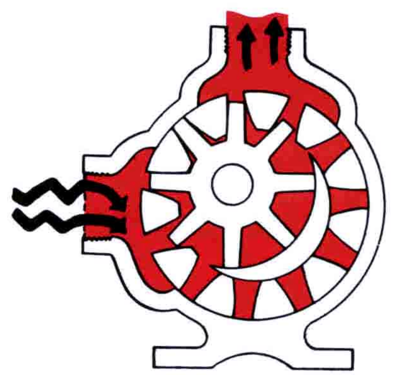
NET POSITIVE SUCTION HEAD

NPSH



AD-19
(ISSUE A)

Few subjects relating to the application of positive displacement (p.d.) pumps are more discussed and less understood than Net Positive Suction Head — NPSH. Until a few years ago it was felt that NPSH was a term reserved for centrifugal pumps handling water and that it was not a subject of concern when applying slow speed p.d. pumps to simple low pressure transfer jobs. This is no longer a valid assumption.

TODAY — with sophisticated processes using p.d. pumps for drawing liquids from vessels under high vacuum (oil purifying and recycling); with users wanting more capacity from smaller pumps (higher speeds); with more customers installing reserve fuel supply tanks far beneath concreted parking lots (long suction lines with a lift); and with a growing interest in installations that are quiet (do not exceed OSHA noise levels), that are highly efficient (conserve energy) and that do not wear out (are inexpensive to maintain) — yes, today we are concerned about NPSH 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.

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 p.d. rotary pumps instead of using NPSH expressed in feet of liquid absolute, the Hydraulic Institute (Viking is a member) uses Available Net Inlet Pressure (ANIP) and Required Net Inlet Pressure (RNIP) expressed in PSIA. NPSH and NIP 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 article.

NPSH **available** 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** is based on everything from the suction port to the point in the pump where the pressure starts to increase; here, everything includes the entrance losses and the friction losses or pressure drops getting into the pumping elements.

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 till they get to a higher pressure region in the pump where they, the bubbles, collapse. This phenomenon is known as cavitation. It is the violent collapse of the bubbles of vapor with the resulting shock 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.

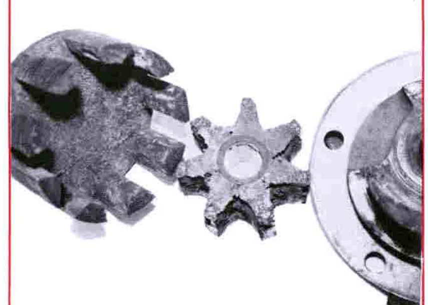


FIGURE 1. Parts from a Viking Internal Gear Pump deliberately run at excessive speeds on an accelerated water test. The appearance is typical of that caused by *severe* cavitation.

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 of the p.d. type, 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 internal gear rotary pumps. Viking will be doing some testing in this area in the near future.

NPSH Available

There is a widely used formula for calculating NPSHa which we should review and understand 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, and provides a basis for comparing systems and selecting pumps and can give an alert to a potentially troublesome installation. As we have indicated before, Net Positive Suction Head available 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 FORMULA – NPSHa = Ha ± Hz – Hf + Hv – Hvp; See Installation 1.

Where

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

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

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

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

Hvp = absolute vapor pressure of liquid at pumping temperature expressed in feet of liquid pumped.

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%.

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

The part that velocity head (Hv) has to play in NPSHa calculations and measurements is somewhat controversial. $Hv = \frac{V^2}{2g}$ 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). The value of Hv is normally quite small, see Table 1. From Table 2 we see that the value of Hv is 1.3 feet of liquid or less for all but two Viking Heavy-Duty pumps. Since the value of Hv is small for normal Viking pump applications, we suggest that it not be included in NPSHa values determined by calculations.

INSTALLATION 1

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.

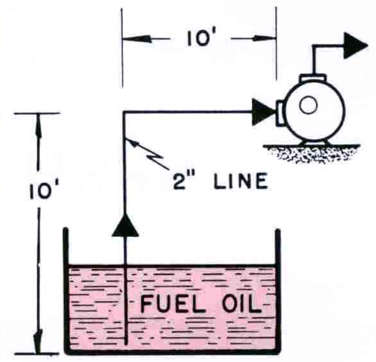


TABLE 1. Velocity-Velocity Head (Hv)

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

TABLE 2. Velocity (V) of Liquid in Feet/Second and Velocity Head (Hv) in Feet of Liquid at the Suction Port for Viking Heavy Duty Internal Gear Rotary Pumps at Nominal Rated Conditions

PUMP MODEL	PORT SIZE, INCHES	RATED CAPACITY, GPM	LIQUID VELOCITY* AT SUCTION PORT FEET/SEC.	VELOCITY HEAD, FEET OF LIQUID (Hv)
H125	1½	15	2.4	.09
HL125	1½	30	4.7	.34
K125	2	60	5.7	.50
KK125	2	80	7.6	.90
L125	2	135	12.9	2.6
LQ125	2½	135	9.0	1.3
LL125	3	140	6.1	.58
LS125	3	200	8.7	1.2
Q125	4	300	7.6	.90
QS125	6	500	5.6	.49
M125	4	420	10.6	1.7
GG4195	1	10	3.7	.21
HJ4195	1½	20	3.2	.16
HL4195	1½	30	4.7	.34
AS4195	2½	35	2.3	.08
AK4195	2½	50	3.4	.18
AL4195	3	75	3.3	.17
N335	6	600	6.7	.70
R335	8	1100	7.1	.78

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

INSTALLATION 1

NPSHa = Ha ± Hz – Hf – Hvp

Ha = absolute pressure on liquid.
= atmospheric pressure in feet of liquid.

$$= \frac{(14.7\ \text{PSIA*})(2.31'\ H^2O/PSI)}{0.88\ \text{S.G. of Fuel Oil}}$$

Ha = 38.8 feet of fuel oil.

