SPE 2904

Practical Hydraulics of Positive Displacement Pumps for High-Pressure Waterflood Installations

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Introduction

As the demand for oil continues its dynamic climb upward, waterflooding will be an increasingly important factor in this country's energy picture. In 1970, waterflooding accounted for 1 of every 4 bbl of oil produced in the U. S. This number will increase to 1 of every 3 bbl by 1980.

To meet this demand, many waterfloods are being installed in fields that were considered nonfloodable just a few years ago because of high fluid injection pressures required. Good engineering is imperative in the design of high-pressure installations and will pay for itself many times over in reduced maintenance costs. The purpose of this paper is to present a summary of the technology used in the hydraulic design of positive displacement pump systems for high-pressure waterflood service.

Of the three flow regions in a positive displacement high-pressure waterflood system — the suction, the pump, and the discharge — the pump is the consistent element. The manufacturers spend a great deal of time and money on the hydraulics and mechanics of their pumps. To increase volumetric efficiency, the effects of path tortuosity, entrance and exit losses, and fluid compressibility in the design of pump fluid ends are minimized. The elementary principles of fluid mechanics used for the design and analysis of fluid ends can also be used for the inlet and outlet systems.

Poor design of suction and discharge piping results in increased pump repair and decreased life of equipment due to overstressing of components by cavitation, excessive pulsation and mechanical vibration. In the design of new systems good sense must be used to avoid these pitfalls and insure the best possible hydraulic and mechanical operating conditions.

Positive Displacement Pumps

The positive displacement pump is the most widely used pump for high-pressure water injection. (In this paper, 2,500 psi and above is considered high pressure.) It is better adapted to high-pressure service than the centrifugal pump because of its high uniform efficiency. (For typical plunger pumps used in oilfield service, the volumetric efficiency is 95 percent and the mechanical efficiency is 90 percent.[1])

A plunger pump, which is the positive displacement pump used in waterflooding, is relatively simple. Fig. 1 shows a cross-section of a typical plunger pump. The fluid is pumped by use of a plunger and a series of valves. The plunger delivers a specific amount of fluid into the discharge line during the discharge stroke. In this part of the operation, the discharge valve is open and the suction valve is closed. During the charge stroke, the specific amount of fluid is taken from the suction line and the valve action is reversed. It is this action of the pump that causes pressure surges. These pressure surges are natural for all positive displacement pumps and if the system is properly designed, should not be troublesome. However, if

Pressure surges, cavitation, and mechanical vibration prevent water injection systems from operating at maximum efficiency. They cause premature failure of the equipment and excessive operating cost. By using the engineering principles outlined here, detrimental effects due to inadequate net positive suction head and poor pipe streamlining can be minimized.

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pressure surges are ignored in the system design, the result will be severe damage to the equipment and piping.

Pressure surges, sometimes called water hammers or pressure pulsations, are localized rises in pressure caused by sudden acceleration or deceleration of the fluid. The pressure surges generated by a typical pump are illustrated in Fig. 2. The intensity or magnitude of the surges is a function of the net change in velocity, the volume, and the length of the fluid column.

Once generated by the pump, a pressure surge travels through the piping system (both suction and discharge) at the speed of sound in water until it strikes a reflection point such as a bend or restriction in the flow line. The surge is then reflected back and repeats the cycle. This can be a critical problem because one pressure surge sometimes coincides with another. When this resonant condition occurs, the magnitude of the waves is additive. The amplitude of the pressure surges and resulting vibration become large and can be destructive.

Pressure surges can be reduced by properly designing the system, considering net positive suction head and flow-line streamlining and using pulsation dampeners or desurgers. Sufficient net positive suction should be provided to insure that each cylinder of the pump is completely filled during the suction stroke. If the cylinder is not properly filled, severe pressure surges and cavitation will result. Even if the system is designed under "ideal" hydraulic conditions, not all pressure surges can be eliminated. In the field it is often necessary to use pulsation dampeners to reduce the effects of these pressure surges. Pulsation dampeners absorb the peak pressure surge and feed it back to the flow stream during the low point in the pressure cycle (see Fig. 2). The net result is a pressure-vs-time curve that no longer shows pronounced high- and low-pressure areas, but has the appearance of a gently waving line. Thus the peak pressure surges, which do maximum damage to the system, are dampened out. Methods to reduce the effects of pressure pulses will be discussed in more detail later in the paper.

