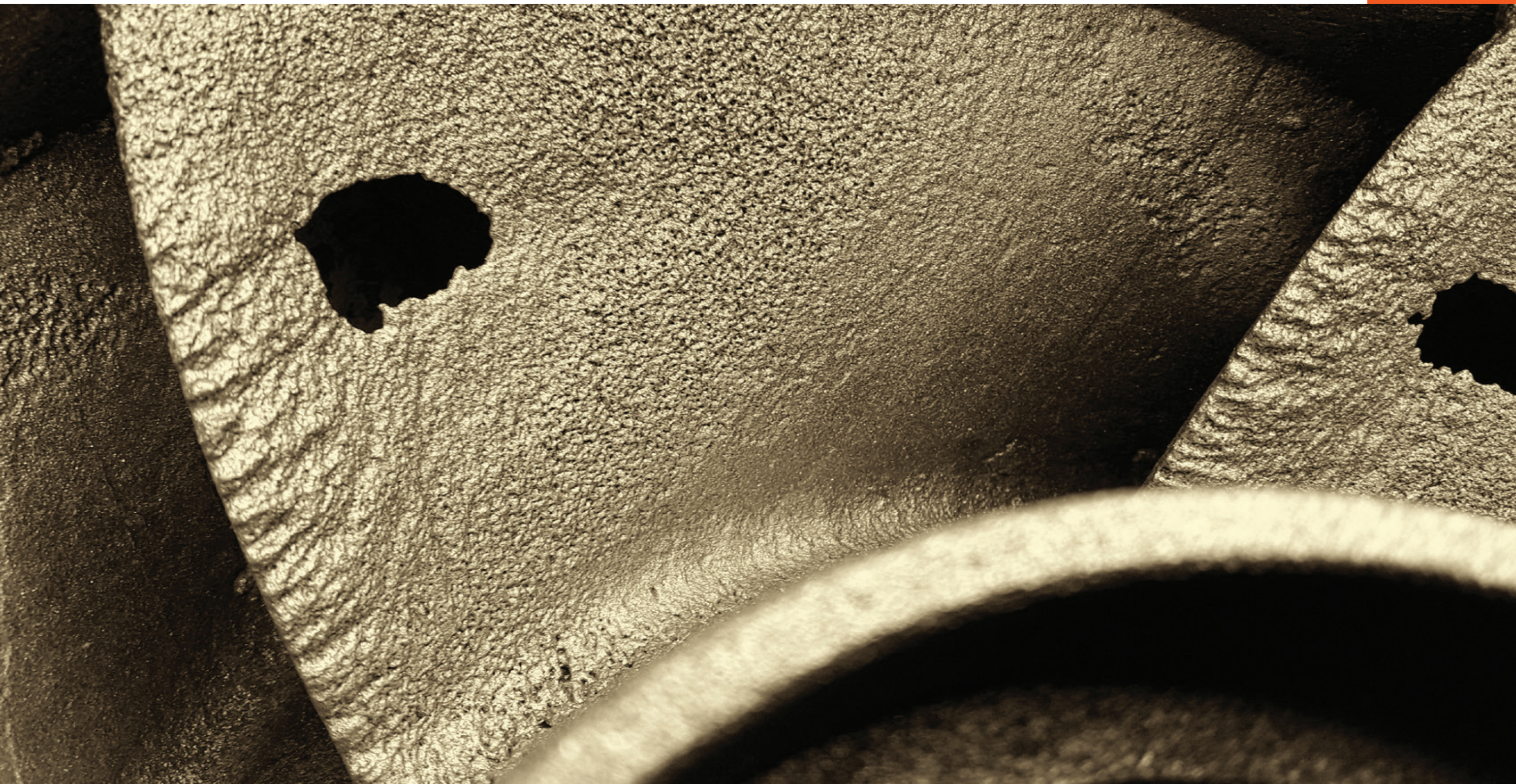
PUMPS&SYSTEMS®

e-book

BEP & CAVITATION: UNDERSTAND THESE CRITICAL PUMP BASICS

Pumps & Systems columnists Jim Elsey and Amin Almasi explain why cavitation occurs, how best efficiency point plays a part, and what to do to avoid damage caused by this problem.

Articles by Jim Elsey and Amin Almasi, originally published in Pumps & Systems



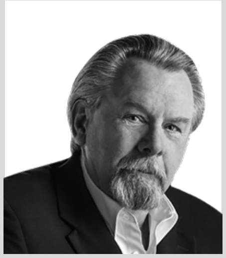
BEP & CAVITATION: UNDERSTAND THESE CRITICAL PUMP BASICS

Pumps & Systems columnists Jim Elsey and Amin Almasi explain why cavitation occurs, how best efficiency point plays a part, and what to do to avoid damage caused by this problem.

Contents

Popular columnist Jim Elsey wrote a column for the November 2021 issue of Pumps & Systems on "how to mitigate the cavitation blues." And he's not exaggerating—damage caused by cavitation can make any user want to sing "woe is me." In this ebook, we have collected four articles from Jim Elsey, and two from columnist Amin Almasi, that explain the basics of BEP and cavitation, how they relate, how to understand net positive suction head (NPSH) and how to keep from facing pump problems that have you feeling blue.

