NPSH: It shouldn't mean not pumping so hot

And it won't if you follow the clear-cut NPSH evaluation procedures set forth here

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It is well-known that liquids boil at defined temperature-pressure relationships. For any given liquid at a given temperature, pressure reduction to some stated value will cause boiling or vaporization.

A pumped liquid can vaporize or flash within the pump itself because of inadequate pressurization. Liquid vaporization within the pump is generally defined as cavitation, and it can cause the following problems:

- Pump impeller damage will occur. This is because low pressure in the impeller "eye" will cause vapor bubble formation. The vapor bubbles then collapse or implode because of pressure increase as they move into higher pressure areas inside the impeller. These hammer-like blows against the impeller can cause destruction within a short time.
- The pump curve will change drastically and in an unpredictable manner. Flow can virtually cease or "slug" because the pump cannot readily deliver both liquid and vapor.
- Pump shafts can be broken because of slugging of the impeller against alternate bodies of liquid, vapor, and air.
- Pump mechanical seal failure can occur because the mechanical seal is asked to work under intolerable conditions; vapor flash around the seal causes "dry" operation and rapid wear.

It is most important to successful pump application that adequate pressures (above vaporization) be maintained within the pump.

Every pump operates at a lower pressure in the impeller eye and inlet to the impeller vanes than the pressure existing at the pump suction flange. Even though pressure at the pump suction flange is measured and known to be above the flash or vaporization point, the pump can still cavitate because a pressure drop occurs from the suction flange to the pump interior when the pump operates.

Internal pump pressure drop occurs because of greatly increased liquid velocities from the pump suction flange to and through the impeller eye, and because of turbulence, vane entrance friction losses, etc. To prevent cavitation, an engineer must *know* how much internal pump pressure drop will occur for his design circumstances and for any of a number of specific pump selection possibilities.

The pump manufacturer's measure of this pressure drop is called required NPSH (net positive suction head), which is shown in Fig. 1.

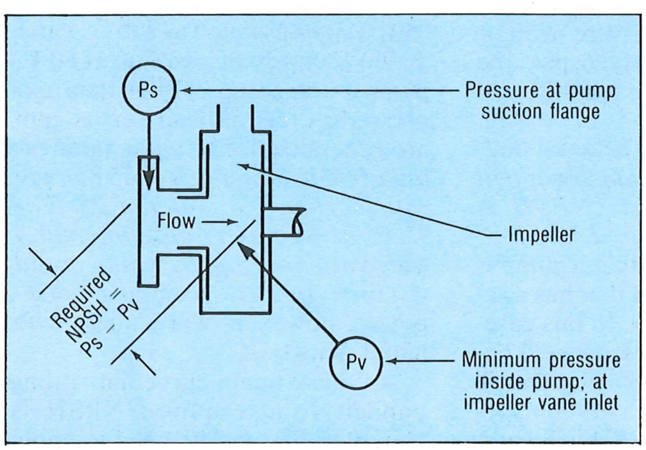
Test procedures for establishing required NPSH have been standardized and are carefully followed by pump manufacturers so as to obtain as true an estimation of internal pump pressure drop as possible. Required NPSH is illustrated on pump curves by several different methods. Fig. 2 shows the method generally used.

Regardless of the illustration method, required NPSH is not a constant value for any pump. Similar to valve pressure drop, required NPSH will increase with increased flow.

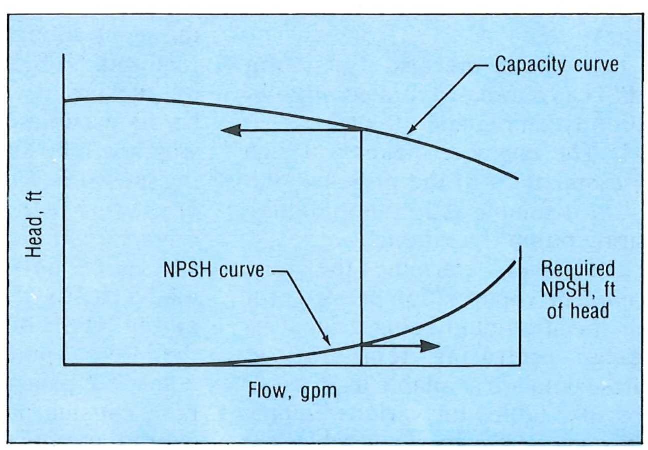
With regard to valves, it is well-known that for a given flow rate, a large valve will cause less pressure drop than a smaller valve. Similarly, a pump can be considered small or large by reference to its impeller eye diameter for the intended flow rate. For the same pumped flow rate, a small pump (small impeller eye diameter) will have a much higher required NPSH than a larger pump. This point is illustrated in Fig. 3.

Fig. 3 provides some important basic pump application points:

