, L_{90} = 0.24 inch (6.1 mm), just 20 percent above his reported 0.200 inch (5 mm) satisfactory lift. For 500 rpm, this produces L_{90} = 0.14 inch (3.6 mm), very near this author's observed 0.15 inch (3.8 mm) for satisfactory pump operation at 500 rpm.

It therefore appears that a smooth-running pump can result from a maximum (L_{90}) lift expressed by the following equation:

$$L_{90} = 72/N \text{ inches} \qquad (6/N \text{ feet}) \qquad (1800/N \text{ mm}) \tag{35}$$

Collier (1983a, pages 183 and 184), describes a 5½ × 12 (140×305) triplex pump, running 120 rpm, with weak valve springs that allowed the valves to lift 1.65 inch (42 mm) from the seats, and allowed reverse flow through the seats before closing. The reverse flow caused a reduction of capacity and rough operation. "Adequate" springs reduced L_{180} to "a few thousandths of an inch," and resulted in an L_{90} of 0.64 inches (16 mm). With an N exponent of one, and for 120 rpm, this produces the equation L_{90} = 77/N, the "77" being only 7 percent above the "72" derived above.

Note that this 90 degree lift L_{90} must be established by the spring, not by a positive stop. The valve must be allowed to accelerate downward from the point of maximum lift. The stiffness of the spring, R, should be kept as low as practical so that the spring continues to push on the valve until it is closed. The closed-valve spring force should be at least one-third of the fully-open-valve spring force.

If the suction and discharge valves are functioning properly, and the fluid friction losses upstream and downstream of the suction valve are negligible (a typical design), the NPSHR is established by the fluid velocity through the valve lift area at the 90 degree crank-angle lift. If the flow is turbulent (the normal case), then the NPSHR is proportional to the square of that velocity. When a pump speed is changed, and valve springs are selected for the new speed, the L_{90} lift is inversely proportional to the speed, so that the velocity through the lift area (at the maximum speed for that spring) is proportional to the square of the rotative speed. Therefore, the maximum NPSHR for each spring (at the maximum speed for that spring) increases to the fourth power of speed, as stronger springs are installed for the higher speeds. (At twice the speed, the L_{90} lift is one-half, the velocity through the valve lift is four times and the NPSHR is 16 times, that at the lower speed.) In equation form:

$$\text{NPSHR} = KN^4 \text{ lb/ft}^2 \tag{36}$$

where:

$$K = (\rho/2)(\pi L_s D_p^2/(1440 c D_3 \sin\alpha))^2 \tag{37}$$

K does not change as the speed and the springs are changed. K includes the density of the pumpage, pump component dimensions, and the orifice coefficient.

This equation has been verified by the group of NPSHR curves from Figure 12. The NPSH requirements on those curves, at the maximum speeds for the different springs, do increase to the fourth power of crankshaft rotative speed, N, rpm (when corrected for the limited valve lift at the lower speeds).

