SECTION 3.6

DISPLACEMENT PUMP PERFORMANCE, INSTRUMENTATION, AND DIAGNOSTICS

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The most common operational and reliability problems in reciprocating positive displacement pump systems are characterized by

- Low net positive suction head (NPSH), pulsations, pressure surge, cavitation, waterhammer
- Vibrations of pump or piping
- Mechanical failures, wear, erosion, alignment
- High horsepower requirements, high motor current, torsional oscillations
- Temperature extremes, thermal cycling
- Harsh liquids: corrosive, caustic, colloidal suspensions, precipitates (slurries)

While any component in a pump system may be defective, most operational problems are caused by liquid transient interaction of the piping system and pump system or by purely mechanical interaction of the pump, drive system, foundation, etc. This section discusses hydraulic and mechanical problems and suggests measurement and diagnostic procedures for determining the sources of these problems.

HYDRAULIC AND MECHANICAL BUMP PROBLEMS

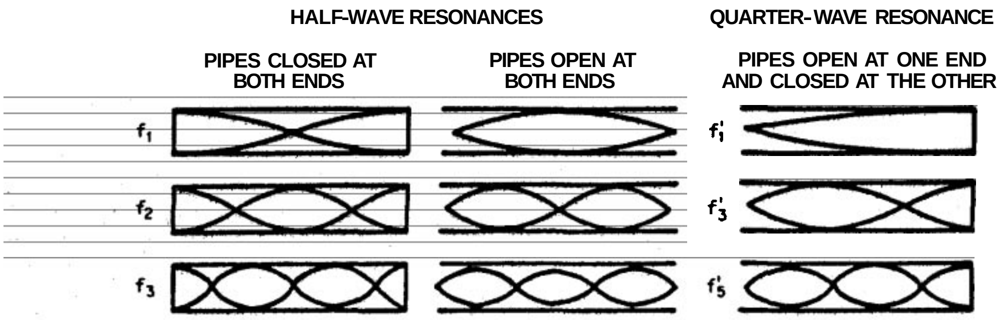
Inadequate NPSH Net positive suction head available (NPSHA) is the static head plus atmospheric head minus lift loss, frictional loss, vapor pressure, and acceleration head available at the suction connection centerline.

Acceleration head can be the highest factor of NPSHA. In some cases it is 10 times the total of all the other losses. Data from both the pump and the suction system are required to determine acceleration head; its value cannot be calculated until these data have been established. Inadequate NPSH can cause cavitation, the rapid collapse of vapor bubbles, which can result in a variety of pump problems, including noise, vibration, loss of head and capacity, and severe erosion of the valves and surfaces in the adjacent inlet areas. To avoid cavitation of liquid in the pump or piping, the absolute liquid static pressure at pumping temperatures must always exceed the vapor pressure of the liquid. The pressure at the pump suction should include sufficient margin to allow for the presence of pulsations as well as pressure losses due to flow.

Positive Displacement Pump Pulsations The intermittent flow of a liquid through pump internal valves generates liquid pulsations at integral multiples of the pump operating speed. For example, a 120-rpm triplex pump generates pulsations at all multiples of pump speed (2 Hz, 4 Hz, etc.); however, the most significant components will usually be multiples of the number of plungers (6 Hz, 12 Hz, 18 Hz, etc.). Resultant pulsation pressures in the piping system are determined by the interaction of the generated pulsation spectrum from the pump and the acoustic length resonances of liquid in the piping. For variable-speed units, the discrete frequency components change in frequency as a function of operating speed and the measured amplitude of any pulsation harmonic can vary substantially with changes in the location of the measurement point relative to the pressure nodes and antinodes of the standing wave pattern.

Piping System Pulsation Response Since acoustic liquid resonances occur in piping systems of finite length, these resonances will selectively amplify some pulsation frequencies and attenuate others. Resonances of individual piping segments can be described from organ pipe acoustic theory. The resonant frequencies of standing pressure waves depend upon the velocity of sound in the liquid being pumped, pipe length, and end conditions. The equations for calculating these frequencies are shown in Fig. 1. All of the integral multiples (N) of a resonance can occur, and it is desirable to mismatch the excitation frequencies from any acoustical resonances. A 2:1 diameter increase or greater would represent an open end for the smaller pipe. Closed valves, pumps, or a 2:1 diameter reduction represent closed ends. For example, a 2-in (51-mm) diameter pipe which connects radially into two 8-in (203-rnm) diameter volumes would respond acoustically as an open-end pipe.

Complex piping system responses depend upon the termination impedances and interaction



$$f = \frac{Na}{2L} \qquad\qquad f = \frac{(2N-1)\,a}{4L}$$

where a = velocity of sound, ft/s (m/s)
L = length, ft (m)
N = 1,2,3 ...

FIG. 1 Organ pipe resonant mode shapes.

