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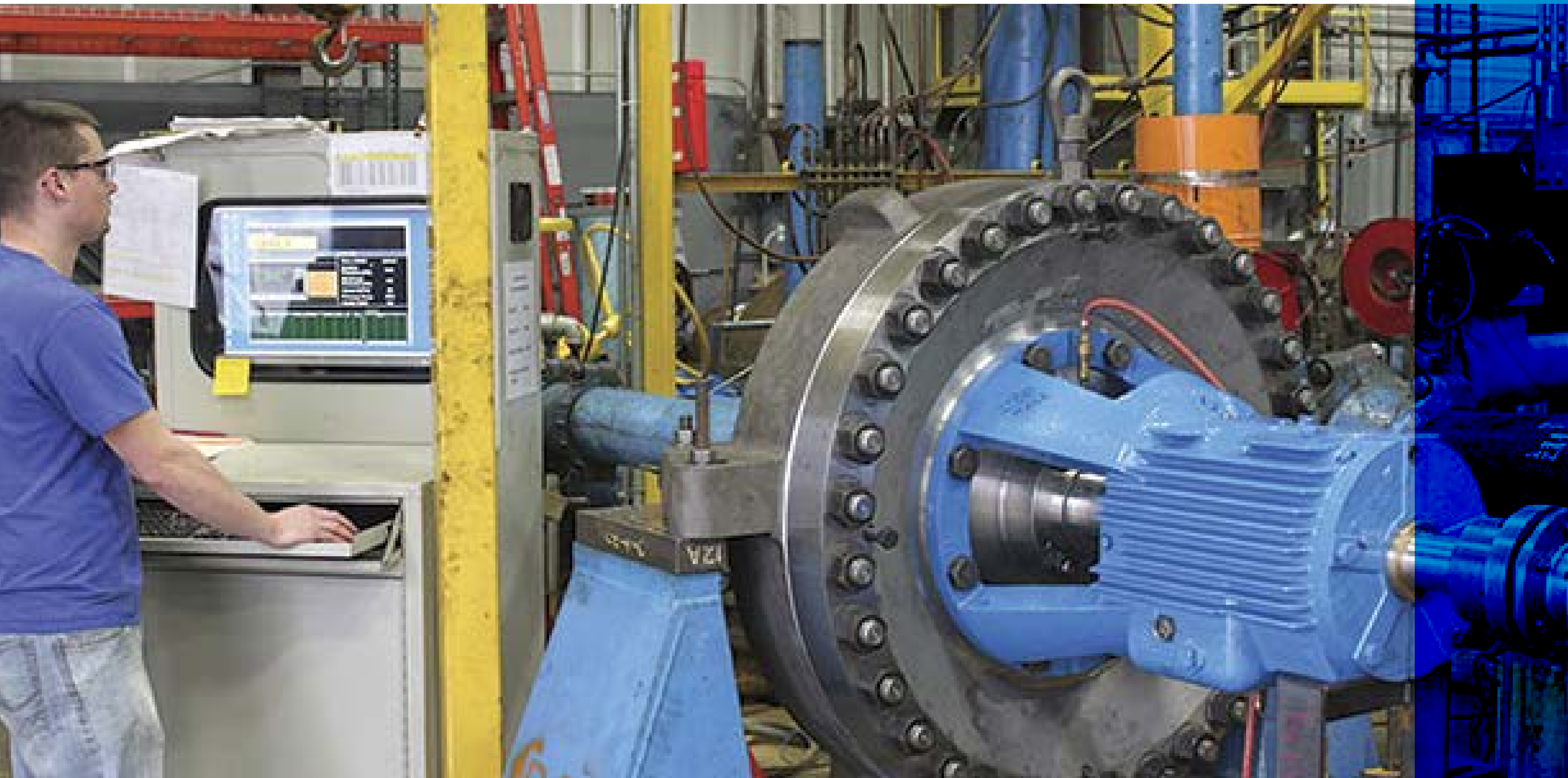
NPSHa Basics

HOW TO BEST CALCULATE NET POSITIVE SUCTION HEAD AVAILABLE IN A VARIETY OF SCENARIOS.

Articles by Jim Elsey, originally published in Pumps & Systems



NPSHa Basics

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Contents

DEAR READER,

Pumps & Systems is known for its highly technical information, and we pride on our place as the industry's best resource for operations and maintenance knowledge.

In recent years, we are seeing an increasing need for more basic articles. Our columnist Jim Elsey—the top-read writer in our publication month after month—addresses these topics with expertise and flair. In this ebook, we have collected Elsey's columns about NPSHa (net positive suction head available) in one complete package. NPSH represents one of the cornerstones of pump system operation, and it is frequently misunderstood. In these pages, NPSHa in a variety of scenarios will become easier to understand. And for more, visit pumpsandsystems.com.

Thanks for reading!

By **Jim Elsey**
Summit Pump Inc.

Jim Elsey is a mechanical engineer who has focused on rotating equipment design and applications for the military and several large original equipment manufacturers for 47 years in most industrial markets around the world. Elsey is an active member of the American Society of Mechanical Engineers, the National Association of Corrosion Engineers and the American Society for Metals.

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By Jim Elsey

Step by Step Calculation of NPSHa

Part 1 of 5

In the world of psychology and specifically in the area of introspective issues, it is often declared that, "what you resist will persist."

You've heard it before, and here we go again.

Allegedly, the most misunderstood concept in the pump world is net positive suction head (NPSH). I have written several articles on the subject and so have many other pump technicians or engineers and so-called experts.

The NPSH name itself, an acronym, confuses most pump neophytes. The subject and required calculations confounds people who are new to the industry, those on the periphery (operators or administrators) and professionals who incorrectly believe they fully understand the subject even after 25 years in the business.

I suggest we need to be concerned about this issue, because mistakes with respect to NPSH available (NPSHa) calculations are all too frequent and expensive to correct.

One of the fun parts of my position is teaching at several pump schools each year and devoting a major portion of the course to the subject of understanding the concept of NPSH and how to complete the calculations.

In the teaching process, I cover the five main examples that you will likely encounter in normal industry applications. The examples are adapted from chapter 1 of the "Cameron Hydraulic Data Book."

I will explain these five examples with the basic optimistic intent that once you learn these five examples and a few variations of each you will be able to handle the applications encountered in the real world.

As background, please re-examine two of my previous columns on the subject, one from *Pumps & Systems* August 2015 and the other from April 2018.

Definition of NPSH, NPSHa and NPSHr

The net positive suction head is the total suction head in feet of liquid (or meters), less the vapor pressure (in feet or meters) of the liquid being pumped.

Think of head as an energy level and not as a force-like pressure. All values are absolute.

NPSHa is measured at the pump centerline or the impeller eye. These two things can be at different places or elevations.

Think of NPSHa as the liquid's available energy level at the inlet of the pump or the eye of the impeller. The liquid will flash to vapor if there is not enough NPSHa.

Do not confuse NPSHa with suction pressure. While suction pressure is in some ways a component in the mix, there is something more complex to the story.

NPSHa is the amount of NPSH that the system has available at the eye of the pump impeller.

This NPSHa value is entirely a function of the liquid, its properties, ambient conditions and the suction system design and geometry. Essentially, the calculation is about the suction system itself and has nothing to do with the pump.

This calculation should be completed by the system owner, the end user and/or their engineer or consultant.

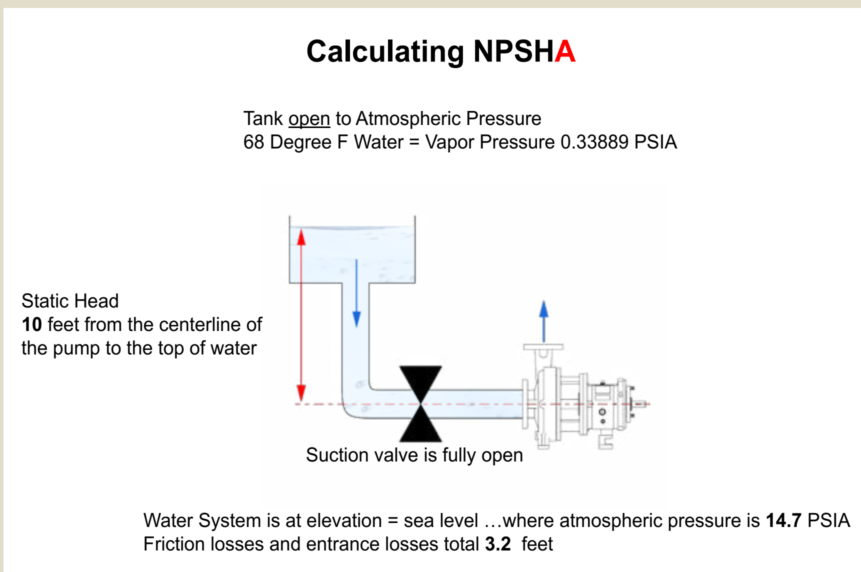

Image 1. Flooded open system with ambient temperature water
(Image courtesy of the author)

For liability reasons, manufacturers are normally directed to not be involved in the customers' calculations; however, as time marches on, the manufacturer is getting more involved mostly as a preservation issue.