Pressure pulses, inherent in positive displacement pump systems, need not be destructive. With proper system design, pressure pulses can be controlled.

Suction Design

As with any hydraulic system, the design of the suction system must be simple. It should assure a positive suction head on each pump cylinder and should assure that pressure pulsations are kept to a minimum. In high-pressure systems, the problems caused by cavitation and pulsation are compounded and result in high pump repair cost and down time.[2]

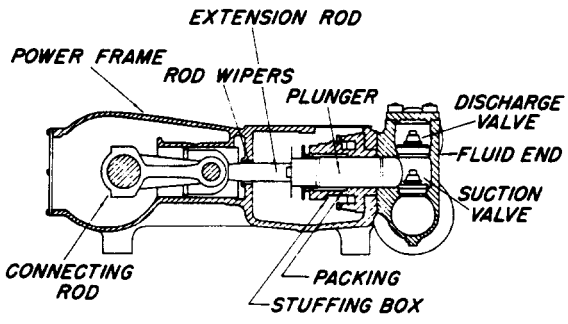

Fig. 1—Cross-section of a typical plunger pump (horizontal model shown).

Net Positive Suction Head (NPSH)

Net positive suction head (NPSH) is defined as the head, in feet of liquid above vapor pressure, required at the suction inlet of the pump to completely fill each cylinder on the suction stroke.[3] If these conditions are met, cavitation and pressure pulses will be reduced.

The NPSH available in a system can be determined by the following equation:

$$\text{NPSH}_{\text{available}} = H_s + H_a + H_v - H_{vp} - H_f - H_{ac} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

H_s = static head (feet of water)
H_a = barometric pressure (feet of water)
H_{vp} = vapor pressure at temperature pumped (feet of water)
H_f = suction pipe friction loss (feet of water)
H_{ac} = acceleration head (feet of water)
H_v = velocity head (feet of water)

The value of the velocity head, H_v, is very small if the suction system is properly designed, and generally it can be ignored in application.

A schematic of these forces is seen in Fig. 3. The positive forces — those that aid in pump cylinder filling — are the static head of water and the atmospheric pressure. Negative forces — the forces that prevent cylinder filling — are the vapor pressure of the water, the friction-head loss of the flow line, and acceleration head.

The values of these factors can be easily determined. Static head is determined from the geometry of the system. Barometric pressure and vapor pressure are found from Figs. 4 and 5, respectively. (It is important to consider the effects of temperature on the vapor pressure. As noted in the figure, a difference of 100°F—from 80° to 180°F—will require an additional 16 ft of head.) Piping losses can be determined from standard friction-loss equations.[4]

Acceleration head, defined as the force required to accelerate the fluid from a static position to maximum velocity during the pump stroke, is determined from an empirical formula. The most widely used equation is:

$$H_{ac} = \frac{C_a L S Q}{D^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Recent laboratory studies have indicated that the effect of acceleration head is not so great as this equation indicates;[3] however, for field use where all NPSH variables are not so well defined, it is recommended that Eq. 2, while somewhat conservative, be used.

The NPSH required to prevent cavitation and pressure pulsing can be determined by the following equation:

$$\text{NPSH}_{\text{required}} = H_a + H_g - H_{vp} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The value of H_g should be obtained from the pump

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manufacturer. This term is a function of the diameter of the plunger, speed of the pump, type of valve used, valve-spring load and tolerance of the equipment;[5] therefore, all pump manufacturers should have this value for their pumps. Some pump manufacturers will supply required NPSH rather than H_g. NPSH determined in Eq. 3 or supplied by the manufacturer is the minimum theoretical suction head required by the system to assure maximum volumetric efficiency of the pump. In field application, a design factor of 10 to 20 percent is commonly included to account for unknown variables that could cause additional suction-head requirements.