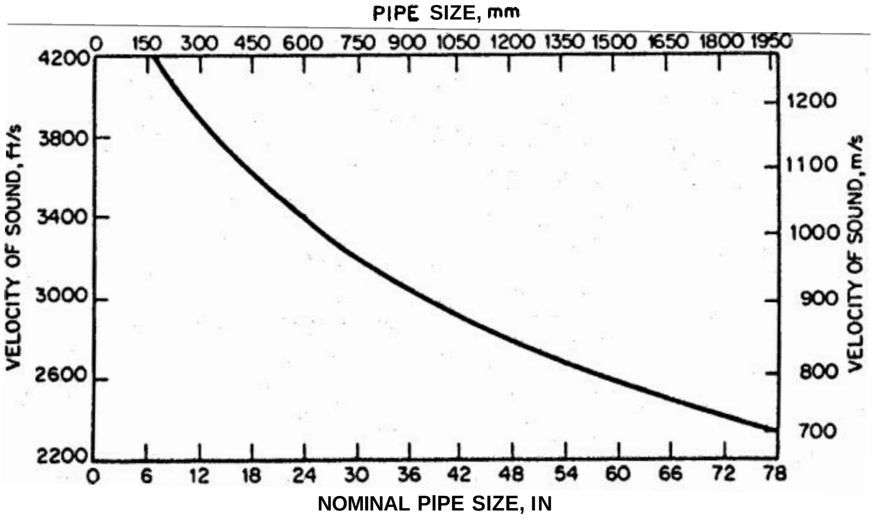


FIG. 2 **Velocity of sound in water at 14.6 lb/in^2 (1 bar), 60°F (15.6°C) versus nominal pipe size with 0.25-in (6.35-mm) wall thickness.**

of amustical resonances and cannot be handled with simplified equations. An electoacoustic analog[1] or digital computer can be used for the more complex systems.

Velocity of Sound in Liquid Piping Systems The acoustic velocity of liquids can be determined by the following equation:

$$a = C_1\sqrt{\frac{K_s}{\text{sp. gr.}}} \tag{1}$$

where a = velocity of sound, ft/s (m/s)
C_1 = 8.615 for USCS units, 1.0 for SI units
K_s = isentropic bulk modulus, lb/in^2 (kPa)
sp. gr. = specific gravity

In liquid piping systems, the acoustic velocity can be significantly affected by pipe wall flexibility. The acoustic velocity can be adjusted by the following equation:

$$a_{\text{adjusted}} = \sqrt{\frac{1}{1 + \dfrac{DK_s}{tE}}} \tag{2}$$

where D = pipe diameter, in (mm)
t = pipe wall thickness, in (mm)
E = elastic modulus of pipe material, lb/in^2 (kPa)

Pipe wall radial compliance can reduce the velocity of sound in liquid in a pipe as shown in Fig. 2'

The bulk modulus of water can be calculated with the following equation[3] for temperatures from 0 to 212°F (0 to 100°C) and pressures from 0 to 4.4 X 10^4 lb/in^2 (0 to 3 kbar*):

$$K_s = K_0 + 3.4P \tag{3}$$

where K_s = isentropic bulk modulus, kbar
K_0 = constant from Table 1, kbar
P = pressure, kbar
(1 kbar = 10^5 kPa = 14,700 lb/in^2)

***1 bar = 10^5 Pa. For a discussion of bar, *see SI Units—A Commentary* in the front matter.**

TABLE 1 Constant K_0 for Evaluation of Isentropic Bulk Modulus of Water from 0 to 3 kbar

$$K_s = K_0 + 3.4P$$

Temperature. °C (°F)	Isentropic constant K_0, kbar[a]
0 (32)	19.7
10 (50)	21.0
20 (68)	22.0
30 (86)	22.7
40 (104)	23.2
50 (122)	23.5
60 (140)	257
70 (158)	23.7
80 (176)	23.5
90 (194)	23.3
100 (212)	22.9

[a] 1 kbar = 14,700 lb/in^2.

SOURCE: Ref.3.

The calculation of the isentropic bulk modulus of water is accurate to ±0.5% at 68°F (20°C) and lower pressures.[4] At elevated pressures (greater than 3 kbar) and temperatures (greater than 100°C), the error should not exceed ±3%.