I suggest we need to be concerned about this issue because mistakes with respect to NPSHa calculations are all too frequent and expensive to correct.

NPSH required (NPSHr) is most commonly determined by the pump manufacturer by empirical methods and using standards and specifications from the Hydraulic Institute (HI). NPSHr values are normally reported on the performance curves for the pump.

Note that NPSHr and NPSH3 are essentially the same thing. At the given operating point of head and flow, the pump is already slightly cavitating due to insufficient NPSH and the developed head has dropped by 3 percent while the flow rate is fixed at some value.

NPSH margin is how much the NPSHa value exceeds the NPSHr.

There are guidelines for recommended or proper margins, and I say the higher the margin, the better.

See ANSI/HI 9.6.1-2012 for more information on this subject.

NPSHa: Formula or the Figure?

I like to think of formulas as my "friends" because once I know the correct formula to use, I can simply fill in the values for the terms or components in the formula, complete the math steps and come up with the right answer.

In college, we referred to this process as "plug and chug." Some friends have told me that the moment I reference a formula in class or my articles, I lose 50 percent of my audience! So, for those of you that glaze over and hate formulas, perhaps you should study Image 1 for a few moments and see what's going on there before we approach the formula.

In Image 1, we have a tank of clear water at ambient temperature (68 F, which also means the specific gravity will be 1.0) and it is open to atmospheric pressure. Here, the tank and pump system are at an elevation near sea level.

The top surface of the water level in the tank is 10 feet above the pump centerline. We call this a "flooded suction" because the source of the liquid is above the pump impeller.

There is adequately sized piping from the tank to the pump suction with an elbow and a fully open isolation valve. We will assume that the water level remains constant at 10 feet for the example, but in the real world you will want to calculate the NPSHa for the worse condition, which will likely be at a lower level.

At this point, with just the information I have given you from the figure, you have all the data you need to calculate the NPSHa, except the friction head.

For this first example I will calculate the friction head to keep the problem simple. We will ignore velocity head since the value is normally small.

In future examples, we will discuss the methods of calculation for the friction head, and I will also calculate and show how the velocity head can affect the outcome or not.

To calculate NPSHa, you need to know:

- absolute pressure on or at the liquid surface
- static head, which is simply the vertical distance from the top of the water surface to the pump impeller eye (or pump centerline if they are the same)
- vapor pressure of the liquid being pumped, which you can easily calculate based on the temperature and is a value that is easy to look up
- friction head, which I have calculated and will advise for this example to be 3.2 feet
- velocity head, normally a negligible factor and for simplicity we will skip it this month and address in a later article in the series

How to Calculate the NPSHa for a Flooded Suction:

1. For our metric friends, these components can also be in units of meters, but stay consistent with the units.
2. The components of vapor pressure and friction will always be a negative value and consequently always work against you.
3. Absolute pressure can be zero, but it cannot be negative (by definition).
4. Suppose the liquid source was below the pump instead of above? In the case of a "lift" where the liquid level is

NPSHa = pressure absolute – vapor pressure absolute + static height of the liquid – friction

Where:
the pressures are in absolute values and all components are in units of feet.

Or: NPSHa = A – V + S – F
Or: NPSHa = h_a - h_{vpa} + h_{st} - h_f

Where:
absolute pressure = h_a
vapor pressure = h_{vpa} (absolute value)
static head = h_{st} (positive value because it is flooded)
friction head = h_f

below the suction of the pump, (not a "flooded" situation) then that value for static height is negative (the situation is referred to as a "lift") and this component in the equation now works against you.

5. We do not need to be concerned with the discharge side of the pump or system for NPSHa calculations.
6. In summary, most of the components in the NPSHa formula are working against you. You begin to understand why suction sources that are elevated, flooded, open to atmosphere and/or pressurized with the liquid at lower temperatures are desired.

Back to the formula (yes I slipped it into the article).

As mentioned before, you have the information to fill these four components

$$NPSHa = h_a - h_{vpa} + h_{st} - h_f$$

in with real numbers and complete the calculation to determine NPSHa.

The first component in the equation (h_a) represents the value for the absolute pressure above the open water tank. It was given earlier that the system is at sea level.

The liquid in the open tank is subject to atmospheric pressure. At sea level, the atmospheric pressure can be assumed to be near 14.7 pounds per square inch absolute (psia), or 0 psi gauge (psig). Note that variations in the atmospheric pressure can and will affect the NPSHa value.

Now convert the atmospheric pressure from psia to feet of head. Multiply 14.7 by 2.31, and the result is 33.957 feet, rounding to 34 feet. The value for the first component in the equation is 34 feet. We will cover the effects of higher elevations and vacuum in later articles.

The second component in the equation is h_{vpa}, or the vapor pressure for the liquid at this given temperature of 68 F. To get the vapor pressure value, simply look it up in a reference book like the "Cameron Hydraulic Data Book." The value (pressure of saturated vapor absolute) will typically be given in units of psia, will vary directly with the temperature and is also different for each liquid type.

The value you should obtain from your search is 0.33889 psia. Multiply that number by 2.31 to convert to head in units of feet. You will obtain the value of 0.7828 feet. We will round off to 0.783 feet. You now have the second value in the equation: 0.783 feet.

The third component in the equation is the static head (h_{st}). This is a vertical measurement from the surface of the liquid to the centerline of the pump (impeller eye). Remember to figure for the worst case (lowest level expected). In our example, it was stated the static height is 10 feet. No need to convert as it is already in the correct units. You now have the value for the third component in the equation: 10 feet.

The fourth component (h_f) in the equation is the friction loss in the piping, which I previously supplied at a value of 3.2 feet. You now have all four values for figuring out the answer.

Note that the given friction factor of 3.2 feet is a function of the liquid properties, the flow rate and the pipe (suction system) materials and geometry. You should understand in simple terms that for a given liquid flow rate, there will be friction losses for the pipe length, the elbow, the valve, the exit loss from the tank (large to small transition) and the entrance loss into the pump (change in diameters from the pipe to the pump nozzle).

We will address some pipe friction fundamentals in the later examples in the coming articles.

Lastly, remember we are not addressing the fifth factor in the formula, which is velocity head (h_{vel}). In a properly designed system (with Newtonian liquids in a nonslurry application), the value of velocity head will typically be under 1 foot in value. The value for the velocity head component is positive.

For now, we have the four values required to fill in for the components and calculate the answer for NPSHa. All units are in feet. The pressures are in absolute values.

Please work through the example below to see if you get the same or similar value.

Next we will calculate NPSHa for a lift condition.

Later we will look at the effects of high elevations, hot liquids, hydrocarbons and pressurized suction tanks. We will also address suction conditions under vacuum. ■

$$NPSHa = h_a - h_{vpa} + h_{st} - h_f$$

NPSHa = 34 – 0.783 + 10 – 3.2 = 40.017 feet—or rounded to 40 feet NPSHa.

NPSHa = 40 feet

References
Cameron Hydraulic Data Book, 16th edition

By Jim Elsey

Calculate NPSHa for a Suction Lift Condition

Part 2 of 5

In the last article, we worked through an example for a simple system calculation of net positive suction head available (NPSHa) based on a 10-foot flooded suction with 68 F water at sea level. The head due to friction in the suction system was 3.2 feet. After working through the formula with the data (calculated or given), the answer for the amount of NPSHa was 40 feet.

This month, we will turn the suction system around and calculate the NPSHa for a 10-foot suction lift. The liquid source will remain open to atmospheric pressure. The location elevation is at sea level and the liquid is water at a temperature of 68 F. The friction head (h_f) for the suction side system is calculated by the author at 3.2 feet. Please refer to Image 1 to see the system arrangement for this second example.