JIM ELSEY
Summit Pump Inc.



Jim Elsey is a mechanical engineer with more than 50 years of design and operating experience, primarily focused on rotating equipment reliability in most industrial applications and markets worldwide. Elsey is GM of Summit Pumps and an active member of the American Society of Mechanical Engineers, the American Society of Metals, the National Association of Corrosion Engineers and the Naval Submarine League. Elsey is also the principal of MaDDog Pump Consultants LLC. He may be reached at jim.elsey@spiohio.com.

Amin Almasi
Principal Machinery/
Mechanical Consultant

Amin Almasi is a principal machinery/mechanical consultant in Australia. He is a chartered professional engineer of Engineers Australia (MIEAust CPEng–Mechanical) and IMechE (CEng MIMechE). He holds a Bachelor of Science and Master of Science in mechanical engineering and is a RPEQ (Registered Professional Engineer in Queensland). He has authored more than 200 papers and articles dealing with pumps, rotating equipment, mechanical equipment, condition monitoring and reliability.

Why Cavitation Occurs & Ways to Treat It

By **Jim Elsey** | Summit Pump Inc.

The old joke about falling from a tall building is that "it isn't the fall that kills you, it is the sudden stop at the bottom." When it comes to cavitation, it is not the formation of the vapor bubbles that kills the pump, it is the subsequent collapse.

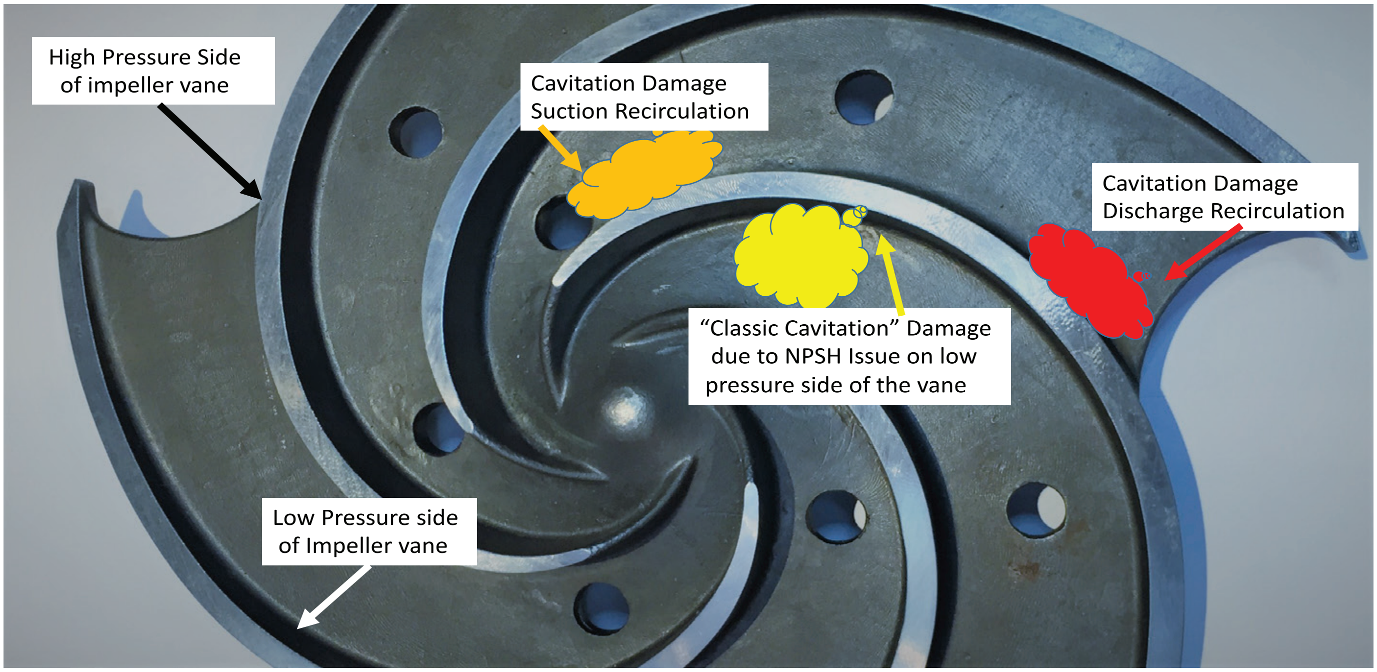
There are plenty of articles on net positive suction head (NPSH) and cavitation that talk about the bubble formation and the consequential pump damage in a broad sense, but the details of the damage mechanism are rarely discussed. The focus of this article is looking into why are we so worried about a few bubbles.

Cavitation causes an increase in pump noise and vibration, but more importantly, a drop in performance, efficiency and impeller erosion. Not all of the damage from cavitation is metal loss or metal damage. Sometimes the issue is shortened bearing and mechanical seal life due to the unsteady flows (surging).

