VIKING PUMP
A Unit of IDEX Corporation

AD-19

Net Positive Suction Head

Few subjects related to the application of positive displacement (PD) pumps are more discussed and less understood than Net Positive Suction Head – NPSH.

Sophisticated processes using PD pumps for drawing liquids from vessels under high vacuum (oil purifying and recycling); users wanting more capacity from smaller pumps (higher speeds); more customers installing reserve fuel supply tanks far beneath concreted parking lots (long suction lines with a lift). These are just a few applications where NPSH needs to be evaluated since a lack of consideration for NPSH could spell TROUBLE in any of the situations just mentioned. Another reason for concern is that more and more spec sheets and requests for quotes are asking for or are giving NPSH values.

WHAT IS NPSH?

Net Positive Suction Head – must be indicated "available" or "required" to be meaningful – is the pressure in feet of liquid absolute measured at the pump suction port less the vapor pressure. Seems simple enough, but the "available" and "required" terms and the "absolute" pressure cause some problems. For PD rotary pumps instead of using NPSH expressed in feet of liquid absolute, the Hydraulic Institute (Viking is a member) uses Net Positive Inlet Pressure Available (NPIPA) and Net Positive Inlet Pressure Required (NPIPR) expressed in PSIA. NPSH and NPIP are the same thing but expressed in different units. To avoid confusion and additional problems, we will stay with the more frequently used NPSH and feet of liquid absolute throughout this document.

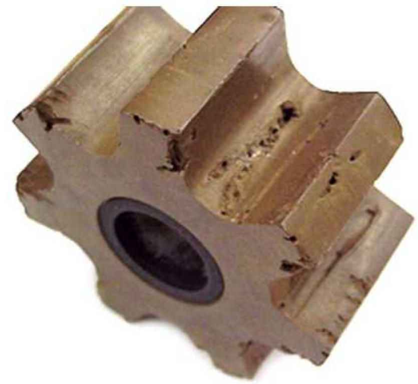
NPSH available (NPSHa) is a function of everything in the system on the suction side of the pump up to the suction port. Everything includes the pressure on the surface of the liquid in the supply tank, the difference between the liquid level and the centerline of the pump port, line losses, velocity head, and vapor pressure. ***NPSH required (NPSHr)*** is based on everything from the suction port to the point in the pump where the pressure starts to increase. Everything includes the entrance losses and the friction losses or pressure drops of getting the liquid into the pumping elements.

CAVITATION

Since NPSHa is the absolute pressure available less the vapor pressure of the liquid, it is logical to reason that the NPSHa should always be greater than the NPSHr. If this were not the case, there would be vapor formed in the suction area of the pump. This reasoning is sound. Thus for a pump to operate properly, the NPSHa must be greater than the NPSHr. With NPSHa less than NPSHr, the pressure at some point in the pump suction area will be less than the vapor pressure of the liquid and CAVITATION will take place in the pump.

With the pressure in the pump below the vapor pressure, bubbles (pockets or cavities of vapor) start to form in the liquid. They are carried along with the liquid until they get to a higher pressure region in the pump where the bubbles collapse. This phenomenon is known as cavitation. It is the violent collapse of the bubbles of vapor with the resulting shock wave that causes many of the damaging effects associated with cavitation – noise, vibration, eroded parts, short service life. These, plus reduced capacity and efficiency, possible pulsations, and an unhappy user, all make it mandatory that NPSHa be greater than NPSHr. See **Figure 1.**

FIGURE 1:
Typical wear caused by severe cavitation. Cavitation damage usually occurs on the face of the rotor, the idler face that was against the rotor, and the root of the idler teeth.



NPSH NOT EXACT

When a value for NPSH – available or required – is given, there is the impression that the value is exact. Unfortunately, this is not always the case. Such items as the following all tend to make stated values of NPSH possibly less exact than we would like.

1. Entrained air or gases in the liquid.
2. The ability of a pump, particularly a PD pump, to handle some vapor* with little adverse effect.
3. The fact that pumps handling different liquids, particularly some hydrocarbons, will operate satisfactorily with less NPSHa than would be required for water or other test liquids. This situation is true for centrifugal pumps per information in the Hydraulic Institute Standards and has been observed as being true for PD pumps.

* *Although the Hydraulic Institute and API 676 define NPSHr as the point where there is a 3% loss in capacity, there is no common agreement among pump manufacturers on the % of drop in capacity that should be used to determine the point at which the NPSHr value is read. (e.g. some use 1%, others 3%.)*

NPSH AVAILABLE – NPSHA

There is a widely used formula for calculating NPSHa which should be reviewed and understood before continuing. Even though the value determined from the formula may not be as exact as we would like, it is the best available short of actual test data, provides a basis for comparing systems and selecting pumps, and can give an alert to a potentially troublesome installation. As we have indicated before, NPSHa is a function of the suction piping system, the operating conditions, and the liquid pumped. For a system at the design stage or for one in use, the NPSHa can be calculated from the following formula – **$NPSH_a = H_a \pm H_z - H_f + H_v - H_{vp}$.** See **Installation 1.**

Where

H_a = absolute pressure on the surface of the liquid in the supply tank expressed in feet of liquid pumped.

H_z = vertical distance in feet from the surface of the liquid in the supply tank to the centerline of the pump suction port. If the liquid is below the centerline, H_z is negative.

H_f = friction losses in suction piping expressed in feet of liquid pumped.

H_v = velocity head at the suction port in feet of liquid pumped.

H_{vp} = absolute vapor pressure of liquid at pumping temperature expressed in feet of liquid pumped.

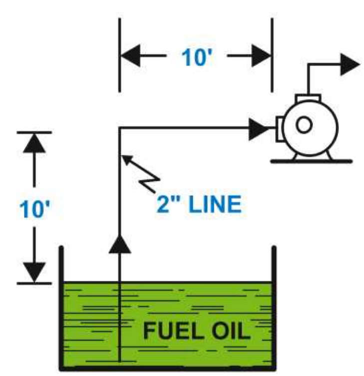
***Remember!!!** All values must be expressed in the same units, feet of liquid pumped.*

The part that velocity head (H_v) has to play in NPSHa calculations and measurements is somewhat controversial.

$H_v = \frac{V^2}{2g}$ where V equals velocity of the liquid at the suction port in feet per second and g equals the acceleration due to gravity (32.2 feet per second squared). The value of H_v is normally quite small. See **Table 1**. **Table 2** shows that the value of H_v is 1.3 feet of liquid or less for all but two Viking Heavy-Duty pumps. Since the value of H_v is small for normal Viking pump applications, Viking suggests that it not be included in NPSHa values determined by calculations.

NPSHA CALCULATION EXAMPLE

We will calculate the NPSHa for a typical installation to show how the formula is applied. Values for vapor pressure and line losses are approximate and are meant for illustration only.



No. 2 Fuel Oil, 90 GPM, 75°F, Sea Level Installation, S.G. 0.88, 38 SSU Viscosity

TABLE 1:

Velocity-Velocity Head (H_v)

Velocity of Liquid at Suction Port, Feet/Second	4	5	6	7	8	9	10	11	12	15
Velocity Head (H_V), Feet of Liquid	.25	.39	.56	.76	1.0	1.25	1.55	1.87	2.24	3.50

TABLE 2:

Velocity Head of Viking Heavy Duty Internal Gear Rotary Pumps at Nominal Rated Conditions

Pump Size	Port Size Inches	Rated Capacity GPM	Liquid Velocity* at Suction Port Feet/Sec.	Velocity Head Feet of Liquid (H_V)
GG	1	10	3.7	.21
H	1.5	15	2.4	.09
HJ	1.5	20	3.2	.16
HL	1.5	30	4.7	.34
AS	2.5	35	2.3	.08
AK	2.5	50	3.4	.18
AL	3	75	3.3	.17
K	2	60	5.7	.50
KK	2	80	7.6	.90
L	2	135	12.9	2.6
LQ	2.5	135	9.0	1.3
LL	3	140	6.1	.58
LS	3	200	8.7	1.2
Q	4	300	7.6	.90
QS	6	500	5.6	.49
M	4	420	10.6	1.7
N	6	600	6.7	.70
R	8	1100	7.1	.78
RS	10	1600	6.5	.66

** Based on flow thru Schedule 40 pipe same size as port.*

INSTALLATION 1:

$NPSH_a = H_a \pm H_z - H_f - H_{vp}$

H_a = absolute pressure on liquid

= atmospheric pressure in feet of liquid

$= 14.7 \text{ PSIA}^* \times \frac{2.31' \text{ of } H_2O}{\text{PSI}} \times \frac{1' \text{ of Fuel Oil}}{0.88' \text{ of } H_2O}$

H_a = 38.6 feet of fuel oil

H_z = vertical distance from liquid level to centerline of suction port (use maximum or worst condition – empty tank). Liquid is below pump so H_z is negative.