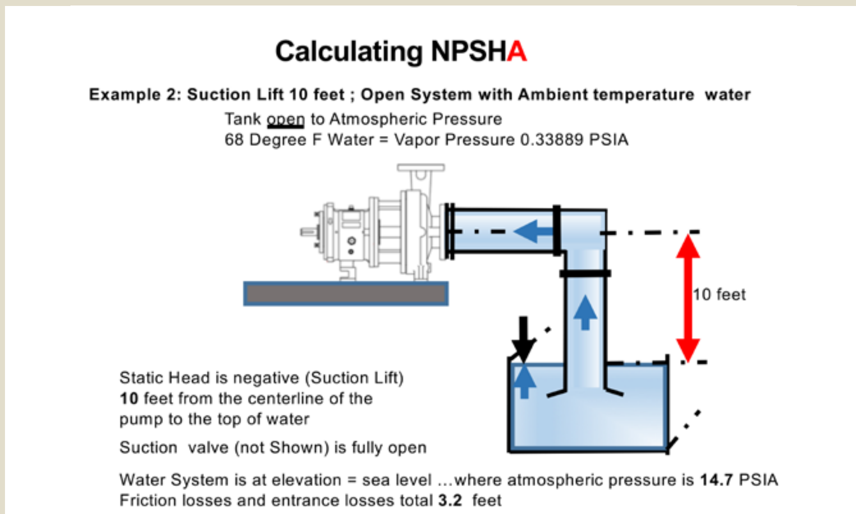

Image 1. The system arrangement for a 10-foot suction lift. *(Image courtesy of the author)*

We call this a "lift" situation because the top surface of the liquid is below the centerline of the pump impeller. Consequently, the static head will be a negative quantity.

The pump is not really "lifting" the liquid because water (along with all other liquids) has no tensile strength, and the pump is not capable of reaching out and pulling the fluid in—a common misconception.

Due to atmospheric pressure pushing down on the fluid, there is a force that causes the liquid to move to the pump when the air in the line is removed (the pump is primed). The pump exerts a small amount of differential pressure on the liquid in front of the impeller that initiates flow.

The actual suction lift is the static head plus the friction head that must be overcome.

In this case, the static head is 10 feet and the friction head is 3.2 feet, so the actual suction lift added together is 13.2 feet.

When calculating the power required to drive the pump, you need to consider the work done in the lift evolution, because the pump is moving liquid of a certain weight per unit volume up a vertical distance (lift or static suction lift). Water at 68 F is 62.32 pounds per cubic foot or 8.33 pounds per gallon.

As I said in the last article, the formula for NPSHa is my "friend" because if I can fill in the values for the formula, then all I need to do is the math to get my result (also known as "plug and chug"). The formula for NPSHa for a suction lift condition is in the box at right.

Notes:

1. (h_a) The system is open to atmosphere so the absolute pressure above the liquid in this example is simply atmospheric pressure. The pump site location/elevation was given as sea level. Consequently, the absolute atmospheric pressure at sea level of 14.7 pounds

$$NPSHa = h_a - h_{vpa} - h_{st} - h_f$$

Where:

h_a = the head from the absolute pressure acting on the surface of the liquid

h_{vpa} = the head from vapor pressure. This is always a negative value

h_{st} = the static head or elevation difference. In this case of a lift, the value is negative. Note the change in sign (+ to -) from the first example (last month) of a flooded suction.

h_f = the friction head. Friction head is always a negative component.

Equation 1

per square inch absolute (psia) can be converted to feet (absolute) by simply multiplying 14.7 psia by 2.31. The answer is 33.957 feet, rounded to 34 feet for this illustration. For elevations above sea level, the absolute pressure (head) will be reduced proportionally with the increase in elevation. Look at reference charts that will provide this information. Also, be aware there are locations that may be below sea level.

2. (h_{vpa}) The head due to the vapor pressure is a direct function of the liquid properties and its temperature. With water at 68 F, you can look up the vapor pressure in a reference source such as the "Cameron Hydraulic Data Book" and it is usually expressed in units of psia. The 68 F water will have a vapor pressure of 0.33889 psia, which will convert to 0.7828 feet when you multiply by 2.31 and divide by the specific gravity (SG) of 1.0.
3. (h_{st}) The static head will be negative because it is a lift situation, meaning the surface of the liquid is below the centerline of the impeller. The difference in elevation is 10 feet.
4. (h_f) The friction head was given as 3.2 feet.
5. ($h_{velocity}$) This is not shown in the equation. The velocity head is not required when calculating the NPSHa. It is accounted for elsewhere in the formula.
6. You should always calculate for the worst condition. The liquid surface level may be a different (lower) level. The temperature of the liquid may be hotter, which will increase the head of vapor pressure. The friction head may also increase with time, fouling, corrosion or with different valve settings.

To solve for NPSHa, fill in the given or calculated values. Again, the static head factor is negative in this example because it is a lift condition, and for last month's example the static head was positive because it was a flooded suction.

The NPSHa formula for a lift is shown in Equation 2 below.

Last month, the NPSHa result for the 10-foot flooded suction Example 1 was 40 feet. The only change made this month for Example 2 was that the flooded suction was changed to a 10-foot suction lift with the NPSHa result of 20 feet.

$$NPSHa = h_a - h_{vpa} - h_{st} - h_f$$

Filling in the factors:
NPSHa for the 10-foot suction lift example = 34 – 0.783 – 10 – 3.2 = 20 feet
NPSHa = 20 feet for this example.

Equation 2

Recognize that simply changing the conditions from a 10-foot flooded suction to a 10-foot suction lift changed the NPSHa by a difference of 20 feet.

Velocity Head ($h_{velocity}$): To Be or Not To Be?

I wish to explain a point of confusion for many of the readers. (I am guilty for not being consistent with this practice across many of my articles.)

The component of velocity head ($h_{velocity}$) in the NPSHa formula is not required when you are calculating NPSHa using the formula method, but it is required when you are measuring for NPSHa.

There are two methods for the determination of NPSHa—either the calculation method or the empirical measurement for NPSHa. The fifth component in the NPSHa formula is velocity head and must be included if you are (measuring) determining the NPSH from a gauge reading on the pump suction. Velocity head is already included if the NPSH is established from a difference in elevation. The velocity head component is a positive value when used in the equation.

Measuring NPSHa = head of inlet + head of velocity – head vapor pressure.

Head of the inlet is equal to the absolute head + or – the static head – the friction head (as shown in Equation 3 below).

It is important to know that when measuring for NPSHa using a suction pressure gauge, the difference in elevation between the gauge and the pump impeller centerline (my proposed datum point for this example) must be corrected for the NPSHa result. If the gauge is above the datum point, add the result (how many feet or fraction thereof) and if the gauge is below the datum, subtract the elevation amount.

$$h_{inlet} = h_{absolute} \ (+ \text{ or } -) \ h_{static} - h_{friction}$$

Where:
You can substitute h_{inlet} for gauge pressure at or near the pump suction and correct for elevation to the datum point.

Equation 3

Most gauges used in the field are not accurate enough or in the wrong range/scale to measure for the specific purpose of NPSHa. Using a calibrated gauge in the proper range, with the proper scale, can yield good results. ■

References
Cameron Hydraulic Data Book, 16th edition

By Jim Elsey

Calculating NPSHa When the Liquid Is Above Ambient Temperature

Part 3 of 5

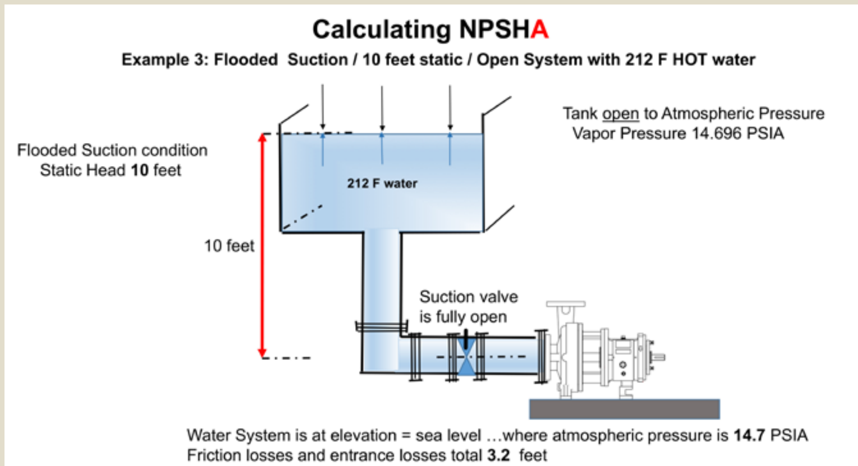

Image 1. Flooded suction with hot water *(Image courtesy of the author)*

In the first part of this five-part series, we defined net positive suction head available (NPSHa) and then walked through a simple calculation based on ambient temperature water for a flooded suction condition.

In part two, we worked through an almost identical situation except the water source was at a level below the pump centerline, which is a suction lift situation. In both cases, the systems were open to atmospheric pressure, located at the elevation of sea level. The liquid temperature was at ambient 68 F.