Simply defined, "classic cavitation" from the perspective of centrifugal pumps is the formation of bubbles in the pump inlet near the eye of the impeller. The bubbles form because local pressure has dropped below the vapor pressure of the fluid (another way to view this is that the NPSH margin is not sufficient). Less than a fractional second later as the bubbles transit along the low pressure side of the impeller vanes, they enter a region of higher pressure and collapse.

I refer to this as classic cavitation to differentiate it from other causes of cavitation, such as suction or discharge recirculation that manifests on the other side of the impeller vane. Recirculation cavitation is typically due to operating the pump to the left side of the pump operating curve (reduced flows) and away from the best efficiency point (BEP). The approach angle of the incoming flow does not match that of the rotating impeller inlet vane geometry. Consequently, eddy currents and turbulence are generated in between the vanes.

Inside the general area of the eddy current, the velocity increases and the pressure decreases as a result. This action occurs due to the laws of conservation of energy as explained by Bernoulli's equation and the pressure-velocity relationship. When the local pressure drops below the vapor pressure, the cavitation bubbles are formed. Recirculation cavitation is typically not caused by insufficient NPSH in the classic sense. You could have more than adequate NPSH margin and still experience



recirculation cavitation because the pump is being operated away from its BEP.

When this situation occurs, there is a mismatch in the flow angle as compared to the impeller inlet incidence angle. The higher the suction specific speed (NSS) of the pump impeller, the more likely recirculation cavitation is an issue.

For more information on the subject of NSS, see my Pumps & Systems February 2019 article.

Cavitation bubbles that break down in the middle of the impeller passageway collapse symmetrically (equally from all directions), so there is less cause for concern other than potential noise and perhaps some vibration. Similar (but different) to boiling water in an open pan on a stove, the bubble forms at the bottom of the pan, rises to the surface and collapses without issue or harmful effects (technically this is a burst and not a collapse so almost no energy is released).

However, when the vapor bubbles in a pump impeller collapse adjacent to the metal surface of the vane, there is a much higher potential for damage and concern due to metal loss from the substrate. When the bubble collapses near the vane surface, it will collapse asymmetrically. Because of its proximity to the vane surface, the bubble geometry changes and makes the action more lethal. When the bubble collapses, it is not just the surrounding fluid that rushes in to fill that void, it is more importantly that the vapor is changing state from a vapor (back) to a liquid.

I repeat for emphasis that the amount of energy transferred for a change of state is very high. You can calculate the energy using enthalpy equations. Additionally, the collapse of a vapor bubble is exponentially more impactful than if it was an air bubble. With vapor bubbles there is a change of state from liquid to vapor and back, while an air bubble creation or dissipation does not involve a change of state. Further, when the vapor bubble collapses asymmetrically, there is a resulting reentrant microjet burst that on a local, nanoscale level is powerful (the local scale is 1 x 10-9, that is 10 to the negative nine exponent or a billionth).

Local pressure forces involved in the microjet burst can have resultant shockwaves higher than 10,000 pounds per square inch gauge (psig). The bubble collapse phenomena can occur with a high periodicity of 300 times per second and all of this action happens at the speed of sound. The resultant microburst jet almost always directs at the adjacent surface in lieu of the fluid stream. The vane material substrate is subjected to a localized surface fatigue failure. The average lifespan of a vapor bubble from creation to collapse is about 2 to 3 milliseconds. Not everyone agrees if it is the shockwave or the reentrant micro jet burst that creates the damage. Likely, it is the combination.

Hopefully, with this perspective, you begin to understand how cavitation can damage an impeller in short order.

On a scientific level, besides the enthalpy equation mentioned earlier, the energy of the bubble collapse is simply a kinetic energy calculation and is a function of the mass and velocity.

kinetic energy = 0.5 x mv2

Where: m is the mass and v is the velocity

Equation 1

Note that vapor bubbles formed in water at ambient temperature are of a much larger size (mass) than if the water temperature was close to and approaching 200 F. The larger the bubble, the more energy and damage. Therefore, cold water cavitation is much more dangerous than hot water cavitation.

The root cause for vapor bubble evolution is often overlooked. Pumps do not so much generate heat to make the water flash to vapor, but instead it is a result of the drop in pressure near the impeller eye. Remember you can boil water at 33 F if you reduce the pressure low enough.

There is some correlation of cavitation noise (intensity) to impeller damage. I am not presently aware of a conclusive formula or method for accurate determination. I am aware that several people are conducting studies in this subject area. Noise level for cavitation falls in the general range of 10 kilohertz (kHz) to 120 kHz. The general accepted range of hearing for humans is only 20 Hz to 20 kHz. Perhaps I will devote a future article to acoustic detection of cavitation. If you hear cavitation noise, the pump is likely cavitating, but just because you do not hear cavitation noise does not mean it is not cavitating. Some of the most damaging cavitation occurs at noise levels outside the audible range. I also witness many people confusing cavitation noise with turbulent or high velocity flow noises.

Sometimes you just cannot have a cavitation-free system, and you may wish to treat the symptom in lieu of the problem. With all of this energy being dissipated near the surface of the impeller vane, it is important to note that all impeller materials react differently to the exerted force.