Hz = vertical distance from liquid level to centerline of suction port (use maximum or worst condition—empty tank). Liquid is below pumps, so Hz is negative.

Hz = 10 feet of fuel oil.

Hf = friction loss is suction pipe size and length and includes elbows, valves, other fittings as equivalent length.

Hf = 2.9 feet of fuel oil (see pressure loss charts in Section 510 of Viking General Catalog).

Hvp = vapor pressure of fuel oil at 75°F. in feet of fuel oil absolute.

Hvp = 1 foot of fuel oil (maximum).

NPSHa = Ha (38.8) – Hz (10) – Hf (2.9) – Hvp (1)

NPSHa = 24.9 feet of fuel oil.

**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 Ha would have been 36.1 feet of fuel oil.*

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	29.93
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

In practice, an installation similar to this would have had a lower NPSHa because of additional losses (greater Hf) 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 .71 and the vapor pressure to 8.5 psia. The formula for calculating NPSHa remains the same but the values change significantly.

NPSHa = Ha ± Hz – Hf – Hvp

$$Ha = \frac{(14.7)\ (2.31)}{.71}$$

Ha = 47.8 feet of gasoline.

Hz = 10* of gasoline.

Hf = 2.9′ of gasoline.

Hvp = 8.5 psia (winter gasoline at 75°F.)

$$= \frac{8.5 \times 2.31}{.71}$$

Hvp = 27.6 feet of gasoline.

NPSHa = Ha (47.8) – Hz (10) – Hf (2.9) – Hvp (27.6)

NPSHa = 7.3 feet of gasoline.

Note the big difference the vapor pressure has made in the NPSH available from the same system, 24.9 feet of fuel oil vs. 7.3 feet of gasoline.

The NPSHa of 7.3′ of gasoline would normally be less by an additional friction losses (Hf) through a shut off valve, foot valve and strainer. It would also be affected by changes in tank level, temperature and source of supply.

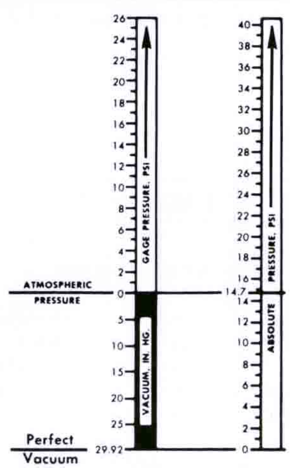


FIGURE 2. Comparison Between Absolute and Gage Pressure Scales.

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.

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

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 relating 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 or 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 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 in Engineering Handbooks and similar references. 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.

VAPOR PRESSURE

In the FORMULA used on page 2 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.

To help visualize another way that vapor pressure can be measured refer to Figure 4. The procedure for finding the vapor pressure of a liquid using the equipment is:

1. Fill **liquid** chamber; close valve V1
2. Pull full vacuum on **vapor** chamber; close valve V2
3. Allow temperature to stabilize; open valve V1
4. Read vapor pressure on the gage or manometer; record temperature.
5. Repeat this procedure at different temperatures until enough points have been recorded to draw a curve.

The Reid vapor pressure for gasoline and other volatile petroleum products is determined using the same basic test vessel arrangement as shown in Figure 4. The procedure is somewhat different from that outlined but the results are for all practical purposes the same. Details of the procedure for determining Reid vapor pressure are spelled out in ASTM standard D323. The Reid vapor pressure is normally understood to be an indication of the vapor pressure of the liquid at 100°F.

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 9.5 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

Figures 5, 6 and 7 show the Vapor Pressure — Temperature Curves for a number of liquids frequently handled by Viking pumps.

The term volatile is sometimes encountered when working with vapor pressures and petroleum products. It is a rather general term which gives some indication of the vapor pressure of a liquid or more specifically, an indication of how rapidly the liquid will vaporize. When comparing two liquids at the same temperature, the one with the higher vapor pressure would be considered the more volatile and would vaporize more rapidly. For example, gasoline has a relatively high vapor

TABLE 4.
Conversion Table.