H_z = 10 feet of fuel oil

H_f = friction loss in the suction side piping. Calculated using suction pipe size, pipe length, viscosity, and flow rate. Includes elbows, valves, other fittings as equivalent length.

* *If the altitude at the installation was 2000 feet instead of sea level, the atmospheric pressure would have been 13.6 PSIA (see **Table 3**) and H_a would have been 36.1 feet of fuel oil.*

H_f = 2.9 feet of fuel oil*

H_{vp} = vapor pressure of fuel oil at 75ºF in feet of fuel oil absolute

H_{vp} = 1 foot of fuel oil (maximum)

$NPSH_a = H_a\ (38.6) - H_z\ (10) - H_f\ (2.9) - H_{vp}\ (1)$

$NPSH_a$ = 24.7 feet of fuel oil

In practice, an installation similar to this would have had a lower NPSHa because of additional losses (greater H_f) through a shut off valve and possibly a foot valve and/or a strainer. Because there is often entrained air in No. 2 fuel oil, the vacuum reading at the pump should not exceed 15" Hg. under the worst conditions, even though the NPSH available might indicate higher lifts could be handled satisfactorily.

If the liquid being pumped in **installation 1** was changed from No. 2 fuel oil to gasoline, the NPSHa would be quite different. With gasoline, the specific gravity would change to 0.71 and the vapor pressure to 8.5 PSIA. The formula for calculating NPSHa remains the same but the values change significantly.

$NPSH_a = H_a \pm H_z - H_f - H_{vp}$

$$H_a = 14.7\ \text{PSIA} \times \frac{2.31'\ \text{of}\ H_2O}{\text{PSI}} \times \frac{1'\ \text{of Gasoline}}{0.71'\ \text{of}\ H_2O}$$

H_a = 47.8 feet of gasoline

H_z = 10 feet of gasoline**

H_f = 2.9 feet of gasoline

H_{vp} = 8.5 PSIA (winter gasoline at 75ºF)

$$H_{vp} = 8.5\ \text{PSIA} \times \frac{2.31'\ \text{of}\ H_2O}{\text{PSI}} \times \frac{1'\ \text{of Gasoline}}{0.71'\ \text{of}\ H_2O}$$

H_{vp} = 27.7 feet of gasoline

$NPSH_a = H_a\ (47.8) - H_z\ (10) - H_f\ (2.9) - H_{vp}\ (27.7)$

$NPSH_a$ = 7.2 feet of gasoline

Note the big difference the vapor pressure has made in the NPSHa from the same system, 24.7 feet of fuel oil versus 7.2 feet of gasoline.

TABLE 3:

Effect of Altitude on Atmospheric Pressure and Barometer Readings

Altitude Above Sea Level in Feet	Atmospheric Pressure - Pounds Per Square Inch	Barometer Reading - Inches of Mercury
0	14.7	29.929
1000	14.2	28.8
2000	13.6	27.7
3000	13.1	26.7
4000	12.6	25.7
5000	12.1	24.7
6000	11.7	23.8
7000	11.2	22.9
8000	10.8	22.1
9000	10.4	21.2
10000	10.0	20.4

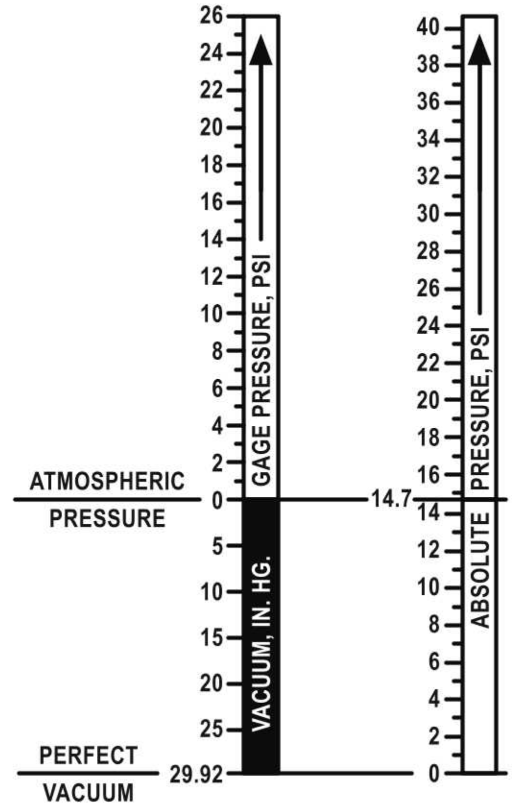
* *See pressure loss charts in Section 510 of Viking General Catalog. The calculations in Installations 2-4 provide additional details on how H_f is calculated.*

** *From a practical standpoint, the normal lift should not exceed 6 feet when handling gasoline.*

The NPSHa of 7.2 feet of gasoline would normally be less due to additional friction losses (H_f) through a shut off valve, foot valve, and strainer. It would also be affected by changes in tank level, temperature, and source of supply.

FIGURE 2:

Comparison Between Absolute and Gage Pressure



***Note:** To convert pressure in psi to feet of liquid, multiply by 2.31 and divide by specific gravity of the liquid.*

To convert inches of mercury to feet of liquid, multiply by 1.133 and divide by specific gravity of the liquid.

Since it is mandatory that all calculations be made in or converted to the same pressure unit, an understanding of the more frequently used units is necessary before doing any work related to NPSH.

The need for converting becomes obvious when it is remembered that **atmospheric pressure** is often given in inches of mercury absolute ("Hg. abs.) or PSIA, **vapor pressure** in mm Hg. absolute (mm Hg. abs.) or PSIA, **elevation** in feet of liquid, and **line loss** in PSI or feet of liquid. **Figure 2** compares the two basic pressure systems – Absolute and Gage. Always keep in mind that the gage pressure system zero point (atmospheric pressure) varies according to the actual elevation above sea level of the pumping site and that the atmospheric pressure at any given site can vary ± 1" Hg. from an average value. Thus the standard atmospheric pressure of 14.7 PSIA of 29.9" Hg. abs. is good as a reference but actually would seldom be the pressure at a particular site at any given time.

TABLE 4:

Conversion Table

Pressure Unit	Atmospheres, Atmos.	Feet of Water, ' H_2O	Inches of Mercury, " Hg	Kilograms per square centimeters, Kg/cm^2	Millimeters of Mercury, mm Hg	kPa	Pounds per square inch, PSI
Atmospheres, Atmos.	1	33.9	29.9	1.03	760	101.3	14.7
Feet of Water, ' H_2O #	0.029	1	0.883	0.030	22.4	2.99	0.433
Inches of Mercury, " Hg	0.033	1.13	1	0.034	25.4	3.39	0.49
Kilograms per square centimeters, Kg/cm^2	0.968	32.8	28.9	1	736	98.07	14.2
Millimeters of Mercury, mm Hg	0.00131	0.0446	0.0394	0.00136	1	0.1335	0.01935
Kilopascal, kPa	0.00987	0.3345	0.295	0.0102	7.4938	1	0.1450
Pounds per square inch, PSI	0.0680	2.31	2.04	0.0703	51.7	6.895	1

To convert from a pressure unit in the left hand column, multiply the numerical value times the factors in the vertical column showing the unit you are converting to. For example, to convert 50 feet of water to PSI, go horizontally to the right from the "Feet of Water" unit to the box under PSI, the factor is 0.433. The 50 feet of water would then be multiplied by 0.433 to get 21.7 PSI.