For impellers, 300 series stainless is better than cast iron. Higher chrome content steels are better yet, while CD4MCu (duplex alloy) is better than high chrome stainless. There is good information and engineering studies completed in this area. Your empirical results may differ.

Finally, note that even with high NPSH margins, where the NPSH available far exceeds the NPSH required, the pump may still experience some cavitation. It is nearly an impossible task to reduce it to zero. ■

Don’t Overlook This Basic Advice, Part 1

By **Jim Elsey** | Summit Pump Inc.

Note from the author: A mistake was published in the May 2021 column. Here is the corrected text from under “Why Would You Want a Variable Speed Pump?”: “The most common reason to apply a VSD is that the pump is improperly sized for some or all of the system requirements. So, in some cases, the pump output is bypassed in some manner to keep the pump operation point away from the left end of the curve that is fraught with high radial thrust and other recirculation related issues. Other times, the opposite condition is true, and the pump requires some type of throttling device to keep it away from the right side of the curve where cavitation and radial thrust are an issue. Throttling or bypassing flow from the pump is inefficient and manifests as wasted energy.”

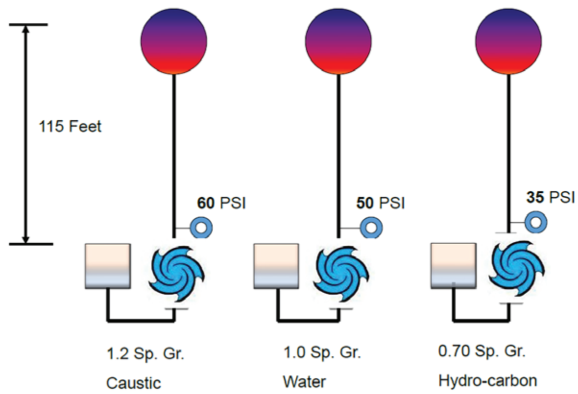


IMAGE 1: The relationship between head and pressure (Images courtesy of the author)

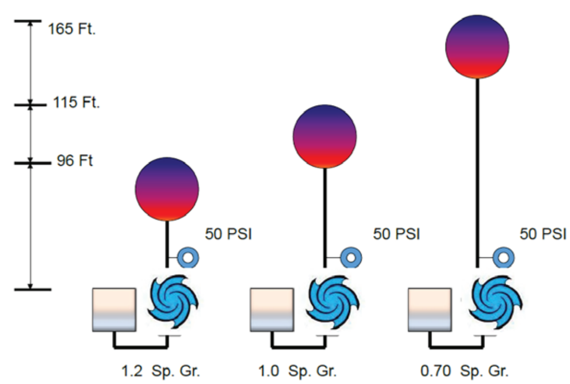


IMAGE 2: The relationship between head and pressure

This month I am approaching the subject of basic pump theory and operational principles from the perspective of an inexperienced person. Based on my 50 years in the pump business, I thought this might be a good way to explain the potholes along the road to pump reliability. If you have read any of my columns, many of these comments will be familiar.

The System Comes First; The Pump Is Second

My advice is to first design the system to meet the needs of the process, and then select the pump that best fits the system. It is the system that tells the pump what to do, not the other way around. You can’t purchase a pump for some flow rate X and head/pressure Y and then expect that the pump will perform to those parameters (refer to my August 2019 column where I explain that “Wishin and Hopin” will not get the pump to work correctly). This column will not instruct you on how to design the system. I will state that almost all pump problems occur on the suction side of the pump (I estimate 80%). This is mostly due to a common misunderstanding that pumps will “suck” the liquid into the pump—they do not. The suction portion of the system must supply the required energy to move the liquid to the pump; this is typically accomplished by gravity or atmospheric pressure.

If you are troubleshooting an existing pump that continues to present problems, the pump is likely misapplied. It is easier to blame the pump manufacturer, but it will not solve the issue. Review the system design specifications and review the pump capabilities. The “bad actor” pump may not be sized properly for the application.

First Things First

Before we get to the actual liquid being pumped, let’s discuss why most pumps are rated in units of head (feet or meters) instead of pressure (pounds per square inch [psi], kilopascal [kPa] or barg—a unit of gauge pressure). The simplest way I can explain this is that the centrifugal pump performance is predictable, measurable and consistent when rated in head regardless of the fluid, the density and the associated temperature (assumes Newtonian nonviscous liquids). If you are pumping clean water at 65 F (18 C), then it would not be an issue. But when water temperature changes, so does the density and so does the performance (pressure).

As an example: a pump is moving ambient temperature water at a pressure of

50 psi, so the corresponding head would be 115 feet. If the liquid was a hydrocarbon like diesel fuel (specific gravity [SG] = 0.70), the corresponding pressure would be 35 psi. If the liquid was a caustic solution like sodium hydroxide with a SG of 1.2, the pressure would be 60 psi. Regardless of the three different fluids and the different pressures generated by the same pump, the head remains the same at 115 feet.