Flooded Suction Pumping Hot Water

In this third part of the series, we will investigate what happens when the liquid temperature is above ambient. Before we get to the formula and the calculation, I recommend a review of the definition for vapor pressure.

I covered vapor pressure in my April 2018 *Pumps & Systems* column should you wish to revisit. A brief summary is as follows.

A liquid in an open container will eventually evaporate to a vapor unless some other force is present to prevent the change from occurring.

In most examples, that force is simply atmospheric pressure. Even a bowl of water left on the kitchen counter will evaporate away over time, but it will happen at a significantly faster rate with each incremental increase in temperature.

The amount of energy associated with the vapor pressure is subtracted from the total energy level available for the NPSHa. The energy associated with the vapor pressure is always a negative quantity. In the context of NPSHa, vapor pressure is not a friend to the pump.

For a given temperature, a liquid exerts a certain pressure to the atmosphere, and the atmosphere exerts a counter pressure in return.

For water temperatures below 212 F, the atmospheric pressure is greater than the vapor pressure. Those two pressures (vapor and atmospheric) would be in equilibrium if the water was at 212 F at sea level, where the water would begin to boil.

If the pump example was at higher elevations (lower ambient pressure and consequently a lower equilibrium temperature) the water would boil at a lower temperature.

Vapor pressures for water at various temperatures are easy to calculate or obtain in a reference source, but it is often very difficult to find reliable information on vapor pressures for other liquids. It is not uncommon for the end user/operator to be using the incorrect vapor pressure in their calculations.

The Formula for NPSHa

Remember that we are calculating NPSHa, so we do not need to include velocity head. Velocity head would be included if we were measuring NPSHa. Please refer to Image 1.

The water level is 10 feet (h_{st}) above the pump centerline, and let us assume the water level will remain at that height for this example. We will assume the supply rate is the same as the demand rate. In a real world situation, you must calculate NPSHa based on the expected worst-case scenario. For example, the application could be a batch process where the tank will be nearly empty at some point.

For ease in working the example, I have calculated the total friction losses (h_f) as 3.2 feet. Note that friction losses are technically lower for hot water than cold, but we will ignore the small difference for the examples in this series.

The tank is open to atmospheric pressure, and the system is located at an elevation of sea level. The absolute pressure in feet of head (h_a) is 35.4 feet as a result. Remember from the formal NPSHa definition for

$$NPSHa = h_a - h_{vpa} + \text{or} - h_{st} - h_f$$

Where:

h_a = the absolute pressure. Absolute pressure as measured in feet of head of the liquid being pumped at the surface of the liquid.

h_{vpa} = the vapor pressure. The head in feet corresponding to the vapor pressure of the liquid at the temperature being pumped.

h_{st} = the static head of the liquid over the pump centerline for a flooded suction in feet (positive value for flooded suction)

h_{st} = the static head of the liquid below the pump centerline for a lift situation in feet (negative value for lift situations)

h_f = the total friction loss in feet of head for the suction side system

Equation 1

absolute pressure (h_a)… "is the absolute pressure as measured in feet of head of the liquid being pumped at the surface of the liquid." Absolute atmospheric pressure is 14.7 pounds per square inch absolute (psia). To convert to feet, multiply by 2.31 and divide by the specific gravity of the liquid being pumped. (See Equation 2.)

In real life examples, the actual atmospheric pressure will be slightly lower and vary with weather/barometric pressure and elevation above or below sea level.

$$\frac{(14.7 \times 2.31)}{0.9580} = 35.4 \text{ or} \approx 35 \text{ feet}$$

Equation 2

Standard pressure at sea level may also be listed as 14.696 psia in lieu of 14.7. Either way, it rounds off to 14.7 psia.

The more important fact is to know the actual elevation for the pump location and adjust your calculations accordingly.

The difference between this third example and the first example is that now the water temperature has been increased from 68 F to 212 F. Both examples are a flooded suction and an open system. Vapor pressure for 212 F water can be obtained from a reference text. Vapor pressure is normally expressed in units of psia, which must then be converted to feet ("for the liquid being pumped") for use in the previous equation. The vapor pressure for water at 212 F is 14.696 psia. When converted to feet, this will round off to 35 feet.

The calculation is shown in Equation 3.

$$\frac{(14.696 \times 2.31)}{0.958} = 35.4 \approx 35 \text{ feet}$$

Equation 3

The significant difference for the current example is that the vapor pressure will be higher because the temperature is higher. The vapor pressure for 68 F water is 0.783 feet. (To obtain that value, convert 0.33889 psia to head by multiplying by 2.31 and dividing by specific gravity, which was 1.0.) But now the vapor pressure for 212 F water is 35 feet. See Equation 3 above. This is a significant difference of almost 34 feet over the 68 F water example and the major point of this article.

Many sales engineers, pump technicians and operators will work for years in the pump application world and will have no pump suction side issues with NPSHa until that first time they encounter an application involving hot liquids (higher vapor pressures), where the previous "forgiving adequacy" of 34 feet NPSHa margin was just "mysteriously stolen" from them by the thief that is vapor pressure. A similar scenario plays out when people who have always worked with applications involving flooded suctions are instead confronted by a project with a lift or a vacuum on the suction side.

To continue the full calculation for this example, the head due to static height (h_{st}) is positive because it is a flooded suction above the pump centerline.

$$NPSHa = h_a - h_{vpa} + h_{st} - h_f$$
$$NPSHa = 35 - 35 + 10 - 3.2 = 6.8 \text{ feet}$$

Equation 4

When the water temperature was 68 F, the NPSHa was 40 feet. Now that the water is 212 F, the NPSHa has dropped to 6.8 feet—a significant difference of 33.2 feet.

Summary

The situation illustrated in this article where the vapor pressure cancels out the absolute pressure is very common in the industry because many processes are designed to operate at or near equilibrium for maximum efficiency.

The only system design aspect working for your pump application in a positive way remains the static head (h_{st}). This is why you see so many elevated tanks in chemical plants and refineries and also why condensate pumps are always in the lowest level of the steam plant. ■

References

Cameron Hydraulic Data Book, 19th Edition

By Jim Elsey

Calculating NPSHa for a Closed & Pressurized System

Part 4 of 5

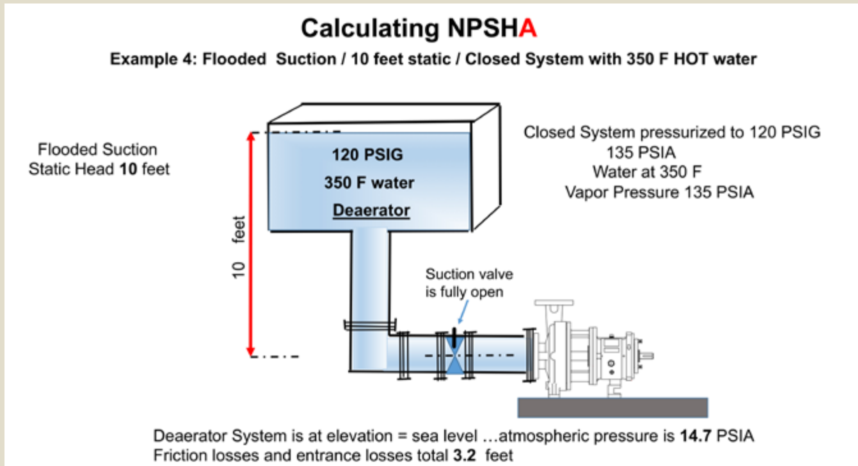

Image 1. Flooded suction in closed system with hot water *(Image courtesy of the author)*

In the first three parts of this five-part series, we covered the calculation of net positive suction head available (NPSHa) for a flooded suction, a lift condition and a third scenario where the water was hot (212 F) on a flooded suction example. Note we did not go over a hot water lift condition example because these situations are not practical or feasible.

The negative aspects of the vapor pressure and the static lift components in conjunction with the friction will negate almost all of the energy that is supplied by atmospheric pressure ($h_{absolute}$). Always be very careful and circumspect of any hot liquid lift situation for these reasons.

Pressurized Suction Source

The majority of pump applications in North America are installed at elevations below 5,000 feet (reference to sea level). When the system is open to atmospheric pressure, we automatically gain the energy from the air column pushing down on the surface of the liquid.

At sea level, we gain 34 feet of absolute pressure and at 5,000 feet above sea level (ASL), we still have 28 feet of energy due to the atmospheric pressure.

So far in this series, the examples have been for open systems. This means the liquid supply source was open and subject to the atmospheric pressure for the elevation and barometric conditions at the pump site.