The bulk modulus for petroleum oils (hydraulic fluids) can be obtained at various temperatures and pressures by using Figs. 3 and 4, which were developed by the American Petroleum Institute

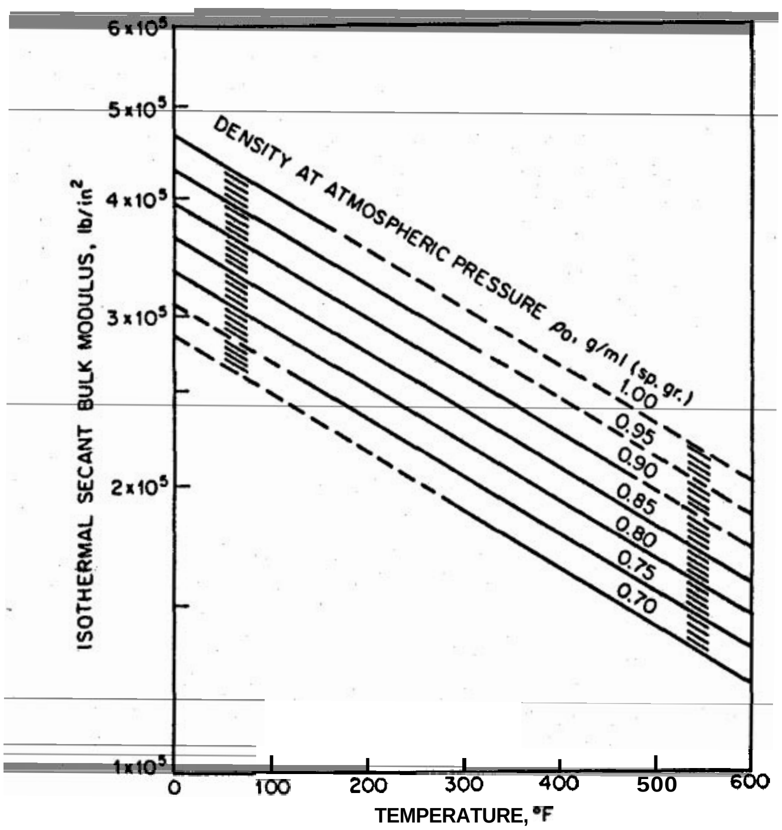


FIG. 3 Isothermal secant bulk modulus at 20,000 lb/in^2 gage for petroleum oils. (1 lb/in^2 = 6.895 kPa; °C = (°F − 32)/1.8)

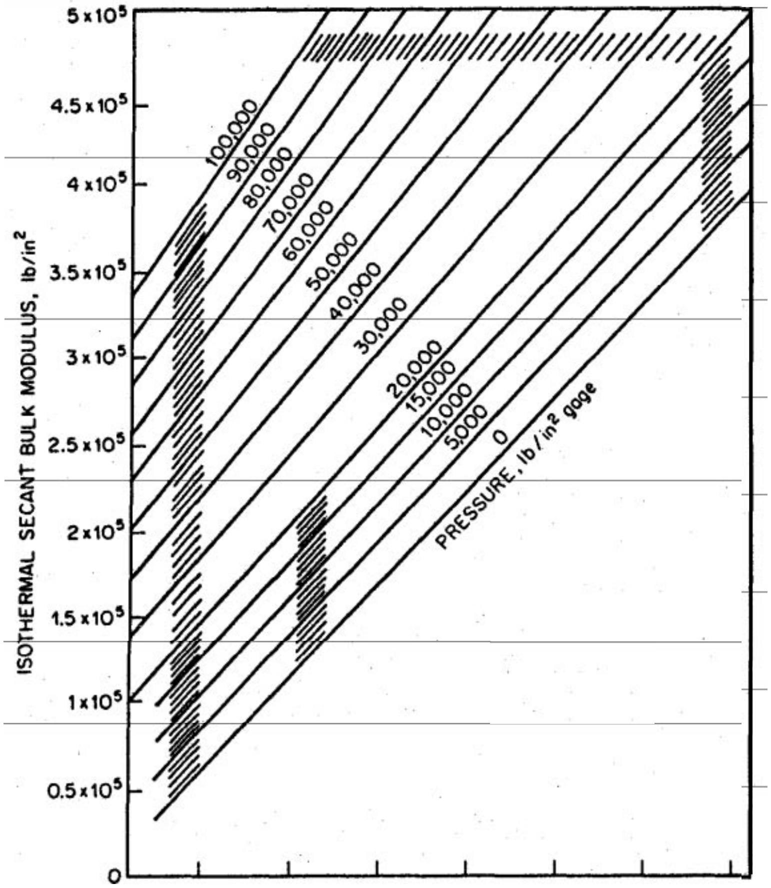


FIG. 4 Pressure correction for isothermal secant bulk modulus for petroleum oils. (1 lb/in² = 6.895 kPa)

(API).[4-7] Figure 3 relates density (mass per unit volume) and temperature to the isothermal secant bulk modulus at 20,000 lb/in² (137,900 kPa), and Fig. 4 corrects for various pressures.

