DETERMINATION OF NPSHR FOR RECIPROCATING POSITIVE DISPLACEMENT–PUMPS: A NEW APPROACH

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ABSTRACT

$NPSH_R$ is a significant pump characteristic; it is often taken as a measure of minimum suction head needed to operate a pump satisfactorily. Hydraulic Institute, among others, specifies a measurement of procedure for $NPSH_R$, which is at the point that a reciprocating pump loses 3.0 percent in volumetric efficiency relative to a stable efficiency at a high suction head. Although noise, vibration, or any sign of mechanical damage would be an indication of reaching the $NPSH_R$ limit on test, they may not be easily observed during a short term factory performance test. Also, the differences in the test system and the field installation can result in an improper correlation with the current methods for predicting suction performance.

In order to better predict successful field operation, a number of additional factors need to be considered, such as:

• The influence of the piping system attached to the pump.

• The need to avoid cavitation damage inside the pump.

• The need to operate reliably, with attention to proper valve dynamics.

• Changes in fluid properties between test and the installation.

• An additional term is defined ($NPSH_F$), which is intended to better predict suction performance in the field.

A description is provided of the valve and pump chamber operation as the suction head is reduced to near the $NPSH_R$ and the factors effecting $NPSH_R$ are discussed. Data are provided for the suction pressure and the pressure observed in the fluid chamber of a diaphragm pump. The inlet pressure *vs* time history of the diaphragm pump inlet with varying lengths of suction piping is also presented. This example demonstrates the deficiency in the standard modelling of the "Acceleration Head" effect.

A new method is proposed to better predict suction performance in the field based on traditional $NPSH_R$ testing and a "Power Density Index" (PDI). Computer modelling is discussed that would be needed for pumps with a high PDI.

INTRODUCTION

Net positive suction head required (NPSH$_R$) is probably the most significant characteristic of any pump next to its head-flow-efficiency (H-Q) performance characteristic. For a positive displacement pump, Hydraulic Institute Standards define it as the suction head where the pump loses three percent flow capacity relative to the capacity at high suction head. The standards also define the test loop configurations for running NPSH$_R$ test. NPSH$_R$ varies with pump speed and flow, and most pump manufacturers provide NPSH$_R$ performance curves for their pumps.

NPSH$_R$ is important because it provides a user guidance on the minimum suction head that the user needs to supply in his system. When the field piping configuration does not match the test loop configuration, which is normally the case, the standards provide a guidance on the adjustment of available head to account for acceleration and friction losses. When the adjusted available suction head at the pump inlet exceeds NPSH$_R$, the assumption is that one can operate the pump properly without excessive performance loss, cavitation, vibration, and noise. Or, can one?

The answer to this simple but critical question remains both unclear and misunderstood by much of the pump community. Using the Hydraulic Institute standards can produce either excessive or inadequate values of suction head requirement, depending on the type, size, and application of the pump.

The authors have several objectives: to provide analytical understanding of pump operation and behavior at low suction heads approaching NPSH$_R$; to present experimental data showing pressure development within and outside the pump chamber with time; to present basis for mathematical models that can be used to predict NPSH$_R$ with reasonable accuracy; and to provide guidelines to pump manufacturers and users for computing NPSH for satisfactory pump operation for various applications.

While the fundamentals for computing or measuring NPSH$_R$ for various types of reciprocating positive displacement pumps are essentially similar, pump type, and operating conditions do have significant influence on the behavior and reliability of pumps operating at suction head close to NPSH$_R$.

The pump speed and power input to the pump, for instance, have a large effect on cavitation damage and piping vibration. The authors set out a new criterion, pump power density, to judge the likely influence of pump operation near NPSH$_R$ on pump reliability.

The authors also present NPSH$_R$ computation and measurement data for two types of positive displacement pumps: pneumatically-driven displacement pumps, and crank shaft-driven power pumps. These two types of pumps cover a wide range of power density spectrum, and are, thus, indicative of the influence of NPSH$_R$ on positive displacement pumps in general.

UNDERSTANDING NPSH$_R$

One of the very desirable characteristics of positive displacement pumps is that they are self priming, and that they generally need low levels of suction head to operate. However, there are many types of positive displacement pumps and the various types, while having many similarities, also have some peculiar differences when operating at low suction head.

What is NPSH$_R$?

NPSH$_R$ is commonly understood to be the minimum head at which a positive displacement pump can be satisfactorily operated. A more precise definition, according to Hydraulic Institute Standards, states that NPSH$_R$ is the suction head measured at the pump's inlet manifold at which the pump's volumetric efficiency falls by 3.0 percent from a value measured at relatively high suction head. A typical NPSH$_R$ curve is shown in Figure 1.
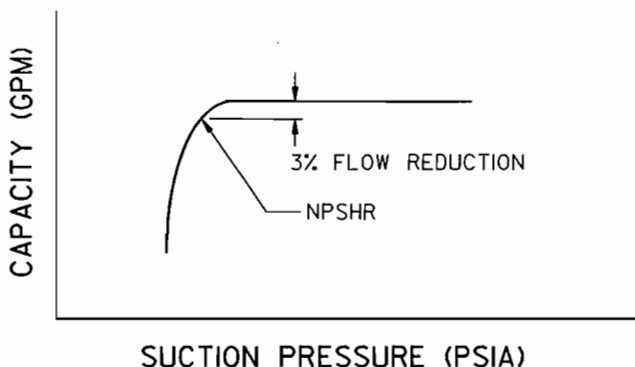

Figure 1. Typical NPSH Test Curve and NPSH$_R$ Definition.