- A pump selected at the end of

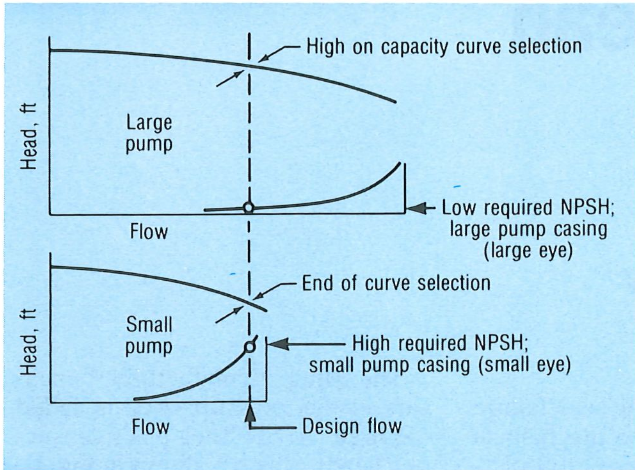


1 Required NPSH is measure of pump pressure drop.

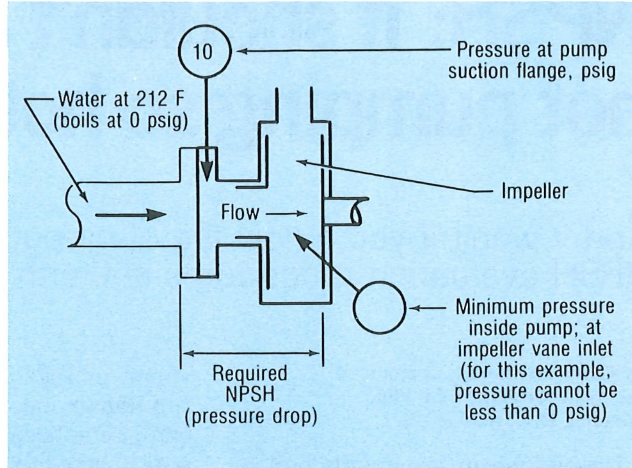


2 Required NPSH increases as flow through pump increases.

NPSH evaluation



3 Required NPSH, for the same flow rate, is a function of pump size.



4 Pumping schematic for example problem; available NPSH equals 10 psi.

its capacity curve (feet of head versus gallons per minute) is being driven to maximum capability and is the smallest pump that can provide design flow rate. The "small" pump, however, establishes a maximum required NPSH (pump pressure drop). While generally lowest in cost, because of minimum size, such a selection establishes maximum trouble potential.

- A pump selected at the midpoint area of its capacity curve is larger; the impeller eye velocity is reduced, and the pump internal pressure drop must be lower. A pump so selected will cost more than the minimal "end of curve" selection, but it will reduce trouble potential when NPSH or cavitational problems are a consideration.

Using NSPH

We have thus far established a basic point: that required NPSH is a description of a specific pump's internal pressure drop as flow rate through the pump changes. How is knowledge of required NPSH used in specific pump application problems?

The basic method for using NPSH as a tool to avoid pump cavitation is simple and direct:

1) The engineer makes a design assessment as to the pressure that will be available at the pump suction during pump operation.

2) He then determines the liquid boiling or vaporization pressure for the specific liquid being used at its design operating temperature. These data are available from vapor pressure tables for various liquids. Keenan and Keyes steam tables are generally used for water.

3) By subtracting the liquid boiling pressure from available pressure at the pump suction, the engineer obtains the "available NPSH." Available NPSH is thus the available suction flange pressure over and above fluid boiling pressure.

4) The engineer now selects a pump so that required NPSH is lower than available NPSH. A pumped liquid cannot boil or cavitate within the pump as long as required NPSH is lower than available NPSH. This can be illustrated by the following simple example:

A system under design is intended to pump 212 F water. The application engineer states his conclusion, after calculation, that pressure at the pump suction flange will be 10 psig during operation. What is the available NPSH?

Since water at 212 F boils at 0 psig, the available NPSH will be 10 psig minus 0 psig, or 10 psi. Pump suction flange pressure will be 10 psi over and above the liquid boiling point. A schematic of the pumping situation is presented in Fig. 4.

From Fig. 4 it can be noted that if the pump internal pressure drop or required NPSH is only 6 psi, the internal pump working pressure will be 10 psi minus 6 psi, or 4 psig, which will be above the liquid boiling pressure. *The liquid cannot boil because the available NPSH is greater than the required NPSH.*

Assume, however, that a pump is inadvertently selected that has a required NPSH of 12 psi. In this case, the internal pump pressure will become −2 psig, and the liquid will boil, causing pump cavitation, because the required NPSH is greater than the available NPSH.

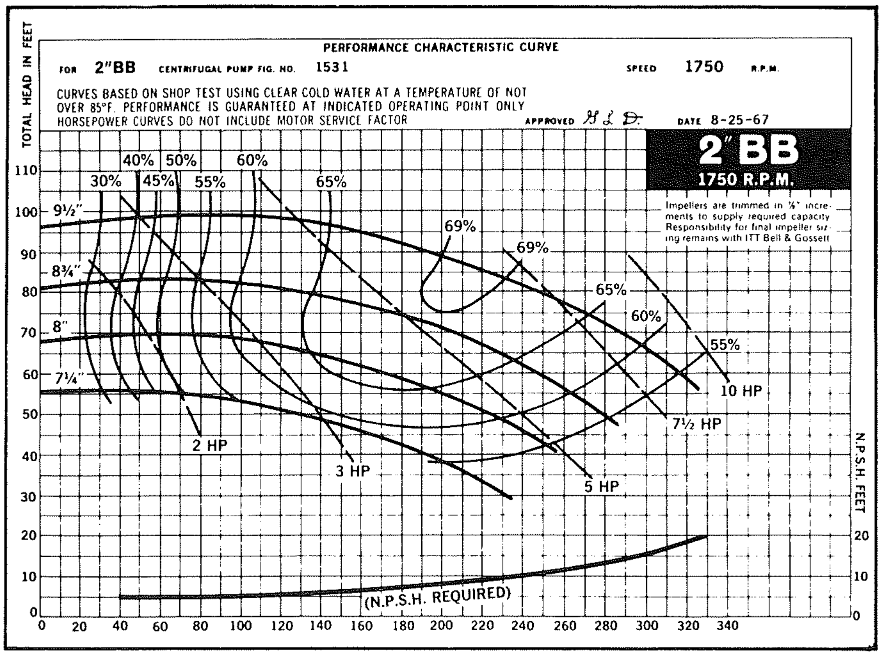
A problem of units

The NPSH explanation and example illustrate the fundamental reasoning behind the NPSH evaluation procedure. It will be noted, however, that the example states NPSH as psi. This is done to clarify fundamental procedural concepts. NPSH, whether available or required, is never expressed in terms of psi; it is always stated in terms of feet of liquid head.

The reason why NPSH is stated in feet of liquid head stems from the need for pump curve generalization. It would be impractical to publish a different pump capacity curve and NPSH curve for an infinite variety of liquids and, in addition, to provide separate capacity and NPSH curves for all temperature variations with each liquid. This would be necessary if pump curves and NPSH data were expressed in terms of psi.

Pump curves and NPSH data are presented as feet of head versus gpm because feet of liquid head means differential energy per unit weight of fluid. Since 1 lb of water at 85 F weighs as much as 1 lb of water at 200 F or 1 lb of gasoline at 60 F, pump curves and NPSH data expressed as feet of head versus gpm are generalized. *The pump data established by water test at 85 F apply without change to water at 200 F or 45 F, as well as to gasoline and a wide variety of liquids within broad viscosity ranges.* (Pumping horsepower, however, will change with liquid density.)