The required static head ($H_{s\ \text{required}}$) to insure maximum pump volumetric efficiency can be determined by setting the $\text{NPSH}_{\text{available}}$ of Eq. 1 equal to the $\text{NPSH}_{\text{required}}$ of Eq. 3 and solving for $H_{s\ \text{required}}$.

$$H_{s\ \text{required}} = H_g + H_{ac} + H_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

If the value of H_g is not readily available, the static head required can be estimated with practical accuracy from the following empirical equation:

$$H_{s\ \text{required}} = H_{vp} + H_f + H_{ac} + \frac{\text{Elevation}}{1{,}000}, \quad (5)$$

where elevation is measured in feet. Simply stated, a level of water equal to the sum of the vapor pressure plus a correction for altitude plus a correction for acceleration head plus a correction for suction pipe friction must be provided above the pump suction. This assumes that to insure maximum volumetric efficiency the suction head must be at least level with the pump suction. The atmospheric pressure (34 ft of water head at sea level) is then available to satisfy the NPSH requirements of the pump. The equation is somewhat conservative but it allows a design factor for the many unknown variables that adversely affect the NPSH of the pump.

In the design of a water suction system, NPSH must be given top consideration. Tanks or charge pumps must be designed to deliver the required suction head at all times. If the NPSH requirement of the pump is not met, cavitation and pulsation may result.

Suction Flowline Design

Properly designed flow lines can minimize the inherent pressure surging of positive displacement pumps. All flow should be as streamlined as possible; simplicity is the watchword in flowline design.

Straight pipe, uncomplicated by restrictions or enlargements, makes the best and least expensive system. Separation of the fluid from the pipe boundary, as well as eddy currents, result when fluid passes around a corner. Boundary separation must be avoided as this is the point where, as a result of the local pressure reduction, cavities or bubbles form. These cavities travel downstream to areas of higher pressure and collapse, creating localized pressure forces. This phenomenon is called cavitation. If severe enough, it will affect pump performance, cause vibration and noise, and erode the metal from the pump and adjacent piping. Eddy currents and boundary separation should be avoided because they both cause energy losses.

Pipe entrance from tanks is a common place for boundary separation and eddies. A series of entrance

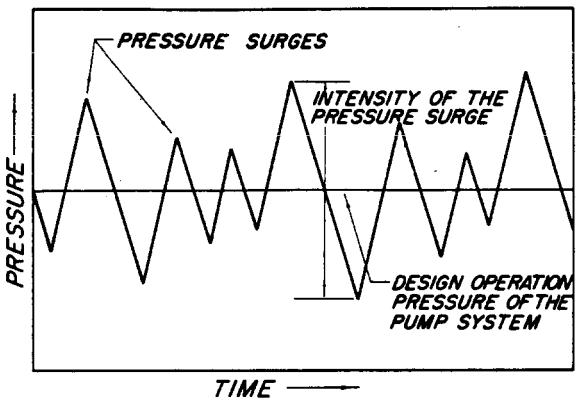
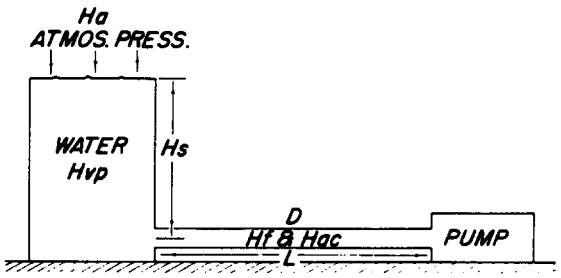


Fig. 2—Pressure-vs-time curve showing the effects of pressure surging.



NPSH available = Hs+Ha-Hvp-Hf-Hac

Fig. 3—Forces affecting net positive suction head (NPSH).

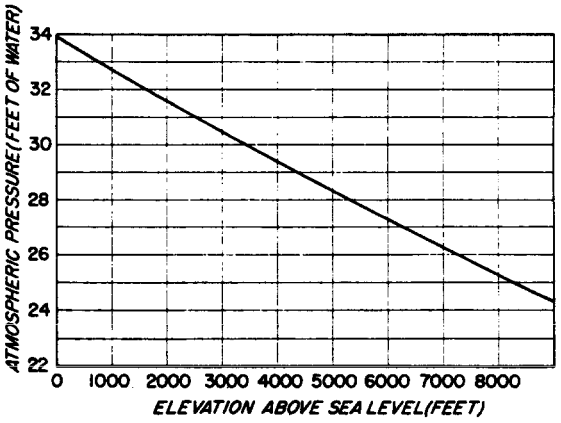


Fig. 4—Atmospheric pressure.

