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Revision Memo

- 12/01** Reciprocating Pumps, pulsation control, modified paragraph to include effect of suction stabilizer on $NPSH_A$; Reciprocating Pumps, Data Sheet Requirements, paragraphs 1 and 3, modified to include latest revision of API and GP requiring Design approach 2 for 50 hp instead of 25, Controlled Volume Pumps, Pressure Pulsations, eliminated redundant heading titles.



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SCOPE

This section presents additional detail and requirements for positive displacement pumps commonly used in process plant services. Positive displacement pumps differ from centrifugal pumps in that they do not rely on imparting energy to the liquid via an increase in velocity. They simply move discrete volumes of the liquid from the suction to the discharge of the pump. As the fluid is pushed into the discharge, it assumes the same pressure as the discharge line to which the pump is connected. The pumps included in this section are:

- Reciprocating, including direct acting and power pumps.
- Controlled volume, including diaphragm and piston pumps.
- Air or other media driven, direct acting diaphragm pumps.
- Rotary, including screw, progressive cavity, gear, and sliding vane pumps.

A description of each pump type, including general application limits and how to select the appropriate type, is presented as the first part of this section. Following this are sections on each pump type describing Design Specification related items.

REFERENCES

DESIGN PRACTICES

(In addition to other parts of [Section X](#))

[Section XV](#), Safety in Plant Design

GLOBAL PRACTICES

- GP 10-2-1 Positive Displacement Pumps, Reciprocating
- GP 10-2-2 Positive Displacement Pumps, Rotary
- GP 10-2-3 Positive Displacement Pumps, Controlled Volume
- GP 3-18-1 Piping Fabrication Shop or Field
- GP 10-11-1 Sizing of Drivers and Transmissions for Compressors, Fans and Pumps

OTHER REFERENCES

API 674, *Positive Displacement Pumps, Reciprocating*

API 675, *Positive Displacement Pumps, Controlled Volume*

API 676, *Positive Displacement Pumps, Rotary*

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Imo Industries Inc., Imo Pump Division, General Catalog GP89-A

DEFINITIONS

Positive Displacement Pump

A type of pump which physically transfers discrete volumes of liquid from the suction region to the discharge region of the pump. Within boundaries of NPSH and viscosity, flow rate is a variable of pump speed only. Unlike centrifugal pumps, the discharge pressure of a positive displacement pump is not affected by the pumped fluid specific gravity. The discharge pressure is only a function of the pressure in the discharge system and the ability of the pump to push the fluid into the discharge. Thus, if the discharge is blocked off, the discharge pressure will rise until the pump stalls or a mechanical failure of the pump or piping occurs. This factor necessitates a pressure relief valve in the discharge of almost all positive displacement pumps.

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DEFINITIONS (Cont)

Reciprocating Pump

A type of pump that has a cylinder or cylinders, each of which has at least one inlet and one outlet valve and which uses a piston or plunger moving in a linear motion in the cylinder to move the pumped fluid. On the suction stroke, the pressure in the cylinder is lowered to below that of the suction system allowing liquid to flow into the cylinder. On the discharge portion of the stroke the piston or plunger pushes the liquid out through the discharge valve and into the system at a pressure equal to or greater than the discharge line pressure.

Piston and Piston Rod

The component of a reciprocating pump that moves linearly within the cylinder to alternately lower the cylinder pressure and then push the liquid out through the discharge valve. The diameter of the piston is equal to diameter of the cylinder and is many times larger than the packing used to seal the liquid in the cylinder. The piston rod connects the piston to the driving mechanism and moves the piston. The packing used to seal the liquid in the cylinder is the same diameter as the piston rod.

Plunger

The component of a reciprocating pump that moves linearly within the cylinder to alternately lower the cylinder pressure and then push the liquid out through the discharge valve. The diameter of the plunger is equal to diameter of the packing that seals the liquid in the cylinder. Plungers provide substantially increased resistance to bending when put in compression as opposed to the rod of a piston and, therefore, are used at higher pressures.

Rotary Pump

A type of positive displacement pump without inlet and outlet valves that uses a rotating shaft which has some geometry or features such as gears, vanes or integrally meshing screw rotors to trap a finite volume of liquid at the suction condition and transport it to the discharge where it is physically pushed into the discharge line with little or no pulsation.

Slip

Leakage flow within a rotary positive displacement pump from the discharge back to the suction caused by the clearances needed between rotating and stationary components. Slip may also occur in a double acting piston type reciprocating pump.