Please note that just because a tank has a top or cover on it does not mean it is a pressured tank.

One of the biggest mistakes I see in new pump applications is the false assumption that suction pressure is equivalent to NPSHa. For more than 47 years, I have unfortunately witnessed this mistake being made time and again.

I talk with other technicians, engineers, industry peers and even industry contacts at competitive companies. This common, yet preventable, mistake is the major point of this article. Suction pressure is not NPSHa.

About This Example

The example for this article (see Image 1) shows a closed and pressurized suction tank at 120 pounds per square inch gauge (psig) and 350 degrees F. The system is at sea level. It is important to state this not because the atmospheric pressure is exerting on the fluid, but because it also affects the gage pressure readings.

- (gage = absolute – atmospheric)
- (atmosphere = absolute – gage)
- (absolute = gage + atmospheric)

The static head is 10 feet (hst). This is the vertical distance from the top of the liquid surface to the centerline of the pump. The value for the static head component is positive because this is a flooded suction situation.

Steam systems that are 100 to 150 psig are common in industrial and commercial applications. If you are involved in pump applications, you will come across this example at some point.

This is specifically an example of a deaerator tank where 120 psig steam is used to both preheat the water and reduce and/or eliminate the dissolved gas level in the feedwater.

The main culprit to be removed from the liquid is oxygen. By reducing the oxygen level in the system, the corrosion is minimized.

The saturation temperature for 120 psig steam (134 5 psia) is in essence 350 F. Consequently the corresponding vapor pressure for the liquid will be the same or very close (h_{vpa}).

By Jim Elsey

Because the system is at or close to saturation, corresponding head from the absolute pressure will negate head from the vapor pressure component in the NPSHa equation. This is common in industrial and commercial applications.

Remember from part one of this series that we need to convert to absolute values when calculating NPSHa. The absolute pressure for the system would be approximately 135 psia. Absolute pressure equals gauge pressure plus atmospheric pressure, therefore 120 + 14.7 = 134.7. We will round off to 135 psia.

Notice that while the suction pressure is almost 120 psig, the result for the NPSHa calculation will be less than 7 feet. This is one reason I instruct all of my pump school students to always calculate the NPSHa. The other reason is that I see the mistake of confusing suction pressure for NPSHa on a regular basis.

The Formula

I know many of you hate this part, but remember the formula (the equation) is your friend in these examples. If you know the formula you can just "plug" in the values and "chug" through the math to get the correct answer.

Remember that we are calculating NPSHa (Equation 1), so we do not need to include velocity head.The water level is 10 feet (10 feet of static head hst) above the pump centerline (Image 1). For this example, let us assume the water level will remain at that height. We will assume the liquid supply rate is the same as the demand rate. In a real world situation, you must calculate NPSHa based on the expected worst case scenario.

For example, the application could be a batch process where the tank will be nearly empty at some point. In this case of a deaerator system the level will likely remain the same.

For ease in working the example, I have calculated the total friction losses (h_f) as 3.2 feet. Note that friction losses are technically lower for hot water than cold, but we will ignore the small difference for the examples in this series.

The tank is closed off to atmospheric pressure and the system is pressurized with steam to 120 psig. The absolute pressure in feet of head (h_a) is 350 feet as a result. Remember the formal NPSHa definition for absolute pressure (h_a): "the absolute pressure as measured in feet of head of the liquid being pumped at the surface of the liquid." Depending on which method of conversion of pressure to feet you use, the answer will still round off to 350 feet (Equations 2 and 3).

The vapor pressure for 350 F water is 134.604 psia. I looked for this value in the "Cameron Hydraulic Data Book," but you can find the information in several technical reference sources. I then converted to units of feet (Equation 4).

$$NPSHa = h_a - h_{vpa} + h_{st} - h_f$$

Where:

h_a = the absolute pressure. Absolute pressure as measured in feet of head of the liquid being pumped at the surface of the liquid.

h_{vpa} = the vapor pressure. The head in feet corresponding to the vapor pressure of the liquid at the temperature being pumped.

h_{st} = the static head of the liquid over the pump centerline for a flooded suction in feet (positive value for flooded suction)

h_f = the total friction loss in feet of head for the suction side system

Equation 1

$$\frac{[(135\text{ psia}) \times (2.31)]}{0.8904} = 350.23 \approx 350 \text{ feet } (h_a)$$

Where:

0.8904 = the specific gravity of water at 350 F

Equation 2

$$\frac{[(135\text{psia}) \times (144)]}{55.59} = 349.703 \approx 350 \text{ feet } (h_a)$$

Where:

55.59 = the specific weight in pounds per cubic foot

Equation 3

The Calculation

At this point, you have all the data to complete the calculation. All units are in feet at absolute values.

$NPSHa = h_a - h_{vpa} + h_{st} - h_f$

NPSHa = 350 - 350 + 10 – 3.2 = 6.8 ft.

Ask yourself if the pump you would select for this application has a net positive suction head required (NPSHr) value of less than 5 feet.

In conclusion, many technicians and operators will not bother to conduct the NPSHa calculations because the suction pressure of 120 psig leads to a false belief that it is not necessary.

Do not let it happen to you. The mistake is very costly to correct.

References
Cameron Hydraulic Data Book, 19th Edition

By Jim Elsey

NPSHa: Calculating for Systems Under Vacuum

Part 5 of 5

In the first four parts of this five-part series, we covered the calculation of NPSHa for a flooded suction, a lift condition, a hot water flooded suction and a pressurized hot water flooded example. In this fifth and final working example, we will investigate what happens to NPSHa when the system is under vacuum.

Understanding a Vacuum

The concept of vacuum is frequently misunderstood and is a source of confusion to many in the pump world. In general, people that manufacture, sell, maintain and operate vacuum pumps and other related vacuum equipment know the terms, units and principles very well, but the rest of us are often confused and dismayed by the subject. Nomenclature in the world of vacuum applications can be confusing and is often counterintuitive; for example, the term "high vacuum" simply implies low pressure. The higher the vacuum, the lower the remaining pressure, and vice versa.

What if I told you that vacuum is pressure? Most people would surely dismiss that statement as silly, but please think about it. In a vacuum, there remains an amount of pressure that is below atmospheric pressure, but is also above absolute zero. Even in a container at "middle to high vacuum" there is some pressure remaining.

The application example for this article is a steam condenser. Condensers operate in a vacuum by design because this approach maximizes efficiency for the steam system. If you are applying pumps in commercial or industrial applications, you will eventually encounter a situation where the liquid on the suction side of the pump is under vacuum. Condensers are not the only applications concerned with vacuum. The pressure in the suction line of a centrifugal pump operating in a lift application will most often be in a vacuum—that is, at a pressure less than atmospheric.

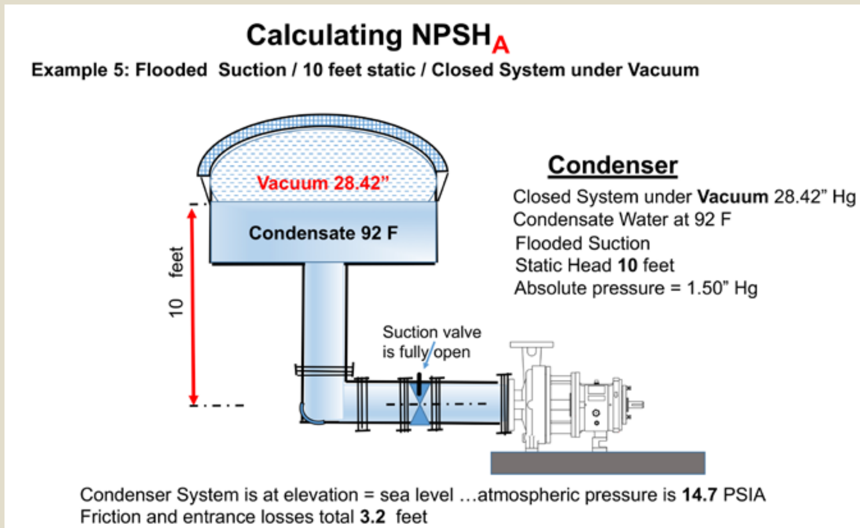

Image 1. Flooded suction/10 feet static/closed system under vacuum *(Image courtesy of the author)*

To be clear, when we state a pressure, we should add the mode to differentiate the pressure measurement we are referencing.

That is, we should state with the units and measurement quantity the correct mode; either vacuum, atmospheric, gauge or absolute.

For this article, we will refer to the area of vacuum as that pressure range (mode) below atmospheric pressure and above zero pressure absolute. Note that atmospheric pressure changes with the weather (barometric pressure) and the elevation above or below sea level.