100 kPa = 1Bar

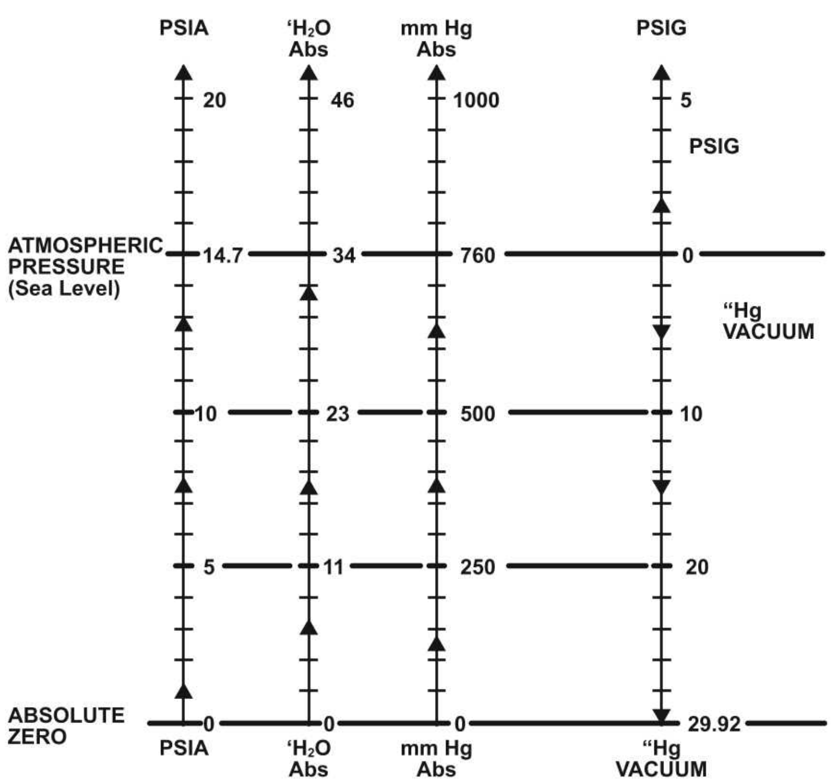
For more information, see Engineering Section 510.22 Conversion Factors.

Table 4 shows the factors to use for converting from one pressure unit to another. Those shown are the most frequently used. Other units are occasionally encountered. Conversions for these other units can be found online or in engineering handbooks. As the SI or modern metric system of measurement becomes more widely used in the United States, we can expect to see pressures expressed in pascals (Pa) or kilopascals (kPa).

Figure 3 gives a visual comparison between the more frequently used pressure units. The conversions are not quite as exact as the chart would indicate, so actual conversion factors should be used when doing calculations for a particular system.

FIGURE 3:

Pressure Unit Comparison Chart



VAPOR PRESSURE

In the formula used to calculate the NPSH available in a system, we included the vapor pressure factor and used typical values for the liquids involved. At that time, we did not define or discuss vapor pressure. We will do that here since an understanding of vapor pressure is as vital to properly comprehending NPSH as is a good understanding of the relationship between the various pressure units.

Vapor pressure is one of the physical properties of a liquid. By definition, it is the pressure exerted by the vapors of a confined liquid. It varies with temperature.

Liquids such as water and fuel oil that can be stored in open containers at ambient temperatures have a relatively low vapor pressure (well below atmospheric pressure). For liquids of this type, the vapor pressure is normally expressed in mm Hg. abs. or PSIA. Liquids such as LP-Gas, Freons, and ammonia, which must be stored in closed containers, have a relatively high vapor pressure (considerably above atmospheric pressure). For such liquids, the vapor pressure is often expressed in PSIG.

For fuel oils and similar liquids, the vapor pressure can be measured by putting some of the liquid above the mercury in a barometer and noting the depression of the mercury column. For LP-Gas, the vapor pressure can be determined by reading a gage attached to the vapor section of a partially filled, enclosed container. The temperature should always be noted whenever such vapor pressure determinations are made.

The Reid vapor pressure is an important factor when selecting a gasoline blend for use during different seasons and for different locales to assure proper starting and performance of automobile engines. The Reid vapor pressure for summer gasoline is approximately 9.0 PSIA while for winter gasoline, it is around 13.5 PSIA. The lower vapor pressure for summer gasoline tends to keep it from vapor locking in the summer heat. The higher vapor pressure for winter gasoline helps it vaporize in the carburetor at wintertime temperatures. Remember that the Reid vapor pressure is taken at 100°F.

For comparison with gasoline, the vapor pressure at 100°F of several solvents and LP-Gases is shown below:

Xylene – 16 mm Hg. abs.

Water – 47 mm Hg. abs.

Toluene – 54 mm Hg. abs.

Acetone – 391 mm Hg. abs.

Butane – 37 PSIG

Propane – 170 PSIG

As mentioned earlier, the factors connected with NPSH determinations are not always precise. While the vapor pressure of a pure liquid is consistent and predictable, it is possible to have entrained or dissolved air or gas in the liquid with the result that the fluid acts as though it had a higher vapor pressure than figures for the pure liquid indicate. In this case as the pressure in the suction system is reduced, any entrained air in the liquid will tend to expand or any dissolved gas will tend to be released. As this occurs, there will be less liquid coming into the pump, resulting in a lower volumetric efficiency and possibly erratic flow. This situation happens most frequently in a system where the liquid receives considerable agitation as it is being transported or unloaded or where the liquid is being continuously recirculated. A good example of a typical pumping application involving these problems is one handling #2 fuel oil. Air is often entrained as the oil is moved from the jobber or distributor to the user in their heating oil system. The vapor pressure of #2 fuel oil at normal ambient temperatures is virtually nil, but because of possible problems resulting from expansion of air entrained during hauling and recirculating, it is seldom wise to operate a pump handling #2 fuel oil with a vacuum greater than 15" Hg.

Refer to AD-22 for additional information on pumping fuel oil.

A similar situation can occur when handling volatile petroleum products since the liquid is probably made up of many different fractions, each with a vapor pressure of its own. For such a liquid, a slight vacuum might cause vaporizing of the lighter fractions.

In summary, if the suction side of a system is so designed that the pump must develop or "pull" a vacuum that results in an absolute pressure less than the vapor pressure of the liquid being handled, the liquid will tend to vaporize (form bubbles or "boil"). This formation of vapor on the suction side of the system reduces capacity and can cause vapor lock. The collapse of the vapor bubbles on the discharge side of the pump causes noise, vibration, and rapid wear. This is the phenomenon which we are trying to avoid by developing an understanding of NPSH, learning to calculate it, and thus encourage system designs which provide an NPSH available greater than the NPSH requirement of the pump. NPSHa > NPSHr.

NPSH REQUIRED – NPSHR

Net Positive Suction Head Required – is another way of indicating the pressure loss within the pump itself. Liquid cannot flow from one point to another without a loss of pressure. For pipe, this pressure loss or drop has been calculated, checked through testing, and tabulated in charts or plotted in curves. The loss is expressed in PSI per foot of pipe length or in feet of head loss per 100 feet of pipe.

Data is available for all standard pipe sizes and for a viscosity range from water thin to several hundred thousand SSU. A study of this information readily shows that the pressure loss increases as the rate of flow increases for a given pipe size and viscosity and that the loss increases as the viscosity increases for the same pipe size and flow rate.

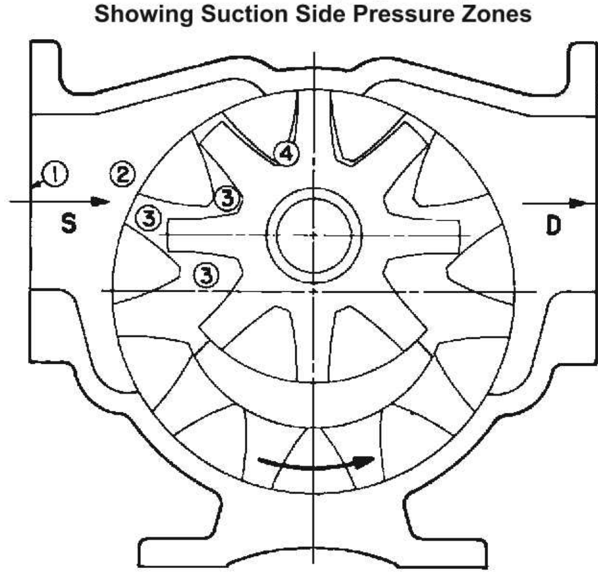
Determining the pressure loss for flow through a pipe is relatively easy. Calculations have also been made and tests conducted to determine the pressure loss through pipe fittings, various types of valves, and other items found in a piping system. This data, as that for pipe, shows increasing loss with increasing flow rate and viscosity.