Controlled Volume Pump

A type of reciprocating pump that provides very precise control of the amount of fluid pumped through control of the stroke length of the diaphragm or plunger.

Diaphragm Pump

A pump which provides pumping action through flexure of a metallic or non-metallic component known as a diaphragm via alternating pressure from a hydraulic drive, a piston or flexure of a drive diaphragm driven usually by air or nitrogen.

Pulsation

The variation of pressure in a line due to the flow variations caused by a piston(s), plunger(s) or diaphragm(s) which are creating a pumping action. Percent Residual Pulsation is customarily used to define existing or allowable levels.

ΔP = Maximum Pressure in Line - Minimum Pressure in Line

Percent Residual Pulsation = $\Delta P \times 100 / \text{Average Pressure in the Line}$

Self-Priming

The ability for a pump to "draw" liquid into itself and start pumping liquid by evacuating the air or vapor in the suction of the pump and suction line of the pump. Positive displacement pumps are inherently self-priming, since they will displace either vapor or liquid. However, they will not self-prime when their suction lift capability has been exceeded. Centrifugal pumps are inherently NOT self-priming because they do not have the ability to "pump" the air or vapor out of their inlets to draw in liquid.

Suction Lift

The term used to describe the amount of negative pressure that a pump can overcome to begin pumping fluid without having been primed (e.g., without there having been liquid in the pump to begin with). Suction lift is measured in feet of head or inches of mercury. Referring to the equations in [Section X-D](#), NPSH, suction lift is the distance in feet from the surface of the liquid when the liquid surface is BELOW the pump inlet.

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GENERAL INSTRUCTIONS AND INFORMATION (GII) CONTENT

Because positive displacement pumps are used as infrequently as possible and when they are applied, the services are generally unique, no specific items are required in the GII. Notes and entries for each specific pump type will be added to the individual pump data sheets as necessary.

LIST OF DESIGN SPECIFICATION REQUIREMENTS

The following Design Specification requirements are needed for positive displacement pumps. Details are provided in later in this section on each type of pump.

- Pump type.
- Flow and flow range.
- Method of flow variation and recycle system design (if recycle is used).
- Pulsation Control design methodology note (reciprocating, metering and diaphragm pumps only).
- Preferred pulsation control device type (reciprocating, metering and diaphragm pumps only).
- Dynamic valve simulation note (Reciprocating pumps only).
- NPSH_A and acceleration head loss note (needed for reciprocating, metering and diaphragm pumps).
- Packing type note (Reciprocating pump only).
- Material Class or materials.
- Self or external flush.
- Utilities requirements.
- Flow plan notes and review items.

PUMP TYPE SELECTION

Selection of a centrifugal versus positive displacement pump is addressed in [Design Practices Section X-C, Pump Types](#). Positive displacement pumps are generally used only when the viscosity is above 200 cSt, when the required head can vary significantly from the design point or when accurate control of flowrate (metering) is needed. If a positive displacement pump has been specified for a service, use the following guidelines to select the type of pump to be used:

Direct Acting Steam Pumps - Generally used for relatively low pressure (below 500 psig/34 barg) services with highly variable fluid properties or high suction lift/run dry service. Examples would be unloading (run dry service), flare drum boot pumpout, slop pumpout.

Reciprocating Power Pumps - Generally used for pressures above 1200 psig (82 barg), almost regardless of fluid conditions including flow rate (multiple pumps are used), temperature (suitable to 750°F/398°C) and solids/viscosity level. Some typical applications are Urea Plant carbamate and ammonia feed services, steam cracker fractionator tar bottoms (alternate standby pump for high viscosity flexibility), high pressure water injection/water blasting or cutting, crude oil pipeline transmission.

Diaphragm Metering Pumps - Used for almost any pressure at generally low flow (up to 50 gpm/11 m³/hr) where the flow rate must be accurately controlled.

AOP Diaphragm Pumps - Air Operated Diaphragm pumps are becoming a substitute for steam pumps in low pressure (125 psig/8.6 barg) services especially for hydrocarbons since they are sealless. Recent applications include portable spill clean up pumps, flare boot pumps, marine cargo unloading hose scavenge pumps.

Timed Rotary Shaft Screw Pumps - Used for services requiring pressure up to 1200 psig (82 barg) and about 8000 gpm (1816 m³/hr) on fluids having high and or variable viscosity or other properties. Typical applications are pipeline transmission, hot bottoms, pitch and tar.