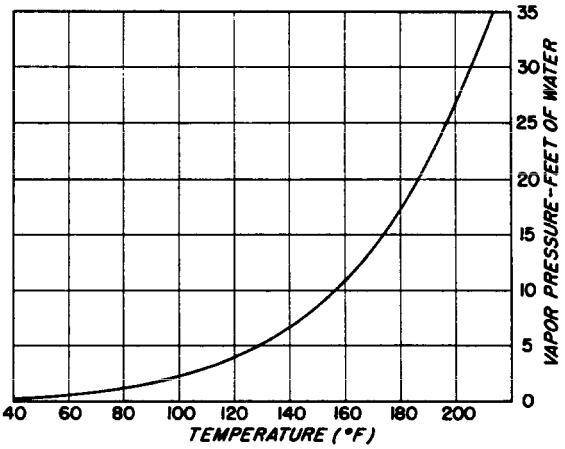


Fig. 5—Vapor pressure of water.

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designs is shown in Fig. 6. Fig. 6a is the preferred design. Rounded corners tend to minimize entrance pressure losses, boundary separation, and eddies. (The minimum acceptable radius of curvature is 1/7 of the pipe diameter or larger: $r \geq D/7$.) The design shown in Fig. 6b generates eddies and boundary separation. The projecting entrance, known as the Borda mouthpiece and shown in Fig. 6c, should never be used because it causes restricted flow and pronounced boundary separation. A rounded entrance can resolve the problems of eddy currents and boundary separation in tank inlets.

Abrupt contraction or restriction in pipe also produces separation. This is illustrated in Fig. 7a. Gradual contractions may be designed to allow for minimum head loss and boundary separation. Separation is particularly to be avoided in the downstream section of the flowline near the pump suction valves. Separation in this area will aggravate cavitation.

Head losses and boundary separation caused by sudden enlargement of the flowline diameter may be considerably reduced if the expansion is made gradually. (Fig. 7b shows the hydraulics of flow through a sudden enlargement.) An optimum divergence angle is 7° or 8°.[7]

In actual field applications, the system should be designed without any elbows, tees, meters or other restrictions. This, of course, is not always feasible; however, 45° elbows or large-radius bends should be used instead of ells.

Individual suction lines from the tank to the pump are recommended. This prevents abnormal pressure pulsations and vibrations in one pump from being introduced into the system where they can affect the operation of other pumps. Pressure surges and vibrations are additive if they are in phase, and under this condition they can cause damage to the system.

To avoid excessive acceleration head losses in the suction line, the fluid velocity should be kept to less than 2 ft/sec. This should help prevent cavitation and reduce pressure pulsations. Also, with this velocity the flow will be laminar.

Flowline valves should be full-opening gate or butterfly valves to reduce flow resistance and turbulence in the system. Valves that are not full opening have the same effect on the fluid flow streams as a sudden pipe contraction and expansion. Boundary separation and eddies are prominent when such valves are employed. Also, check valves should not be used.

In summary, the suction system should be designed with simplicity, eliminating all unnecessary bends, contractions and expansions. There are numerous examples[2] of systems with poor operating records traceable to designing that has ignored these basic hydraulic principles. With proper planning and sound hydraulic engineering, relatively trouble-free high-pressure water injection systems can be built.

Discharge Design

Methods of building a safe, reliable system at the lowest practical cost must be considered in the design of high-pressure piping systems. It is desirable to select and fabricate components that will not have to be replaced or maintained during the life of the project. As pointed out previously, the positive displacement plunger pump is a pulse generator. The discharge of the pump is an amplified version of the suction conditions. A pump operating with inadequate suction will have higher discharge pulsations than a pump with good suction. Since pulsating flow cannot be totally eliminated, the engineer must be aware of its effects. Systems designed with only the static pressure taken into account may fail because of metal fatigue caused by pressure pulsation and mechanical vibration.