NPSH$_R$ is not a single number characteristic. It varies strongly with pump speed, n, (increasing as n^2), and weakly with discharge pressure, Pd (increasing with Pd). Yet, the absolute value of the NPSH$_R$ is generally quite small, mostly substantially below the ambient pressure. The low value of the NPSH$_R$ allows positive displacement pumps to be used for high suction lift.

While NPSH$_R$ is an important characteristic, it is hardly a useful one for many real life pump applications, particularly those involving large (high horsepower) pumps. For example, a 100 hp triplex power pump may have an NPSH$_R$ rating of 5.0 psia, but in the real application, the pump may not operate satisfactorily below 75 psia suction head.

There are several reasons for this apparently large discrepancy. One reason for the difference is that the pump in a real application is often attached to a piping system, and, sometimes, to other pumps through the piping system. The other significant reason is that pump operation at NPSH$_R$ is often rough, and unacceptably damaging to the pump over the long term. Available suction head, thus, must be substantially increased to assure pump life and reliability.

Hydraulic Institute Standards do recommend adjustments for the presence of piping (called acceleration head), and friction losses. It will be shown later that these adjustments are too simplistic, and they either grossly overestimate or underestimate the value of the minimum suction head needed to operate the pump satisfactorily.

Pump Operation At and Near NPSH$_R$

NPSH$_R$ tests are generally conducted in a test loop that meets Hydraulic Institute recommendations for the loop design. The loop has short suction and discharge piping in order to minimize the effect of acceleration head. During the test, the suction head is gradually reduced and the pump flow is recorded until the flow shows a sharp drop (3.0 percent or higher) from its normal value. Depending on the pump type and loop design, the pump may or may not show symptoms of rough operation as evidenced by high vibration, and loud, banging noise.

What happens within the pump chamber when the suction pressure is gradually reduced? The following scenario applies to virtually all reciprocating positive displacement pumps with minor variations:

• Fluid cavitation is the primary reason for pump flow reduction. When the local pressure at any point in the pump or piping system falls below the vapor pressure, the fluid begins to vaporize, and form vapor-filled cavities. The cavitation begins within the pump chamber as the suction head is reduced, and the chamber pressure falls below the vapor pressure.

• The least pressure within the pump chamber normally occurs at the beginning of the suction stroke, close to the point when the suction valve just begins to open. Due to valve inertia, there is a

slight delay in valve opening, during which time the piston continues to move, and the chamber volume continues to increase. The chamber pressure falls off rapidly, and continues to fall off until the flow through the suction valve equals or exceeds the rate of increase in the chamber volume. If the pressure falls below the vapor pressure, the fluid begins to cavitate, and a small amount of fluid turns into vaporous cavities. The pump chamber now has a two-phase mixture of fluid and fluid vapor cavities.

• Normally, the flow through the suction valve is adequate so as to compensate for the missing flow, and to keep up with the chamber volume expansion. Once the pressure rises above the vapor pressure, the vapor cavities collapse, and the chamber is filled with the fluid only. The collapse of the cavities can often produce sharp pressure spikes that, under the extreme circumstances, cause serious damage, and inhibit safe pump operation. The intensity of pressure spikes depends on pump speed, fluid properties, and pump type. Some pumps only experience mild cavitation, and can run for a long time without any ill effects.

• If the suction head is so low that the flow into the chamber is not enough to fill expanding chamber volume, cavitation persists. Whether the cavitation continues throughout the suction stroke or not depends on suction pressure, and also on the rate of chamber volume expansion. The sinusoidal nature of crank shaft motion, for example, helps fill the pump chamber in power pumps as the piston slows down toward the end of the suction stroke. If the pump chamber is not filled at the end of the suction stroke, the volumetric efficiency falls.

• From that point on, the fall in volumetric efficiency is generally very steep (Figure 1) with the drop in suction pressure. The reason for the steepness is that the suction head at the VE fall-off point is already very low, and any further drop represents a large fractional reduction in pressure difference across, and flow through the suction valve. NPSH$_R$ is defined as the suction head at which the VE drops by 3.0 percent.

A test arrangement for determining the suction performance of a diaphragm pump is shown in Figures 2 and 3. The installation consists of a pump mounted about two feet below the water level from a 400 gal open water tank. The valveless suction line consists of 6.5 ft of 3.0 in ID tubing connected to 1.5 ft of 2.0 in NPT line to the pump suction. The suction lines length can be varied.
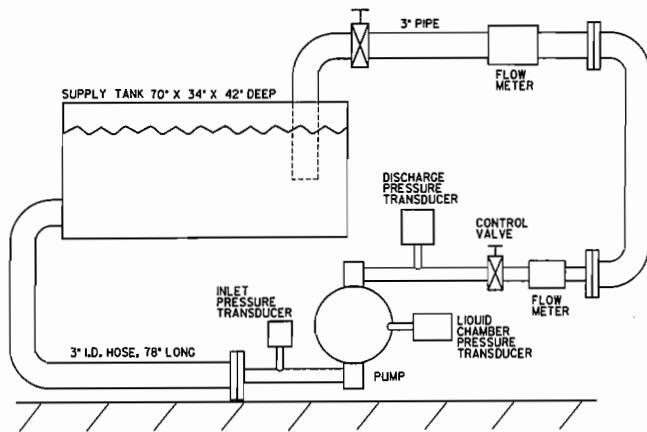
Figure 2. *Diaphragm Pump Test Stand.*