The isentropic tangent bulk modulus is needed to calculate the speed of sound in hydraulic fluids and can be readily obtained from Figs. 3 and 4 as follows:

1. Read the isothermal secant bulk modulus at the desired temperature from Fig. 3.
2. Using the value for isothermal secant bulk modulus obtained from Fig. 3, go to Fig. 4 and locate the intersection of the pressure line with that value. Move vertically to the pressure line representing twice the normal pressure. Read the adjusted isothermal secant bulk modulus for the double value of pressure.
3. Multiply the adjusted isothermal secant bulk modulus by 1.15 to obtain the value of the isentropic tangent bulk modulus (compensation for the ratio of specific heats).

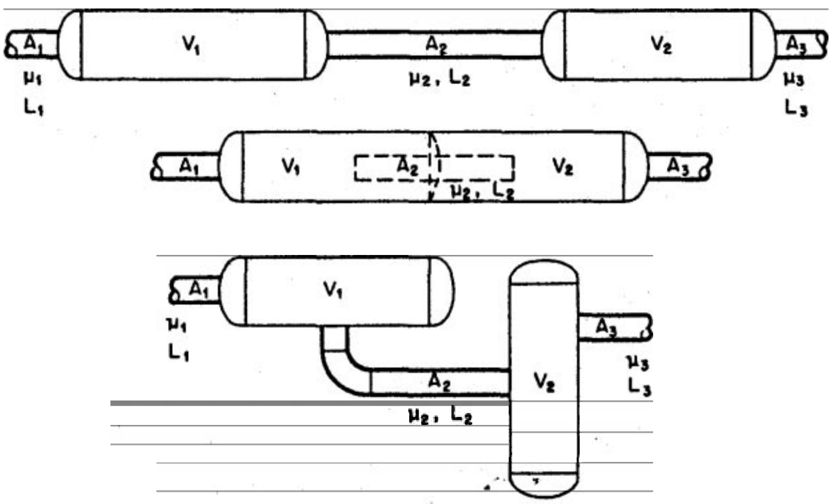
The isothermal tangent bulk modulus has been shown to be approximately equal to the secant bulk modulus at twice the pressure[5] within ±1%. The relationship between isothermal bulk modulus K_t and isentropic bulk modulus K_s is

$$K_s = K_t \overline{c_p/c_v} \tag{4}$$

The value of $\overline{c_p/c_v}$ for most hydraulic fluids is approximately 1.15.

PULSATION CONTROL

Pulsation control can be achieved by judicious use of acoustic filters and side branch accumulators. Acoustic filters are liquid-filled devices consisting of volumes and chokes which use reactive fil-



$$\mu_j = \frac{A}{L_j + 1/2\sqrt{\pi A_j}} \quad \text{for } j = 1, 2, 3$$

$$f = \frac{a}{2\pi}\sqrt{\frac{1}{2}\left[\frac{\mu_1 + \mu_2}{V_1} + \frac{\mu_2 + \mu_3}{V_2} \pm \sqrt{\left(\frac{\mu_1 + \mu_2}{V_1} - \frac{\mu_2 + \mu_3}{V_2}\right)^2 + \frac{4\mu_2^2}{V_1 V_2}}\right]}$$

FOR EQUAL VOLUMES, THE RESONANT FREQUENCY IS APPROXIMATELY:

$$f = \frac{a}{\sqrt{2}\,\pi}\sqrt{\frac{\mu_2}{V_1}}$$

FIG. 5 Two-chamber resonator system with both ends open. V = volume, ft^3 (m^3); f = resonant frequency. Hz; L = choke tube length, ft (m); A = choke tube area, ft^2 (m^2); a = acoustic velocity, ft/s (m/s); μ = acoustic parameter.

tering techniques to attenuate pulsations. Side branch resonators are of two types: quarter-wave. length resonant stubs and gas-charged accumulators. (The terms accumulators, dampeners, and dampers are used interchangeably in the liquid filter industry.)

Acoustic Filters An acoustic filter consisting of two volumes connected by a small-diameter choke can significantly reduce the transmission of pulsations from the pump into the suction and discharge piping systems. The equations given in Fig. 5 can be used to calculate the resonant frequency for a simple volume-choke-volume filter. The filter should be designed to have a resonant frequency no more than one-half the lowest frequency desired to be reduced, referred to as the cutoff frequency. Such a filter is called a low-pass filter since it attenuates frequencies above the cutoff frequency.