Untimed Rotary Shaft Screw Pumps and Gear Pumps - Used primarily for clean services up to 4500 psig (310 barg) where viscosity is above 200 cSt. Loading, unloading, fuel oil, lube oil and other clean applications are typical.

Rotary Vane Pumps - Mostly loading and transfer applications below 200 psig (14 barg) and special applications such as wax boot pumps.

Rotary Gear Pumps - Primarily used for clean oil and fuel services including high viscosity services.

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RECIPROCATING PUMPS

DESCRIPTION

Direct Acting Pumps

The first of the two reciprocating pump styles is the direct acting pump **Figure 1**. On one end of the pump is a liquid cylinder with a piston and valves (inlet and outlet) and on the other end is a power cylinder with a piston and inlet and outlet valves. A rod connects the two pistons directly thus giving rise to the name direct acting pump or rod pump. On the rod, outside of the pistons are linkages and levers to alternately open and close the power end inlet and outlet valves. Almost any gas or vapor may be used to provide power but steam is almost always used and will be the only motive power discussed in this section.

Direct acting steam pumps are arranged as "simplex", having one steam cylinder and one liquid cylinder end or more commonly as "duplex" with two parallel sets of steam and pumped fluid cylinders. Whether the pump is simplex or duplex, the liquid end is usually "double acting" meaning there is a chamber for pumping on each side of the liquid end piston. Liquid is pumped whether the rod is moving toward or away from the steam cylinder.

In a duplex steam pump the pistons are timed by the linkages on the rods so they are 180° out of phase. Thus, the duplex pump has four pumping actions per cycle. Although there are more pressure pulsations (e.g., for a given pump speed there are more pressure spikes in the duplex pump), the magnitude of the pulsations is much lower since the instantaneous flow rate does not change as much as in a simplex pump. Direct acting pumps will require some form of pulsation suppression device on both the suction and discharge. See the paragraphs later in this Design Practice related to PULSATION CONTROL for additional information.

Power Pumps

The second type of reciprocating pump is the power pump, **Figure 2**. In this pump the rotary motion of a shaft is translated into linear motion through a crankshaft, connecting rods and crossheads. The crossheads drive the plungers or pistons inside the cylinders which have an inlet and outlet valve much like the steam pump. Plungers are commonly used for high pressure applications since they provide a "stiffer" design and less lateral rod motion in the stuffing box area while pistons may be used for lower pressure. Piston type power pumps are rarely used in process plants and will not be covered further in this section.

Up to about 350 hp (270 kW) plunger power pumps are usually made in triplex (meaning 3 cylinder) or quintuplex (meaning 5 cylinder) configurations mounted horizontally. They are typically driven by a motor through pulleys and belts giving a speed reduction. At higher power levels, plunger pumps are oriented with the plunger vertical and the crankcase below the liquid end. This eliminates the weight of the plungers (which can be considerable) from the guide bushings in the liquid end. Vertical plunger pumps are typically supplied with 3, 5, 7, or 9 plungers and can be rated as high as 1500 hp (1118 kW). They are typically driven by an electric motor through an external gear or gear which is integral with the crankcase.

Power pumps are usually applied to high-pressure services over 1200 psi (83 barg). They are costly to purchase and maintain making them less desirable than centrifugal pumps whenever a centrifugal can be used. However, they are frequently more cost effective than a direct acting steam pump due to higher efficiency. Their drawback compared to steam driven pumps is the difficulty in achieving variable flow capability and generally higher NPSH required.

FLOW AND FLOW VARIATION

Simplex, direct acting steam pumps are commercially available in sizes up to 400 gal/min (90.8 m³/hr). Models are claimed to be available at higher flow rates but consultation with Machinery Specialists is advised. Specialists should also be consulted when the flow approaches (within 30%) the upper limits stated above. The actual flow rate of a direct acting pump is directly influenced by the speed. Maximum allowable speed for any given size cylinder and stroke is fixed by **GP 10-2-1**. These speeds are then adjusted downward by factors accounting for the size of the pump, NPSH margin, temperature, pressure differential and viscosity.

Capacity control of direct acting steam pumps is accomplished via changes in the pump speed by throttling the inlet steam valve. This is amenable to automatic control and virtually any flow range from zero to maximum can be achieved. The unique feature of a steam pump which sometimes makes it an attractive choice is that reductions to the pump speed also significantly reduce the pulsation levels and required NPSH.