Inlet and liquid chamber pressure *vs* time traces for two operating points are shown in Figures 4 and 5. The inlet pressure rises as the suction valve on one end is closing (the suction valve on the other end is in the closed position), and is then depressed at the beginning of the suction stroke as the filling chamber valve

opens. Once the initial transient has been completed, the suction pressure becomes fairly constant at a value consistent with the steady-state NPSH value. Note that the downward transient is less noticeable when the pump is operating at near the vapor pressure in the inlet line (Figure 5).
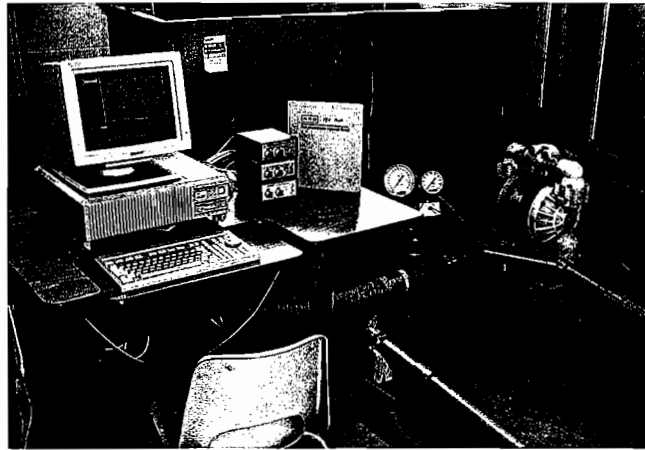
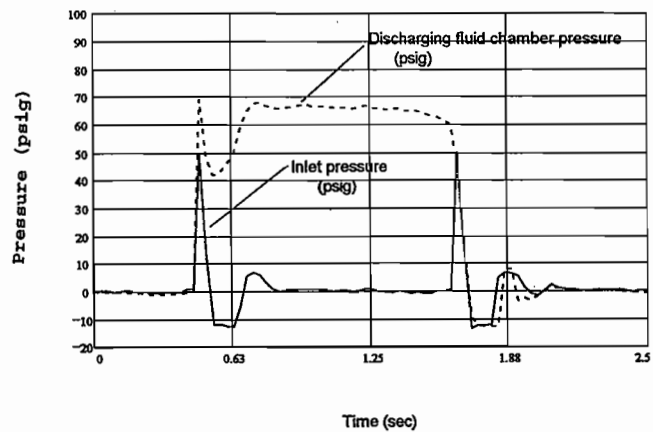Figure 3. *Diaphragm Pump Test Stand.*

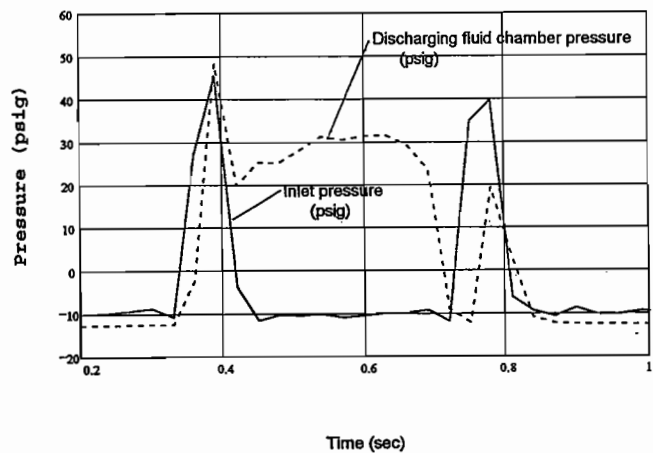Figure 4. *Two Inch Diaphragm Pump Inlet Pressure* vs *Time at 26 CPM and 37 GPM.*

Figure 5. *Two Inch Diaphragm Pump Inlet Pressure* vs *Time at 75 CPM and 109 GPM.*

Factors Influencing NPSH$_R$

NPSH$_R$ can be reasonably estimated without any measurements once the following factors influencing NPSH$_R$ can be properly understood.

Fluid Vapor Pressure: Fluid vapor pressure, the pressure at which the fluid begins to vaporize into cavities, is the baseline for NPSH$_R$. The vapor pressure for most fluids is generally quite low (1.0 psia for 70°F water). It is influenced by temperature and gas content. For some volatile fluids, the vapor pressure can be quite high.

Suction Valve Pressure Drop: Beyond the fluid vapor pressure, the suction valve design is the single most important factor influencing NPSH$_R$. The pressure drop is influenced by net valve flow area, and the rate at which the valve can open and close. While large valve opening is desirable, the valve must also be able to close rapidly following the end of the suction stroke. Otherwise, the fluid will backflow out of the suction valve during the beginning of the discharge stroke, resulting in VE loss. The valve loss typically increases as a square of the pump speed.

Pump Type and Design: Positive displacement pumps come in many types, and each type may have slightly different NPSH$_R$ characteristics. A primary source of this variation is the difference in pump chamber volume expansion rate. For example, power pumps have nearly sinusoidal piston motion while in pneumatically-driven diaphragm pumps, the piston motion is nearly linear. NPSH$_R$ is also mildly influenced by the number of pump chambers, suction manifold design, discharge valve characteristics, and fluid viscosity.

For most cases, NPSH$_R$ can be estimated within +/− 1.0 psi by adding suction valve pressure drop to the fluid vapor pressure. The valve pressure drop can be estimated by computing pressure loss through the valve for the mean suction flow. For highly viscous flows, frictional pressure drop in the suction piping based on the mean suction flow should also be added. In case this estimate is not precise enough, NPSH$_R$ can be computed with good accuracy through mathematical modelling, as described later.

Piping System and Acceleration Head

In the field, the pumps rarely have a piping system representative of the test loop in which NPSH$_R$ measurements are made. Piping can have a significant influence on the pump's suction characteristics as some of the head must be used by the fluid to accelerate itself in the suction piping to meet the pump's flow demand. The influence of piping on pump behavior can be quite complex. For the sake of simplification, the Hydraulic Institute recommends the use of acceleration head to account for the piping influence.