For low viscosity liquids with turbulent flow, the loss through a fitting or valve is expressed as the loss in so many feet (equivalent length) of straight pipe. For laminar flow, the pressure loss through the same fitting or valve may be expressed as a percentage of the equivalent length of pipe loss for turbulent flow, the percentage decreasing as the viscosity increases. See Engineering Section 510.12. A study of the loss information for fittings and valves indicates that it may not be as accurate and easy to determine as that for straight pipe. This is logical when we consider that the liquid flowing through a fitting or valve may experience a drastic change in direction and may go through sudden contractions and/or enlargements.

The point of this discussion is that the more complex the path of liquid flow, the more difficult it is to accurately determine the pressure loss between any two points short of conducting tests under actual conditions.

As there is a pressure loss between any two points in a piping system so there is a pressure loss or drop between any two points within a pump. In addition to a short run of straight flow at the entrance to the suction port, the flow within most positive displacement pumps, depending upon principle, will next experience a change of direction from a few degrees to more than 90º. PD pumps may also have a sudden enlargement or contraction at the entrance to the suction cavity. There may be a disruption to smooth liquid flow caused by the moving pumping elements, the extent again depending on the pumping principle.

FIGURE 4:
Internal Gear Pumping Principle Showing Suction Side Pressure Zones



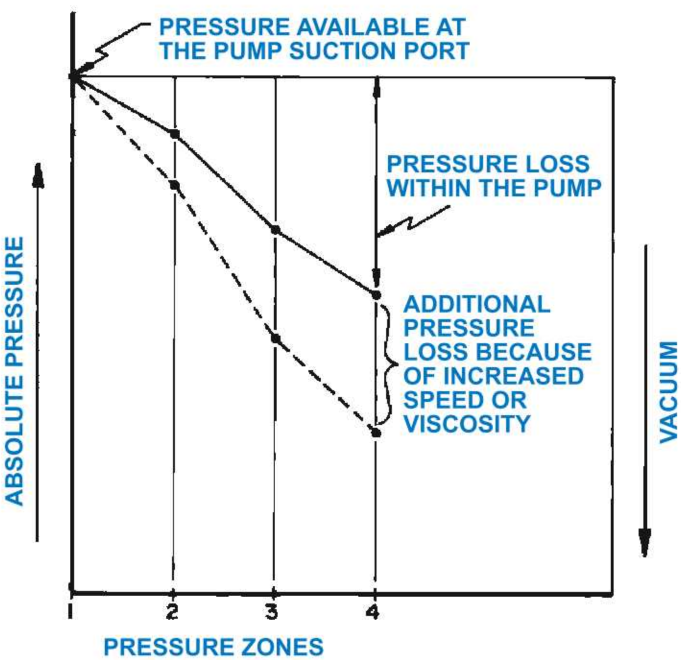
As stated earlier, NPSH required is another way of stating the pressure loss in the suction area of a pump for a given set of conditions. **Figure 4** shows the internal gear pumping principle with numbered points or zones of progression from the suction port to the point of minimum absolute pressure (maximum vacuum).

1. Zone 1 is at the suction port of the pump.
2. Zone 2 is where the port throat opens into the suction area of the casing.
3. Zone 3 is in the area where the pumping elements (rotor and idler) are being filled with liquid.
4. Zone 4 is where the rotor and idler teeth are coming out of mesh; the point of lowest absolute pressure (highest vacuum).

The curve in **Figure 5** shows in a generalized manner the gradual loss of pressure as liquid progresses from the suction port (Zone 1) to the point where the rotor and idler teeth come out of mesh (Zone 4).

FIGURE 5:

Absolute pressure vs pump side pressure zones



Visualizing the pump in operation, it is easy to see that as the pump RPM increases the pressure loss will also increase. As there is an increase in pressure loss in a pipe with increased flow so there is in the throat area (Zone 1) of the pump suction port. As the pressure loss increases with increased flow of liquid through the zig zag path in a valve so it also increases as more liquid flows into the faster moving pumping elements. And finally, since the time for the liquid to fill the void at Zone 4 is shortened as the pump RPM goes up, the liquid must move faster, which requires more pressure (experiences a larger pressure loss). All of these factors explain why increasing pump RPM does increase the pressure loss in the pump.

As all pipe line loss charts show, increasing viscosity results in greater line loss for the same pipe size and flow rate, which also holds true for within a pump as well. For a given RPM, the pressure loss will increase as the viscosity (a liquid's resistance to flow) increases. See the dashed line in **Figure 5**.

Any time a pump is operating, there is a pressure loss within the pump as has just been discussed and as shown in **Figure 5**. Under normal conditions on a well-designed system, even though there is a pressure loss between Zone 1 and Zone 4, the pump pressure at Zone 1 is high enough so that the pressure at Zone 4 does not drop below the vapor pressure of the liquid. As long as this is the case, cavitation does not occur and there is no pumping problem. For a system handling a viscous liquid as long as the pressure at Zone 1 is high enough to assure a complete filling of the pumping elements in Zone 3, there will be no "starvation" (incomplete filling of the pumping elements) and thus no problem.

It is when the pressure above the vapor pressure of the liquid (NPSHa – Net Positive Suction Head Available) at Zone 1 is not sufficient to overcome the pressure loss within the pump that cavitation or "starvation" will occur.

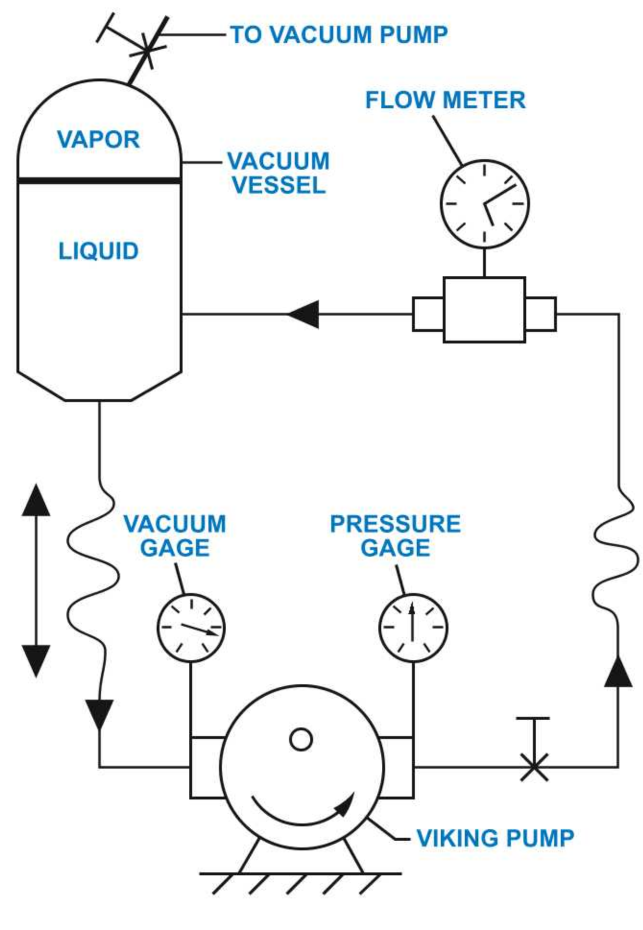
An engineer or designer, when laying out a piping system, should determine the amount of pressure (NPSHa) that will be available at the pump port. It is also important to know what the pressure loss (NPSHr) is for the various pumps available so that a proper selection can be made. Thus, a line of pumps for which there is no NPSHr information may not be given consideration when a pump selection is being made.

Viking has accumulated a wealth of practical field experience over the past 100+ years in applying pumps to applications with limited NPSHa. Through R&D testing, Viking has also gathered NPSHr data for a wide range of speeds and viscosities on the complete pump line.

The drawing in **Figure 6** shows the arrangement of the various pieces of equipment actually used in conducting the NPSHr tests in the R&D Lab.