If metal components are subject to cyclic or fluctuating stresses they will often fail even though careful analysis shows the maximum actual stresses to be below the ultimate strength, and often the yield strength, of the material. When a part fails under static loading, the failure usually appears first as a large deflection because the yield strength has been exceeded, and replacement may be possible before total failure occurs. Fatigue failures, however, usually develop suddenly as cracks, with no visible warning. They appear as fractures of a brittle material such as glass when it fails in tension. Although fatigue failures are not completely understood, enough is known about them to make sound engineering designs. Failure by fatigue is related to both the number of stress reversals and the magnitude of the stresses. Generally, the higher the stresses the fewer the cycles required to cause failure. All steels used in the oil field have an endurance limit, which is the stress level below which infinite life occurs. In design, the stress at which 10 mil-

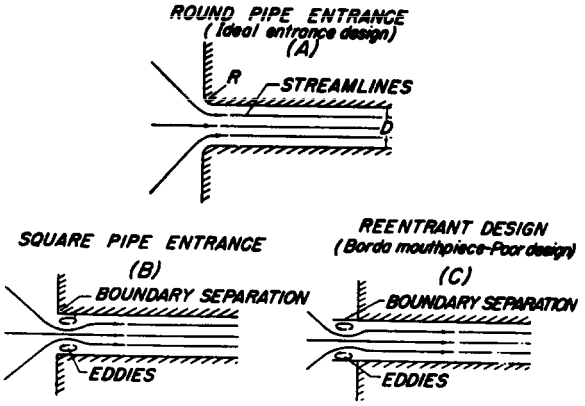

Fig. 6—Streamlines of different pipe entrances.

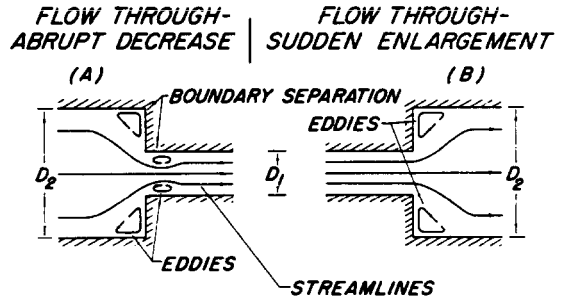

Fig. 7—Streamlines of flow through abrupt decreases and sudden enlargement.

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lion or more cycles are developed without failure can normally be used as the endurance limit. A single quintuplex pump operating at 300 rpm will produce 10 million plunger strokes in 111 hours. It is imperative that the piping be designed to keep stress levels below the endurance limit. Factors that can adversely affect the endurance limit of parts include residual stresses, notches, corrosion, and rough surface finish.

Pulsation Dampeners

Keeping the amplitude of pressure fluctuations and mechanical vibrations to a minimum reduces fluctuating stresses in the discharge piping. Lower stress levels reduce initial investment by avoiding excessive safety factors, result in designs that will not fail, and invariably reduce maintenance and repair costs. The use of pulsation desurgers, or dampeners as they are frequently called, can be very effective in reducing pressure pulsations. The need for these devices is not always apparent. Under normal operating conditions they may not be needed to protect the piping and valves downstream of the pump. Extensive tests have shown[10] that poor pump-valve performance can increase peak-to-peak pulsation amplitude from about 8 percent of line pressure to more than 25 percent of line pressure. A properly installed bladder-type pulsation desurger used in the tests was able to reduce pulsations from 8 to 1.5 percent of line pressure. Since it is probable that during the life of a project any given pump will run for a considerable number of hours with faulty valves, pipe failure may be prevented by using desurgers.

The two basic types of discharge pulsation dampeners are (1) the bladder-type desurger, in either the flow-through or appendage configuration, and (2) the flow-through acoustical type (see Fig. 8). This acoustical type utilizes interference to smooth out pulsations. The device is so designed that the two flow paths through the unit recombine 180° out of phase, cancelling the surges. Pump speed, number of cylinders, and capacity are used for design purposes. Once installed, the acoustical unit requires no maintenance.

The in-line, bladder-type dampener has proved to be more effective than the appendage bladder-type.[10] Important in selecting any bladder desurger is the amount of precharge that can be used. For example, take the following system:

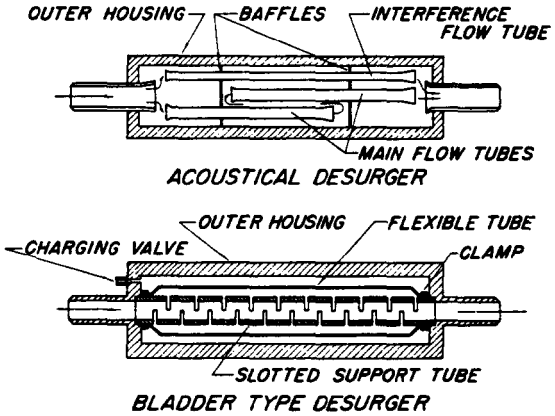

Fig. 8—Typical desurgers used in water injection systems.