Head and pressure can be used interchangeably if they are expressed in the proper units. This relationship is shown in Equation 1 and in Images 1 and 2.

$$(\text{psi})\ (2.31) = (\text{SG})\ (\text{feet})$$

Equation 1

The Liquid to Be Pumped

Liquid personality

Many pump problems are created because someone in the selection process thought all liquids pump the same. During the course of problem-solving, I always ask the users what the liquid is and its physical properties. If all you are going to do is pump clean water at ambient temperatures, then life is good. Otherwise, be aware that pumping any liquid other than clean water at 65 to 70 F (18 to 21 C) may require a modified pump, a different pump or even a different type of system. For example:

Solids

If solids are present in the liquid stream, a standard pump with an enclosed impeller may not work. An impeller (and associated pump) must be selected that can pass solids without clogging the vanes, and it should be of a geometric design and construction materials that mitigate the exponential wear that will come with the entrained solids. If solids are present, consider a more robust or open impeller design, a recessed impeller pump, or a slurry pump. I am keeping this column in the centrifugal pump world, but as an exception, you may want to also consider a progressive cavity pump or some other type of positive displacement pump.

If the pump will handle suspended solids, be prudent in the material selection. The rheology of slurry applications can be overwhelming, and I suggest that you investigate harder materials such as high chrome iron or materials that work harden like CD4MCu. Depending on the type and size of solids in the slurry, you may also consider rubber-lined pumps.

pH levels

If the liquid is an acidic solution with associated low pH levels, then standard pump materials will probably not hold up. Corrosion is always your enemy, but the acidic solutions, especially when accompanied by higher temperatures, will exacerbate and accelerate the destructive processes. Consult with the manufacturer or a knowledgeable materials person to match the materials to the liquid properties.

Acid solutions will normally require higher noble metals, and the more aggressive the application, the higher up the noble scale (and cost) you will need to go. A 300 series austenitic stainless steel is a good start, but check compatibility because the application may require alloy 20, Hastelloy, Monel or titanium.

Further, the liquid may require a nonmetallic pump. The solution may be a fluoroplastic, like a perfluoroalkoxy/polytetrafluoroethylene (PFA/PTFE) lined, nonmetallic and/or mag drive pump. Sometimes a nonmetallic or mag drive pump can be less expensive than a high alloy metal pump with an associated mechanical seal and support system.

Density and specific gravity

Density is the mass of a liquid in a specified unit of volume, for example, pounds per cubic foot. Specific gravity (SG) is a ratio—the density of the liquid relative to that of water. SG may not be an issue for the pump per se but the associated drive motor will notice. Refer to the brake horsepower (BHP) equation for pumps (Equation 2) and you will see that the liquid's SG will have a direct effect on the power required.

$$\text{BHP} = \frac{\text{Head x Flow x Specific Gravity}}{\text{3960 x efficiency}}$$

Equation 2

As an example, calculate the BHP required for a pump that is 75% efficient when moving 500 gallons per minute (gpm) at 160 feet of head. First, calculate the BHP with a liquid at SG 1.0 and then change the SG to 1.3. The difference in SG will change the BHP from 27 to 35. If you had a 30-horsepower (hp) motor to drive the pump it would be operating in the service factor, on overload or tripping the breaker.

Kryptonite for Pumps: aka, Viscosity

Viscosity is the kryptonite of centrifugal pumps. In the lower viscosity ranges of 1 to 100 centipoise (cP), there are some noticeable and negative effects on pump performance, but at higher viscosities, the pump performance will deteriorate markedly. Pump performance curves are based on water, and if the fluid to be pumped is more viscous than water, the performance must be corrected. Consult with the manufacturer to get this information.

The main negative effect of increased viscosity is the pump efficiency, but the flow and head are also marginalized. At 30 to 40 cP or greater, viscosity corrections are needed or you risk adverse performance effects. In the area of 5 to 10 cP, you must at least be aware of the effects, however minor.

The decrease in pump efficiency and the viscosity corrections needed to attain a water-based condition point for the desired head and flow rate all combine to require more horsepower. Consequently, the driver (motor) will need to be bigger. However, the pump power frame may not be able to handle the additional horsepower and torque requirements.

All pumps have a shaft and bearing frame BHP limitation, usually expressed in a maximum BHP per 100 rotation per minute (rpm) format.

If the liquid viscosity for your application is approaching 2,000 cP and you are still considering a centrifugal pump, reconsider and look at a positive displacement pump. ■

Don't Overlook This Basic Advice, Part 2

By **Jim Elsey** | Summit Pump Inc.

Last month, we discussed that the system (not the pump) dictates where the pump will operate on its performance curve. We also discussed "liquid personality" (the properties of the liquid) such as specific gravity, suspended solids, pH and viscosity and the mostly negative effects on the pump and system.

If you have been around the pump world for more than a few days, I am sure you have heard the term best efficiency point (BEP). In essence, all centrifugal pumps are designed for just one operating point of flow and head on the curve. This one design point for flow X and head Y is commonly referred to as the BEP or best operating point (BOP). All other possible operating points are, to some varying degree, a counter compromise with efficiency, cavitation, radial thrust (shaft deflection) and recirculation issues. Ignoring these stress issues will shorten the life of the bearings and mechanical seals, making the pump less reliable and more costly to operate.