PRESSURE UNIT	ATMOS-PHERES, ATMOS.	FEET OF WATER, ' H_2O	INCHES OF MERCURY, " Hg	KILOGRAMS PER SQUARE CENTIMETERS, Kg/CM²	MILLIMETERS OF MERCURY, mm Hg	kPa	POUNDS PER SQ. INCH, PSI
Atmospheres, Atmos.	1	33.90	29.90	1.033	760	101.3	14.70
Feet of Water, ' H_2O	0.0295	1	0.8826	0.0305	22.41	2.99	0.4336
Inches of Mercury, " Hg	0.0334	1.133	1	0.03453	25.4	3.39	.4912
Kilograms per square centimeter, Kg/cm²	0.9678	32.81	28.96	1	735.0	98.07	14.22
Millimeters of Mercury, mm Hg	0.00132	0.0446	0.0394	0.00136	1	0.1335	0.01935
kPa	0.00987	0.3345	0.295	0.0102	7.4938	1	0.1450
Pounds per square inch, PSI	0.06805	2.307	2.036	0.0703	51.67	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 which shows 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 is then multiplied by 0.433 to get 21.6 PSI.

When working with liquids other than water, multiply the conversion factors shown by the Specific Gravity of your liquid.

100kPa = 1 Bar

See Eng. Sect. 510.22 Conversion Factors, and Catalog Sect. 141.17.

Round off Conversion Factors to 4 significant places & no more than five places after decimal.

Example check.

1.0335 Kg/cm² × 14.223 = 14.699 = 14.70 PSI

101.35 kPa × 0.145038 = 14.699 = 14.70 PSI

759.5 mm Hg × 0.01935 = 14.696 = 14.70 PSI

33.90 Ft. H_2O × 0.4336 = 14.699 = 14.70 PSI

29.90 " Hg × 0.492 = 14.710 = 14.70 PSI

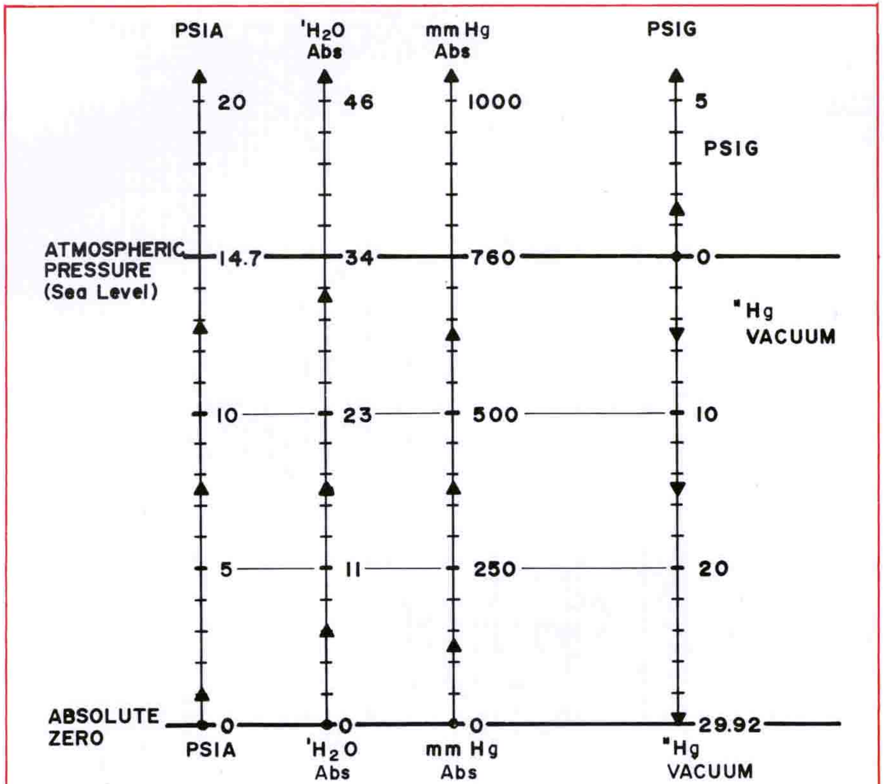


FIGURE 3.
Pressure Unit Comparison Chart.

pressure and is considered more volatile than #2 fuel oil which has a very low vapor pressure.

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.

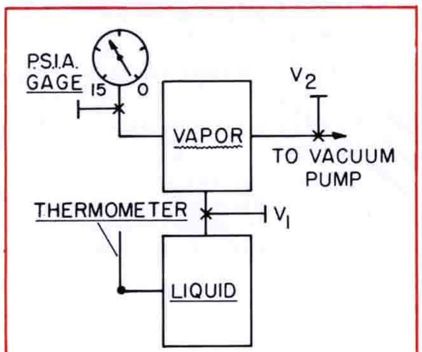


FIGURE 4.

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 a jobber or distributor to the user in his 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, Fuel Oil Systems.

Somewhat the same 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 known as cavitation. It is this 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.

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 by test 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.