Absolute Pressure = Gauge Pressure + Atmospheric Pressure

To help in understanding the next part of this example, I will add a few comments. Atmospheric pressure at sea level equals 14.7 pounds per square inch (psi) and that equates to a pressure of 29.92 inches of Hg (mercury). To convert inches of mercury to units of feet, multiply by 1.1349. For those that deal in SI units or work professionally in the vacuum arena, note that a Torr is defined as 1/760 of an atmosphere and may be expressed as 1 mm-Hg, where 760 mm-Hg also equals 29.92 inches Hg.

Also in what appears to be an evil plot just to confuse the neophytes, we reverse the scale when we switch from pressure to vacuum.

When we use the expression "full vacuum," we are referring to the vacuum level of 29.92 inches of Hg. Note we state full vacuum and not a perfect vacuum. At full vacuum, there is no pressure remaining for the NPSHa calculations.

The Mistake

The mistake most often made in vacuum applications is thinking the level of vacuum is the same as the pressure and that the units just need to be converted. Further, when the person calculating the NPSHa for the system is looking to determine the value of the first component in the NPSHa equation (which is h_a or $h_{absolute}$) many people just convert the vacuum measurement to feet of water and use that result in the equation. That approach will yield the wrong answer. For example, 28 inches of mercury vacuum converts to approximately 32 feet.

The correct answer is discussed later.

[28 x 1.1349 = 31.7772 ≈ 32]

Equation 1

The Formula

For the correct answer, we first need to determine the amount of residual pressure in the condenser. Then use the formula to calculate the resultant NPSHa. I know many of you hate formulas, and I will remain insistent they are your friends.

Refer to Image 1, which depicts the condenser application for this example. The application is at sea level. The condenser is operating at a vacuum of 28.42 inches of mercury (28.42" Hg). Note we are measuring vacuum so the scale is now reversed from our normal perception of pressure. The higher the vacuum, the closer we approach the maximum or full vacuum of 29.92" Hg. At zero or low vacuum, the measurement would be 0" Hg.

Using the information I provided Using the information above, you now know there is still a small amount of pressure remaining in the condenser. You also know that a full vacuum is accepted to be 29.92" Hg. Consequently the difference between the full vacuum and the measured vacuum is therefore 1.5" Hg or 1.7 feet [29.92 – 28.42 = 1.5" Hg].

The 1.5" Hg converts to 1.7 feet of head absolute (1.5 x 1.1349 = 1.7 and note the units are now feet). Now you have the first component in the NPSHa formula as 1.7 feet. The second component in the formula is the vapor pressure (h_{vpa}). I provided the information that the water was at approximately 92 F. Water at this temperature has a vapor pressure of 0.741457 pounds per square inch absolute (psia) and that pressure converts to a head of 1.73 feet which rounds to 1.7 feet.

Caution: remember from the definition of NPSHa formula the vapor pressure is not simply a conversion, but is defined as the head in feet corresponding to the vapor pressure of the liquid at the temperature being pumped. Another way to calculate would be [1.5" Hg x 1.1349 = 1.7 feet].

The third component in the formula is the head due to static height (h_{st}), which was provided in the figure as 10 feet. The fourth component in the equation is the head due to friction (h_f), which was provided by the author as 3.2 feet.

Now you simply need to insert the values in the equation and complete the math steps (see Equation 3).

$$NPSHa = h_a - h_{vpa} + h_{st} - h_f$$

$$NPSHa = 1.7 - 1.7 + 10 - 3.2 = 6.8$$

NPSHa = 6.8 feet

Equation 3

When you conduct your calculations for your own applications, you may have results that differ slightly due to temperature conversions, rounding and conversions between units.

Another explanation for result variance can be different techniques such as using specific weights versus specific gravities when converting between pressure and head; both methods are correct, but may yield slightly different answers.

You can see from the example that the positive component of static head minus the friction becomes the main contributor to the NPSHa total. The head due to absolute pressure and the vapor pressure components cancel each other out because the system is at, or near, equilibrium. This condition is also referred to as saturation.

The friction component takes an additional toll on the total leaving some portion of the static head as the main factor. Pumps applied in low NPSHa situations such as condensate service are always going to be in the lowest level of the plant for the aforementioned reasons. In a refinery application, the tower will need to be at a higher elevation so the pumps have

$$NPSHa = h_a - h_{vpa} + h_{st} - h_f$$

Where:

- h_a = the absolute pressure. Absolute pressure as measured in feet of head of the liquid being pumped at the surface of the liquid. This will be barometric pressure if suction is from an open tank; or the absolute pressure existing in a closed tank such as a condenser hotwell or deaerator.
- h_{vpa} = the vapor pressure. The head in feet corresponding to the vapor pressure of the liquid at the temperature being pumped.
- h_{st} = the static head of the liquid over the pump centerline or impeller eye for a flooded suction in feet (positive value for flooded suction). Not all impeller centerlines correspond to the pump centerline.
- h_f = the total friction loss in feet of head for the suction side system.

Equation 2

By Jim Elsey

adequate NPSHa. At this value of 6.8 feet, you can also see why condensate pumps operate at slow speeds, are physically placed well below the condenser, often have special impellers, enlarged eyes or inducers on the first stage and are frequently dual suction.

The NPSHa in this example is 6.8 feet, but also think of it as the net static submergence. If you refer to the “Cameron Hydraulic Data Book” and view the NPSHa calculation for condenser applications, the editor also points out the vacuum application is similar to, and can also be equated as, a lift situation. The equivalent suction lift is equal to the difference between the “vacuum effect” and the net submergence. In this case it would be 25.45 feet. That is, this vacuum application can be equated to a lift situation of 25.45 feet. The text example is a different result because the friction factor was lower.

28.42 Hg of vacuum x 1.1349 = 32.25 feet	
Static submergence	= 10 feet
Friction losses	= 3.2 feet
Net static submergence	= 6.8 feet
Equivalent suction lift	= 32.25 – 6.8 = 25.45 feet

Equation 4

Conclusion

Remember, static height and the NPSHa formula are your friends, vapor pressure is not. Next I will summarize the subject and lessons learned from the five-part series in combination with a discussion on how to address the issue if you do not have sufficient NPSHa. ■

References

Cameron Hydraulic Data Book, 19th Edition

By Jim Elsey

NPSHa: A Summary & How to Solve Problems

The five-part series of articles covered the calculation of net positive suction head available (NPSHa). I summarized the NPSHa concept and said that firmly grasping the theory and completing the calculation can often be a tricky process. However, with an understanding of the basics and some practice, you can gain confidence and work through most applications. The five examples in the series were selected to cover almost every aspect you will encounter in the real world.

In most real-life NPSH issues, we are not the person that is conducting the initial NPSHa calculation for a system and initially selecting the pump. The more likely scenario is that we are stuck with an existing system problem, and the associated pump is cavitating toward a short and very expensive life ending. The guilty parties are gone or not talking.

Why Cavitation is a Bad Thing

If there is insufficient NPSHa, the pump will cavitate. Cavitation causes pump damage and a reduction in performance. The pump damage manifests as mechanical seal and bearing damage. In the later stages, it can also destroy an impeller. All damage is expensive.

Most readers know that cavitation (classic) is the formation of vapor bubbles in the liquid.

These bubbles form because the pressure on the liquid has dropped below the vapor pressure (NPSH required [NPSHr] exceeds NPSHa). This issue normally occurs near the eye of the impeller since this is the lowest pressure area in the suction system. The bubbles subsequently collapse when they reach an area of higher pressure at about one-third to one-half the distance along the underside of the impeller vane. The formation of the bubbles does little physical damage. Cavitation will affect the pump hydraulic performance. The collapse of the bubbles potentially creates serious damage to the impeller.

NPSH Margin

To preclude or mitigate cavitation, you must have more NPSHa than the pump requires.

How much NPSH margin you need to preclude cavitation varies with each application. The more margin, the better. Guidelines and rules of thumb are as plentiful and reliable as urban myths. I recommend you read American National Standards Institute/Hydraulic Institute (ANSI/HI) specification 9.6.1 to gain a better understanding. The liquid properties and the suction energy level are the differentiating factors.

$$\text{NPSHa} \div \text{NPSHr} = \text{NPSH margin}$$

Where:
NPSHr is also equal to NPSH3

Equation 1

How to Fix a Cavitating Pump

I am frequently asked this question, and I normally suggest a look at the NPSHa formula and its four components for the solution.

Using each of the four components from the formula, you can map potential solutions to solve the existing NPSHa problem.