FIGURE 6:

Drawing of Test Set-up for Checking NPSHr



In theory, the procedure being used to collect data for making NPSHr determinations would be as follows; in practice some revisions have been made to the procedure to stay within the limitations of the equipment:

1. Use the vacuum pump to reduce the pressure in the vacuum vessel to the vapor pressure of the liquid (when using fuel oils or lube oils as test liquids, the pressure in the vacuum vessel approaches zero absolute [29.92" Hg. vacuum] since their vapor pressures are almost nil).
2. Raise the vacuum vessel above the pump.
3. Start the pump and set the speed.
4. Adjust the height of the vacuum vessel to a point that gives a vacuum gage reading (approximately 3.4" Hg. vacuum) that is equivalent to an absolute pressure of 30 feet of liquid (specific gravity of 1.0). Set discharge pressure at 50 psig. Record capacity.
5. Lower the vacuum vessel to a point that gives a vacuum gage reading equivalent to an absolute pressure of 25 feet of liquid. Record capacity.
6. Continue lowering the vacuum vessel and recording capacity until an absolute pressure of approximately zero feet is reached. Capacity at this point will also be zero.

   As the vacuum vessel is lowered, there will be a point – normally between 10 and 2 feet of absolute pressure – where the capacity will start to drop off, which indicates that the pump is cavitating or "starving". It is at this point that more pressure is required to get the liquid into the pumping elements than is available at the suction port and as a result, the elements are not being completely filled with liquid.
7. Repeat the above procedure at various speeds and viscosities.

The curves shown in **Figure 7** are plotted from actual test data taken according to the just reviewed procedure for a Viking Model K124A pump handling 38 SSU liquid. The actual NPSHr value for the various speeds is indicated by the "X" on the capacity curve. It is determined as the point at which the pump capacity deviates from a straight line.

NPSH VS NPIP

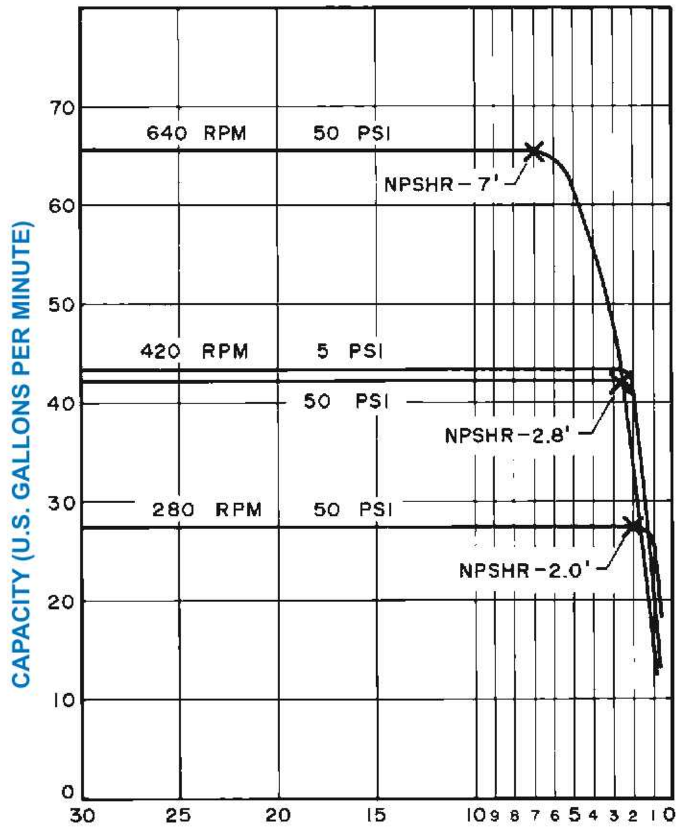
The Hydraulic Institute is advocating the use of the term Net Positive Inlet Pressure (NPIP) in place of NPSH when working with positive displacement pumps. As more spec sheets are written specifically for positive displacement pumps, there will be a growing use of the term Net Positive Inlet Pressure. Net Positive Inlet Pressure is expressed in PSIA. The Net Positive Inlet Pressure Required (NPIPR) is the counterpart to NPSHr. The Net Positive Inlet Pressure Available (NPIPA) is used in place of NPSHa. At the present time, information on Viking internal gear pumps will be given as NPSHr expressed in feet of liquid with a specific gravity of 1.0. To convert this to NPIPR in PSIA, multiply the value by 0.433.

NPSHA CALCULATIONS

The basic formula, $NPSH_a = H_a \pm H_z - H_f - H_{vp}$, was discussed earlier. As before, H_a is the absolute pressure on the surface of the liquid. H_z is the distance from the surface of the liquid to the centerline of the suction port. H_f is the pipe friction or line loss. H_{vp} is the vapor pressure of the liquid. All terms need to be expressed in the same units, feet of liquid pumped.

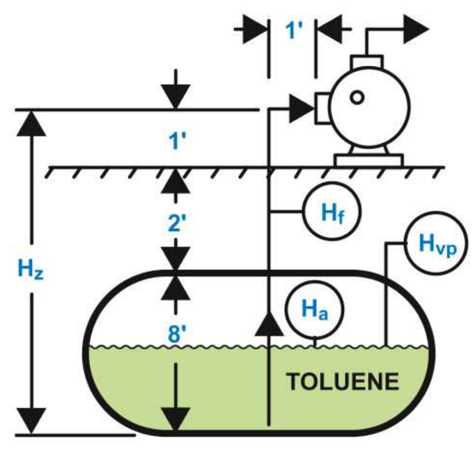
FIGURE 7:

Curves for Viking Pump Model K124A.
Plotting capacity vs. NPSHa when handling 38 SSU liquid.



In the following examples, the basic NPSHa formula is applied to three typical pump installations. These three are representative of the applications which most frequently have NPSH problems.

INSTALLATION 2 – SUCTION LIFT:



Transfer 50 GPM of Toluene from an 8' diameter horizontal buried tank 2' below grade. Normal tank temperature is 60ºF, specific gravity is 0.87, viscosity is 0.8 cPs (about the same as cold water), and the vapor pressure is 0.36 PSIA at 60ºF. Pump is to be installed at grade above the tank, line size is 2", and the job site is in the upper Midwest with an elevation of 2000'.

$NPSH_a = H_a \pm H_z - H_f - H_{vp}$

H_a = 27" Hg. absolute; based on an elevation of 2000' and a low barometer.

$$H_a = 27\text{" Hg. abs} \times \frac{1.13' \text{ of } H_2O}{\text{" Hg.}} \times \frac{1' \text{ of Toluene}}{0.87' \text{ of } H_2O}$$

H_a = 35 feet of Toluene

H_z = 8' + 2' + 1' = 11 feet of Toluene.

This is the distance from the surface of the liquid to the centerline of the pump. However, always figure worst condition with the tank empty. The liquid is below the pump; therefore, H_z will be negative.

H_f = Use the line loss through 35' of pipe [12' actual (11' vertical plus 1' horizontal) plus 23' equivalent for elbows, a gate valve, and a check valve]. Line loss in PSI per foot of pipe = 0.02 PSI/ft from information in Section 510 of General Catalog.

$$H_f = \frac{0.02 \text{ PSI}}{\text{ft. of piping}} \times \frac{2.31' \text{ of } H_2O}{\text{PSI}} \times \frac{1' \text{ of Toluene}}{0.87' \text{ of } H_2O}$$

H_f = 0.053' of Toluene/ft of piping

$$H_f = 35' \text{ of piping} \times \frac{0.053' \text{ Toluene}}{\text{ft. of piping}}$$

H_f = 1.9 feet of Toluene

H_{vp} = 0.36 PSIA at 60°F; use 1.7 PSIA, the vapor pressure at 120°F, which is the maximum possible summertime temperature at the pump.

$$H_{vp} = 1.7 \text{ PSIA} \times \frac{2.31' \text{ of } H_2O}{\text{PSI}} \times \frac{1' \text{ of Toluene}}{0.87' \text{ of } H_2O}$$

H_{vp} = 4.5 feet of Toluene

$NPSH_a$ = 35' – 11' – 1.9' - 4.5'

$NPSH_a$ = 17.6 feet of Toluene

$NPIPA = H_a \pm H_z - H_f - H_{vp}$

H_a = 27" Hg. absolute

$$H_a = 27\text{" Hg.} \times \frac{0.49 \text{ PSI}}{\text{" Hg.}}$$

H_a = 13.2 PSIA

$$H_z = 11' \text{ of Toluene} \times \frac{0.87' \text{ of } H_2O}{1' \text{ of Toluene}} \times \frac{1 \text{ PSI}}{2.31' \text{ } H_2O}$$

H_z = 4.1 PSIA

$$H_f = 35' \text{ of piping} \times \frac{0.02 \text{ PSI}}{\text{ft. of piping}}$$

H_z = 0.7 PSIA

H_{vp} = 1.7 PSIA

NPIPA = 13.2 – 4.1- 0.7 – 1.7

NPIPA = 6.7 PSIA

The NPSHr for a KK size Viking at 420 RPM is 3.3 ft (S.G. 1.0).

$$NPSH_r = 3.3' \text{ } H_2O \times \frac{1' \text{ of Toluene}}{0.87' \text{ of } H_2O}$$

NPSHr = 3.8 feet of Toluene

$NPSH_a$ = 17.6 feet of Toluene

The Net Positive Inlet Pressure Required (NPIPR) is equal to:
Ft of H_2O × 0.433

$$NPIPR = 3.3' \text{ } H_2O \times \frac{0.433 \text{ PSI}}{\text{ft of } H_2O}$$

NPIPR = 1.4 PSIA

NPIPA = 6.7 PSIA

Any self-priming pump with an NPSHr less than 17.6' should work.