To determine 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 be-

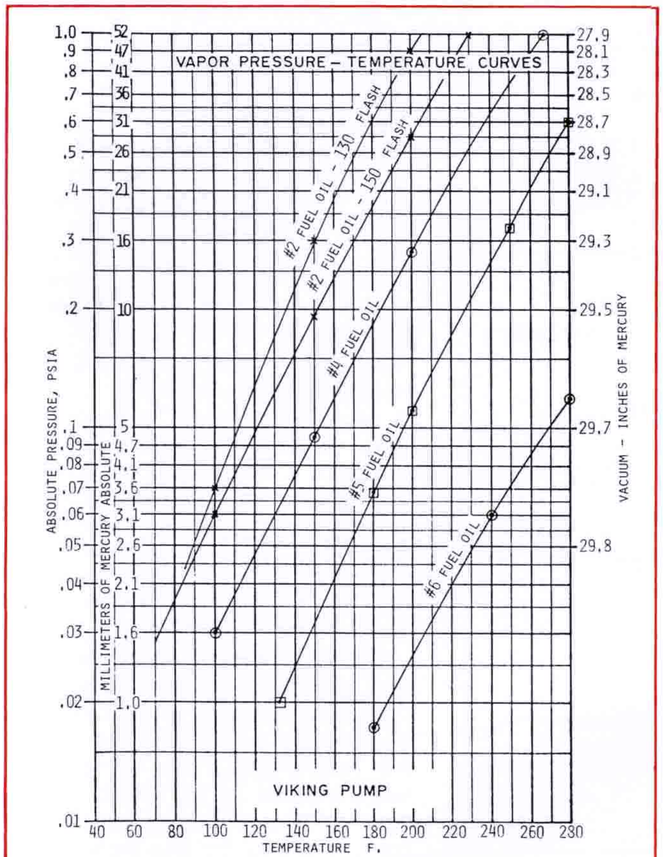


FIGURE 5.

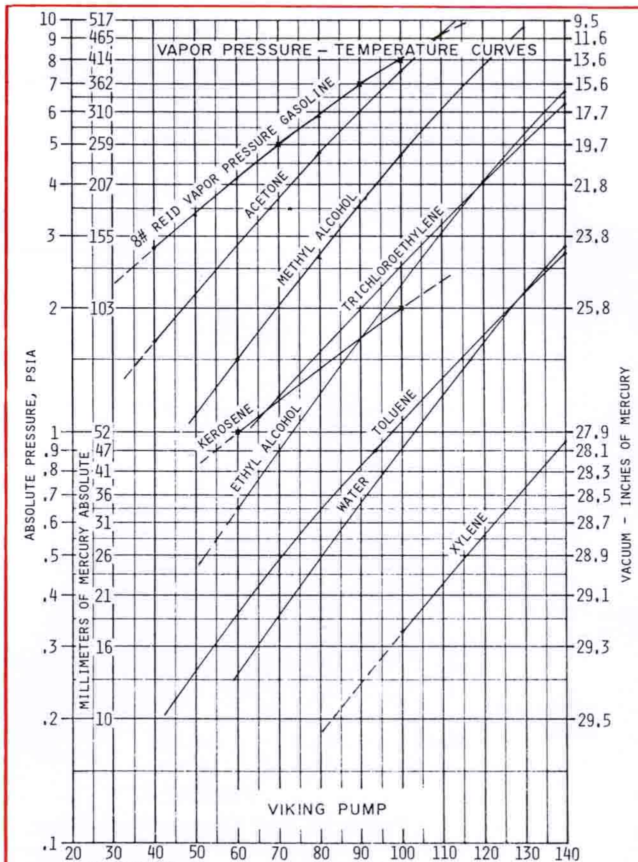


FIGURE 6.

tween 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 90°, may 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.

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 8 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 9 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).

Visualizing the pump in operation it is easy to see that as the pump RPM increases, the pressure loss will also increase.

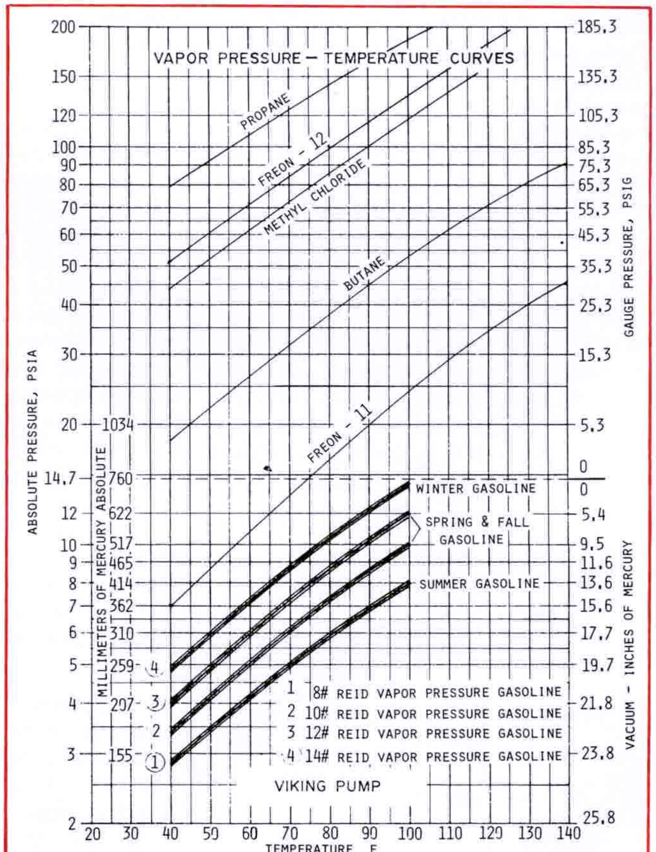


FIGURE 7.