A special case for symmetric liquid-filled filters can be obtained by choosing equal chamber and choke lengths. This reduces the equation in Fig. 5 to

$$f = \frac{ad}{\pi\sqrt{2LD}} \tag{5}$$

where d = choke diameter, in (mm)
L = chamber and choke length, ft (m)
D = chamber diameter, in (mm)

Normally, a good filter design will have a resonant frequency less than one-half the plunger frequency and will have a minimal pressure drop. For example, a triplex pump running at 600 rpm generates pulsations at all multiples of 10 Hz. The largest amplitudes would normally be at 30 Hz, 60 Hz, 90 Hz, etc. The filter resonant frequency should be set at 15 Hz or lower. For water, in which the velocity of sound is 3200 ft/s (976 m/s), and a volume bottle size inner diameter of 19 in (482.6 mm),

in USCS units
$$\frac{L}{d} = \frac{3200}{\pi\sqrt{2}\,(15)(19)} = 2.53\,\frac{\text{ft}}{\text{in}} \tag{6}$$

in SI units
$$\frac{L}{d} = \frac{976}{\pi\sqrt{2}\,(15)(482.6)} = 0.03\,\frac{\text{m}}{\text{mm}}$$

If the choke diameter is selected to be 1.049 in (26.64 mm), then the length of each volume bottle and choke tube is 2.65 ft (0.81 m). See also Ref. 8.

Side Branch Accumulators Liquid-filled, quarter-wavelength, side branch accumulators reduce pulsations in a narrow frequency band and can be effective on constant-speed positive displacement pumps. However, in variable-speed systems, accumulators with or without a bladder can be made more effective by partially charging them with a gas (nitrogen or air) since the gas charge cushions hydraulic shocks and pulsations. If properly selected, located, tuned, and charged, a wide variety of accumulators (weight- or spring-loaded; gas-charged) can be used in positive displacement pump systems to prevent cavitation and waterhammer, damp pulsations, and reduce pressure surges.''.'' Improper sizing or location can aggravate existing problems or cause additional ones. Typically, the best location for accumulators is as close to the pump as possible.

Gas-charged dampeners, or accumulators, such as those depicted in Fig. 6, are most commonly used and can be quite effective in controlling pulsations. These devices are commercially available from several sources. Important to their effectiveness are their location and volume and the pressure of the charge. When gas-charged dampeners are used, the gas pressure must be monitored and maintained since the gas can be absorbed into the liquid. The system pressure can sometimes be lower than the gas charging pressure, such as on start-up; therefore, a valve should be installed to shut off the accumulator during start-up to eliminate gas leakage to the primary liquid. When the valve is closed, the accumulator is decoupled from the system and is not effective. Accumulators with integral check valves should be adjusted so that pressure transients do not close the check valve and render the accumulator ineffective. Accumulators which have bladders (Figs. 6b, d, and e) to separate the gas charge from the liquid have some distinct advantages, particularly if gas absorption is a problem. Accumulators with flexible bladders must be carefully maintained since failure of a bladder could release gas into the liquid system and could compromise the effectiveness of the dampener.

The in-line gas dampener (Fig. 6e) has a cylinder around the pipe containing a gas volume and bladder. The liquid enters the dampener through small holes in the circumference of the pipe and impinges upon the bladder, which produces the same acoustic effect as a side branch configuration.

It is not always possible to design effective pulsation control systems using simplified techniques. For complicated piping systems with multiple pumps, an electroacoustic analog' is recommended for designing optimum filters or accumulators. This tool has become widely accepted for designing reliable piping systems for reciprocating liquid pumps and gas compressor units. The reduction in pulsations in a liquid pump system in a nuclear plant is shown in Fig. 7. These results were obtained with a two-volume acoustic filter system designed with the electroacoustic analog. The volume diameter was 19.3 in (49 cm), and the length was 4 ft (1.2 m). The choke tube diameter was 0.8 in (2 cm), and its length was 7 ft (2.1 m). The speed of the triplex (2-cm) pump was 360 rpm, and the filter resonant frequency was set at 8.1 Hz for a velocity of sound of 4550 ft/s (1390 m/s).

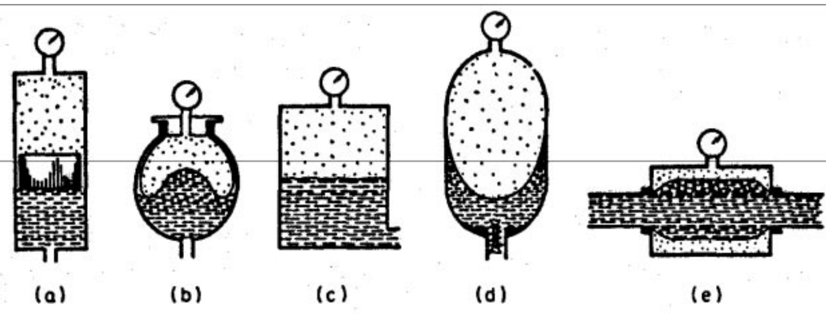
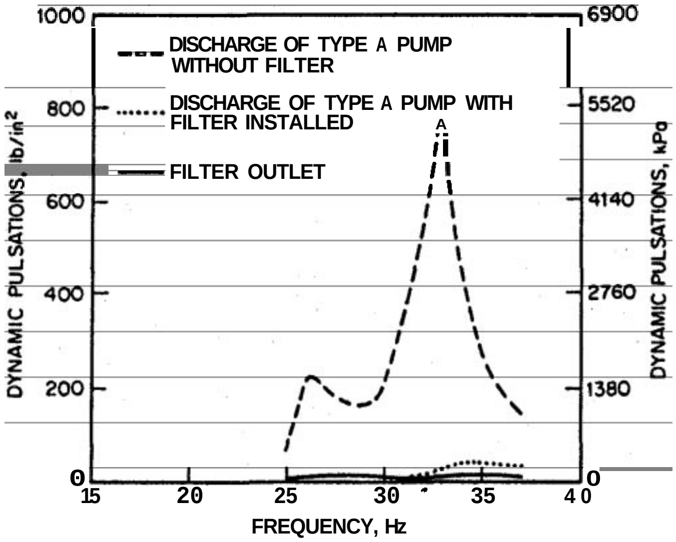