A typical pump curve illustrating capacity and required NPSH is shown in Fig. 5. The need to apply the developed pump curve un-



5 Typical pump characteristc curve including NPSH. Vertical scale at left, total head in feet, means differential energy input per lb of liquid pumped.

changed for a broad variety of liquids is neatly solved by use of the term *head*. Since the pump's NPSH curve is stated in terms of head, the classical NPSH evaluation procedure has been to convert all pressures to head to determine available NPSH. The simple concept behind NPSH evaluation, as illustrated previously, now tends to become blurred by abstractions because of a need to convert atmospheric pressures and liquid vapor pressures to *absolute* feet of head terms.

It is difficult to visualize sea level atmospheric pressure as equivalent to 34 ft of 60 F water head or to 48.6 ft of liquid head when the liquid's specific gravity is 0.7. The statements of atmospheric pressure as related to feet of liquid head are abstract engineering truths that are difficult to correlate with working systems.

The conventional or classical NPSH design evaluation will be avoided in this discussion because of its abstract nature and because classical procedures and formulas are not truly intended for closed loop pumping applications. Conventional NPSH evaluation can be a very confusing, time consuming procedure for the majority of plant engineers, whose NPSH evaluation needs, while important, are only occasional (see appendix).

Our proposed NPSH evaluation procedure is based on an NPSH evaluation chart and is as theoretically correct as the conventional method. It differs from the conventional in that the calculation reference is to pump suction flange pressure expressed as an easily visualized gauge pressure (psig). The NPSH evaluation procedure will be described for both open and closed loop pumping systems.

Open system NPSH evaluation

Fig. 6 illustrates an open pumping system typical of a cooling tower. Water temperature is 85 F, and the tower pan is elevated 2.3 ft above the pump suction. Flow friction loss from the tower pan to the pump suction flange is 4.6 ft. This flow friction loss includes piping, fittings, etc.

The procedural aim is to determine the expected pump suction flange gauge pressure reading when the pump is in operation. For the circumstances shown in Fig. 6, and starting with atmospheric pressure at 0 psig, a static liquid head of 2.3 ft would cause 1 psig to be registered at Gauge A. A suction pipe flow friction loss of 4.6 ft is equivalent to 4.6/2.3 = 2 psi pressure drop, so that the calculated suction flange gauge pressure would become 0 + 1 − 2 = −1 psig.

The NPSH evaluation chart is shown in Fig. 7. For our example conditions, the chart is entered at a calculated pump suction gauge pressure of −1 psig. A line is then extended vertically to intercept the appropriate liquid vapor pressure line. For 85 F water, vapor pressure will be on the order of 0.6 psia. The interception is shown as Point 1.

From Point 1, a line is extended horizontally to intercept the liquid specific gravity line. In this case, the specific gravity is 1.0, and the available NPSH is read as 31 ft at Point 2.

What has chart accomplished?

The NPSH chart has simply taken available suction pressure and deducted liquid vapor pressure to establish available pressure over and above the liquid boiling point. This available pressure has then been converted to feet of liquid head at the liquid's specific gravity. This is in liquid head over and above the liquid boiling point and is defined in conventional pumping terms as *available NPSH*.

Our example problem now states that we have 31 ft of available NPSH. For liquid to flash or cavitate inside the pump, the pump internal pressure drop (required NPSH) must exceed 31 ft. To provide a satisfactory pumping system, we need only provide a pump that has less than 31 ft of required NPSH.

The chart can be used for NPSH evaluation of any pumping system handling any liquid. As an example, an exotic liquid is to be pumped from an open tank in Denver. The manufacturer states that at its pumping temperature, the liquid has a vapor or boiling pressure of 5 psia and that its specific gravity will be 0.6. Determine the available NPSH for this pumping situation, which is illustrated in Fig. 8.

For this example, we need to establish atmospheric pressure at Denver's 5000 ft elevation. This is shown in Table 1 as −2.5 psig (12.2 − 14.7 = −2.5).

Table 2 shows the feet of head to psi equivalencies for liquids of various densities or specific gravities. In this case, the liquid has a specific gravity of 0.6, so that a column height of 3.85 ft of liquid is necessary to cause a pressure of 1 psi.

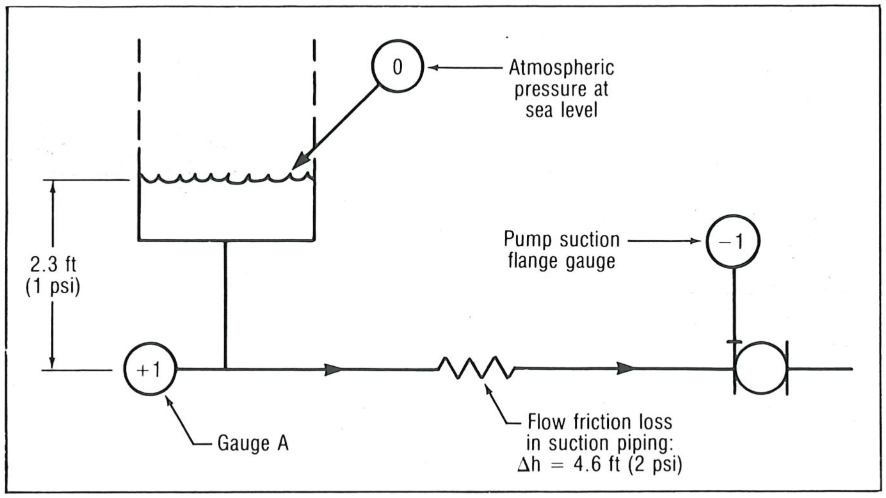
Since the tank is below the pump suction, the 5 ft suction lift will cause a negative static pressure: −5/3.85 = −1.3 psi.

The pump suction flange pressure can now be calculated by combining the atmospheric pressure, static pressure, and pipe loss: −2.5 − 1.3 − 6/3.85 = −5.4 psig.

NPSH evaluation

The NPSH evaluation chart can now be entered at −5.4 psig. Since our liquid has a 5 psia vapor pressure, a line is extended vertically to intercept the line for this value; and from this point, a line is extended horizontally to intercept the 0.6 liquid specific gravity line. An available NPSH of 17 ft is shown for this system.

A pump selected for a required NPSH of less than 17 ft will operate without cavitation problems.