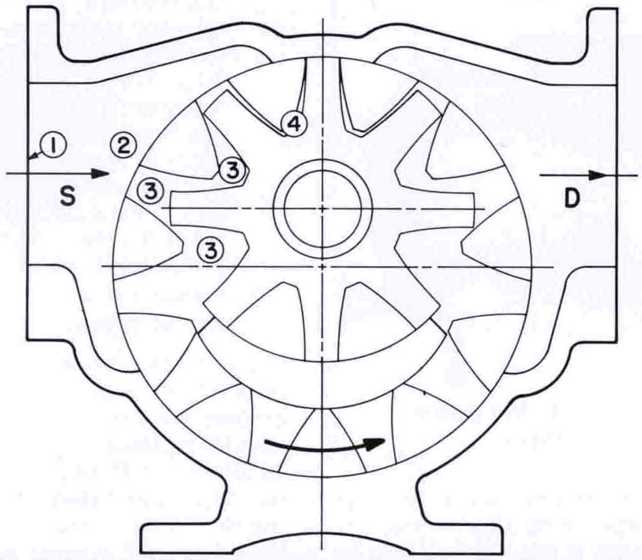


FIGURE 8. Internal gear pumping principle showing suction side pressure zones.

As there is a 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 and to do so, requires more pressure (experiences a larger pressure loss). Increasing pump RPM does increase the pressure loss in the pump.

As all pipe line loss charts show, the higher the viscosity the greater the loss, other conditions remaining the same; so it is within a pump. 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 9.

Any time a pump is operating there is a pressure loss within the pump, as has just been discussed and as shown in Figure 9. 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.

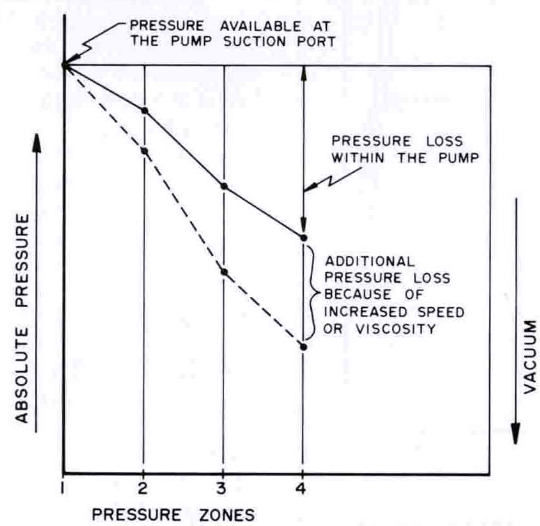


FIGURE 9. Absolute pressure vs. pump suction side pressure zones.

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 then important for him 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 75 years in applying pumps to applications with limited NPSHA. During recent years an aggressive program of R&D lab testing has been underway to accumulate NPSHR data for a wide range of speeds and viscosities on the complete pump line.

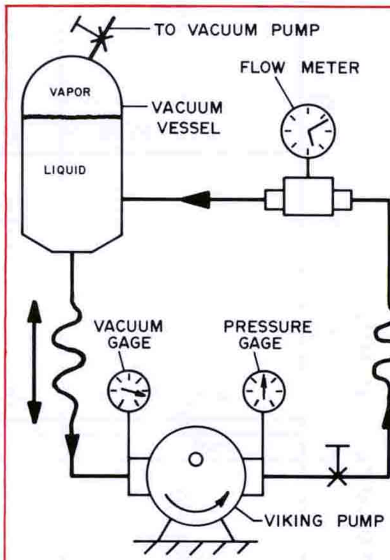


FIGURE 10. Drawing of test set-up for checking NPSHR.

The drawing in Figure 10 shows the arrangement of the various pieces of equipment actually used in conducting the NPSHR tests in the R&D Lab. The 400 gallon tank shown in Figure 11 is the vacuum vessel used when testing at 50 GPM and higher flow rates.

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 gauge 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 gauge reading equivalent to an absolute pressure of 25 feet of liquid. Record the 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, depending upon pump speed and liquid viscosity, the capacity will start to drop off, indicating 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.

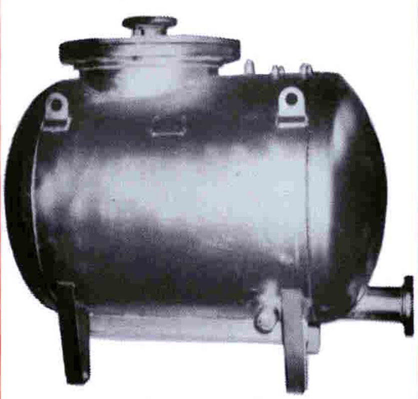


FIGURE 11. 400 gallon vacuum vessel.

The curves shown in Figure 12 are plotted from actual test data taken according to the just reviewed procedure for a Viking Model K125 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.