If time and money were not an issue, the pump OEM would be happy to design and build a pump specifically for the user's unique operating point. Yes, it does happen, but not very often.

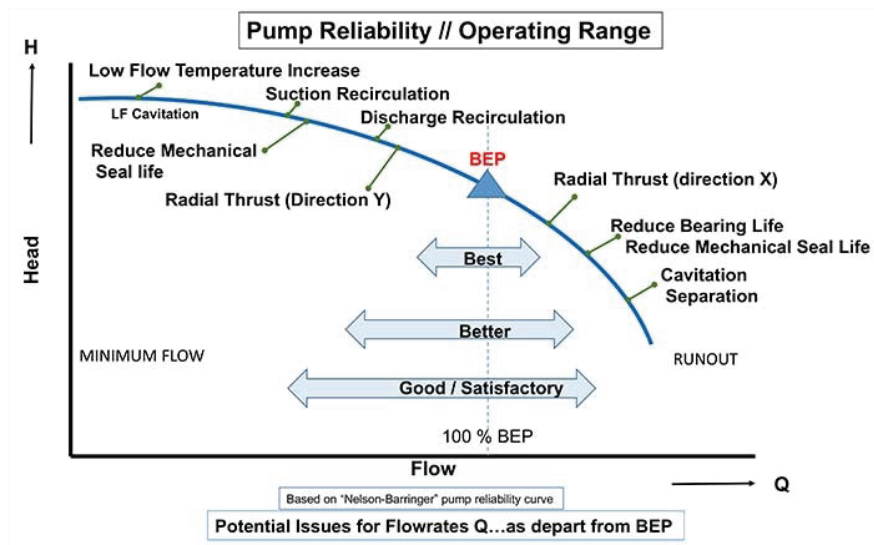


IMAGE 1: Pump reliability/operating range (Images courtesy of the author)

Allowable Operating Region

Of course, most end users don't have just one operating point—normally they want to operate in a wide area of the curve that is commonly referred to as a safe or allowable operating region (AOR). The presumption is that the end user knows where the pump is operating on its curve and fully understands that the pump will operate where the system curve compels it to perform. If you are experiencing pump failures, perhaps the problem is that the pump was selected incorrectly and/or the system curve was miscalculated?

Assuming the pump selection was the best choice compromise for the application, and since many pump applications require operation away from the design area of BEP, there are methods to manage the negative effects. All of the mitigation methods are burdened with the added cost of pump efficiency reduction, but that increased cost may often be an acceptable trade-off for reliability and reduced maintenance costs.

You can explore the numerous methods to reduce or eliminate the negative effects with your pump salesperson, technician/engineer or a knowledgeable systems design person. If you have no means to determine the differential pressure across the operating pump, such as a set of simple pressure gauges or transducers, then your first check box on the road to pump reliability will be to install a set (one on the suction side and one on the discharge side) and then calculate where the pump is operating on the curve.

What's the Big Deal With Operating the Pump Away From BEP?

The simple answer is that if you run too far right—that is, at or near the end of the curve—the pump will cavitate and the result will be high vibration levels that will damage the mechanical seal and bearings in quick fashion. The impeller may also suffer cavitation damage that is dependent on several variables not covered in this column.

As I often state in my role as the master of the obvious during my pump training classes, "the end of the curve ...is the end of the curve." If pump manufacturers thought you could or should operate there, they would extend the curve. Everyone wants more coverage, but the laws of physics keep getting in the way.