6 Schematic for example problem.

Design precautions

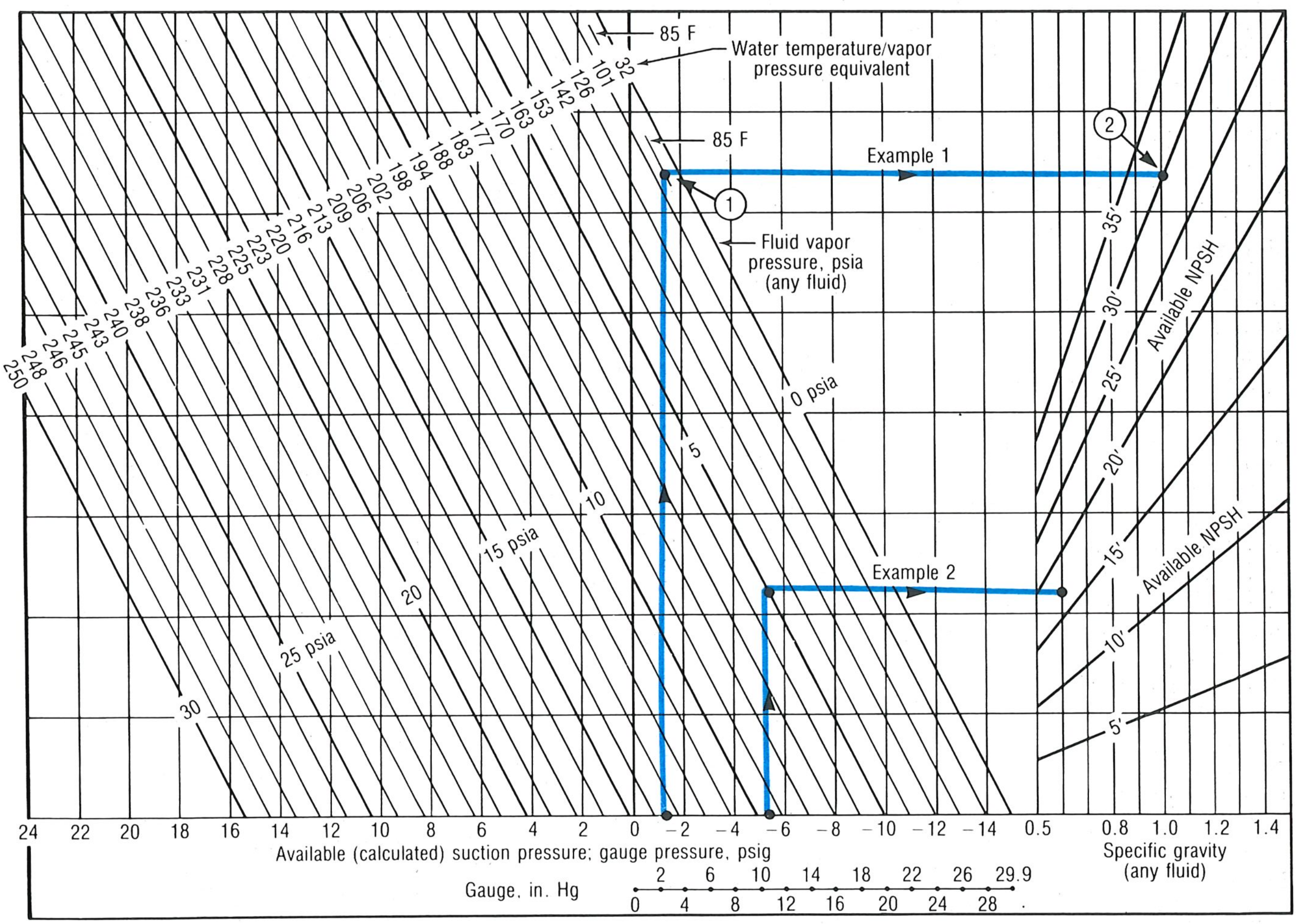
In an open pumping system, the pump pumps away from a tank open to atmosphere. To maintain as high a pump suction flange pressure as possible, the pump should be placed as close to the tank as possible. The pump suction pipe should be as large as possible and preferably free of check valves, control valves, and strainers.

Foot valves, of course, are needed for pumps lifting liquids from tanks. When foot valves are used, care must be taken in sizing. Strainers in suction lines should be avoided since a clogged strainer will negate the most careful NPSH evaluation. When a strainer is used, it should be carefully monitored and should be large mesh since most centrifugal pumps will pass objects up to ¼ in. in diameter.

Figs. 9 and 10 show some generalized dos and don'ts for open pumping system applications.

Closed system NPSH evaluation

A closed loop pumping system differs from an open loop in that the pumped liquid is continuously recirculated. Pressurization of a closed loop system is provided by either a closed compression tank or, in some cases, an elevated expan-



7 NPSH evaluation chart can be used for any pumping system, involving any liquid. Colored lines show solutions to examples in text.

Appendix

The classical method of evaluating available NPSH expresses available head over and above the flash point of the liquid entering the pump suction by the following formula:

$$NSPH_A = P_a \pm H_z - h_f - h_{vpa}$$

where

$NPSH_A$ = available NPSH, ft of liquid

P_a = absolute atmospheric pressure at liquid surface level, ft of liquid

H_z = elevation difference of liquid surface above or below pump suction, ft

h_f = friction losses in pump suction piping, ft of liquid

h_{vpa} = absolute vapor pressure at pump suction temperature, ft of liquid

It will be noted that velocity head ($V^2/2g$) is not included in the formula for a *proposed* system being evaluated for available NPSH. Velocity head is included, however, as a positive value when an already installed system is being evaluated by means of gauge readings from the operating system.

Given the basic formula, we can determine available NPSH for the example problem illustrated as Fig. 6 in the main body of this article:

- P_a, at sea level, is 14.7 lb per sq in. times 2.31 ft per lb per sq in., or 34 ft.
- H_z is given as +2.3 ft.
- h_f is given as 4.6 ft.
- h_{vpa} is 0.5959 (from Keenan and Keyes steam tables, 85 F water has a vapor pressure of 0.5959 psia) times 2.31 ft per lb per sq in., or 1.37 ft.

Thus,

$$NPSH_A = 34 + 2.3 - 4.6 - 1.37 = 30.33 \text{ ft}$$

The close correlation between the 30.33 ft available NPSH calculated and the 30 ft determined from the evaluation chart should be noted.