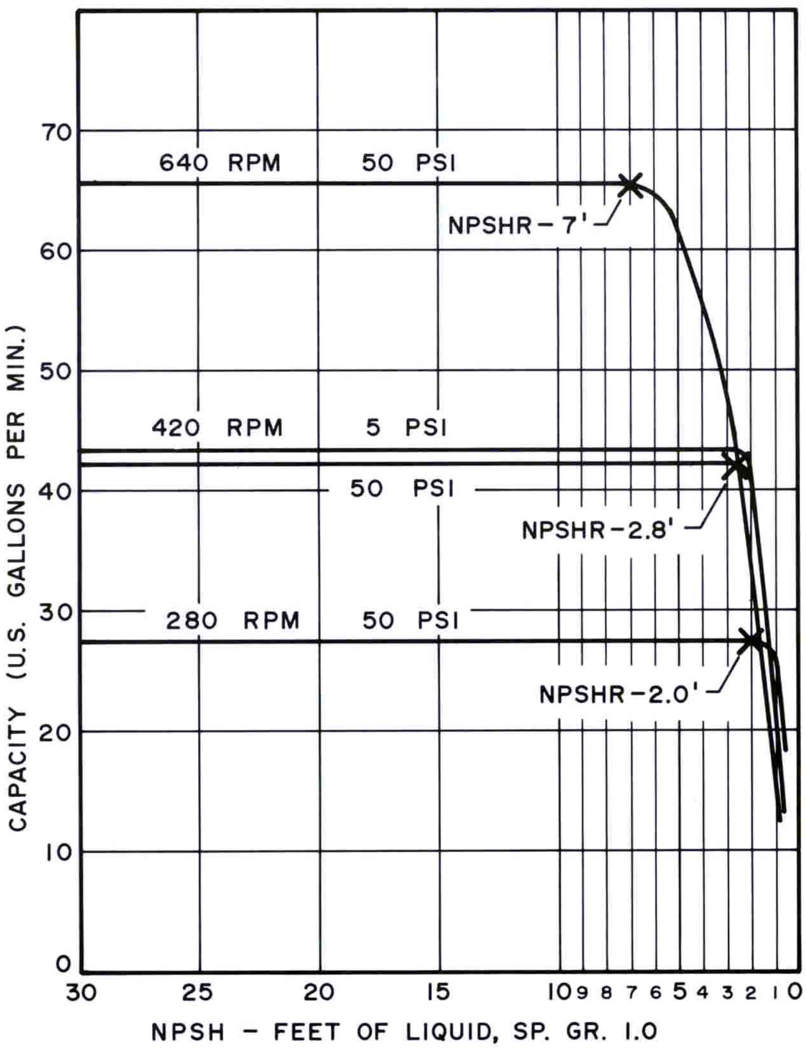


FIGURE 12. Curves for Viking Pump Model K-125. Plotting capacity vs. NPSHA when handling 38 SSU liquid.

NPSHA Calculations —

The basic formula, NPSHA = Ha ± Hz – Hf – Hvp, was discussed earlier.

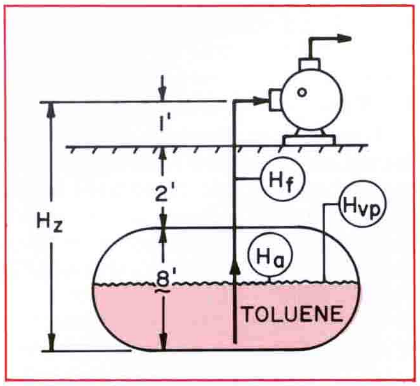
The Hydraulics Institute is advocating the use of the term Net Inlet Pressure (NIP) in place of NPSH when working with positive displacement pumps. As more spec sheets are written specifically for positive displacement pumps, there will be a growing use of the term Net Inlet Pressure. Net Inlet Pressure is expressed in PSIA. The Required Net Inlet Pressure (RNIP) is the counterpart to NPSHR. The Available Net Inlet Pressure (ANIP) 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 RNIP in PSIA multiply the value by 0.433.

As before, Ha is the absolute pressure on the surface of the liquid, Hz is the distance from the surface of the liquid to the centerline of the suction port, Hf is the pipe friction or line loss and Hvp is the vapor pressure of the liquid. All terms to be expressed in the same units, feet of liquid pumped.

In the following examples the basic NPSHA formula is applied to three typical pump installations. These three are representative of the types 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 60°F, 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 60°F. Pump is to be installed at grade above the tank, line size is 2″, the job site is in the upper midwest.



INSTALLATION 2

NPSHA = Ha ± Hz − Hf − Hvp

Ha = 27″ Hg. abs., based on an elevation of 2000′ and a low barometer.

Ha = $\frac{(27)(1.13)}{0.87}$ = 35′ Toluene.

Hz = 8′ + 2′ + 1′ = 11′ 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, Hz will be negative.

Hf = 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 feet of Toluene per foot of pipe = .02 × $\frac{2.3}{.87}$ × .87 = .046; from information in Section 510 of the General Catalog.

Hf = 35 × 0.046

Hf = 1.6′ of Toluene

Hvp = 0.36 PSIA at 60°F; use 1.7 PSIA, the vapor pressure at 120°F which is possible summertime temperatures at the pump.

Hvp = $\frac{(1.7)(2.31)}{0.87}$ = 4.5′ Toluene

NPSHA = 35′ − 11′ − 1.6′ − 4.5′
= 17.9′ Toluene

ANIP = Hg ± Hz − Hf − Hvp

Ha = 27″ Hg. abs. × 0.49

Ha = 13.23 PSIA

Hz = $\frac{\text{(Ft. of Toluene)(SpG)}}{2.31}$

Hz = (11)(.87) ÷ 2.31

Hz = 4.14 PSIA

Hf = (PSI/Ft.)(SpG) times (Total Equiv.)