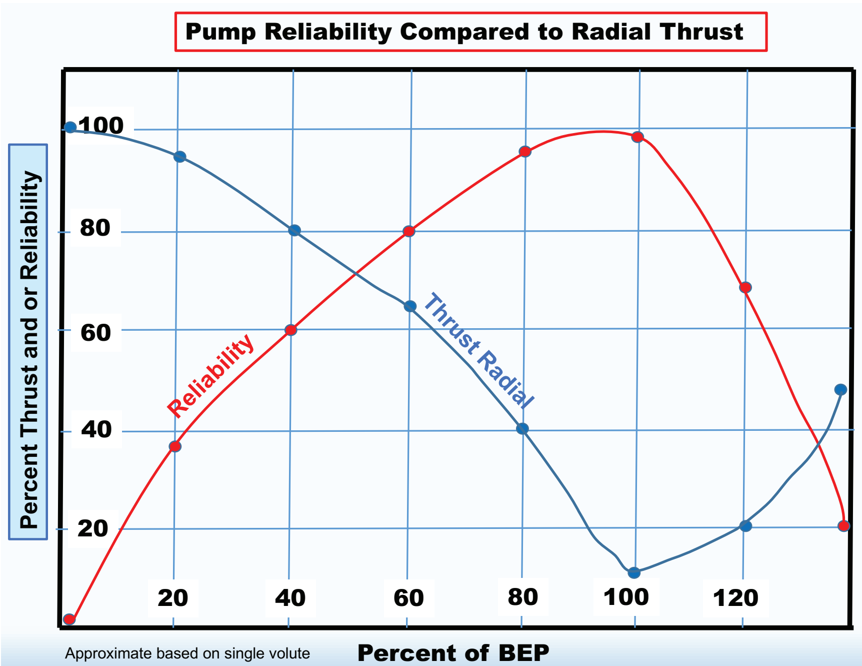


IMAGE 2: Pump reliability compared to radial thrust

Another important factor to consider near the end of the curve is the radial thrust, which will increase exponentially as you depart from BEP and move toward runout. Depending on the shaft rigidity factor (L3/D4 ratio or think robustness factor), the shaft may deflect some amount. Shaft deflection is a dynamic bending of the shaft while in motion that occurs two times per revolution. Understand that a shaft rotating at 3,550 rotations per minute (rpm) will have 7,100 deflections per minute. Shaft deflection will damage mechanical seals and bearings. More importantly, excessive deflection can often lead to shaft cyclic stress fatigue and breakage. Also, be aware that the shaft would measure perfectly straight if you stopped and removed it from the pump. As I mentioned before, deflection is a bending phenomenon that may occur during operation. If the shaft is already bent and/or the impeller is out of balance, the situation is critically exacerbated. For more details, see my January 2021 column on radial thrust and my February 2017 column on shaft breakage.

Operating the pump to the left side of the curve also has negative consequences. When we state operating to the "left," we mean operations between the BEP and shutoff (sometimes just abbreviated as SO). Shutoff is the point of no (zero) flow such as closing the discharge valve or a blocked system component. Operating near shutoff will also increase the radial thrust and deflect the shaft—the same phenomenon we discussed above at the far right side runout condition and with the same penalties. The only difference from the radial thrust experienced at runout (right) when compared to the left side of the curve is the thrust is now applied from the opposite side (180 degrees of opposition). The negative effects are the same.

A Note About MCSF

Note that for a given size pump, if one manufacturer states its minimum continuous stable flow is lower than another manufacturer, this does not mean it is necessarily a better pump. It may just mean the manufacturer is more conservative in its approach to reliability.

Most pump manufacturers will advise you where the recommended minimum flow point is on the left side of the curve. This is frequently referred to as the minimum continuous stable flow (or allowable flow). Minimum continuous stable flow (MCSF) is defined as that flow rate below which the pump should not be operated for any length of time. What is not defined is the amount of time, and I would suggest minimizing the time as much as possible. The higher the pump energy (brake horsepower [BHP]) the shorter the time.

One rudimentary way to think about this is to convert the driver's BHP to British thermal units (Btu), and then realize that much of that energy is working to heat the liquid in the pump casing while simultaneously the shaft is bending twice per revolution.

There are four factors to be calculated/evaluated when determining the acceptable minimum flow point. For this column, we will just examine the main two factors. The first is from a pump mechanical perspective: How much dynamic load from a radial and axial thrust perspective can the pump take? And the second is from a thermodynamic aspect: At what point does the liquid convert/flash to vapor? The pump OEM will determine a minimum continuous flow rate for the pump for both mechanical and thermal factors—the higher of the two will become the minimum flow rate for that pump. For more details, see my Pumps & Systems column from November 2015.

By Design, the System Must Supply the Liquid to the Pump

Stated another way, centrifugal pumps do not suck liquids. I mentioned in Part 1 of this series that most pump problems occur on the suction side of the system due to this misunderstanding about centrifugal pumps. "This is mostly due to a common misunderstanding that pumps will 'suck' the liquid into the pump—they do not. The suction portion of the system must supply the required energy to move the liquid to the pump; this is typically

Notes on Cavitation

1. You may think the liquid is not hot enough to form vapor bubbles and likely it is not at ambient pressure, but remember from science class that you can boil water at room temperature if you reduce the pressure sufficiently. You can boil water at 70 F if you reduce the pressure to 0.363 absolute pressure (psia).
2. Do not confuse vapor bubbles with air bubbles. The collapse of entrained air bubbles in the liquid stream does little damage to the impeller. It is possible to air-bind the pump where the air bubbles block the flow of liquid. Vapor bubbles, on the other hand, possess high energy and can severely damage an impeller in a short period of time. Actual damage will vary with the energy level, liquid properties and impeller materials.
3. Just because you have adequate NPSH margin does not mean that there will be no cavitation or damage. Even with high margins, there may still be cavitation occurring, albeit probably not of a very destructive nature.
4. You may also see what appears as cavitation damage on the convex side (aka, the working side or high-pressure side) of the impeller vanes and this is typically due to recirculation cavitation. This damage is not due to insufficient NPSH margin, but to operating the pump in a nonstable region between BEP and approaching shutoff. The actual operating point (area) where this will occur is a function of the impeller's suction specific speed (Nss). Nss in its simplest form is an expression of the design geometry for the suction side of the impeller. Consider the number of vanes, inlet vane angle, curvature and pitch of the vane, the amount of vane overlap and effective impeller eye diameter, to name a few factors.