TIPS FOR CALCULATING NPSHa

1. Always calculate the NPSHa when choosing, applying or troubleshooting a pump.
2. Always work in absolute values.
3. Keep the units consistent. I recommend working in feet of head if you are working in U.S. customary (USC) units or meters of head if using metric International System (SI) units.
4. Use the NPSHa formula. It is your friend.
5. Always calculate for the worst condition (most restrictive) in the system.
6. Suction pressure is not NPSHa.
7. Do not confuse submergence with NPSHa. You need to calculate for both.
8. Almost every pump problem is on the suction side.
9. Vapor pressure is not your friend. Always know the liquid properties.
10. In a vacuum, there is still some pressure. It is just at a level below atmospheric pressure.
11. For a given pump, the same flow rate (Q) using a smaller impeller will require more NPSH. Look at using a larger impeller if feasible. Note the total dynamic head (TDH or TH) will be different.
12. When in doubt, revert back to this series of articles or call your "pump phone a friend."

By Jim Elsey

The first factor in the formula is absolute pressure (h_a). This factor is always positive. If the suction source is already open to atmosphere, there is little you can do as it is both unlikely and unrealistic to change anything within your control. You cannot change atmospheric pressure or move the pump/system location to a lower elevation in regards to sea level. However, if there is an issue, it will help you understand why the pump is cavitating. If the system is closed and under pressure, there is a possibility you can increase the pressure (consequently the absolute head [h_a]) in some manner. My experience with plant owners and operators is that raising the system suction pressure is almost never going to happen due to overriding and/or higher priority constraints.

The second factor in the formula is the vapor pressure (h_{vpa}). The higher the temperature, the higher the vapor pressure and the higher the negative effect. In my experience, I have only witnessed one case where the customer was willing or able to reduce the system temperature, but it is still a question that must be asked. Even a few degrees can have a significant effect.

The third component in the formula is the static head (h_{st}). Sometimes you can convince the system owner to keep the supply tank (flooded situation) or the sump (lift condition) at a higher level. If you are lucky, the few feet the static head is increased can make a big difference. I have been involved in a few cases where the pump was moved to a lower level and in one case, a lower level was created for the pump. These solutions are expensive.

The fourth component in the formula is the friction factor (h_f). Of all the factors in the formula, I have had more "luck" convincing the system owner to replace or modify the suction piping in an effort to reduce the friction component. You can increase the pipe size and possibly reduce the number of elbows, tees and other components in the suction system to minimize the friction.

Other Possibilities Outside of the Formula

If you cannot increase the NPSHa, perhaps you can reduce the NPSHr.

Look for different pump or impeller options that require less NPSH. It is not uncommon for a manufacturer to have different impellers for the same pump with different NPSH requirements. Some manufacturers will offer an inducer that works in conjunction with the impeller to reduce the NPSHr. Do not add an inducer without consulting with the manufacturer, because inducers must be matched to the impeller. Sometimes a different pump altogether is required.

Moving to a double suction impeller (two eyes) will have significant effect on the issue since the NPSHr will be reduced by 50 percent.

Reduce the pump speed either by incorporating variable speed or simply using a pump that will complete the service (flow [Q] and head [TH]) at a lower speed. The caveat is that the pump will likely be twice as big (physically) as the initial pump with an associated higher cost.

In many cases, the solution is to add a booster pump on the suction of the initial pump. In power plants and other steam systems, it is not uncommon to have a condensate pump that pumps to a feed booster pump before the liquid gets to the actual feed pump.

Materials

Sometimes there is nothing you can do to prevent the pump from cavitating, so your option is to treat the symptom in lieu of the problem. Different materials offer varying ranges of resistance to cavitation damage. Additionally, some materials offer better protection than others during the course of a phenomenon referred to as cavitation induced erosion-corrosion.

Cavitation damage resistance is defined as the reciprocal of the rate of volume loss for a given metal. The material's mechanical properties that are part of this equation are ultimate tensile strength, yield strength, ultimate elongation, Brinell hardness, modulus of elasticity and strain energy.

The most important property from this list is the fracture strain energy of the metals. It is for this reason that variations of aluminum bronze and duplex stainless steels offer better resistance than other materials such as regular carbon steel and iron. Note, as a post-OEM fix, there are also several coatings that can be applied. When using coatings, I recommend the decisive phrase and advice for the day be "caveat emptor," from the Latin for "buyer beware."

With coatings, there are good ones and bad ones and good ones applied poorly.

$$NPSHa = h_a - h_{vpa} + h_{st} - h_f$$

Where:

h_a = the absolute pressure. Absolute pressure as measured in feet of head of the liquid being pumped at the surface of the liquid. This will be barometric pressure if suction is from an open tank, or the absolute pressure existing in a closed tank such as a condenser hotwell or deaerator.

h_{vpa} = the vapor pressure. The head in feet corresponding to the vapor pressure of the liquid at the temperature being pumped.

h_{st} = the static head of the liquid over the pump centerline or impeller eye for a flooded suction in feet (positive value for flooded suction). Not all impeller centerlines correspond to the pump centerline.

h_f = the total friction loss in feet of head for the suction side system.

Equation 2

Proximity to Best Efficiency Point (BEP)

Look at where you are operating on the pump curve (head and flow). If too far to the right, there is a mismatch with the system and the pump. The NPSHr increases exponentially as you move right. Operating too far to the left on the curve can have similar issues. NPSHr actually increases as you approach areas of low and minimum flow rates. This is not published on most pump curves.

Suction Specific Speed (NSS)

Back in the 1970s, new plants or systems were designed with an ever-increasing strict mandate to save money (sometimes over reliability), especially on the initial construction and material costs. As a cost-cutting measure, the NPSHa of systems was reduced (think smaller and lower tanks and pumps at higher levels). The system owners/buyers subsequently placed increasing pressure on the pump manufacturers to design pumps with lower NPSH requirements.

The simplest and quickest solution for the pump manufacturers was to increase the size of the impeller eye. The good news was that the NPSHr was reduced, but the bad news was the hydraulic stability of the pump was also markedly reduced if and as the operating point departed from BEP.

Conclusion

No matter what, you will be involved in pump applications whether new or existing from some aspect where NPSH will be a factor. At least now you will know why impellers have big eyes, tanks have long legs and pumps hang out in low places. ■

By Jim Elsey

Best Practices for Strainer Location

A clogged strainer…on the suction side of a pump, will starve the pump, cause it to cavitate and eventually fail.

—Jim Elsey, 1979, in an engineering field service report at a large Midwest refinery

System designers frequently place strainers on the suction side of a pump. This practice, ostensibly based on good intentions, is rarely a good idea and will create serious issues for the pump from the aspect of reduced and turbulent flow, inadequate net positive suction head available (NPSHa) and eventually blocked flow.

The consequential reduction in NPSHa will create deleterious effects for the pump and the downstream system.

These effects will always manifest as negative and expensive issues. Due to vibrations from cavitation and poor seal face lubrication from mixed phase fluids, the mechanical seals, like canaries in the coal mine, are typically the first component to fail, followed shortly by the bearings. These events create reductions in efficiency and add to unscheduled downtime, which all add to the total cost of ownership.

There are several acceptable methods and alternative designs to avoid these issues. There are also situations where placing the strainer on the suction side is the right thing to do, but it must be done correctly.

Background

Normally the reasons for placing the strainer on the suction side of a pump seem like good ones, as it is prudent to protect the pump and the other ancillary components in the system.

This preventative design approach can work to preclude or mitigate fouling of heat exchangers, valve blockages and other component issues with small operating clearances and annulus voids, to name a few. In the case of some positive displacement pump types such as gear and screw pumps, it is imperative.

General Comments

There are several issues with placing the strainer in the suction line.

The first issue is that the strainer will inevitably clog. The clogging will reduce, and at some rate of closure, completely shut off flow to the pump suction, creating serious damage to the pump. The blockage also creates issues with the system and downstream components that rely on the flow provided by the pump. Example: another pump in series with the first pump.

The second issue is a marked reduction in NPSHa. The formula for NPSHa is shown below for reference. Every pump has a requirement for a given amount of suction energy that is referred to as net positive suction head required (NPSHr). The specific requirement is determined by the manufacturer and is published on their performance curves. The NPSHr data is determined by empirical means.

If you have read some of my previous articles, you already know that the suction side of the system must provide energy to deliver the liquid to the pump. The pump does not reach out and pull the fluid into the impeller.

The pump does not suck the fluid into itself, as fluids do not have tensile strength. The suction system must provide a level of NPSHa. The level of NPSHa is either calculated or measured by the owner or operator of the system. It is imperative that there is more NPSHa than NPSHr. This difference is referred to as the margin. The Hydraulic Institute (HI) and American National Standards Institute (ANSI) have a published standard that covers this subject—ANSI/HI 9.6.1-2012.