Hf = (.02)(.87)(12 + 23)
= (.0174)(35)
= 0.61 PSIA

Hvp = 1.7 PSIA

ANIP = 13.23 − 4.14 − 0.61 − 1.7

ANIP = 6.78 PSIA

The NPSHR for a KK-size Viking at 420 RPM is 3.3 Ft. (SpG − 1.0).
NPSHR = (3.3) ÷ (.87) = 3.79 Ft.
NPSHA = 17.9 Ft.

The required Net Inlet Pressure (RNIP) is equal to: (Ft. Liq.)(0.433) ÷ SpG
RNIP = (3.3)(0.433) ÷ .87
RNIP = 1.64 PSIA
ANIP = 6.78 PSIA

Any self-priming pump with an NPSHR less than 17.9′ 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 Hz factor would be plus instead of minus. 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 startup. 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.

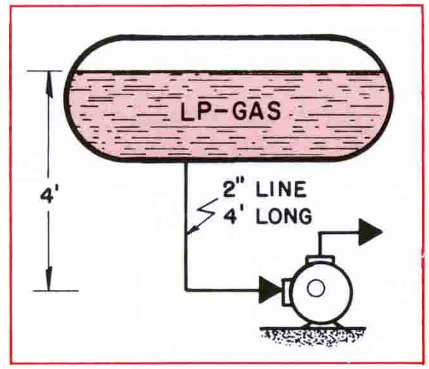
"Flooded" suction means positive pressure at the pump suction port. Many times the suction port is not "flooded" because of long suction line, small piping, high viscosity, a fine mesh strainer, or other fittings which together restrict the flow of liquid and cause NPSH problems.

NPSHA or ANIP should always be calculated on 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 to start with.

INSTALLATION 3— High Vapor Pressure

Transfer 30 GPM of LP Gas (Propane) at 65°F from a large storage tank. Specific gravity 0.50, viscosity is 0.1 Cps, vapor pressure 100.7 PSIG.



INSTALLATION 3

NPSHA = Ha ± Hz − Hf − Hvp

Ha = absolute pressure at surface of liquid.
= atmospheric pressure + vapor pressure.
= 14.7 + 100.7 = 115.4 PSIA.
= $\frac{(115.4)(2.31)}{0.50}$

Ha = 533′ of LP Gas.

Hz = distance from liquid level to port centerline.

Hz = + 4′, liquid is above pump.

Hf = suction line loss from pressure drop charts.

Hf = 1.5′ of LP Gas.

Hvp = vapor pressure of LP Gas at 65°F.
= 100.7 PSIG + atmospheric pressure of 14.7 psia.
= 115.4 psia.

Hvp = 533′ of LP Gas.

NPSHA* = 533′ + 4′ − 1.5′ − 533′
NPSHA = 2.5′ of LP Gas.

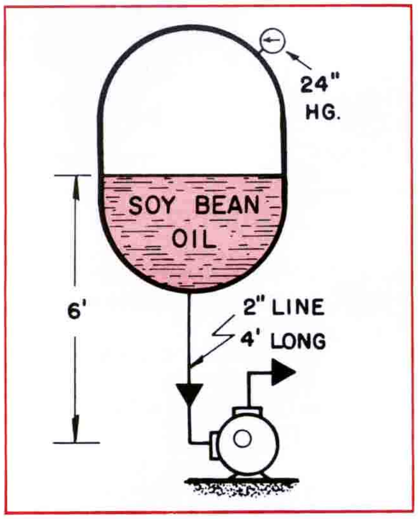
**Note that for a high vapor pressure liquid (a liquid which has to be stored in a closed vessel to keep it from boiling away) the Ha and Hvp terms cancel out and that NPSHA becomes the difference between the elevation head and the line loss; Hz − Hf.*

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 per cent 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 Manual, TSM442 and 443.

INSTALLATION 4 — Vacuum Vessel

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



INSTALLATION 4

NPSHA = Ha ± Hz – Hf – Hvp

Ha = absolute pressure at surface of liquid. Assume a 1000′ elevation installation with a low barometer (use 27.8″ Hg. abs. as barometric pressure).

$= \frac{(\text{"Hg abs.})(\text{Ft.}H_2O \text{ per "Hg})}{\text{S.G.}}$

Ha $= \frac{(27.8 - 24.0)(1.133)}{0.88}$

Ha = 4.9 feet of soybean oil.

Hz = distance from liquid level to port centerline.

Hz = 6′ of soybean oil, this value is plus since the liquid is above the pump.

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

= (.02)(4)(.88)(2.31)

Hf = 0.2′ of soybean oil.

Hvp = vapor pressure of soybean oil at 240°F.

Hvp = 2′ (estimate) of soybean oil

NPSHA = 4.9′ + 6′ – 0.2′ – 2′

NPSHA = 8.7′ of soybean oil

As with Installation 2 it is well 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 startup 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 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.