A classical NPSH evaluation for the second example, shown as Fig. 8, illustrates more clearly the complexities and need for care with classical procedures.

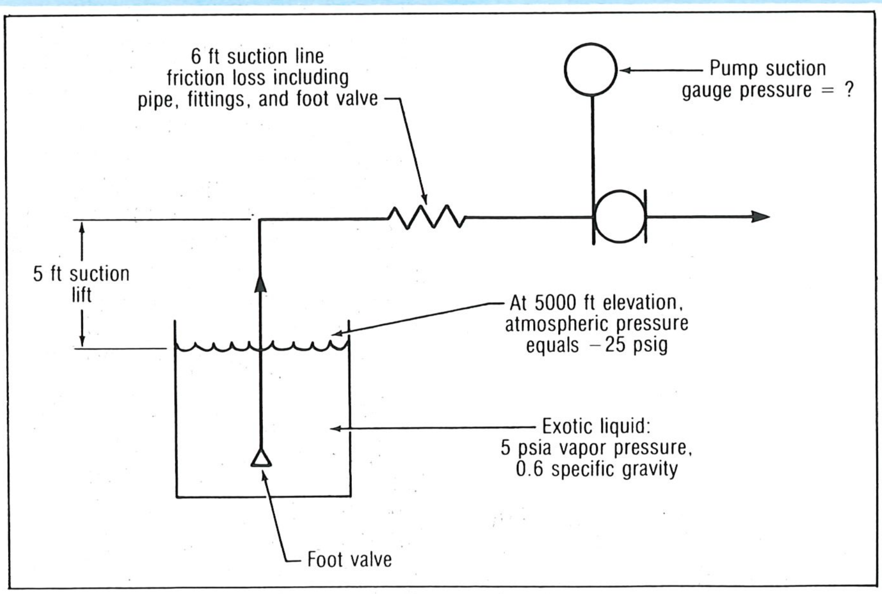
From Fig. 8,

- P_a is equal to 12.2 (absolute pressure at the 5000 ft elevation) times 2.31/0.6 (the conversion of psi to ft for the 0.6 specific gravity), or 47 ft.
- H_z is given as −5 ft.
- H_f is given as 6 ft.
- H_{vpa} is 5 (the liquid vapor pressure, psia, in this case) times 2.31/0.6 (the same conversion factor used above) or 19.25 ft.

Thus,

$$NSPH_A = 47 - 5 - 6 - 19.25 = 16.75 \text{ ft}$$

The need for conversion of absolute pressures to an abstract feet of liquid head at the density of the liquid involved has caused numerous application problems for engineers faced with only occasional demands for NPSH evaluations. The problem is neatly solved by the method proposed in the accompanying article because the abstractions have been eliminated.



8 Schematic for example problem involving exotic liquid and high elevation.

sion tank that is open to atmosphere.

The NPSH evaluation procedure for a closed loop system is identical to that previously described for an open system except for the starting pressure reference point. In an open pumping system, the starting pressure reference is atmospheric pressure pressing on the liquid surface in the open tank. In a closed loop pumping system, the starting pressure reference is compression tank pressure. An NPSH evaluation for a simple closed loop hot water heating system operating at 200 F can be used for illustration.

For this example, shown in Fig. 11, the pump suction flange gauge pressure is the compression tank fill pressure, 12 psig, minus the flow friction loss in the piping between the air separator and the pump suction, or *A-B* in Fig. 11. Flow friction loss *A-B* is shown as 2.3 ft, or approximately 1 psi. The pump suction flang gauge pressure then becomes 12 − 1 = 11 psig.

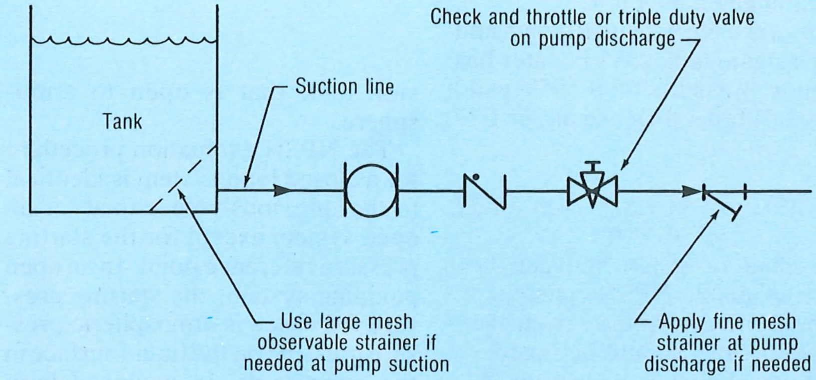
Available NPSH can now be determined from the chart (Fig. 7) by entering at 11 psig, proceeding vertically upward to intersect with the line for 200 F water, and then going horizontally to intersect with the line for a liquid specific gravity of 1. An available NPSH of 34 ft is shown.

Pump selection will be simple because most centrifugal pumps operate with less than 34 ft of required NPSH.

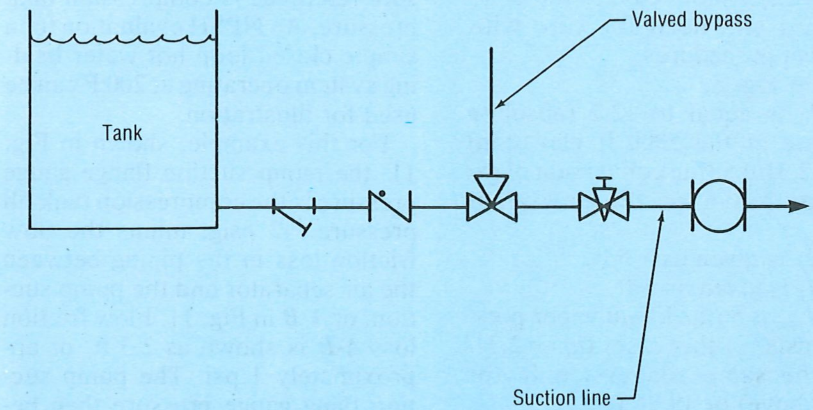
Let us now increase the supply water temperature from 200 to 240 F. Following the same procedure in Fig. 7 (11 psig to 240 F intersection to unity specific gravity intersection), we find that available NPSH is reduced to approximately 2 ft. It will be extremely difficult to find a centrifugal pump to operate under these conditions (although a specially designed feedwater pump that will operate at 2 ft NPSH is available).