The acceleration head is calculated as follows:

$$h_{acc} = LvnC / K \qquad (1)$$

where

L = Suction piping length

v = Average velocity

n = Strokes per minute

C = A constant whose value depends on pump type and dimensional units

K = Relative compressibility factor; 1.0 for cold water

While used frequently by pump application engineers to specify suction requirements, the use of acceleration head to simulate pump suction characteristics is totally inappropriate in most cases. In fact, Equation (1) to calculate acceleration head does not even apply beyond a limited range.

The use of acceleration head has the following limitations:

• Equation (1), despite the compressibility factor, strictly applies only to incompressible fluids. Although liquids have a very high bulk modulus, they can not be assumed to be incompressible for computing dynamic pressure changes.

• Equation (1) is still applicable for compressible fluids, but only in a narrow range. It is best to identify these limitations by starting with a complete solution for a straight piping section. In *Appendix 1*, it is shown that Equation (1) for calculating acceleration head is valid only if

$$L < 3 \ c/N \ n \qquad (2)$$

where

c = speed of sound

n = strokes per minute

N = number of fluid chambers or cylinders

Using typical values of c = 4000 fps, n = 300 rpm, and N = 3 for a triplex power pump in Equation (1), the acceleration head can be used validly if the suction piping length is less than 13 ft. However, Equation (1) is routinely applied to piping sections of much longer length.

Miller [2], without providing a theoretical foundation, recommends use of Equation (1) for pipe lengths less than 10 ft. He cites a good example of misapplication of acceleration head computation. In a triplex power pump with a 90 ft long suction piping, the acceleration head calculations showed NPSH requirements of over 150 psia; yet, the pump ran with good efficiency with only 4.9 psia.

NPSH$_A$ and Acceleration Head

NPSH$_A$ is defined as the head available at the pump inlet after allowing for the acceleration head and the friction losses. Although the acceleration head calculation may indicate the need for significant additional suction head, the measured NPSH$_R$ is not affected by the acceleration head. The inlet pressure *vs* time histories with three different lengths of inlet piping length for the diaphragm pump, and test loop (Figures 2 and 3) are shown in Figure 6. The data show that there is little impact of pipe length on the NPSH$_R$, as measured indirectly from the fluid chamber pressure at the beginning stroke.

Why? First, there exists a deceleration head (pressure recovery) as much as the acceleration head. When the flow in response to the pump demand slows down, the pressure is recovered, and the flow through the suction valve increases, filling the pump chamber faster. In most cases, the acceleration and deceleration balance to a great extent, and the impact on NPSH$_R$ (3.0 percent VE loss) is minimal.

Does it imply that the piping length has no influence on pump behavior? Far from it. If the authors move away from the 3.0 percent VE loss definition, the piping has considerable influence. Such influence can be seen in Figure 6 where pressure measurements in the suction piping are shown for three suction piping lengths, varying from 1.5 ft to 25 ft.

For the 1.5 ft suction piping length, the pressure signal has an upward spike of a short duration followed quickly by a down spike of equally short duration. The upward spike is caused by the deceleration of the fluid when the suction valve in the second pump chamber closes. The downward spike is the acceleration head loss in the piping resulting from rapid opening of the suction valve. When the piping length is increased almost 10 times to 13 ft, the suction piping pressure profile remains similar, except the down spike has a longer duration, and has a flat bottom due to cavitation. When the piping length is doubled again, the magnitude of the

spikes does not change, but the duration of the low pressure spikes increases. In all cases. however, the pressure recovers to a steady level towards the end of the stroke.
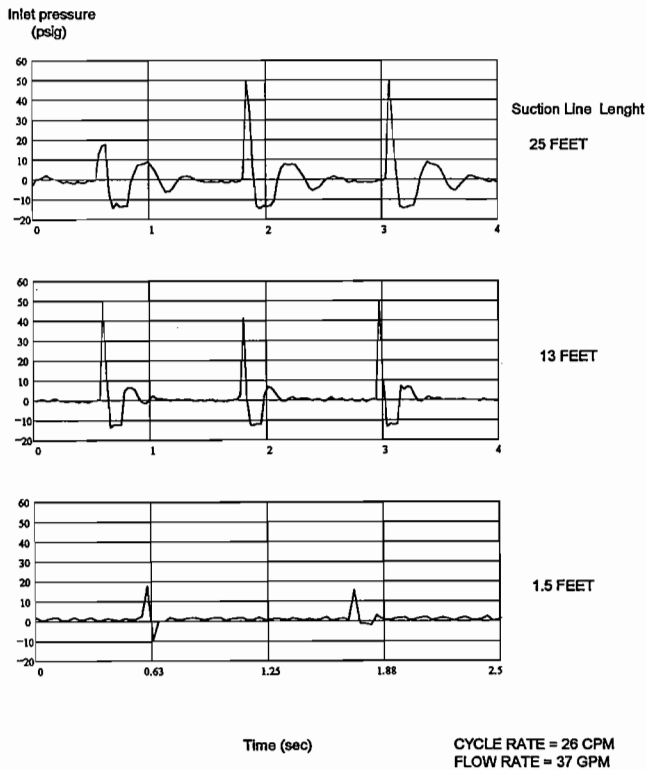

Figure 6. Two Inch Diaphragm Pump Test Data with (25, 13 and 1.5 feet of Suction Piping).

The pressure within the pump chamber reflects the pressure in the suction piping. Although the pressure approaches vapor pressure at the beginning of the suction stroke, the pressure recovers to nominal suction pressure minus valve pressure drop at the end of the stroke. There is no cavitation at the end of the stroke and no VE loss. $NPSH_R$ remains unchanged despite a sixteen fold increase in suction piping length.