Power Pumps are generally available in sizes ranging up to 1500 gpm (340 m³/hr). As with direct acting pumps, their speeds are limited by **GP 10-2-1** based on stroke, pressure differential, viscosity, temperature and NPSH margin. Consultation with Machinery Specialists is advised whenever the required flow exceeds 200 gpm (45 m³/hr).

Capacity control of power pumps is more difficult than for direct acting pumps since the driver is usually an electric motor. Flow changes can be accomplished by recycle or installation of variable speed drivers. Recycle systems should be designed for the maximum required flow of the pump (if the maximum is greater than the normal flow) assuming 100% efficiency in the pump.

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RECIPROCATING PUMPS (Cont)

Recycle systems are prone to these drawbacks:

1. The bypass valve operates with a high ΔP tending to have a small opening which is prone to plugging and erosion.
2. Flashing of the liquid may occur due to the high-pressure drop of the recycle valve.

To alleviate these concerns a variable speed driver such as a variable speed electric motor or variable speed fluid coupling should be considered. Flow variations in the range of 50% to 100% may be easily attained. In services where solids are present, variable speed should always be used instead of a recycle system. This prevents erosion damage to the recycle valve and minimizes pump speed helping to improve life.

PRESSURE AND TEMPERATURE

Simplex steam pumps are available for pumping services to 500 psig (34.5 barg) and 500°F (260°C). Models are claimed available at higher pressure and temperature but Machinery Specialists should be contacted due to the special nature of the pump. Duplex pumps are available up to 300 psig (21 barg) and 250°F (121°C).

Power pumps are found in services with differential pressures from 1000 psi to 30,000 psi (69 bar to 2069 bar) though they tend to develop lower flows as the pressure is increased. Power pumps have been used (at the Exxon Coal Liquefaction Pilot Plant, for example) up to 750°F (400°C) though they are typically found in services below about 250°F (121°C) due to limitations on packing materials.

MAXIMUM DISCHARGE PRESSURE AND PRESSURE RELIEF

Unlike centrifugal pumps, the maximum discharge pressure available from a reciprocating pump is not limited by the performance curve pressure rise to shutoff and the discharge pressure is not affected by the fluid specific gravity. The maximum pressure that can be achieved is related only to the ability of the pump to push the liquid into the discharge before the pump runs out of power (stalls) or something breaks.

For a direct acting steam pump:

$$\text{Maximum Pressure} = \text{Maximum Suction Pressure} + \frac{(\text{Steam Cylinder Diameter})^2}{(\text{Liquid Cylinder Diameter})^2} \times \text{Maximum Steam Differential Pressure}$$

Since steam cylinders are normally selected one standard size larger than the liquid end, the maximum pressure is generally 130% to 150% of the **rated differential pressure plus maximum suction pressure**. This should be accounted for in the design by including a pressure relief valve in the discharge piping as noted below.

For power pumps, the maximum discharge pressure attainable is a function of the maximum motor torque, stroke length and number of cylinders. There are no general rules on the maximum pressure related to the rated differential pressure.

For either a direct acting pump or power pump, since the pump is capable of generating pressure well in excess of that normally required, a Pressure Relief Valve (PRV) should always be used in the pump discharge line. See the subsection below entitled **OVERPRESSURE PROTECTION** for additional information.

SOURCES OF SUCTION AND DISCHARGE PRESSURE PULSATION

Reciprocating pumps are well known for their ability to generate pressure pulsations, induce piping vibration and make control systems and instrumentation inoperable. The key to obtaining a reliable, minimum vibration installation (vibration will never be eliminated) is to understand the characteristics of the pulsations generated and how they interact with the piping and other systems connected to the pump.

All reciprocating pumps generate 3 distinct and unrelated types (e.g., frequency and magnitude) of pressure pulsations:

- Fluid flow induced pulsation.
- Fluid acceleration induced pulsation.
- Shock induced pulsation.

In reaction to these pulsations, the system response is generally:

- Mechanical vibration due to the changing forces and lack of support or restraint.
- Vibration due to mechanical resonance with the forcing frequency of the pulsation.
- Acoustic resonance of the system with the pulsation frequency.
- Vibration at the natural frequency of the component due to the ringing effect from shock loading.
- Pump cavitation leading to increased pulsation, vibration and pump damage.