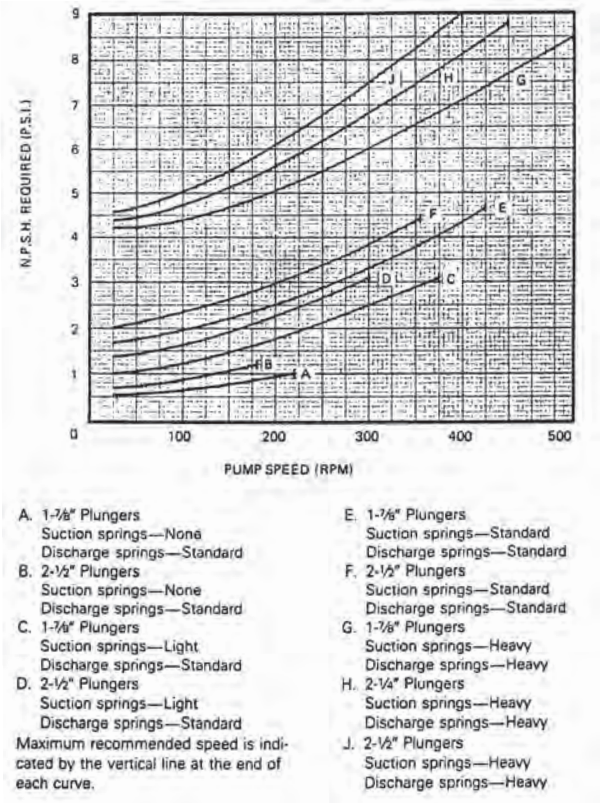

Figure 12. A Set of NPSH Curves for a 3 Inch-Stroke Horizontal Power Pump Illustrating the Significant Increase in NPSHR as Stronger Springs Are Provided for Higher Rotative Speeds. (Courtesy Henshaw, 1987)

MAXIMUM ACCEPTABLE SEATING VELOCITY

If the valve impacts the seat with an excessive velocity, not only will an excessive hydraulic shock be created, but the valve and/or seat can be damaged. Such damage has been observed by the author, and both Collier (1983b) and Worster (1954) report such damage. What is a maximum acceptable impact velocity?

If the closing lag of the valve is not too large (less than about 14 degrees of crank rotation), the displacement curve can be approximated with a sine curve, and the velocity curve would approximate a cosine curve.

From Equation (33), the maximum valve closing velocity occurs as the valve hits the seat, so:

$$V_C = -L_{90}\omega = -L_{90}\pi N/30 \tag{38}$$

Solving for L_{90}:

$$L_{90} = 30\ V_C/(\pi N)\ \text{ft} = 360\ V_C/(\pi N)\ \text{inches} \tag{39}$$

If:

$$L_{90} = 72/N\ \text{inches, then } 360\ V_C/(\pi N) = 72/N \tag{40}$$

and:

$$V_C = 72\pi/360 = 0.63\ \text{ft/s} \tag{41}$$

Therefore, for smooth operation and optimum life of pump and system components, the velocity at which the valve hits the seat should not exceed about ⅝ ft/s (0.19 m/s).

From the 5½ × 12 (140×305) triplex pump discussed above, with an acceptable maximum lift of 0.64 inches (16 mm), the closing velocity calculates to be 0.67 ft/s (17 m/s), only about 6 percent above the 0.63 (0.19) from Equation (41).

CONCLUSIONS

- When the valve assembly contains springs of adequate force for the pump speed and capacity, the last half of the valve displacement (lift) curve, i.e., when closing, can be closely approximated as a sine curve. The velocity can then be closely approximated as a cosine curve, which allows the closing impact velocity to be closely approximated at $L_{90}\omega$ (= $L_{90}\pi N/30$).
- For smooth operation, the maximum valve impact velocity (when the valve hits the seat) is about ⅝ ft/s (0.19 m/s).
- To achieve the above impact velocity, the displacement (lift) of the valve, when the plunger is at its maximum velocity (L_{90}, about midstroke), is 6/N ft (1800/N mm).
- When the speed of a power pump is increased, the valve spring force required to achieve (limit) the above lift, and impact velocity, can increase as much as the crankshaft rotative speed (rpm) to the fourth power.
- When the speed of a power pump is increased, and the valve spring force is increased to obtain the above desired midstroke lift, the pump NPSH requirement increases as the *fourth* power of crankshaft rotative speed (rpm) (absent significant losses upstream or downstream of the suction valve) (NPSHR = KN^4).