Cavitation and Cavitation Damage

While large pressure fluctuations and cavitation may have no influence on $NPSH_R$, they often do cause other forms of unacceptable pump behavior such as piping and pump vibrations, pitting on pump components, and in extreme forms, pump failure. Usually, the cavitation damage manifests itself first in pitting on the suction valve since the valve experiences the lowest pressure in the pumping system and the cavitation collapse occurs very close to the valve elements.

Although cavitation is often associated with cavitation damage, some positive displacement pumps experience cavitation during a part of the suction stroke, and yet operate without any ill effects. There are several reasons for this, as follows:

• Cavitation damage depends on cavitation intensity, and the pump materials' propensity to resist impact erosion. Cavitation intensity is a strong function of cavitation volume, and the rate of change of ambient pressure during cavitation collapse. In flow-induced cavitation as seen in centrifugal pumps, cavitation damage has been found to be proportional to the fifth or sixth power of the fluid velocity (indicative of pressure variation) in the cavitation collapse region. Similar correlation can be expected for positive displace-

ment pumps where the rate of pressure change is governed by the rate of change of the chamber volume, and fluid bulk modulus.

• The rate of change of chamber volume is proportional to pump rpm, and stroke (or piston velocity in most pump types). Thus, the propensity for cavitation damage depends strongly on the fluid bulk modulus and piston velocity, and weakly on the pump pressure rating. In general, pumps rated at high horsepower, and running at high rpm and low suction pressure are much more prone to cavitation damage than low horsepower pumps. In high power pumps, cavitation induced pressure spikes are also likely to cause vibrations, and in extreme cases, violent shaking of the pump.

• Pump speed is a key factor in smooth pump operation at low suction head. Although $NPSH_R$ goes up only as a square of the pump speed, the vibration, and damage intensity can go up by a power factor of five to six. In high horsepower rated pumps, there is virtually no correlation between $NPSH_R$ measurements, and the suction head required in the field to run the pump without damage.

$NPSH_F$: WHAT IT MEANS

While $NPSH_R$ is often included as a part of pump characteristics, its significance in real life use is very limited. $NPSH_R$, as defined by Hydraulic Institute Standards, has a strict definition as the lowest suction head at which pump experiences a 3.0 percent VE loss. Practically, it defines the maximum suction capability of the pump. However, the field installations rarely match the test loop set up, and it is neither wise nor practical to run the pump fully cavitating during most of the suction stroke. Many pump application engineers attempt to adjust $NPSH_R$ to field conditions by including acceleration head. As shown earlier, the common method of computing acceleration head is so limited in scope that it is often misapplied, making the discrepancy even worse.

The problem lies with misunderstanding what $NPSH_R$ really means. To a manufacturer, it is a pump characteristic whose value should be included in the pump manual. To a pump user, however, it provides a measure of minimum suction head at which he can expect smooth pump operation for his particular application.

As a pump characteristic, $NPSH_R$ is a valid measure of the flow capacity of the pump (actually, suction value). However, depending on the pump type, it may or may not have much relevance to pump suction head requirement in the field.

As if $NPSH_R$ and $NPSH_A$ were not confusing already, the authors add to the complexity by introducing a new concept, and defining a new term: $NPSH_F$.

$NPSH_F$ is the minimum suction head required in the field to run the pump satisfactorily and reliably. $NPSH_F$ has the following characteristics:

• $NPSH_F$, unlike $NPSH_R$, is neither pump specific nor a characteristic of the pump. $NPSH_F$ is application specific.

• $NPSH_F$ is also not the same as or similar to $NPSH_A$, a point that will discussed in greater length in the next section.

• $NPSH_F$ is a field parameter, and thus its value varies with field operating conditions. (Values for $NPSH_R$ and $NPSH_A$ also vary with operating conditions such as rpm.)

• The value of $NPSH_F$ depends to a great extent on the piping system attached to the pump. Thus, $NPSH_F$ can be substantially reduced by proper piping design, the use of properly selected and appropriately placed dampeners, and other pressure pulsation damping devices.

• $NPSH_F$ can be defined and measured (for a given set of operating conditions) as the minimum average pressure at which the cavitation disappears in the liquid chamber. $NPSH_F$ can also be calculated with reasonable accuracy using computer models, as discussed later.

Since $NPSH_F$ is not a pump parameter, it is not the pump manufacturer's responsibility to determine $NPSH_F$. Then, it becomes the responsibility of the pump system designer, and ultimately the customer, to make sure that adequate suction pressure is available to run the pump satisfactorily.

Obviously, it is not economical to measure or calculate (through computer modelling) $NPSH_F$ for low power or inexpensive pumps. Therefore, the authors have established an index, called Power Density Index (PDI), to determine when proper determination of $NPSH_F$ is critical and when it is not.

$NPSH_A$ AND $NPSH_F$

Some experienced pump users may wonder what is the difference between $NPSH_A$ (NPSH available) and $NPSH_F$. $NPSH_A$ is presumed to take into account for the field conditions through the use of acceleration head, and friction losses.

The primary difference between $NPSH_A$ and $NPSH_F$ is that $NPSH_F$ is a dynamic pressure concept, while $NPSH_A$ is a static pressure concept. Reciprocating positive displacement pumps, unlike rotary pumps, create significant flow fluctuations that normally result in substantial pressure pulsations that depend on the piping attached to the pump. The concept of $NPSH_A$ does recognize the influence of piping through the use of a static acceleration head whose shortcomings have already been delineated earlier.