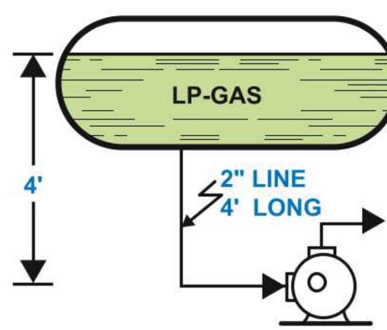
If in **Installation 2** the tank was installed above grade, the liquid level would be above the pump suction port. In this case the NPSHa would be significantly increased because the H_z factor would be positive instead of negative. This is not to say that such an installation should not be reviewed closely when at the design stage or that it should not be checked if pump problems develop at start up. It is quite possible to fall into the trap of believing that for an installation with a "flooded" suction (liquid level above the pump) there is no need to check the suction side of the pump.

"Flooded" suction means positive pressure at the pump suction port. Many times the suction port is not "flooded" because of long suction line, small piping, high viscosity, a fine mesh strainer, or other fittings, which together restrict the flow of liquid and cause NPSH problems.

NPSHa or NPIPA should always be calculated with extreme conditions to prevent start up problems.

You will note from the calculations for the NPSHa for **Installation 2** that several of the terms were figured on a conservative or extreme condition. As pump manufacturers, we feel that this is always the preferred approach. If problems develop after the system is installed, no amount of checking and verifying data or calculations will correct a situation that might not have developed if more conservative values had been used in the original NPSH calculations.

INSTALLATION 3 – HIGH VAPOR PRESSURE:



Transfer 30 GPM of LP Gas (Propane) at 65°F from a large storage tank. Specific gravity is 0.50; Viscosity is 0.1 cPs; Vapor pressure is 100.7 PSIG.

$NPSH_a = H_a \pm H_z - H_f - H_{vp}$

H_a = absolute pressure at surface of liquid

H_a = atmospheric pressure + vapor pressure

H_a = 14.7 + 100.7 = 115.4 PSIA

$$H_a = \frac{(115.4 \text{ PSIA}) \left(\frac{2.31' \text{ of } H_2O}{\text{PSI}}\right)}{0.50 \text{ S.G. of LP Gas}}$$

H_a = 533 feet of LP Gas

H_z = distance from liquid level to port centerline

H_z = +4 feet of LP Gas; this value is positive since the liquid is above the pump.

H_f = suction line loss from pressure drop charts (0.008 PSI/ft of pipe)

$$H_f = \frac{(4')\left(\frac{0.008 \text{ PSI}}{\text{ft of pipe}}\right)\left(\frac{2.31' \text{ of } H_2O}{\text{PSI}}\right)}{0.50 \text{ S.G. of LP Gas}}$$

H_f = 0.1 feet of LP Gas

H_{vp} = vapor pressure of LP Gas at 65°F

H_{vp} = 100.7 PSIG + atmospheric pressure of 14.7 PSIA = 115.4 PSIA

$$= \frac{(115.4 \text{ PSIA})\left(\frac{2.31' \text{ of } H_2O}{\text{PSI}}\right)}{0.50 \text{ S.G. of LP Gas}}$$

H_{vp} = 533 feet of LP Gas

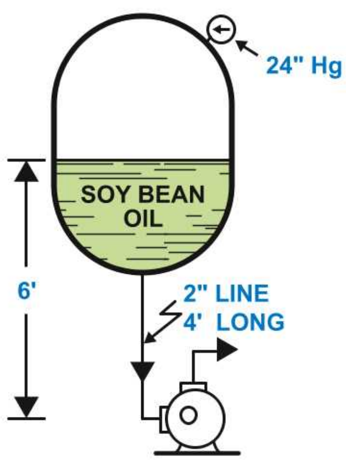
$NPSH_a$* = 533' + 4' – 0.1' – 533'

$NPSH_a$ = 3.9 feet of LP Gas

In addition to the LP gases, propane and butane, other high vapor pressure liquids that are frequently handled with Viking pumps are anhydrous ammonia and Freon.

Adequate NPSHa when handling the high vapor pressure liquids is obviously a must. There is evidence and some test data that indicates that these liquids, particularly the hydrocarbons, can be pumped satisfactorily with less NPSHa than would be required for cold water or other low vapor pressure test liquids. Under some conditions and with certain liquids, pumps will operate satisfactorily with as little as 50 percent of the NPSHa that might be required if handling cold water. Some guidelines for the NPSHa required for Viking LP Gas pumps are given in the General Catalog Section 440 and Technical Service Manuals TSM 442 and 443.

INSTALLATION 4 – VACUUM VESSEL:



Transfer 40 GPM of soybean oil at 240°F from a vessel under 24" Hg. vacuum. Viscosity is 40 SSU; specific gravity is 0.88.

$NPSH_a = H_a \pm H_z - H_f - H_{vp}$

H_a = absolute pressure at surface of liquid. Assume a 1000' elevation with a low barometer (use 27.8" Hg. abs. as barometric pressure).

* *Note that for a high vapor pressure liquid (a liquid which has to be stored in a closed vessel to keep it from boiling away) the H_a and H_{vp} terms cancel out and that NPSHa becomes the difference between the elevation head and the line loss; $H_z - H_f$.*

$$H_a = \frac{(27.8 - 24\text{" Hg})\left(\frac{1.13' \text{ of } H_2O}{\text{" Hg}}\right)}{0.88 \text{ S.G. of Soybean Oil}}$$

H_a = 4.9 feet of soybean oil

H_z = distance from liquid level to port centerline

H_z = +6 feet of soybean oil; this value is positive since the liquid is above the pump.

H_f = suction line loss in feet of soybean oil. Ref. Sec. 510 of Viking General Catalog for line loss charts.

$$H_f = \frac{(4')\left(\frac{0.008 \text{ PSI}}{\text{ft of pipe}}\right)\left(\frac{2.31' \text{ of } H_2O}{\text{PSI}}\right)}{0.88 \text{ S.G. of Soybean Oil}}$$

H_f = 0.2 feet of soybean oil

H_{vp} = vapor pressure of soybean oil at 240°F

H_{vp} = 2 feet of soybean oil (estimate)

$NPSH_a$ = 4.9' + 6' – 0.2' – 2'

$NPSH_a$ = 8.7 feet of soybean oil

As with **Installation 2**, it is recommended when making calculations at the design stage for a system of this type to be conservative wherever possible.

Installation 4 is also quite typical of applications involving transformer oil purifying, lube oil reclaiming, and solvent recovery. One important point to remember on this type of application is that the pump is at the low point in the system and is, therefore, wetted at start up and able to develop a good initial vacuum. A pump with a lift often has to evacuate air or vapor from the suction line before it becomes wetted with the liquid being pumped.

Another point to keep in mind is that on some applications, the liquid is quite viscous and that it is possible to have more friction loss per foot of vertical pipe than there is gain because of elevation. Larger pipe size will correct this situation.