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RECIPROCATING PUMPS (Cont)

Given all the permutations and combinations of the above forcing functions and responses, it is not difficult to understand why most reciprocating pumps experience vibration problems at some time during their operation. Each of these items will be briefly discussed to make the designer aware of the importance of specific control measure during the plant conception.

Flow Induced Pulsation - Since the motion of a plunger or piston plotted against time is basically sinusoidal (whether a direct acting pump or power pump), the flow that is being pumped varies in a sinusoidal fashion as well. As each plunger or piston reaches its maximum velocity, the flow rate from the suction line or into the discharge is maximized. Since pressure drop due to friction is a function of the square of velocity, the flow variations develop into pressure variations or pulsations. These flow-induced pulsations occur in both the suction and discharge lines. However, since the friction losses in the suction are deliberately designed to be low, they are mostly a concern in the pump discharge where pulsation control devices must be used to filter out the pulsation.

In a duplex double acting pump the frequency of these flow-induced pulsations is 4 times the pump stroke speed. For a power pump, the primary pressure pulsation frequencies are in the 2 Hz to 15 Hz range and can be calculated as follows:

$$\text{Primary Pulsation Frequency, Power Pump} = \frac{\text{Number of Cylinders} \times \text{Pump Speed (rpm)} \times 2 \text{ (if Double Acting)}}{60}, \text{ Hz}$$

For power pumps, there is also a small pressure change (a decrease actually) preceding the maximum velocity from each cylinder due to a combination of the flow from other cylinders. Therefore, for power pumps a second primary frequency equal to two times the first primary frequency will also develop. **Figure 3** plots the flow from each cylinder of several different pump types and relates the average pressure to the pulsation pressure. **Figure 4** is the actual pressure trace from a triplex power pump. The upper trace shows the discharge pressure downstream of the discharge pressure pulsation control device. Clearly visible in the upper trace are both the peaks and the small dips in the pressure. The lower trace in **Figure 4** shows the suction pressure upstream of the suction pulsation suppressor/stabilizer.

Acceleration Induced Pulsation - The second source of pressure pulsation occurs as each plunger or piston begins either the suction or discharge portion of the stroke. This is the point of maximum fluid acceleration and it creates a pressure wave which is generally much higher in frequency than those generated by the flow variations. The acceleration-induced pulsation is generally a function of the square of the pump speed.

On the suction side, the sudden acceleration of the liquid followed by the rapid pressure rise behind it induces a high frequency wave generally in the 75 to 150 Hz region. The frequency and magnitude of this pulsation is unpredictable and has no relationship to the theoretical acceleration wave form. However, it is usually present in most reciprocating pumps. It is the acceleration-induced pulsations that dominate the suction behavior of recip pumps. **Figure 5** shows the suction of the same triplex power pump as **Figure 4** but between the suction stabilizer and the pump inlet manifold. Note the high frequencies generated in the suction (lower trace). (Each dip in the upper trace signals a marker on the shaft or one revolution.) Also note the lower trace in **Figure 4** (also the suction of the same pump) does not show these frequencies because it is outside (upstream) of the stabilizer.

At the discharge, if the pressure created by the pump is due primarily to overcome friction or restrictions in the downstream piping, then the flow induced residual pulsation is usually of sufficient magnitude to greatly exceed the acceleration pulsation. However, if the downstream system is such that very little friction or restriction exists (such a 4000 ft vertical pipe from the bottom of a mine shaft) then the acceleration induced pulsations can be significant or exceed the flow-induced pulsation. In fact, the acceleration induced pulsation in systems with long vertical runs can be amplified as the entire column of liquid must be accelerated for every change in fluid velocity. Even though the flow induced pulsation may dominate the discharge of the pump, there is usually evidence of the acceleration pulsation. The upper trace in **Figure 4** clearly shows the higher frequency acceleration pulsation superimposed on the flow pulsation.

Shock Induced Pulsations - The third source of pressure pulsations is similar to water hammer and occurs for a variety of reasons. Shock waves are created by the normal action of the pump valves opening and closing. These are generally low magnitude and fairly high frequency and do not appear to cause significant harm. However, if the valve action is delayed in any way these shock loads can become very significant as the valve suddenly opens and slams against the valve guard. In addition the resultant pressure and flow variations increase since there is a finite volume of liquid to ingest into or expel from the cylinder and a shorter time in which to do it. Causes of valve delay include improperly designed valves, valve hang up due to mechanical interference and hydraulic friction (liquid between the valve components makes them more difficult to open, like two flat plates pressed together with oil between them).