1. 4,000 psi operating pressure
2. 500 psi peak-to-peak pulsation amplitude
3. Two choices for pulsation dampeners:
   A. 100 cu in. gas volume unit, 1,000 psi maximum charge
   B. 100 cu in. gas volume unit, 2,000 psi maximum charge

Assuming a simple gas law behavior — that for a closed system with constant temperature and mass of gas $- p \times V =$ constant. So, for dampeners A and B,

$p_A V_A = 100 \text{ cu in.} \times 1000 \text{ lb/sq in.} = 1 \times 10^5 \text{ lb-in.}$
$p_B V_B = 100 \text{ cu in.} \times 2000 \text{ lb/sq in.} = 2 \times 10^5 \text{ lb-in.}$

With the pump running, peak pressure is 4,250 psi, minimum pressure is 3,750 psi. Bladder volumes for the two systems are

$$V_A \text{ (at } p_{max}) = \frac{p_A V_A}{p_{max}}$$

$$V_B \text{ (at } p_{max}) = \frac{p_B V_B}{p_{max}}$$

$$V_A \text{ (at } p_{min}) = \frac{p_A V_A}{p_{min}}$$

$$V_B \text{ (at } p_{min}) = \frac{p_B V_B}{p_{min}}$$

The net "cushion" effect from the dampener is the difference in volume at maximum and minimum pressures. These values are calculated to be

$V_A = 3.1$ cu in., (with 25 percent precharge)

$V_B = 6.2$ cu in., (with 50 percent precharge)

To get equal effectiveness, then, with System A as with System B, A should be a 200-cu in. unit. It has been found that 70 percent of line pressure for precharge[10] gives maximum performance for the in-line desurgers tested.

Piping Design

Piping design, with or without the use of desurgers, should take into account the stress endurance limit of the pipe in relation to the corrosivity of the water being pumped. Since the effects of corrosion on metal fatigue are not well understood, we must turn to empirical derating factors to achieve a satisfactory design. The following method for determining the maximum allowable operating pressure under various severe environment conditions with the presence of pressure pulsations was developed from Ref. 11.

1. The allowable (endurance) stress in the steel pipe is

$$S_A = E_s \times 3/(2-R), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6a)$$

or

$$S_A = \text{YS} \times F \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6b)$$

2. The allowable maximum operating pressure is

$$p = 2 \times S_A \times (t/D_o) \times J \times T_f \quad . \quad . \quad . \quad . \quad (7)$$

3. The desired wall thickness of the pipe is

$$t = \frac{2 \times S_A \times J \times T_f}{D_o \times p} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

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For this method we must first determine the size and grade of the pipe to be used. To reduce turbulence, it is recommended that the velocities in the discharge piping be kept below 8 ft/sec and that the pipe not be reduced in size for at least 15 ft from the pump manifold. The corrosivity of the fluid to be pumped must then be used to determine the corrosion factor C_c from Table 2. The maximum peak-to-peak pulsation amplitude should be estimated from pump manufacturers' data or from data obtained from a similar system. E_s, determined from Table 3, and R can now be used to find the allowable stress, S_A, using Eq. 6a. This should be compared with the second value of S_A obtained using the yield strength of the pipe material. The lesser of these two S_A values is then used in Eq. 8 to determine the desired wall thickness, t. In this way, existing systems can be analyzed to determine if operating conditions should be modified by use of pulsation control equipment so as not to exceed allowable pipe stress levels (Eq. 7).