FIG. 6 Types of accumulators: (a) piston, (*b*) diaphragm, (*c*) gas-charged, (*d*) bladder, (*e*) in-line.



PIC. 7 Effect of acoustic filter on pulsations.

PIPING VIBRATIONS

When mechanical resonances are excited by pulsations, vibrations in the pump and piping can sometimes be 20 times higher than under off-resonant conditions. When the mechanical resonances coincide with the acoustic resonances, an additional amplification factor as high as **300** can **be** encountered.

Piping system mechanical natural frequencies can be calculated using simplified design procedures to provide effective detuning from known excitation sources. A nomogram (Fig. 8) for calculating the lowest natural frequency of uniform steel piping spans'' can be used in designing piping systems and in diagnosing and solving vibration problems. For example, welded between two bottles, a 4-in (102-mm) pipe that is 10 ft (3 m) long and has an inner diameter of 3.826 in (97.18 mm) would have a mechanical frequency of 74 Hz. If the pipe was an equal-leg L bend (L = 5), the natural frequency would be 50 Hz.

To minimize piping vibration problems, all unnecessary bends (considering routing and thermal flexibility)should be eliminated since they provide a strong coupling point between pulsation excitation forces and the mechanical system. When bends are needed, use the largest enclosed angle possible and locate restraints near each bend. Piping should also have supports near all piping size reductions and at large masses (valves, accumulators, flanges, etc.). Small auxiliary piping connections (vents, drains, pressure test connections, etc.) should be designed such that the mass of the valve and flange is effectively tied back to the main piping, thus eliminating relative vibration.

DIAGNOSTICS AND INSTRUMENTATION

The diagnosis of vibrations in positive displacement pumps should usually include dynamic pressure measurements in the cylinders and piping near the pump. These measurements can be obtained by the use of piezoelectric or strain gage pressure transducers. If cavitation or flashing is suspected, a pressure transducer capable of measuring static and large dynamic pressures should be used; even then, cavitation-produced pressure shocks may damage the transducer.

Accelerometers with low frequency characteristics may be used with electronic integrators to obtain accurate vibration displacement data from the pump case, cylinders, or piping. Similarly, velocity or seismic pickups may be employed. Maximum vibrations usually occur at the middle of piping spans and at **unsupported** elbows (out of plane).

Real-time analyzers and oscilloscopes may be used to display the resulting signals, A field