Figure 6 plots seven different measurements from a quintuplex power pump experiencing valve hang up due to friction. Traces 1 and 2 show the actual pressure inside Cylinders No. 1 and 5, respectively. Note the very high spike above the average discharge pressure and below the average suction pressure. Also note the high levels of vibration (acceleration) on the valve covers (Traces 3, 4) and high-pressure pulsations in the suction and discharge manifolds (Traces 5, 6) (within the boundary of pulsation control bottle and suction stabilizer). This pump broke several piping connections within weeks of initial startup. Most of the failures were due to excitement of the piping component natural frequency from the high shock loads of the improperly operating valves.

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RECIPROCATING PUMPS (Cont)

Figure 7 plots the same variables but after modifications to the pump valves. The over/underpressure spikes are still present (further valve design changes were made) but are significantly reduced. There is also a substantial reduction in the pulsation in the manifolds.

System Responses to Pulsation - As mentioned above, there are five general types of responses to pulsation. First, since there are pressure and flow changes taking place, piping that is not well supported or restrained will move. Efforts must be taken during design to ensure that the piping support system will account for the expected alternating forces. As an example, in one case the discharge pipe on a triplex power pump broke approximately 100 yards (91 m) away from the pump. The break occurred at a point where the pipe came off a support, went vertically up 15 ft and then 8 ft horizontally into a vessel with no intermediate supports. The excessive movement due to lack of support and the alternating flow simply fatigued the elbow.

Piping system components may at times have mechanical resonances near the primary pump pulsation frequencies. Remembering that these fundamental frequencies are in the 2 to 15 Hz range, typical natural frequencies for a 30 in. (0.76 m) length of 2 in. to 6 in. pipe range from 2.9 Hz to 12.5 Hz depending on how it is supported. Therefore, it would be wise during detail design to determine the length of line having a natural frequency equal to pulsation frequency and avoid using this length or multiples in the suction and discharge. See the **Pump Data Sheet Requirements for Pulsation Control** later in this section for a description of pulsation control design approaches.

Thirdly, the liquid in the system has acoustic natural frequencies based on the bulk modulus of the fluid. If acoustic natural frequencies coincide with the pulsation frequencies, pressure pulsations will be greatly amplified. To determine the acoustic frequencies, a piping analysis according to Design Approach 2 of API 674 must be done (see below).

Fourth are vibrations at a component natural frequency due to the shock pulsations. This is similar to repeatedly striking a pipe connection with a small but very persistent hammer such that the take off "rings" at its natural frequency. Shock loading will always occur in recip pump installations at some time. Valves fail causing shock loads, cavitation occurs intermittently even in the best systems and a host of other causes induce shocks. Failures can be eliminated by deleting all unnecessary connections and branches and strict adherence to standards (**GP 3-18-1**) in gusseting connections.

Lastly, when excessive pulsation occurs on the suction side, the absolute pressure can be low enough that the liquid vaporizes and cavitation occurs. This will damage the pump and severely increase the pulsation and vibration of the entire system. As the plunger or piston compresses the vapor and it suddenly collapses, the effect is like hitting the entire liquid discharge with a sledgehammer. Only proper system design and operation within the design limits can correct this problem.

DISCHARGE PULSATION CONTROL DEVICES

Discharge pressure pulsation control devices are generally either a gas damped design or a totally liquid filled acoustic filter. The gas-damped design can be either a bladder type (such as an accumulator having an elastomer bladder on the inside) or a small vessel with continuous injection of gas to provide a "bubble" on top. These devices generally provide a high degree of pulsation dampening at fairly low frequencies where the liquid mass can actually move to compress/expand the gas. They tend to work well on flow-induced pulsations but have little effect on acceleration or shock induced pulsation. Their drawback is frequent replacement of the bladder and availability/use of high-pressure gas.

The liquid filled systems operate on the principle of de-tuning an acoustic frequency that matches the pressure pulsation. Both the Ingersoll-Rand Cameron Hydraulic Data Book, Pump Handbook, and The Reciprocating Pump, Theory, Design and Use references provide excellent descriptions of these devices. They can be designed to filter single frequencies (the Helmholtz resonator), discrete multiple frequencies (quarter wave stub) or a range of frequencies (Helmholtz Filter).