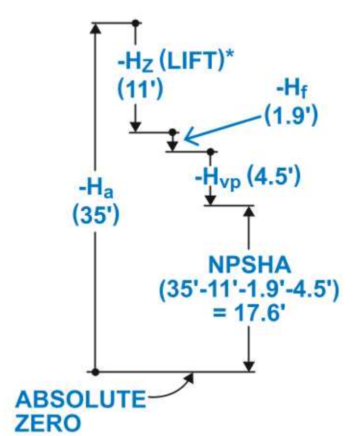
Expressing the information in the formula by a line chart can also be of help in visualizing how the actual NPSHa figure is determined. **Figure 8** shows the steps taken in arriving at the NPSHa for **Installation 2**.

FIGURE 8:

Line chart showing steps taken in calculating NPSHa.

ALL VALUES ARE IN FEET OF TOLUENE

REFERENCE INSTALLATION 2 - SUCTION LIFT HANDLING TOLUENE



* *If the liquid level was 11' above the pump H_z would be plus and would be added to instead of being subtracted from H_a. The NPSHa would then be 39.6'.*

DETERMINING MAXIMUM SUCTION LIFT

The NPSHa formula is most often used for making calculations in conjunction with existing or completely designed systems as we have just done for **Installations 2, 3, and 4**. It can also serve as an effective tool when used in the early stages of system design to determine limiting conditions.

One of the admonitions stated several times is that to have an acceptable installation, the NPSHa of the system must be greater than the NPSHr of the pump. If a pump is selected in the early stages of system design, it is then possible to determine the maximum allowable lift by equating NPSHa to NPSHr and solving for H_z. As an example consider for **Installation 2** if the question had been raised, "What is the maximum vertical lift possible?"

From data available for the pump selected, determine the NPSHr. For this example, assume that the NPSHr for the pump selected is equal to 3.5' of Toluene; use 5.0' to be conservative. Then $NPSH_a = NPSH_r = 5.0' = H_a - H_z - H_f - H_{vp}$. Values for the different factors are taken from the calculation for **Installation 2**.

H_a	= 35 feet of Toluene
H_z	= unknown
H_f	= 0.053* (H_z + 24'**)
H_f	= $0.053H_z + 1.3$
H_{vp}	= 4.5 feet of Toluene
5.0'	= $H_a - H_z - H_f - H_{vp}$
5.0'	= 35' – H_z – (0.053H_z + 1.3') – 4.5'
5.0'	= 35' – 1.053H_z – 1.3' – 4.5'
1.053H_z	= 35' – 1.3' – 4.5' – 5.0'
H_z	= 23.0 feet of Toluene

A vertical lift of 23.0' is higher than would normally be encountered, but the exercise does illustrate an additional use for the NPSH formula. The same approach applied to other systems would permit determination of 1) the minimum liquid leg required above a pump when handling a high vapor pressure liquid or when pulling from a vacuum vessel; 2) the proper line size (determined by the H_f factor) when the lift or liquid leg is predetermined.

DETERMINING NPSHA BY GAGE

It is possible by using a pump suction port gage to determine the NPSHa of an operating system without making all of the calculations shown for **Installations 2, 3, and 4**.

By definition from an earlier chapter, NPSH equals "the pressure in feet of liquid absolute measured at the pump suction port less the vapor pressure." If we introduce a new term, Hi, and define it as the absolute pressure of the liquid at the pump suction port expressed in feet of liquid, we can set it equal to the first three terms of the NPSHa formula:

$$\mathbf{H_i = H_a \pm H_z - H_f}$$

* *Line loss in feet of Toluene per foot of pipe.*

** *One foot of horizontal pipe plus 23' of equivalent length equals 24'.*

If you restate the NPSHa formula using the Hi term, it becomes: $\mathbf{NPSH_a = H_i - H_{vp}}$

The value of Hi can be determined by converting a suction port gage reading into feet of liquid absolute. When making the conversion use the local absolute barometric pressure, the specific gravity of the liquid, and the appropriate factors. The final step is to subtract the vapor pressure of the liquid at the pumping temperature to arrive at the NPSHa for the system.

To illustrate, refer to **Installation 2** and assume the unit is operating under the following conditions: liquid temperature is 60°F, the tank is one-half full, and the barometer is 27" Hg. absolute. The vacuum gage reads 6" Hg. Determine the NPSHa of the system.

The suction port gage reading of 6" Hg. vacuum subtracted from an absolute barometric pressure reading of 27" Hg. gives a pressure reading at the pump of 21" Hg. absolute. The calculations below convert the 21" Hg. absolute to 27.3 feet of Toluene absolute; this is the value for the Hi term.
Calculation –

$$21\text{" Hg} \times \frac{1.13\text{' of } H_2O}{\text{" Hg.}} \times \frac{1\text{' of Toluene}}{0.87\text{' of } H_2O}$$

The vapor pressure at 60°F is 0.36 PSIA or one foot of Toluene.

$NPSH_a = H_i - H_{vp} = 27.3' - 1' = 26.3'$ of Toluene

This is considerably greater than the 17.6' determined by calculations for **Installation 2**. The difference, of course, relates to the fact that for calculation purposes we used the extreme conditions of having an empty tank and 120°F operating temperature with its resulting higher vapor pressure.

RECOMMENDATIONS FOR REDUCING NPSH PROBLEMS

In the discussion following the calculations for **Installations 2, 3, and 4**, the desirability of using conservative values for the factors affecting NPSH was mentioned several times. Some specific comments and suggestions regarding each of the factors follows:

$\mathbf{H_a}$ – the absolute pressure on the surface of a liquid – does not lend itself to much change or increase since it is a function of either atmospheric pressure or the process.

$\mathbf{H_z}$ – an indication of the location of the surface of the liquid with respect to the pump suction port – is relatively easy to change at the design stage; it may be very difficult to change after an installation is in operation.

$\mathbf{H_f}$ – line loss or pressure drop – is relatively easy to reduce when designing the system. For example, larger line sizes, fewer elbows, larger strainers, different type shutoff valve, etc. all tend to reduce the line loss. Such changes may result in added initial expense, but the added expense is justified if the changes are the difference between a system that works and one that does not.

To make changes in the piping system after it is installed to reduce the line loss value is difficult and very expensive. For an operating system, it may be possible to reduce line loss by increasing the temperature and thus reducing the viscosity. This is okay if the reduction in line loss is not more than offset by an increase in vapor pressure. It may also be possible to reduce the line loss (H_f) by reducing the flow rate. For a continuous operation, this may not be practical. For a batch type operation or a transfer job, reducing the flow rate may be a practical means of increasing NPSHa.

H_{vp}- vapor pressure of the liquid at operating temperature – is tied to ambient conditions or to the process. In those few cases where it may be possible to reduce the vapor pressure by lowering the temperature, it should only be done if the necessary reduction in temperature does not cause a more than offsetting increase in the line loss (H_f) because of higher viscosity.

If changes to the system are not practical to reduce NPSH problems, the NPSHr of the pump should be reviewed. For a given pump, reducing the speed (and thus lowering the capacity) will reduce the NPSHr; it will also increase the NPSHa because the line loss (H_f) will be smaller. If it is not practical to reduce the capacity because of system or process requirements, a larger pump running slower will be able to deliver the same capacity with a lower NPSHr figure.

THINGS TO CONSIDER

When an actual NPSHa increase of one foot of liquid can spell the difference between an acceptable installation and one that is troublesome, it seems worthwhile to pursue all avenues that might reduce potential problems. The following points or variables can prove to be a source of trouble if not given proper consideration.

1. Liquid level in the supply vessel. In a storage tank, this level goes up and down as the product is used and as the tank is refilled. If the NPSHa is marginal, problems may cycle with the level in the tank.

   Seldom is liquid in the bottom quarter of a buried tank used, therefore, the suction pipe does not extend clear to the bottom of the tank. On those occasions when the liquid level is unusually low, it is possible to get a bit of swirling or vortexing at the inlet to the suction pipe with resulting air entrainment and reduced capacity.

   Many supply vessels and vacuum vessels use float actuated controls to maintain liquid level. Sometimes there is more variation in liquid level than was intended. Often location of the start and stop levels can be changed to help an NPSH problem.

2. Ambient temperatures. Higher temperatures than anticipated may cause the product to vaporize in the suction line or may raise the vapor pressure. Lower temperatures may cause problems because of increased viscosity and line losses.

3. Suction gage location. A suction gage on the tank side of a strainer does not give the proper picture of conditions at the pump, although it may only be a couple of feet from the suction port.