As an example, one way to increase $NPSH_A$ is to use a pulsation dampener close to the pump inlet. Many dampener suppliers will suggest that the use of a properly sized dampener will reduce the fluid acceleration length to a short section between the dampener and the pump inlet. This static view of the dampener use may be a good approximation in some cases, but it falls short of the true dampener behavior. In an extensive investigation of pulsation dampener performance, Singh and Chaplis [3] tested five pulsation dampeners from different manufacturers in the same triplex pump test loop configuration. The dampeners were specified by the dampener manufacturers, and were installed at an identical location in the test loop. Although all dampeners reduced pressure pulsations, the level of reduction and the frequency of pulsations varied substantially among the dampeners. While a static approach like $NPSH_A$ would have predicted nearly identical results, the actual results varied markedly.

In the dynamic sense, the dampener acts like an electrical filter, reducing pulsations below the cut-off frequency. However, the dampener itself becomes a part of the compliant piping system, creating its own response frequency. If this frequency is a close multiple of the fundamental rotational frequency, the pressure pulsations may actually intensify rather than subside.

To reduce $NPSH_F$, one needs to find a way to minimize pressure pulsations. The use of a pulsation dampener is one obvious way to achieve a reduction in pulsation. However, what is even more important is the need to optimize the choice of the dampener through the selection of right type, size, and placement of the dampener.

Another key aspect of the pressure minimization strategy is to increase the acoustic damping in the piping circuit. Such damping can be increased, for instance, by incorporating a properly located orifice in the piping circuit. The use of an orifice to help suction performance is totally counter-intuitive to the $NPSH_A$ concept, since the pressure drop induced by the orifice would reduce $NPSH_A$. However, in the dynamic realm, the reduction in pressure pulsations with the orifice may be an order of magnitude larger than the static pressure drop.

The difference between $NPSH_A$ and $NPSH_F$ can be illustrated through a field case study reported by Walchel, et al. [4], and in which the senior author was involved. Four triplex pumps rated at 275 rpm, 368 gpm, and 1800 psig discharge pressure were operated in parallel in a pumping station to pump crude oil. Pump $NPSH_R$ was less than 1.0 psig, and each pump was equipped with a centrifugal charge pump, capable of a charge pressure up to 90 psig, and a bladder type dampener. Standard $NPSH_A$ calculations would indicate that the pump would require a charge pressure of less than 20 psig.

However, the pumps could not operate satisfactorily even at a charge pressure of 60 to 70 psig. Measurements showed that peak-to-peak magnitude of pressure pulsations exceeded 200 psi in some cases. Even at 70 psig charge pressure, the pumps cavitated producing pressure spike of over 800 psi. Analyses of test data indicated that the high pressures pulsations resulted from acoustic resonance. Under these field conditions, the charge pressure would need to exceed 100 psig ($NPSH_F$ = 100 psig).

The solution, however, was not to use a larger booster pump, but to design and install acoustical dampeners. The dampeners were designed with the help of computer modelling of the piping system. The results were dramatic, and the pumps operated smoothly with the existing booster pumps, while reducing estimated $NPSH_F$ to less than 50 psig. Similar case studies have been reported by Parry [5], and Singh and Madavan [6].

Desirable as it may be, it is neither necessary nor economical to use computer modelling for all pump installations to determine $NPSH_F$. High pressure pulsations and cavitation do a lot more damage in high energy pumps than in low power pumps. The authors introduce an index, called Power Density Index, that can be used as an indicator of when detailed calculations or computer modelling of $NPSH_F$ would be desirable.

Power Density Index

Power density index is a dimensional number defined as follows:

$$PDI = FHP/ (N\ D^2)$$

where FHP is the liquid horse power, N is the number of pump chambers, and D is the representative piston diameter in inches. PDI is a measure of fluid power conveyed per unit area of the fluid chamber volume.

$$FHP = Q\ H/3960$$

$$Q = \pi\ D^2\ S\ N\ n\ \eta_v\ /231$$

$$PDI = 2.\ 10^{-6}\ n\ S\ H$$

where

S   =   Stroke in inches

H   =   Head rise in ft

n   =   Strokes per minute per fluid chamber

PDI is, thus, proportional to pump speed, piston stroke and head rise. PDI values can vary from 0.01 (diaphragm pumps) to 5.0 (large reciprocating pumps) and even higher.

Following guidelines relating PDI to $NPSH_F$, although preliminary at the moment, can become the basis for further refinement with the help of the pump industry:

• For PDI less than 0.02, cavitation damage or piping vibrations is not much of concern. Although the pump is cavitating, damage is rare. $NPSH_F$ is only slightly higher than $NPSH_R$. Allowance should be made for friction losses in the case of high viscosity fluids. With a PDI between 0.02 and 0.2, a suction dampener would be desirable.

• For PDI between 0.2 and 1.0, a minimum margin of 5.0 to 10 psi above $NPSH_R$ is recommended. Higher margins may be needed to

completely suppress cavitation. The influence of short suction piping may be estimated by the proper use of the acceleration head calculations. A suction pulsation dampener is recommended.

• For PDI between 1.0 and 2.5, $NPSH_F$ would depend on the piping system. It is recommended that a computer prediction model (described in the next section) be used for computing $NPSH_F$. If such a model is not available, past experience with similar applications can serve as a guide. Typical margins above $NPSH_R$ may lie between 15 to 40 psi. Suction pressure dampener may be required to keep $NPSH_F$ within reasonable limits.

• For PDI above 2.5, the use of computer prediction models is highly recommended. Without such modelling, it is nearly impossible to estimate $NPSH_F$, unless the user has experience with a very similar installation. In many cases, the model can suggest very inexpensive ways such as the use of a properly located orifice to reduce $NPSH_F$. Margins above $NPSH_R$ may reach as high as 80 psi. A suction dampener is then a must.

When a pump installation has multiple pumps that are located close enough (within approximately 50 suction pipe diameters) to lie within the range of pump interaction, PDI for each such pump should be added, and the new PDI value used in following the above guidelines.