Liquid filled acoustic devices work better at higher frequencies than gas type dampers. At the lower frequencies of flow-induced pulsation, they can be just as effective as gas damped types but they tend to become large. This makes them expensive since they must be designed for the discharge pressure of the pump. However, they require essentially no maintenance. The critical item is that the fluid properties must be well known and fairly constant to achieve success. Since they have no moving parts and do not "fail" as when the gas types do, well-engineered acoustic filters are preferred.

SUCTION STABILIZER AND PULSATION CONTROL DEVICES

- On the suction side of the pump, the device used to control pulsations is often called a suction stabilizer. However, it may, in fact, be identical to the device installed on the discharge. The function of a suction stabilizer is the same as for a pulsation dampener on the discharge, to reduce the pressure pulsations and isolate them from the connected equipment. Keeping in mind that the primary pulsation in the suction is of the acceleration variety, the design of the suction stabilizer must be to attenuate higher frequencies. Both liquid filled and gas types are used on the suction. Additional functions of the stabilizer are to provide a relatively large volume just ahead of the pump to collect any vapor or bubbles in the line and provide a pressure "reservoir" to help ease acceleration of the liquid. The suction stabilizer influences NPSH available by significantly reducing the term for the length of line to the nearest upstream vessel (see NPSH Available discussion later in this section).

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RECIPROCATING PUMPS (Cont)

RECIPROCATING PUMP DATA SHEET REQUIREMENTS FOR PULSATION CONTROL

Several items pertaining to pulsation control may be required on the Pump Data sheet depending on Owner Preference and the size of the pump.

- ▶ 1. **API 674** (Second Edition June, 1995) states that the purchaser shall use either Design Approach 1 (good engineering practices and pulsation control devices, if needed) or Design Approach 2 (acoustical and mechanical piping simulation) to control pulsation. **GP 10-2-1**, 2001 states that Design Approach 2 shall be used for all reciprocating pumps over 50 hp (35kW). When specifying a reciprocating pump this selection of Design Approach should be re-examined. If the following conditions exist for pumps below 50 hp (35kW) consult Machinery Specialists to review the need for Design Approach 2:
 - The stream has a Respiratory Exposure Class of R1,
 - OR**
 - Reciprocating pumps will be operated in parallel
- 2. If the Owner has a preference for either specific type of dampener (Gas Filled Bladder, Bladderless Gas Type or Liquid Filled Acoustic Filter), this should be added as a note on the Pump Data Sheet. A typical note for the Design Specification might read as follows:

“Pulsation control and suction stabilization devices shall be of the liquid filled, Helmholtz filter type.”
- ▶ 3. **API 674** (Second Edition - June 1995) has an optional Dynamic Valve Simulation Study as specified in Appendix E, Item E.1.4. This study simulates the valve motion to better predict the performance and potential problems of the pump. The Owner should be asked and the Design Specification should indicate whether this study is desired. In general, Machinery Specialists recommend that a valve study be done whenever the pump is in a critical service (unsparred or high economic impact) or rated over 50 hp (35 kW). If a study is to be done, the following paragraph is suitable for inclusion in the Design Specification.

“A pump valve dynamic simulation study shall be performed to model the valve motion and determine the dynamic interaction with the flow, pressure, pulsation and valve impact characteristics. The results shall be used to modify the valves, if necessary. The effect of valve friction shall be included in the analysis”.

VISCOSITY

Direct acting steam pumps are well suited to services in which the viscosity changes over a wide range. By lowering the speed as viscosity increases, the pump can be used for services from water (1 cSt at 68°F) to heavy oils (1700 cSt). As the viscosity decreases, wear in the cylinder increases and slip increases such that the pump efficiency is reduced. This is accounted for in **GP 10-2-1** by the requirement that the pump be sized using a volumetric efficiency of 90% or less (this will be done during detailed design, an increase on the pump data sheet is not required).

Power pumps are not as well suited to high viscosity since their valve areas are generally smaller than direct acting pumps and they tend to operate at higher speeds. Consult Machinery Specialists and the pump vendor if the service is over 50 cSt.

NPSH AVAILABLE ($NPSH_A$) - ACCELERATION HEAD

Design Practices Section X-D, *NPSH* provides the details on calculation of $NPSH_A$ (available). For reciprocating pumps all terms are the same as for centrifugal pumps. However, there are two differences for reciprocating pumps: 1) inclusion of the acceleration head loss term, h_a , and 2) the datum level used for power pump *NPSH* calculation.