4. Light ends vaporizing. Some liquids, particularly fuels oils, may have light fractions which will vaporize under slight vacuum conditions and cause cavitation and noise. The problem may come and go when handling heavy fuel oils depending on the source of supply and whether the particular oil was a blend or a straight distilled product. Lowering the oil temperature a few degrees may help.

5. Air in the liquid. Dissolved or entrained air in a product may also cause problems if allowance is not made for this possibility in the design of the system. The extent of potential problems can best be determined by discussing the actual liquid with the user to get as much information about it as possible. If a liquid containing entrained air can be left undisturbed for several hours, some of the air will rise to the surface and escape.

GENERAL COMMENTS

As was mentioned earlier, working with NPSH involves a number of factors and variables, often interrelated, which must be given consideration. Thus Viking, as a pump manufacturer, tends to be somewhat conservative when making recommendations for a system or a pump on an application which may involve potential NPSH problems. As you can appreciate, if the system doesn't work, the first thing that is suspect is the pump, which is the heart of the system.

If you have followed through the calculations, you can appreciate that it is necessary to understand the conversion between the various pressure units.

The Viking R&D Laboratory has completed the NPSHr testing of all the pump models currently catalogued. It is one more tool in the hands of the Viking marketing organization which will permit applying Viking pumps to the fullest extent of their capabilities while giving assurance that the pump will perform satisfactorily.

NET POSITIVE SUCTION HEAD REQUIRED BY VIKING PUMPS

TABLE 5: NPSHR FOR VIKING INTERNAL GEAR PUMPS

NPSHr - Feet of Liquid S.G. 1.0

PUMP SIZE	PUMP SPEED [RPM]															
	100	125	155	190	230	280	350	420	520	640	780	950	1150	1450	1750	2900
C												1.7	1.9	2.2	2.4	
F, FH										1.8	1.9	2.1	2.3	2.8	3.4	
GS, G, GG								1.8	2.0	2.2	2.6	3.1	3.9	5.6	7.6	
H, HJ, HL					1.7	1.8	1.9	2.1	2.4	2.8	3.4	4.5	6.2	9.5	13.5	37.0
AS, AK, AL			1.6	1.7	1.8	2.0	2.3	2.7	3.2	3.9	5.5	7.7	11.2	17.0	23.3	
J, KS, K, KK		1.7	1.8	1.9	2.1	2.3	2.8	3.3	4.4	6.3	9.1	24.4				
L, LQ, LL, LS	1.7	1.8	2.0	2.2	2.5	3.0	3.8	5.0	7.3	10.8	17.7	32.0				
Q, QS	1.9	2.1	2.3	2.7	3.3	4.2	6.1	8.4	12.7							
M	2.1	2.3	2.8	3.4	4.3	6.0	9.0	12.7			Catalog Speed Rating[4]					
N	2.1	2.5	3.5	4.5	6.3	9.5	15.0	29.5								
R	6.1	7.1	8.3	10.1	12.1	15.2										
RS	7.0	8.5	10.4	13.1	17.2	22.4										

1. *NPSHa (Net Positive Suction Head Available) must be greater than the NPSHr (Net Positive Suction Head Required) given in the above table.*
2. *VISCOSITY - Above chart applies to viscosities up thru 750 SSU. Consult factory or Viking representative for viscosities above 750 SSU.*
3. *For liquids other than water, divide by specific gravity.*
4. *Limit for most models. Some models exceed this limit. See Viking Pump catalog for details.*
5. *Hygienic Series Pumps (157B, 4157B, 257B, 4257B) have different NPSHr values than shown in Table 5. NPSHr tables for the Hygienic Series can be found in TR-126.*
6. *Motor Speed Pumps have different NPSHr. Refer to Table 6 below.*

TABLE 6: NPSHR FOR VIKING MOTOR SPEED PUMPS

NPSHr - Feet of Liquid S.G. 1.0

PUMP SIZE	PUMP SPEED [RPM]							
	420	520	640	780	950	1150	1450	1750
GG	1.1	1.6	2.2	2.6	3.1	3.9	5.6	7.6
HJ, HL	1.4	2	2.8	3.4	4.5	6.2	9.5	13.5
AS, AK, AL	1.9	2.8	3.9	5.5	7.7	11.2	16.8	23.3
KE, KKE	4	4.4	4.9	5.7	7	8.9	12.8	17.9
LQE, LSE	7.3	8.1	9.3	10.9	13.1	16.1		
Q, QS	5.5	6.8	9				Catalog Speed Rating	

1. *NPSHa (Net Positive Suction Head Available) must be greater than the NPSHr (Net Positive Suction Head Required) given in the above table.*
2. *VISCOSITY - Above chart applies to viscosities up thru 750 SSU. Consult factory or Viking representative for viscosities above 750 SSU.*
3. *For liquids other than water, divide by specific gravity.*

TABLE 7: NPSHR FOR VIKING EXTERNAL GEAR PUMPS

NPSHr - Feet of Liquid S.G. 1.0

PUMP SIZE	PUMP SPEED [RPM]						
	640	950	1150	1450	1750	3000	3450
- 04 (All Sizes)	N/A	N/A	N/A	N/A	9.5	11.5	13.0
- 0518	3.2	3.2	3.2	3.2	3.2	5.0	6.2
- 0525	3.2	3.2	3.3	3.4	3.4	5.4	6.6
- 0535	3.3	3.4	3.5	3.6	3.7	5.8	7.1
- 0550	3.4	3.6	3.8	4.0	4.3	6.6	8.0
- 0570	3.6	4.1	4.4	4.9	5.5	8.2	9.8
- 0510	2.5	2.7	2.8	3.7	4.3	8.2	9.8
- 0514	2.5	2.7	2.8	3.7	4.3	9.0	11.5
- 0519	2.6	2.8	3.2	4.0	5.0	10.8	13.5
- 0528	3.0	3.4	3.9	4.9	6.2	14.0	17.5
- 0729	3.0	3.1	3.2	3.8	4.8	11.1	15.6
- 0741	3.0	3.1	3.2	3.8	4.8	11.3	16.0
- 0758	3.0	3.1	3.2	3.8	4.8	11.8	16.6
- 0782	3.0	3.1	3.2	3.8	4.8	12.8	17.6
- 0711	3.0	3.1	3.2	3.8	5.0	15.2	20.6
- 0716	3.0	3.1	3.2	4.5	6.5	21.0	27.6
- 0722	3.0	3.1	3.2	3.8	5.0	-	-
- 0732	3.0	3.1	3.2	4.5	6.5	-	-
- 1009	5.5	6.3	7.0	8.4	10.3	-	-
- 1013	5.6	6.6	7.5	9.2	11.5	-	-
- 1026	5.8	6.9	8.0	10.1	12.8	-	-
- 1420	6.9	8.2	9.4	11.8	15.0	-	-
- 1436	7.0	8.7	10.2	13.3	17.5	-	-
- 1456	7.2	9.3	11.2	15.3	20.9	-	-

1. *NPSHa (Net Positive Suction Head Available) must be greater than the NPSHr (Net Positive Suction Head Required) given in the above table.*

2. *VISCOSITY - Above chart applies to viscosities up thru 750 SSU. Consult factory or Viking representative for viscosities above 750 SSU.*

3. *For liquids other than water, divide by specific gravity.*

TABLE 8: NPSHR FOR VIKING VANE PUMPS

NPSHr - Feet of Liquid S.G. 1.0

PUMP SIZE	PUMP SPEED [RPM]													
	100	125	155	190	230	280	350	420	520	640	780	950	1150	1450	1750
01									4.7	5.3	6.1	7.3	9.0	12.3	16.4
02									4.7	5.3	6.1	7.3	9.0	12.3	16.4
05							6.2	6.9	8.0	9.8	12.4	16.3	21.9		
08							9.8	11.0	13.0	16.0	20.3	26.9			
19			3.2	3.4	3.8	4.5	5.9	8.0	12.5						
23			3.9	4.3	5.0	6.2	8.8	12.8	20.8						

← *Catalog Speed Rating*

1. *NPSHa (Net Positive Suction Head Available) must be greater than the NPSHr (Net Positive Suction Head Required) given in the above table.*

2. *VISCOSITY - Above chart applies to viscosities up thru 100 SSU. Consult factory or Viking representative for viscosities above 100 SSU.*

3. *For liquids other than water, divide by specific gravity.*

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