The solution to the problem is evident: we must increase the compression tank fill pressure to allow operation without pump cavitation.

What is the required pressure increase? Let us assume that for the required system flow rate, a pump that has 15 ft required NPSH has been selected. Working in reverse



9 Right: Leave suction line alone.



10 Wrong: High pressure drop strainer and check, control, and balance valves in suction line invite problems.

Table 1 — Gauge pressures for various altitudes.

Altitude, ft above sea level	Atmospheric Pressure psia	Gauge Pressure psig
Sea level	14.7	0
1,000	14.2	−.5
2,000	13.7	−1
3,000	13.2	−1.5
4,000	12.7	−2
5,000	12.2	−2.5
6,000	11.8	−2.9
8,000	11	−3.7
10,000	10.2	−4.5

Table 2 — Liquid elevation heads equivalent to 1 psi.

Specific gravity	Liquid head, ft
0.5	4.62
0.6	3.85
0.7	3.3
0.8	2.9
0.9	2.56
1.0	2.31
1.1	2.1
1.2	1.93
1.3	1.78
1.4	1.65
1.5	1.54

on the NPSH evaluation chart, we enter at the point established by 15 ft available NPSH and specific gravity of 1, proceed horizontally to the left to intersect with the 240 F water line, and then drop vertically to the bottom scale to see that pump suction flange gauge pressure must be at least 17 psig if cavitation is to be avoided.

NPSH evaluation

Since there is a 1 psi pressure drop between the pump suction flange and the compression tank connection to the system, the compression tank fill pressure must be increased to *at least* 18 psig if cavitation is to be avoided.

Pump location

In open loop pumping practice, the accepted design rule is to minimize pipe suction flow friction loss between the open tank and the pump suction. The pump is placed as close as possible to the tank. A similar rule applies to closed loop pumping. In a closed loop pumping system, the pump should pump away from the connection to the compression tank and into the piping system. In addition, the suction piping flow friction loss between the tank connection and the pump suction should be kept as low as possible. The pump *should be close to* and *should always pump away from* the compression tank connection to the system.

The disastrous effect of a simple change in pump location — pumping into the tank connection — is illustrated in Fig. 12.

Many a pump has been placed on the return side of a system, discharging into the compression tank, because of the consideration that return water is not as hot as supply water and the pump is thus less likely to cavitate. This proposition is false because a pump discharging into a compression tank causes a system pressure decrease when it operates. The system pressure decrease shows up as a pressure reduction at the pump suction flange, which increases pump cavitation potential. Additional problems, such as air suction into the system through automatic air vents and consequent system corrosion, are caused by improper pump location.

The reason why improper pump location causes pump cavitation and other problems is that the compression tank junction with the system is a "point of no pressure change" regardless of whether or not the pump operates. When the pump discharges into the tank connection, its discharge pressure is fixed by tank pressure. Since the pump must cause a pressure difference across itself, the fact that discharge pressure is fixed means that pump suction pressure must decrease. This is shown in Fig. 12.

In Fig. 12, compression tank pressure is set at 12 psig. A pressure reduction of 46 ft (20 psi) occurs because of flow friction through the piping system to the pump suction. The pump suction flange pressure must then become 12 psig minus 20 psi or −8 psig during pump operation.

The NPSH evaluation chart indicates that at −8 psig and with 180 F water, a negative available NPSH results. This means that liquid will flash in the piping system prior to pump entry and that the pump will surely cavitate.

There are two solutions to the problem shown in Fig. 12:

1) The pump can be returned to the location shown in Fig. 11. In this case, available NPSH is 34 ft, as previously noted.

2) The pump can be left in its return piping location, but with the compression tank and separation unit relocated at the pump suction. In this case, available NPSH increases to more than 40 ft.

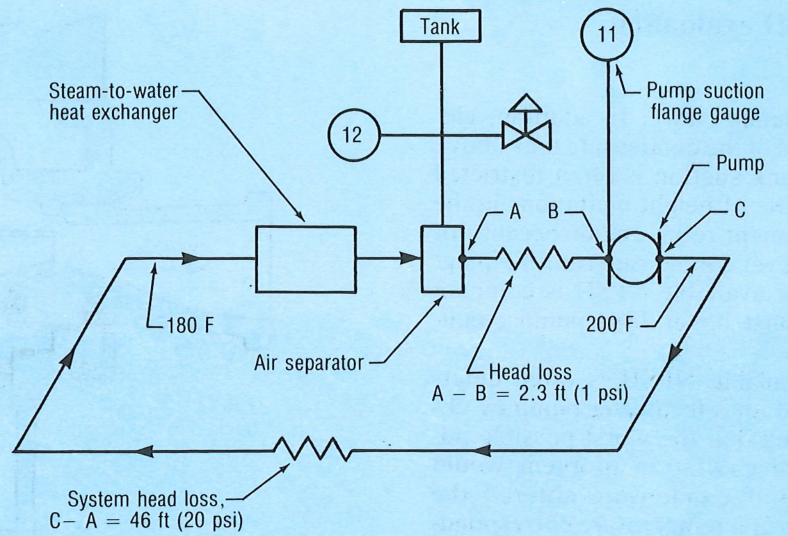
Note that a basic solution to the problem of low available NPSH for closed loop pumping systems is to *pump away from the compression tank*. This principle should be followed by engineers for all closed loop pumping applications, regardless of the liquid being pumped.