Acceleration Head

Acceleration head loss occurs because at the instant that the suction valve opens on a reciprocating or controlled volume pump, the liquid must be accelerated into the cylinder. The acceleration loss term can be estimated using the formula below. However, before proceeding there are several cautions. First, the equation assumes that the pump type is known. During Design Specification Data Sheet preparation, this is not the case. Therefore, Designers should consult with Machinery Specialists at the inception to make a preliminary pump selection after consultation with vendors. To do this, all the other terms of the *NPSH* calculation must be known.

The second problem is that the acceleration head loss calculation is based on the theoretical acceleration curve and not on the fact that the high frequencies dominate the suction pulsation of the pump. Therefore, the result is at best an estimate.

Thirdly, the calculation technique is known to be invalid for long and/or complicated suction piping systems. It overpredicts the acceleration loss term.

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RECIPROCATING PUMPS (Cont)

The only method to accurately calculate $NPSH_A$ for reciprocating and controlled volume pumps is to do an acoustic study of the system once it is designed. For the purposes of the Design Specification, however, the following formula in conjunction with Machinery Specialist and pump vendor consultation should be used. Then, the following note should be added to the Design Specification Data Sheet for the $NPSH_A$.

" $NPSH_A$ listed is the total value including acceleration losses calculated according to the Hydraulic Institute procedures and assuming a [enter pump type here] pump. The value listed is expressed in feet or meters and may require changing by the Contractor for entry onto the API pump data sheet. The $NPSH_A$ without the acceleration head loss is [enter $NPSH_A$ without acceleration loss here]. The purchaser shall reconfirm the $NPSH_A$ once the piping has been designed. Acceleration losses in the suction shall be minimized through use of the straightest, shortest suction line possible."

$$\text{Estimated Acceleration Head Loss, } h_a = \frac{L V n C}{K g}, \text{ ft / m}$$

- where:**
- L** = Length of the suction line from the nearest upstream vessel (or suction stabilizer) to the pump, (ft/m)
 - V** = Average velocity in the suction line, (ft/sec or m/sec)
 - n** = Pump speed, rpm
 - C** = A constant to account for pump geometry, units and other factors.
 - = 0.4 for simplex single acting
 - = 0.2 for simplex double acting
 - = 0.2 for duplex single acting
 - = 0.115 for duplex double acting
 - = 0.066 for triplex single or double acting
 - = 0.040 for quintuplex single or double acting
 - = 0.028 for septuplex single or double acting
 - K** = A factor representing the relative compressibility of the liquid.
 - = 2.5 for hot oils
 - = 2.0 for most cool hydrocarbons
 - = 1.5 for amine solutions, glycols and plain tap water
 - = 1.4 for deaerated water
 - = 1.0 for urea and liquids with minimal air entrained.
 - g** = Gravitational constant (32.2 ft/sec² or 9.82 m/s²)

This formula loses validity when the suction line exceeds 50 ft in actual length (15.3 m), has more than 3 elbows, has more than 2 pumps operating on the same suction line or has complex liquid mixtures. The estimate also does NOT account for the high frequency pulsations commonly found in reciprocating pump suction systems. The only method which will accurately calculate $NPSH_A$ is an acoustic simulation of the suction system.

If the service is critical or pumps will be operated in parallel, Machinery Specialists recommend that the system receive an acoustic analysis to re-confirm adequate $NPSH_A$. This recommendation is reinforced by the fact that increasing the $NPSH_A$ may not solve cavitation problems in a reciprocating pump as it would with a centrifugal pump. As explained previously, the suction condition of a recip pump is dominated by the acceleration effect of the liquid. Regardless of how high the $NPSH$ margin is, the fluid acceleration required does not change. The same mass of fluid must be accelerated as each cylinder takes suction. If the acceleration head losses are high, very large increases in $NPSH_A$ would be needed to prevent cavitation. To overcome this concern, only straight, simple and short suction lines should be used.

Elevation convention

The Design Specification customarily uses a 2 ft elevation as the point where $NPSH_A$ is listed. This should be satisfactory for small (under 50 hp/39 kW) power pumps and direct acting pumps. However, larger power pumps often have their inlets much higher than this (5 to 7 ft/1.5 to 2 m) above grade. This must be accounted for in the detailed design by the contractor. No Design Specification changes are needed, but the Designer should not assume that $NPSH$ margins can be made "tight".