NOMENCLATURE

a = Acceleration of valve (disc), ft/sec^2 (m/s^2)
a_{90} = Acceleration of valve (disc) near midstroke of the plunger, ft/sec^2 (m/s^2) = $-L_{90}\omega^2$
A_{en} = "Escape" area (lift flow area) at entrance to disc, ft^2 (m^2) = $(\pi)(D_3)(x)(\sin\alpha)$
A_{ex} = "Escape" area (lift flow area) at exit of disc, ft^2 (m^2) = $(\pi)(D_4)(x)(\sin\alpha)$
A_p = Cross-sectional area of plunger or piston, ft^2 (m^2) = $(\pi/4)(D_p)^2$
A_s = Flow area through valve seat (wings and webs are ignored), ft^2 (m^2) = $(\pi/4)(D_3^2 - D_2^2)$
A_v = Same as A_4
A_2 = Valve area acted upon by P_2, ft^2 (m^2) = $(\pi/4)(D_3^2 - D_1^2)$ = $A_4 - A_3$
A_3 = Seating surface area exposed to P_3, ft^2 (m^2) = $(\pi/4)(D_4^2 - D_3^2)$
A_4 = Area of top of valve (that exposed to P_4), ft^2 (m^2) = $(\pi/4)(D_4^2 - D_1^2)$
C = Clearance volume. The dead space in pumping chamber w/plunger at end of discharge stroke, ft^3 (m^3)
c = Orifice coefficient of valve "escape" area
D = Plunger displacement. The volume swept by the plunger during one suction stroke or one discharge stroke, ft^3 (m^3) = $(\pi/4)D_p^2L_S$
D_p = Diameter of plunger or piston, ft (m)
D_1 = Diameter of hole in center of valve disc, ft (m)
D_2 = OD of inner seating surface, ft (m) (ID of port opening in seat)
D_3 = ID of outer seating surface, ft (m) (OD of port opening in seat)
D_4 = OD of valve outer seating surface, ft (m)
f_L = Fraction of plunger stroke lost to backflow caused by valve closing lag
f_{LD} = Fraction of plunger suction stroke lost to backflow caused by the discharge valve closing lag
f_{LS} = Fraction of plunger discharge stroke lost to backflow caused by the suction valve closing lag
f_{OD} = Fraction of stroke plunger moves on discharge stroke before discharge valve opens
f_{OS} = Fraction of stroke plunger moves on suction stroke before suction valve opens
F_c = "Clinging" force. The force that pushes the valve toward the seat due to velocity between the seating surfaces, lb (N)
F_i = Force imparted to upstream side of valve due to the turning of the pumpage (inertial force), lb (N)
F_o = Force from spring(s) when valve is closed, lb (N)
F_p = Force on valve due to static pressures, lb (N)