Low NPSH pumps; boiler feed

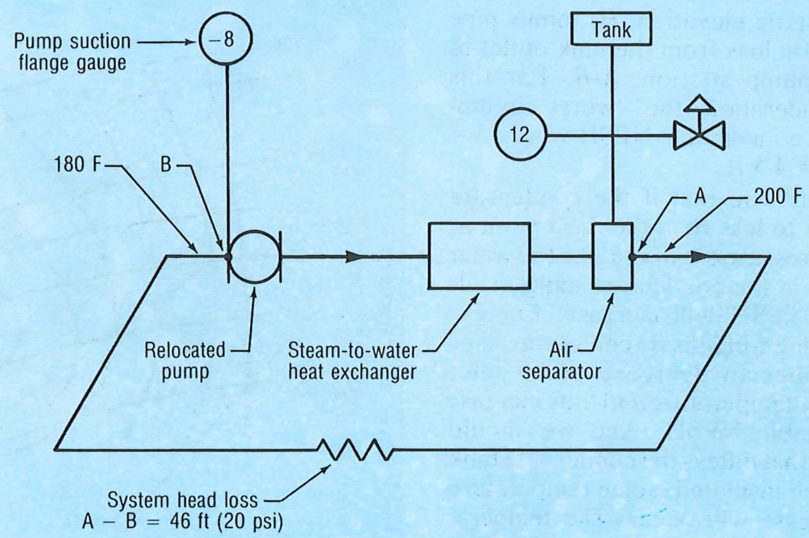
Many pumping situations require pumps with extremely low required NPSH characteristics. This is particularly true for boiler feedwater pumping, where pumps work with low available NPSH. A typical low NPSH pump is shown in Fig. 13.

The low NPSH pump uses an axial flow impeller to pressurize the eye of the main centrifugal impeller. Actual operating pressure is provided by the main centrifugal impeller. Very high discharge pressures (more than 200 psig) can be obtained with two-stage centrifugal impeller design. Required NPSH for such pumps is remarkably low over the entire flow range. They can easily be selected for less than 2 ft of required NPSH. A typical two-stage low NPSH pump curve is shown in Fig. 14.

Low NPSH pumps are widely applied for boiler feedwater pumping because liquid entering the pump is generally close to satura-



11 Closed loop pumping system serving as NPSH evaluation example. System is pressurized to 12 psig at compression tank.



12 Location of pump on return, discharging into compression tank (same as Fig. 11 except for pump location) has disastrous effect.

Table 3 — Available NPSH gains, for vented condensate tanks at sea level, for various pump inlet temperatures less than saturation (212 F).

Pump inlet temperature, F	Available NPSH gain, ft
212	0
211	0.5
210	1.4
209	2
208	2.6
207	3.5
206	4
205	4.5
204	5.2
203	5.6
202	6.3
201	6.9
200	7.5
190	10.2
180	12.5

tion temperature. In addition, elevation of the condensate tank above the tank suction is often restricted because of height limitations in the equipment room and/or because of low level condensate return piping. A low available NPSH is common for most boiler feed pump conditions.

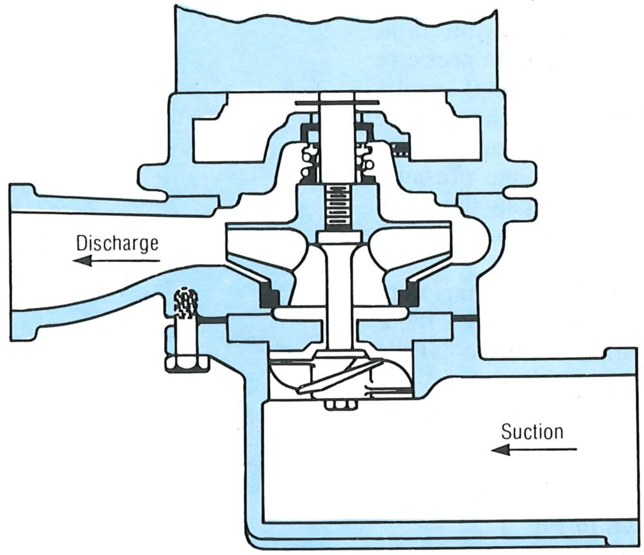
Available NPSH is often determined for a feedwater pump by observing that the worst possible potential cavitation problem would occur if condensate entered the pump at a temperature corresponding to the liquid flash point at the top water level in the condensate tank. A simplified condensate return tank and pump are shown in Fig. 15.

Note that for the worst circumstance, the available NPSH will be the static elevation, *H*, minus pipe friction loss from the tank outlet to the pump suction, *A-B*. For this consideration, the "worst circumstance" available NPSH will be 5 − 0.5 = 4.5 ft.

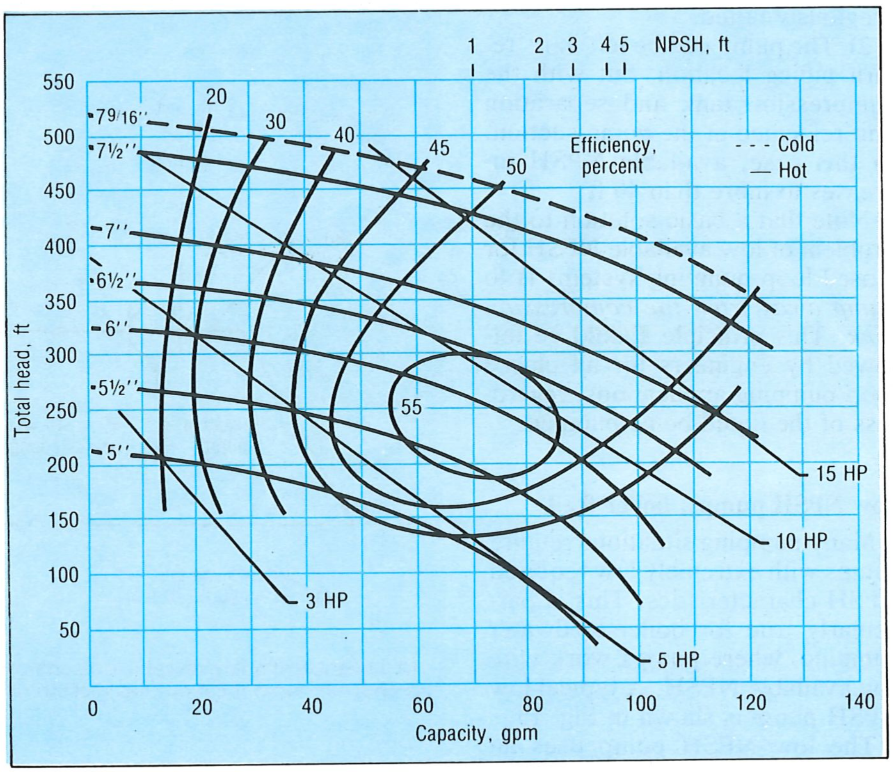
It is true that if the condensate cools to less than the flash point at the pressure existing at the top water level in the condensate tank, available NPSH will increase. Energy-wasting condensate coolers are used at times to decrease pump inlet water temperature and thus increase available NPSH. And we should note that unless the condensate tank is well insulated, some temperature decrease will occur. The temperature decrease will vary as a function of system load and boiler firing rate as well as tank heat loss. The temperature decrease effect on available NPSH is often treated as a safety factor because of estimation difficulties and because of the importance of maintaining noncavitating operation of the feed pump at *all* system operating points.