Mechanical vibrations in pipes can also cause problems if the supports are so spaced as to allow the pipe to vibrate in one of the frequencies set up by the pumps.[8] The following plan spaces pipe supports so that the piping frequency is four times higher than the pump pulse frequency. The estimated highest pulsation frequency is

$$F_P = \frac{(S \times N \times A)}{60} \quad \ldots\ldots\ldots \quad (9)$$

$$f = F_P \times 4 \quad \ldots\ldots\ldots \quad (10)$$

The distance between pipe supports, L, can now be calculated:

$$f = \frac{\propto}{L^2}\sqrt{\frac{EI}{W}} \quad \ldots\ldots\ldots \quad (11)$$

and then

$$L = \sqrt{\frac{\propto \left(\frac{EI}{W}\right)^{0.5}}{f}} \quad \ldots\ldots\ldots \quad (12)$$

The proper installation of these supports is as critical as the spacing. Supports should be as short and sturdy as possible to prevent rocking. Also, supports should be staggered varying amounts, up to 10 percent below calculated spacing, so that identical spacing will not produce synchronous vibrations between adjacent pipe spans. All bypass piping, heavy flanges, pulsation dampeners and heavy valves should be solidly supported.

High-pressure discharge piping cannot be properly designed without considering pressure pulsation and mechanical vibration. Pulsation can be controlled with dampeners, and vibration can be controlled with proper supports. By keeping to a minimum the level of fluctuating stresses in the pipe, failures should not occur.

TABLE 1—PUMP CONSTANTS FOR DETERMINING ACCELERATION HEAD[3]

Constant (C_a)	Type of Pump
0.00181	Single-acting duplex
0.00110	Double-acting duplex
0.00508	Single- or double-acting triplex
0.000362	Single- or double-acting quintuplex
0.000254	Single- or double-acting septuplex

Conclusions

Pressure surges, cavitation, and mechanical vibration prevent water injection systems from operating at maximum efficiency. They cause premature failure of the equipment and excessive operating cost. By using the engineering principles outlined, detrimental effects due to inadequate net positive suction head and poor pipe streamlining can be minimized.

Nomenclature

A = number of pumps on manifold
C_a = a constant that varies with pump types (see Table 1)
C_c = corrosion factor (see Table 2)
D = inside diameter of pipe, in.
D_o = outside diameter of pipe, in.
E_s = (TS /3) C_c (see Table 3)
E = modulus of elasticity (use 30×10^6), psi
F = constant = 0.72 for oil and water lines
$f = F_P \times 4$ = natural frequency of pipe
F_P = pulsation frequency, cycles/sec
H_a = barometric pressure, feet of water
H_{ac} = acceleration head, feet of water
H_f = suction pipe friction loss, feet of water
H_g = minimum pressure required for acceptable pump performance, feet of water
H_s = static head, feet of water
H_v = velocity head, feet of water
H_{vp} = vapor pressure at temperature pumped, feet of water
I = moment of inertia (see Table 4), in.^{4}
J = longitudinal joint factor = 1.0 for seamless and electric resistance welded pipe, and 0.60 for furnace butt welded pipe
L = pipe length, ft
N = number of plungers per pump
p = allowable maximum operating pressure, psi. This value is estimated to be the operating pressure (gauge) + ½ maximum pulsation pressure amplitude
p_A = peak-to-peak pulsation pressure amplitude, psi

TABLE 2—VALUE OF CORROSION FACTOR[11]

(C_c = endurance limit in a specific environment ÷ endurance limit in a completely deaerated salt water.)

Value of C_c	Environment	Corrosive Action
1.0	Air	Normal air corrosion
1.0	Salt water, deaerated	Very mild corrosion
	Salt water saturated with:	
0.80	H_2S	Cracking and hydrogen blister
0.59	CO_2	Multiple surface cracking
0.59	CO_2 + air*	Excessive surface cracking
0.52	H_2S + air**	Pitting, severe corrosion, cracking
0.38	H_2S + CO_2†	Pitting and surface cracking
0.35	Aerated salt water‡	Severe corrosion pitting

*Gas flow: 20 cc/min CO_2, 15 cc/min air (estimated: 500 ppm CO_2, 3 ppm O)
**Gas flow: 20 cc/min H_2S, 10 cc/min air (estimated. 2,000 ppm H_2S, 3 ppm O)
†70 ppm H_2S, 250-300 ppm HCO_2, 700-800 ppm CO_2
‡7-8 ppm O

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TABLE 3—VALUE OF YS × F AND VALUE OF E_s.[11]