The above guidelines are only estimates based on the authors' experience; they should be examined by the pump community, and revised as needed based on a much broader experience base.

COMPUTER PREDICTION MODELS

With the notable exception of a few companies, the pump industry has been lagging in the use of computer models to predict pump performance, and the influence of piping system on pump behavior including suction pressure requirements. The reciprocating compressor industry recognized the need for modelling piping over 40 years ago, and has been using analog computers, and lately digital computers including PCs, to predict piping pressure pulsations and design piping systems to reduce such pulsations, [7, 8, 9]. Since the magnitude of pulsations is directly proportional to fluid bulk modulus, and since liquids have a much higher bulk modulus than gases, pumps normally face much more severe pulsation problems than compressors.

Singh, and Madavan [6] developed and presented the first of such comprehensive models for power pumps in 1984. Since then, the model has been upgraded, and used in tens of field installations with great success. Experience shows that computer models can predict the following with reasonable accuracy:

• Pump performance
• Suction and discharge valve dynamics
• $NPSH_R$
• Piping pressure pulsations
• $NPSH_F$

In regards to $NPSH_R$, such models can provide a wealth of information. First, $NPSH_R$ can be calculated with good accuracy, obviating the need for expensive tests except those required for occasional verification. The model can also provide a detailed time history of pressure within and flow into and out of the pump chamber, and dynamics of the suction valve as the suction head is lowered. The model can reasonably predict at what suction pressure the cavitation is likely to begin, and how long the cavitation persists during the suction stroke. Comparisons are shown in Figures 7 and 8 between test data, along with computer predictions for $NPSH_R$, and pressure pulsations at one point in the piping [3] for a power pump.
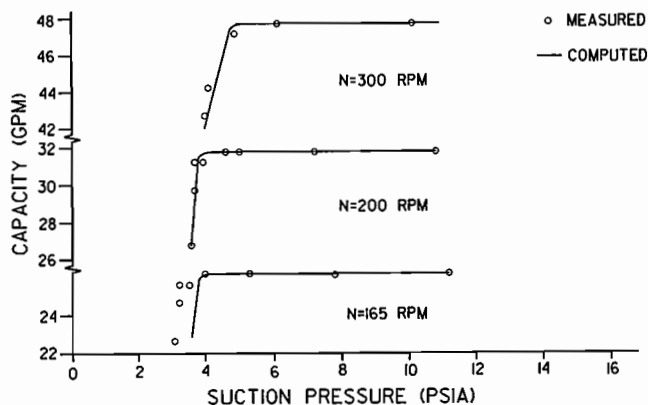

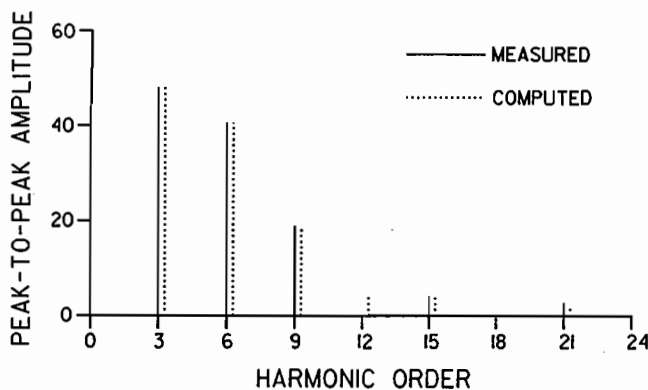Figure 7. Power Pump Predicted and Measured $NPSH_R$.

Figure 8. Power Pump Predicted and Measured Pressure Pulsations at a Point Near the Discharge Flange. The Peak-To-Peak Pressure Amplitudes are in psi.

It is highly recommended that pump manufacturers use computer modelling techniques in addition to test, and historical data for pumps with a PDI higher than two. The Hydraulic Institute can take a lead in this area, and provide guidelines on the use and the range of applicability for such models. The Institute can also test commercially available models, and recommend a list of independent providers of such models.

CONCLUSIONS

$NPSH_R$ as currently defined has a limited validity as an indicator of suction performance of a pump for most types of positive displacement pumps. For most pump applications, reliable pump operation in the field at or near $NPSH_R$ is nearly impossible. The use of acceleration head to correct $NPSH_R$ for piping influence is fundamentally flawed, except for very small sections of pipe.

The authors argue the need for another indicator, $NPSH_F$, that defines the minimum suction head required to assure proper performance, and long-term reliability of the pump in a particular field application. Since $NPSH_F$ can vary from one application to the other for the same pump, the determination of $NPSH_R$ can pose undue burden on the pump manufacturer. The authors offer guidelines on various techniques to estimate or compute $NPSH_F$, depending on a pump's likely behavior at low suction head. The latter is characterized by a dimensional index, called power density index or PDI.

One of the best and cost-effective techniques to determine $NPSH_R$ and a pump's suction performance is to use computer modelling. Such models have been shown to be very effective in

predicting pump $NPSH_R$ and the suction pressure level needed to run a pump free of cavitation and vibration.