Table 3 shows the gains in available NPSH, for atmospherically vented tanks at sea level, for various pump inlet temperatures below saturation temperature (212 F).

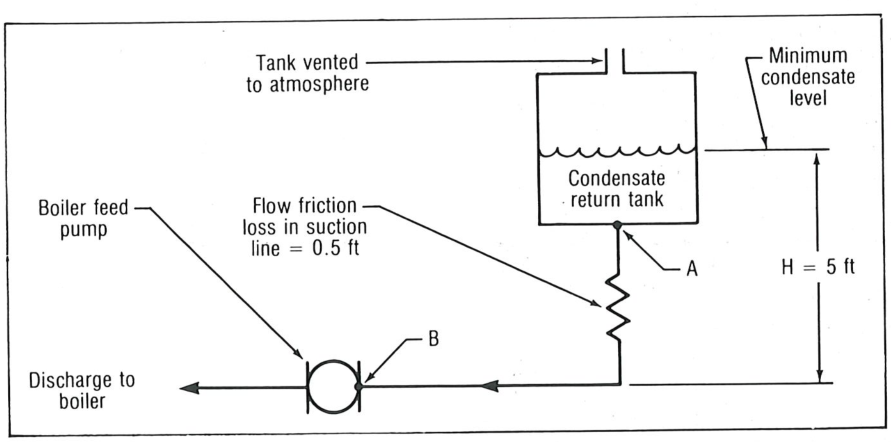
For the example shown in Fig. 15, assuming a pump inlet temperature of 208 F, we add the 4.5 ft "worst circumstance" available NPSH previously determined to the 2.6 ft available NPSH gain due to temperature loss found in Table 3 and obtain a total available NPSH at 208 F inlet temperature of 7.1 ft. Ω



13 Low NPSH pump.



14 Two-stage, 2 ft NPSH, centrifugal pump curve.



15 Simplified condensate return tank and pump.

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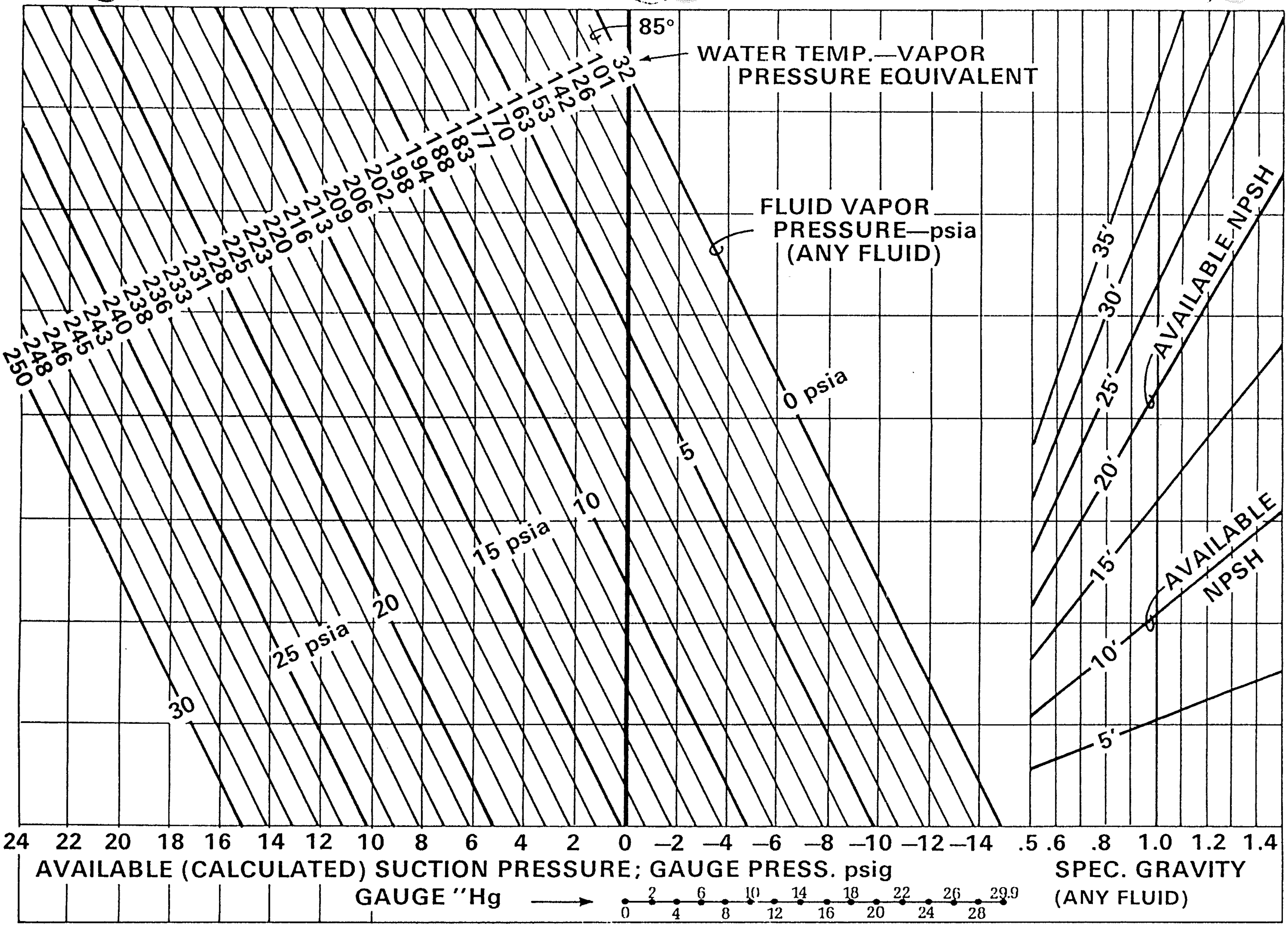

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