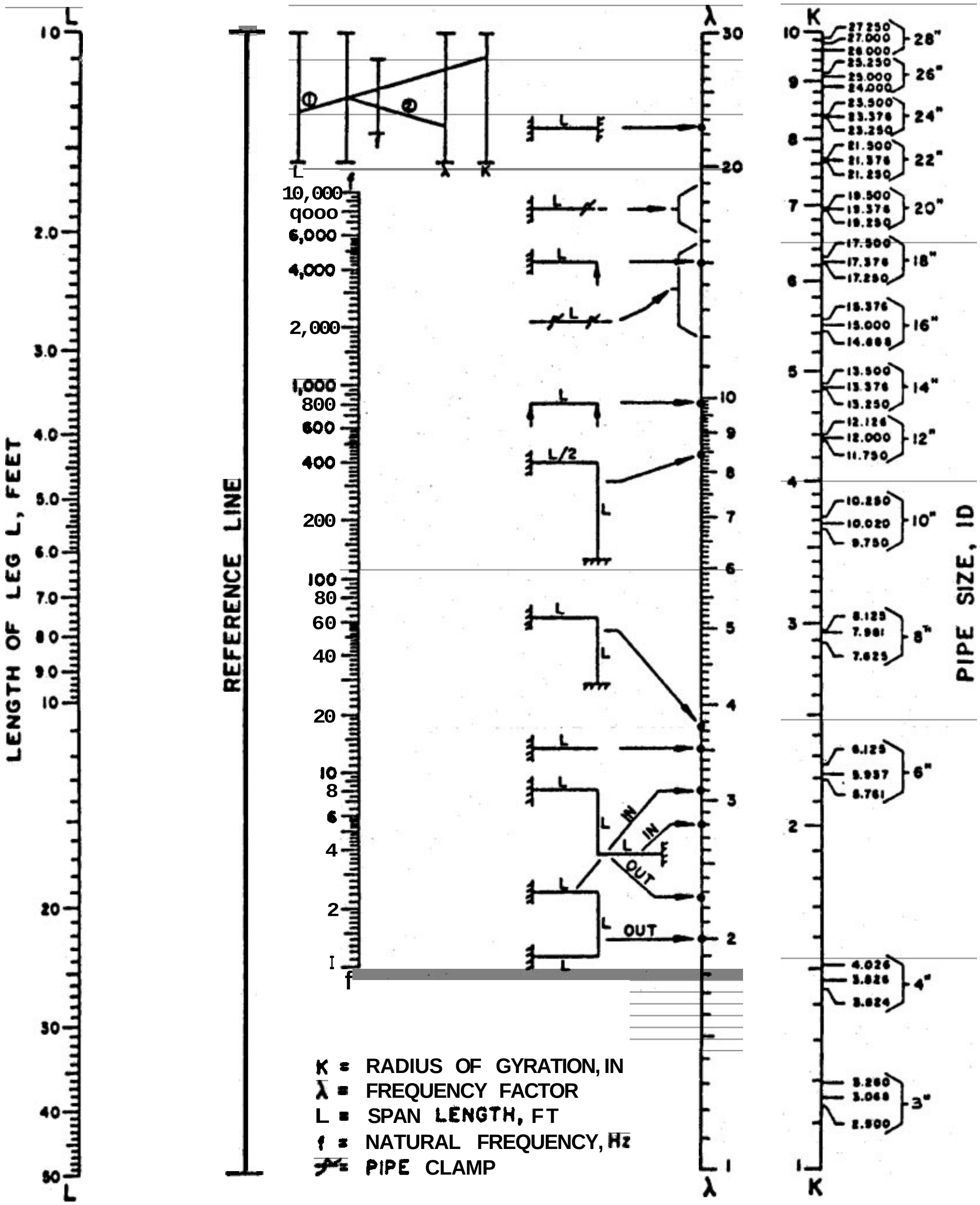


FIG. 8 **Natural frequency of uniform steel piping spans. (1 ft = 0.3048 m; 1 in = 2.54 cm)**

example showing the diagnosis of cavitation at the pump suction using a strain gage diaphragm pressure transducer is given in the oscilloscope trace of Fig. 9. The vapor pressure (gage) for this system was 25 Ib/in^2 (172 kPa). Note that the negative half of the cycle is flattened when vapor pressure is reached and that very high amplitude pressure spikes are apparent. For liquids with dissolved gases, lower pulsation amplitudes can produce cavitation; however, the cavitation is usually less severe.

Typical field pulsation data obtained on a three-plunger pump is shown in Fig. 10, which is a frequency spectrum of the pulsations made by a real-time analyzer. Each spike represents a frequency multiple of running speed. Note that the third spike, representing the plunger frequency, has the largest amplitude; however, the components at one and two times pump speed are also

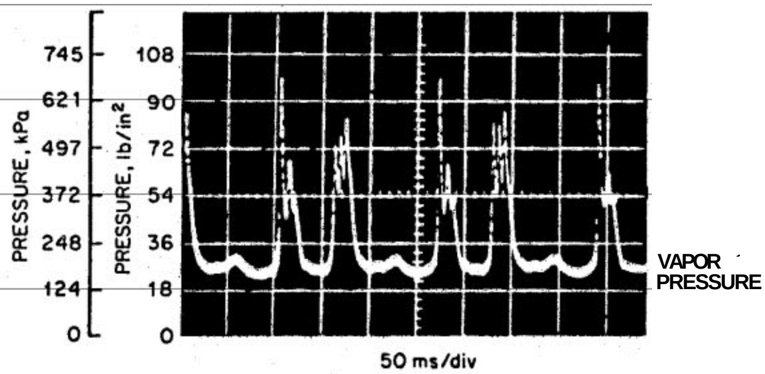


FIG. 9 Complex wave data showing cavitation effects of pressure wave in a liquid piping system.

significant, which means that these frequency components should be considered for pulsation control.

The vibration frequencies of the piping should be compared with pulsation frequencies to evaluate potential pulsation excitation of mechanical resonances. A check of the piping mechanical natural frequencies from the nomogram (Fig. 8) should be made to evaluate the possibility of a mechanical resonance. High vibrations produced by low-level pulsations at a particular frequency are indicative of a mechanical resonance, which can usually be corrected by additional piping restraints or snubbers. Once the causes of the pulsations and vibrations are diagnosed, the techniques presented above can be used to develop solutions.