Possibly expressing the information in the formula by line chart would be of help in visualizing how the actual NPSHA figure is determined. Figure 13 shows the steps taken in arriving at the NPSHA for Installation 2.

Determine 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

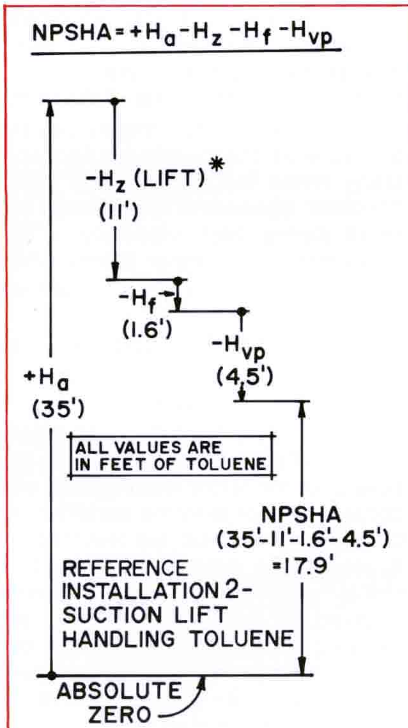


FIGURE 13.

*Line chart **showing** steps taken in calculating NPSHA.*

**If the liquid level was 11′ above the pump Hz would be plus and would be added to instead of being subtracted from Ha. The NPSHA would then be 39.9′.*

must be greater than the NPSHR of the pump. If, in the early stages of system design, a pump is selected, it is then possible to determine the maximum allowable lift by equating NPSHA to NPSHR and solving for Hz. As an example, consider that for Installation 2, the question had been raised "What is the maximum vertical lift possible?"

From data available for the pump selected, determine the NPSHR; as an example consider that the NPSHR for the pump selected is equal to 3.5′ of Toluene; use 5.0′ to be conservative. Then NPSHA = NPSHR = 5.0′ = Ha – Hz – Hf – Hvp

Values for the different factors are taken from the calculations on Installation 2.

Ha = 35′

Hz = unknown

Hf = 0.046* (Hz + 24′**)

Hf = 0.046 Hz + 1.1

Hvp = 4.5′

5.0′ = Ha – Hz – Hf – Hvp

5.0′ = 35′ – Hz – (0.046 Hz + 1.1′) – 4.5′

5.0′ = 35′ – 1.046 Hz – 1.1′ – 4.5′

1.046 Hz = 35′ – 1.1′ – 4.5′ – 5.0′

Hz = 23.3″

A vertical lift of 23.3′ 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 Hf factor) when the lift or liquid leg is predetermined.

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

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

Determining NPSHA By Gauge—

It is possible by using a pump suction port gauge 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 —

Hi = Ha ± Hz – Hf

Restating the NPSHA formula using the Hi term it becomes —

NPSHA = Hi – Hvp

The value of Hi can be determined by converting a suction port gauge 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 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, the barometer is 27″ Hg. absolute. The vacuum gauge reads 6″ Hg., determine the NPSHA of the system.

The suction port gauge reading of 6″ Hg. vacuum subtracted from an absolute barometric presure 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 —

$$21\ (''\text{Hg.}) \times 1.13\ \frac{(\text{feet of } H_2O)}{(''\text{Hg.})} \times \frac{1}{0.87}\ \frac{(\text{feet of Toluene})}{(\text{feet of } H_2O)} = 27.3'\ \text{of Toluene}$$

The vapor pressure at 60°F is 0.36 PSIA or 1′ of Toluene.

NPSHA = Hi – Hvp = 27.3′ – 1′ = 26.3′ of Toluene

This is considerably greater than the 17.9′ 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.

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:

Ha — 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.

Hz — 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.

Hf — 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 if they are necessary to afford the difference between a system that works and one that does not then, of course, the added expense is justified.

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 (Hf) 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.

Hvp — 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 Hf (line loss) term because of higher viscosity.

If to reduce NPSH problems, changes to the system do not seem the most practical, 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 Hf (line loss) factor 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.

Variables —

When an actual NPSHA increase of 1 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 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 fuel 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 problem 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.

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 conversions between the various pressure units.

The Viking R & D Laboratory has completed the NPSHR testing of all the internal gear pump sizes 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

NPSHR — Feet of Liquid Sp. Gr. 1.0

PUMP SIZE	PUMP SPEED R.P.M.														
	100	125	155	190	230	280	350	420	520	640	780	950	1150	1450	1750
C												1.7	1.9	2.2	2.4
F, FH										1.8	1.9	2.1	2.3	2.8	3.4
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
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, K, KK		1.7	1.8	1.9	2.1	2.3	2.8	3.3	4.4	6.3	9.1				
L, LL, LS	1.7	1.8	2.0	2.2	2.5	3.0	3.8	5.0	7.3	10.8					
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							
N	2.1	2.5	3.5	4.5	6.3	9.5	15.0								
R	2.7	3.2	4.2	5.8	8.2	11.9									

CATALOG SPEED RATING

NOTES:
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 Sp. Gr.

FIGURE 14.

Viking Pump Inc., A Unit of IDEX Corporation
Cedar Falls, Iowa 50613 • Telephone (319) 266-1741 • FAX (319) 273-8157

  Printed in U.S.A.