F_S = Total spring force on valve, lb (N) = F_O + Rx
F_{S90} = Spring force on valve when crank angle, θ = 90 degrees, lb (N)
F_t = Total of all forces on the valve, lb (N)
F_{90} = Total closing force on valve when crank angle, θ = 90 degrees, lb (N)
9 = Acceleration of gravity, 32.2 ft/sec^2 (9.8 m/s^2)
h_f = Friction loss through valve assembly, ft (m)
K_c = Clinging coefficient: for an OD-flow valve = $((D_4/D_3)^2+(D_3/D_4)^2-2)/8\pi(\sin\alpha)^2$; for an ID-flow valve = $(2-(D_2/D_1)^2-(D_1/D_2)^2)/8\pi(\sin\alpha)^2$
K_i = Coefficient of inertial impact of pumpage on upstream side of valve
K_L = The numerator in the fraction for $L_{90} = K_L/N$, ft-rev/min (m-s/min)
K_N = The pump and pumpage constant in equation. $F_{S90} + W_2 = K_N N^4$; $K_N = (\sigma/8\pi)(\pi^2 L_S Dp^2/240cK_L \sin\alpha)^2$
K_O = Coefficient of cracking pressure. The ratio of cracking pressure to pressure loss across valve assembly at peak flow rate
L_m = Maximum lift of valve, ft (m). (The maximum lift is normally equal to the L_{90} lift, but can be higher for a late-opening valve that “overshoots” when it is “jerked” open.)
L_S = Stroke length of plunger, ft (m)
L_{SL} = Distance plunger moves, after reversing, before valve closes, ft (m)
L_V = Close approximation of maximum valve lift, determined without considering acceleration, ft (m)
L_{90} = The lift of the valve at about 90 degrees of crank rotation, ft (m)
L_{180} = The lift of the valve at 180 degrees of crank rotation (the point at which the plunger reverses), ft (m)
m = Mass of the valve, slugs (kg) = $(W_1)/g$
n = Number of plungers in the pump
N = Rotative speed of pump crankshaft, rev/min
P_f = Friction loss through valve assembly, lb/ft^2 (Pa)
P_1 = Stagnation pressure upstream of valve assembly, lb/ft^2 (Pa)
P_2 = Static pressure acting on upstream disc area A_2, lb/ft^2 (Pa); $= P_1 - \rho(V_S)^2/2$
P_3 = Static pressure in lift flow area between valve and seat, lb/ft^2 (Pa)
P_4 = Stagnation pressure downstream of valve assembly, lb/ft^2 (Pa)
Q = Pump capacity. The flow rate in the pump inlet pipe, ft^3/sec (m^3/s) = $\eta_V Q_D$
Q_D = Total average pump displacement flow rate. The volume rate swept by all the plungers, ft^3/sec (m^3/s) = $\pi NnL_sD_p{}^2/240$
Q_e = “Escape” flow rate. The flow rate between the seating surfaces of the valve and the seat, ft^3/sec (m^3/s) = $Q_2 - Q_V$
Q_V = Rate of volume displacement of valve, ft^3/sec (m^3/s) = $(A_4)(V_V)$
Q_1 = Flow rate created by plunger at midstroke (the peak) = Q_{90}, assuming pure sinusoidal movement of plunger, ft^3/sec (m^3/s) $= \pi^2 NL_sD_p{}^2/240$
Q_2 = Instantaneous flow rate created by the plunger (plunger volume displacement), ft^3/sec (m^3/s) = $(Q_1)(\sin\theta)$
R = Spring rate, lb/ft (N/m)
S = Specific gravity of pumpage
t = Time increment, sec = 30θ/πN (for θ in radians); = θ/6N (for θ in degrees)
t_{LC} = Time lag closing (the time between θ = 180 degrees and valve hitting seat), sec
T = Thickness of valve disc, ft (m)
V_C = The velocity at which the valve hits the seat when closing, ft/sec (m/s)
V_{en} = Entrance “escape” velocity. The pumpage velocity between the valve and seat at the point of maximum (instantaneous) velocity, ft/sec (m/s) = $Q_e/\pi D_3 x \sin\alpha$
V_{ex} = Exit “escape” velocity. The pumpage velocity between the valve and seat at the exit, ft/sec (m/s) = $Q_e/\pi D_4 x \sin\alpha$
V_p = Velocity of plunger or piston, ft/sec (m/s)
V_S = Velocity of pumpage moving through seat, ft/sec (m/s) = $(Q_2)/(A_S)$
V_V = Velocity of valve, ft/sec (m/s)
W_1 = Weight of valve (in air), lb (N)
W_2 = Weight of valve if axis is vertical, less the bouyant effect of the pumpage, lb (N) = $W_1(1 - W_3/W_4)$ (if axis is horizontal, $W_2 = 0$)
W_3 = Specific weight of pumpage, lb/ft^3 (N/m^3) = ρg
W_4 = Specific weight of valve, lb/ft^3 (N/m^3)
x = Distance valve has lifted from seat, ft (m) (linear displacement of valve)
α = Angle between valve seating surface and valve axis, degrees
η_v = Volumetric efficiency of pump, as a fraction = $1 - f_{OS} - f_{OD}$
θ = Angle of crank rotation from start of stroke of plunger, radians = ωt = πNt/30
θ_{LC} = Valve closing lag. Angle of crank rotation from start of return stroke of plunger before the valve closes, radians or degrees
θ_{OD} = Angle of crank when discharge valve opens, radians or degrees
θ_{OS} = Angle of crank when suction valve opens, radians or degrees
Δθ = Increment of crank rotation, radians = ωΔt = πNΔt/30
λ = Compressibility of pumpage = (W_3 at discharge)/(W_3 at inlet) − 1
ρ = Density of pumpage, slugs/ft^3 (lb-s^2/ft^4) (kg/m^3) = W_3/g
ω = Angular velocity of pump crankshaft, radians/sec = πN/30

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