Thermal Problems In pump systems having high thermal gradients, large forces and moments on the pump case can cause misalignment of the pump and its driver as well as pump case distortion resulting in vibrations, rubbing (wear), bearing failure, seal leakage, etc. High stresses can be imposed on the piping, resulting in local yielding or damage to the piping restraints. snubbers, or support system. Misalignment problems commonly exhibit a high second-order component of the shaft vibrations. Proximity probes can be used at the bearings to measure movement of the shaft relative to the bearing centerlina.

Diagnosis of Shaft Failures Pump and driver shafting can experience high stresses during start-up and normal operation because of the uneven torque loading of the positive displacement pumping action. Shaft failures are strongly influenced by the torsional resonances of the system, which are the angular natural frequencies of the system.

Torsional vibrations can be measured using velocity-type torsional transducers which mount

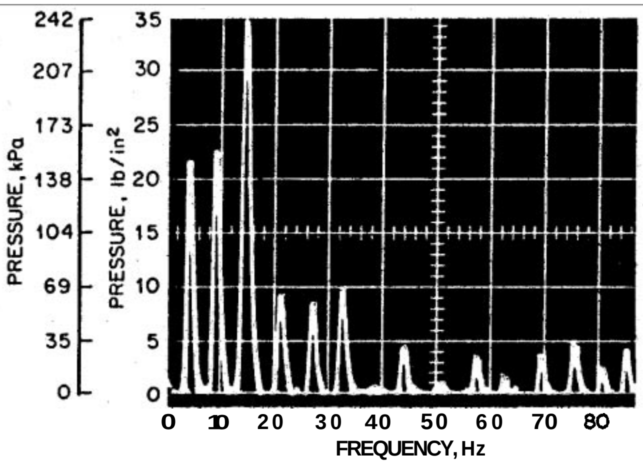


FIG. 10 Typical field data recorded on a three-plunger pump.

on a stub shaft or by measuring the gear tooth passing frequency with a magnetic transducer or proximity probe and using frequency-to-voltage converters to give the change in tooth passing frequency (the torsional vibrational velocity). Spectral analysis of these signals defines the torsional amplitudes and natural frequencies. The stresses can be calculated by using the mode shape of the specific resonant natural frequency and combining all the torsional loads. Torsional natural frequencies, mode shapes, and stresses can be calculated by using either the Holzer technique or digital computer programs.[12]

Torsional problems can usually be solved by changing the coupling stiffness between the driver and pump or by using a flywheel in an effective location. The addition of a flywheel will tend to smooth the torque oscillations. Pumps with a greater number of cylinders and equal cylinder phasing usually operate more smoothly with lower shaft stresses.

REFERENCES

1. Von Nimitz, W. W.: "Reliability and Performance Assurance in the Design of Reciprocating Compressor and Pump Installation," 1974 Purdue Compressor Technology Conference.
2. Sparks, C. R., and J. C. Wachel: "Pulsation in Centrifugal Pump and Piping Systems," Hydrocarbon Processing, July 1977, p. 183.
3. Hayward, A. T. J.: "How to Estimate the Bulk Modulus of Hydraulic Fluids," Hydraulic Pneumatic Power, January 1970, p. 28.
4. Wright, W. A.: "Prediction of Bulk Moduli and Pressure-Volume-Temperature Data for Petroleum Oils, ASLE Transactions 10:349 (1967).
5. API Technical Data Book, Petroleum Refining, 2d ed., Washington, D.C., 1972.
6. Klaus, E. E., and J. A. O'Brien: "Precise Measurement and Prediction of Bulk-Modulus Values for Fluids and Lubricants," Journal of Basic Engineering, September 1964, p. 469.
7. Noonan, J. W., "Ultrasonic Determination of the Bulk Modulus of Hydraulic Fluids," Materials Research and Standards, December 1965, p. 615.
8. Hicks, E. J., and T. R. Grant: "Acoustic Filter Controls Reciprocating Pump Pulsations," Oil and Gas Journal, January 15, 1979, p. 67.
9. Hydraulic Institute Standards for Centrifugal, Rotary and Reciprocating Pumps, 14th ed., 1983.
10. Machine Design, Fluid Power Reference Issue, vol. 52, no. 21, Penton Publications, Cleveland, 1980.
11. Wachel, J. C., and C. L. Bates: "Techniques for Controlling Piping Vibration and Failures," ASME paper 76-Pet-18, 1976.
12. Szenasi, F. R., and L. E. Blodgett: "Isolation of Torsional Vibrations in Rotating Machinery," Proceedings of the National Conference on Power Transmission, vol. II, Illinois Institute of Technology, 1975.

FURTHER READING

Positive Displacement Pumps: Reciprocating, API Standard 674, 1st ed., Washington, D.C., 1980.

Positive Displacement Pumps: Controlled Volume, API Standard 675, 1st ed., Washington, D.C.. 1980.