Line Pipe Grade	Minimum Tensile Strength (psi)	Minimum Yield Strength (psi)	Value of YS × F, when:		Value of E_s,* when:					
			$F = 0.72$	$F = 0.50$	$C = 1.0$	$C = 0.80$	$C = 0.59$	$C = 0.52$	$C = 0.38$	$C = 0.35$
A	48,000	30,000	21,600	15,000	16,000	12,800	9,400	8,300	6,100	5,700
B	60,000	35,000	25,200	17,500	20,000	16,000	11,800	10,400	7,600	7,000
X 42	60,000	42,000	30,200	21,000	20,000	16,000	11,800	10,400	7,600	7,000
X 46	63,000	46,000	33,100	23,000	21,000	16,800	12,400	10,900	8,000	7,300
X 52	66,000	52,000	37,400	26,000	22,000	27,600	13,000	11,400	8,370	7,700
X 60	75,000	60,000	43,200	30,000	25,000	20,000	14,700	13,000	9,500	8,700
Tubing Grade										
H-40	60,000	40,000	28,800	20,000	20,000	16,000	11,800	10,400	7,600	7,000
J-55	95,000	55,000	68,300	27,500	31,600	25,300	18,700	16,500	12,000	11,100
N-80	100,000	80,000	72,000	40,000	33,333	**	19,600	**	**	11,650

*Where E_s = (TS/3) C, psi
**Not recommended for use in H_2S systems

TABLE 4—DESIGN PROPERTIES OF PIPE

Nominal Size (in.)	Schedule 40		Schedule 80		Schedule 160	
	Weight (lb/ft)	Moment of Inertia (in.⁴)	Weight (lb/ft)	Moment of Inertia (in.⁴)	Weight (lb/ft)	Moment of Inertia (in.⁴)
1	1.679	0.0874	2.172	0.1056	2.844	0.1252
1½	2.718	0.310	3.632	0.391	4.866	0.483
2	3.653	0.666	5.022	0.868	7.445	1.163
3	7.580	3.02	10.25	3.90	14.33	5.04
4	10.79	7.23	14.99	9.61	22.51	13.27
6	18.98	28.1	28.58	40.5	45.30	59.0
8	28.56	72.5	43.40	105.7	74.70	165.9
10	40.50	160.8	64.30	244.8	115.7	399.4
12	53.50	300	88.50	475	160.3	781
14	63.40	429	106.1	687	189.1	1,117
16	82.80	732	136.5	1,157	245.1	1,894
18	104.7	1,171	170.8	1,834	308.5	3,020
20	122.9	1,704	208.9	2,772	379.1	4,585
24	171.1	3,420	296.4	5,670	542.0	9,455

p_o = design operating pressure (gauge), psi
Q = pump flow rate, gal/min

$$R = \text{stress ratio} = \frac{p_o - \frac{1}{2} p_A}{p_o + \frac{1}{2} p_A}$$

S = pump speed, rpm
S_A = allowable stress, psi
t = new pipe wall thickness, inches, minus the corrosion allowance and thread or groove depth, inches; i.e., t = new pipe wall thickness − (project life, years) × (corrosion rate, MPY) × 1,000
T_f = temperature operating factors: 1.0 for −20° to 250°F; 0.967 for 300°F; 0.930 for 350°F; 0.900 for 400°F; etc.
TS = minimum tensile strength, psi
YS = minimum yield strength, psi
V = volume
W = weight per foot of pipe (see Table 4), lb
α = coefficient that varies with end support condition and piping configuration (Fig. 9).

FREQUENCY COEFFICIENT FOR STRAIGHT SECTIONS OF PIPE

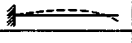
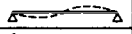
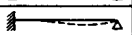
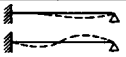
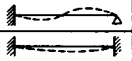
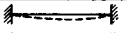
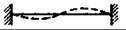
SUPPORT	MODE OF VIBRATION		α
CANTILEVER	FUNDAMENTAL (1st.)		0.265
	SECOND MODE		1.66
BOTH ENDS SIMPLY SUPPORTED	FUNDAMENTAL (1st.)		0.743
	SECOND MODE		2.97
ONE END SIMPLY SUPPORTED, ONE FIX.	FUNDAMENTAL (1st.)		1.16
	SECOND MODE		3.76
BOTH ENDS FIXED	FUNDAMENTAL (1st.)		1.69
	SECOND MODE		4.64

Fig. 9—Frequency coefficients for straight sections of pipe.

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