NOMENCLATURE

Bulk modulus compressibility of a given liquid; (the change in pressure/change in density) x (the density at a given temperature).

c	speed of sound
C	A constant whose value depends on pump type and dimensional units
D	The representative piston diameter in inches
FHP	The liquid horsepower
H	Head rise in feet
hacc	Acceleration head
L	Suction piping length
K	Relative compressibility factor; 1.0 for cold water
n	Strokes per minute
N	The number of pump chambers
NPSH	The total suction head in feet (or psia) of the liquid being pumped less the absolute vapor pressure of the liquid.
$NPSH_A$	The NPSH available from the system at the pump inlet.
$NPSH_R$	The NPSH required by the pump to operate a given flowrate with no more than a 3.0 percent loss in volumetric efficiency, noise, vibration, unstable operation or mechanical damage while being tested.
$NPSH_F$	The NPSH required to operate the pump properly in the field without loss in volumetric efficiency, noise, vibration, unstable operation or mechanical damage.
PDI	Power density index , a dimensional number defined as $PDI = FHP / (N D^2)$.
Q	Flowrate, gal/min
S	Stroke in inches
v	Average velocity
VE	Volumetric Efficiency: Volume being displaced per stroke/volume of fluid chamber. To account for compressibility effects, see Millers book [2].
ϖ	Angular frequency
j	Imaginary constant
k	Wave number
λ	Wave length
A	Pipe area

APPENDICES

Appendix 1

Pressure pulsations at any point in a piping network linked to one or more positive displacement pumps are caused by flow variations induced by opening and closing of valves. The pulsations can be calculated by multiplying each flow fluctuation harmonic with the corresponding piping impedance at that point and summing the resulting values for all harmonics. The details of the underlying transfer matrix theory are presented in [3].

For a simple case of a suction pipe of length L, with one end attached to a large tank and the other end to a pump valve, the pressure pulsations at the valve from the transfer matrix theory are given by:

$$P_1 (\varpi) = Z_{11} (\varpi) Q_1 (\varpi) \qquad (A-1)$$

where Z_{11} is the transfer impedance at the valve, and P_1 and Q_1 are the pressure and flow fluctuation harmonics at angular frequency ϖ. The value of Z_{11} for undamped acoustic pulsations in an open end pipe [3] is given by:

$$Z_{11} = j \rho c \tan kL / A \qquad (A-2)$$

where

$$k = \varpi / c = 2 \pi / \lambda \qquad (A-3)$$

is the wave number and λ is the wave length. If $L << \lambda$, $kL << 1$, $\tan kL \approx kL$, then, Equation (A-2) simplifies to,

$$Z_{11} = j \rho c kL / A = j \rho \varpi L / A \qquad (A-4)$$

$$P_1 (\varpi) = j \rho \varpi L Q_1 (\varpi) / A \qquad (A-5)$$

$$P_1 (t) = (\rho L / A) dQ_1 / dt \qquad (A-6)$$

Equation (A-6) is the transformation of Equation (A-5) from frequency domain to time domain. Equation (A-6) is the genesis of the definition of acceleration head used in Hydraulic Institute standards. This derivation shows that the Hydraulic Institute definition is valid only when $L << \lambda /2\pi$. In practice, L should not exceed $\lambda/10$, and preferably, should be less than $\lambda/20$.

As an example, for a triplex power pump running at 300 rpm, the basic frequency of pulsations, f_n, is given by:

$$f_n = 300 * 3 / 60 = 15 \text{ Hz}$$

The wavelength for the pulsation at 15 Hz in water (speed of sound = 3900 ft/sec) is given by,

$$\lambda = 3900 / 15 = 260 \text{ ft}$$

The acceleration head calculation is then valid when suction pipe length is less than 13 ft ($\lambda/20$), and somewhat valid when the length is less than 26 ft ($\lambda/10$). These calculations apply only to basic frequency. For power pumps, the highest flow fluctuations occur at twice the basic frequency, limiting the applicability of acceleration head to suction pipe length less than 7.0 ft. Other flow frequencies can be as high as twenty times the basic frequency.

Appendix 2

Diaphragm pump suction tests with varying inlet piping lengths

The inlet pressure *vs* time histories with three different lengths of inlet piping length for the diaphragm pump and test loop (Figures 2 and 3) is shown in Figure 6. The suction line contains 6.5 ft of 3.0 in ID tubing connected to a 1.5 ft long 2.0 in NPT line to the pump suction to make the "short suction" system. A 13.5 ft section replaces the 1.5 ft section of 2.0 in NPT pipe to make the "mid suction" system. A 25 ft section is used to make the "long suction" system.

The acceleration pressure calculations for these installations are found Hydraulic Institute Standards, Section 7.3.8 [1]:

Pacc = LvnCs/(2.31* K*g)

Where:

L = Length of pipe in feet

C = coefficient = 0.628

K = compressibility factor of the liquid

     (K = 1 for cold water)

v = average velocity, ft/sec

n = strokes per minute

g = gravitational constant = 32.2 ft /sec

s = specific gravity (s =1 for cold water)

In this example:

v_2 = 37 gal/min in 2.0 in pipe

    = 3.5 ft/sec

v_3 = 37 gal/min in 3.0 in pipe

    = 1.6 ft/sec

L_3 = 6.5 ft (3.0 in ID)

L_short = 1.5 ft (2.0 in NPT)

L_mid = 13.5 ft (2.0 in NPT)

L_long = 25 ft (2.0 in NPT)

n = 26 cycles per minute

Short Suction:

Pacc = Pacc (3.0 in section) + Pacc (2.0 in section)

    = (6.5)(1.6)(26)(0.628)/((2.31)(1.0)(32.2)

    + (1.5)(3.5)(26)(0.628)/((2.31)(1.0)(32.2)

    = 2.3 + 1.2 = 2.5 psi

Mid Suction:

Pacc = Pacc (3 in section) + Pacc (2 in section)

    = (6.5)(1.6)(26)(0.628)/((2.31)(1.0)(32.2)

    + (13.5)(3.5)(26)(0.628)/((2.31)(1.0)(32.2)

    = 2.3 + 10.4 = 12.7 psi

Long Suction:

Pacc = Pacc (3 in section) + Pacc (2 in section)

    = (6.5)(1.6)(26)(0.628)/((2.31)(1.0)(32.2)

    + (25)(3.5)(26)(0.628)/((2.31)(1.0)(32.2)

    = 2.3 + 19.3 = 21.6 psi

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