FLOW COEFFICIENT FORMULA

Strainer manufacturers typically rate the strainer by size and $\mathbf{C_V}$ number.
$\mathbf{C_V}$ is the Flow Coefficient. It is a value based on testing and empirical data.
A simple definition, $\mathbf{C_V}$ …is equal to the number of gallons of water that can flow through the strainer with a 1 PSI pressure differential.
The formula for $\mathbf{C_V}$ can be helpful in many ways to better understand the flowrate through the strainer and subsequent effects on the differential pressure.

- **GPM** is the flowrate in gallons per minute.
- **S** is the Specific Gravity
- **DP** is the differential; pressure in psi.

$$C_V = GPM\sqrt{S/(DP)} \qquad GPM = C_V\sqrt{(DP)/S}$$

$$(DP) = (S)(GPM / C_V)^2$$

By Jim Elsey

STRAINERS & FILTERS

One person's strainer may actually be another's filter. Some people use the terms interchangeably, but typically the term "strainer" is actually a subset of "filter." I have used the word strainer throughout the article, but note it could also be a filter.

A strainer removes solids from a solution using a purely mechanical and particle size-based sieving process. If the particle is larger than the size of the holes, it will not pass. If the particle is smaller than the holes, it will pass.

Filter is really a general term. It describes a device that removes solids from a fluid that is passing through the filter media. The solid could be removed by straining, but it can also be removed by other methods.

Technically, some people define a filter at less than 75 microns and a strainer at 75 microns and larger. In my experience, the colloquial expression is to use the word "strainer" if the device has a metal screen, regardless of the mesh size. Otherwise, the word "filter" is used if the device has bags or cartridges, regardless of the micron rating. You may have other definitions in your facility.

The issue with insufficient margin is cavitation and its subsequent deleterious effects, which include material damage (usually on the impeller), vibration, mechanical seal and bearing damage, loss of efficiency and loss of flow or partial flow.

The formula for NPSHa

Remember that we are calculating NPSHa, so we do not need to include velocity head. Velocity head would be included if we were measuring NPSHa.

$$NPSHa = h_a - h_{vpa} + \text{ or } - h_{st} - h_f$$

Where:

h_a = the absolute pressure. Absolute pressure as measured in feet of head of the liquid being pumped at the surface of the liquid.

h_{vpa} = the vapor pressure. The head in feet corresponding to the vapor pressure of the liquid at the temperature being pumped.

h_{st} = the static head of the liquid over the pump centerline for a flooded suction in feet (positive value for flooded suction)

h_{st} = the static head of the liquid below the pump centerline for a lift situation in feet (negative value for lift situations)

h_f = the total friction loss in feet of head for the suction side system

Equation 1

Issues with a Reduced Margin

The friction component of the NPSHa formula becomes increasingly large when the strainer becomes clogged. This is a negative component in the formula and as a result the reduction in available NPSH will cause the pump to cavitate.

If you look deeper into the cavitation phenomena, the reduction in flow caused by the restricted strainer will result in a higher velocity across the device. The fluid in the suction line must obey the law of conservation of energy, which states energy can neither be created nor destroyed, but it can be altered in form. This is best summarized in Bernoulli's equation.

Bernoulli's Law, simplified for this article, explains why the pressure will drop correspondingly as the velocity increases. The clogged strainer is why the velocity has increased in the first place. As the pressure drops, the vapor pressure will also be affected, and it is possible at some point for the liquid to change state and form a vapor. Centrifugal pumps are not capable of pumping air, vapor or non-condensable gases. If there is as little as 4 percent entrainment in the liquid, it can bind (vapor lock) the pump.

The higher the temperature of the liquid, the higher the possibility for this phenomena to occur.

I have covered details of these phenomena in four other articles in *Pumps & Systems* magazine.

- How to Reduce or Eliminate Air Entrainment (December 2017)
- Most Common Reasons for Air Entrainment in Pump Systems (December 2017)
- Guidelines for Submergence & Air Entrainment (April 2016)
- 10 Common Self Priming Pump Issues (September 2015)

Oil & Gas Applications

In oil and gas applications where the pump is upstream at the well head, there is simply no way to avoid the introduction of solid containments and dual phase fluids to the pump suction.

It is extremely rare that a fluid coming out of the ground will be 100 percent gas free. A centrifugal pump cannot pump (compress) air or gas.

The comparative difference in the range of fluid densities is a factor of approximately 800. Different designs and sizes of pumps handle this issue in better ways than others, but at some level all centrifugal pumps will vapor lock and fail. The impeller eye will become blocked by the air, vapor or gas.

Lift Situations

If the pump is a self-primer or simply a centrifugal pump placed in a lift condition application (externally primed), there can or will be performance issues should any restriction be placed in the suction line.

Note that "lift" signifies that the level of the source liquid to be pumped is below the centerline of the pump impeller. The available NPSH will already be low because the static head component in the formula is now a negative quantity due to the lift condition. Further restrictions will add to the negative component of friction and for any temperature above ambient the vapor pressure component will also work against the pump.

"OK, I understand you, but I need to have strainers on the suction side."

Assume you must have strainers on the suction side of the pump. Now what?

If there are strainers on the suction side of the pump, the best step you can take is to add instrumentation and continuously monitor the differential pressure (DP) across the strainer. The strainer will have a resistance coefficient assigned by the manufacturer. This is a great place to use an automated alarm system. The DP across the strainer for both clean and dirty conditions should be known. There must be a low value of DP for the new and clean strainer; I prefer less than 2 pounds per square inch gauge (psig). Also, you can compare the low clean value of DP to a higher value for a clogged strainer as indication for action.

Note: *You have to know the head loss across the strainer anyway to do the NPSHa calculation. How else would you know if there is sufficient margin in your design?*

If the DP across the strainer is not automatically monitored and alarmed, then an operator must check on a scheduled basis. All changes must be recorded and action must be taken if the DP is out of specification. Even a difference of 1 or 2 psig DP can be the difference between success and failure.

I suggest using a duplex gauge rather than two separate gauges, due to differences in system losses and gauge inaccuracies. A differential pressure transducer is normally better than a duplex gauge.

Some of the better designs incorporate a duplex strainer arrangement so that one strainer can be offline for cleaning and maintenance with no disruption in service. Some designs automatically change over and clean with no operator action required.

The size of the suction pipe must have an adequate diameter to keep friction losses down and velocities in an acceptable range. I always recommend to keep liquid velocities on the suction side below 2 meters per second (6.6 feet per second) at the maximum, and one meter per second (3.3 feet per second) is better. Just as the suction pipe size must be adequate, so must the strainer size.

Strainers should be engineered and selected to keep the pressure drop down to an acceptable low and safe level. The industry rates strainers using "CV"—known as the flow coefficient. When selecting strainers, I suggest you work with a knowledgeable expert, because the selection process can be tricky for the uninitiated. Most strainer resistance coefficients are based on water and a given mesh size for the strainer screen. You may need to correct for viscosity and a different mesh depending on your fluid properties. You will also need to decide the capacity ratio known as open area ratio (OAR). The OAR will indicate how long you can operate the strainer before it will require cleaning or replacement.

In a good system design, the pump should be positioned close to the suction source, but there should also be at least 5 to 10 pipe diameters worth of straight, unobstructed piping connecting to the pump. Never connect any components such as an elbow, reducer, valve or strainer within the final run of pipework. A clogged strainer will present a turbulent flow profile to the pump suction.

If you connect an elbow directly to the pump flange, the fluid will be forced toward the outside of the elbow and will not be directed into the center of the impeller. Most pumps are designed for fluid to be evenly loaded at the center of the impeller. Otherwise, the imbalance creates stress on the pump's bearings and seals that leads to wear and premature failure.

Acquiesce on Strainers & Filters on Suction Side

It is noted and acknowledged that many processes are purposely designed to have strainers on the suction side of the pump. Some processes actually rely on the reduced pressure (vacuum) created in the process. An example would be a filter press. Note these systems and pumps are designed and instrumented for the additional stress.

Conclusion

Every rule usually has exceptions, but from the perspective of an industry best practice, I suggest thinking long and hard before the placement of strainers on the suction side of the pump—and you "exceptions" already know who you are.

Many pumps are designed to handle some amount and size of solids. The manufacturer can or will advise what size solids the pump will handle. It is typically a better idea to place the strainer on the discharge side of the pump if possible. If you must place the strainer on the suction side for a valid design reason, then take the proper steps to design the system to prevent pump issues caused by clogging.

For more than 48 years, I have watched in amazement as operators destroyed their pumps by starvation. ■