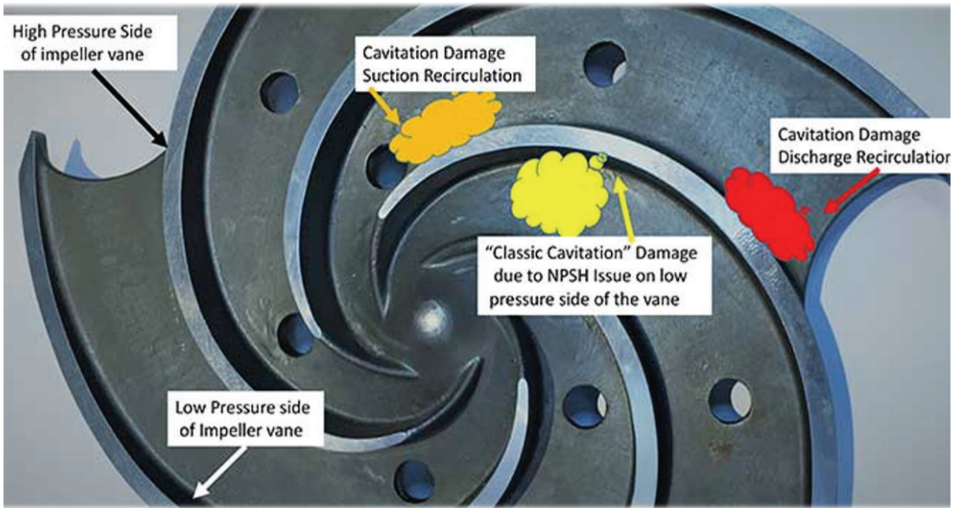


IMAGE 3: Examples of cavitation damage

accomplished by gravity or atmospheric pressure." (Some external source other than the pump.)

Note from a technical perspective that the pump impeller does create a small differential pressure directly in front of the impeller, but that energy level is in no way sufficient to overcome gravity and the friction required to initiate and sustain the flow rate. Also, note that liquids do not possess tensile properties, and so the pump is not capable of pulling the fluid into itself.

One way to describe the energy required on the suction side of the system at the pump suction flange is called net positive suction head available (NPSHa). I understand this is a return-to-pump-basics column and many neophytes will dismiss and ignore this uncomfortable subject for as long as possible. I would suggest that the sooner you can wrap your head around the subject, the better off you will be.

For reference and assistance, please refer to a detailed series of five columns I authored on this subject starting in July 2018.

NPSH: A Primer

Yes, the pun is intended. The pump OEM/manufacturer will design and test its pump to determine net positive suction head required (NPSHr). That is, for several condition points on the pump curve there is a corresponding amount of energy required at the pump suction that must be satisfied by the system. It is the pump manufacturer's responsibility to conduct this test and report/publish the results in accordance with industry standards.

Conversely, for any given system design and corresponding flow rate, there will be an amount of NPSHa. It is the responsibility of the system designer to accurately determine/calculate the NPSHa. There must be more NPSHa than NPSHr, and the margin required will vary based on several circumstances not covered here but can be referenced in American National Standards Institute/Hydraulic Institute Standard 9.6.1 (2012).

If there is insufficient NPSH margin, the pump will cavitate and be short-lived. Cavitation is the formation of vapor bubbles in the liquid stream (normally just in front of the impeller) and then the subsequent collapse of those bubbles some distance along the impeller vane. Normally the bubbles collapse within the first 25% to 33% of the vane length. The bubbles will collapse on the underside (concave side) of the vane. ■

12 Ways to Mitigate the Cavitation Blues

By **Jim Elsey** | Summit Pump Inc.

Here I go again on the tedious subject of net positive suction head (NPSH). I know it seems like this is my favorite subject, but before you turn the page, NPSH is a critical subject that pops up daily in my work. (And for the record, anyone who knows me will tell you my favorite topic is maximizing small block Chevy engines.)

This column will address possible mitigation steps to correct an existing situation where the NPSH margin is not sufficient. My comments are based on pumping water.

There is good news and bad news. The good news is there are several potential ways to correct situations with inadequate NPSH issues. The bad news is that most mitigation measures will be cost prohibitive and objectional to the equipment owner. It is best to avoid the issue in the first place because the corrective solutions, if any, are typically painful.

NPSH Margin

The suction side of the pump system must somehow supply the energy required by the pump (above vapor pressure) to avoid cavitation issues. Contrary to urban myth, the pump is not capable of reaching out and pulling the liquid into the eye of the impeller. The energy will typically come from gravity (static head) or atmospheric pressure but may also be induced by an external pressure source.

It is incumbent on the system owner or their proxy to determine the amount of NPSH available (NPSHa). The pump

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

Equation 1

manufacturer will in turn test the pump and publish the amount of NPSH required (NPSHr). To avoid/mitigate cavitation issues, you must have more NPSHa than NPSHr. The ratio of NPSHa to NPHSr is referred to as the NPSH margin.

A general rule of thumb is to have the system provide a minimum of more than 3 to 5 feet available than the pump requires. Designing by this thumb rule will eventually land you in trouble, and I recommend you read and follow Hydraulic Institute 9.6.1 guidelines and standards for proper margin. Sometimes even a 30-foot margin (ratio 1.5) will not be enough. Also, realize that even with a high margin, you are not truly eliminating cavitation—you are just hopefully lowering it to a level that will not cause pump performance or material degradation issues.

Tip: If you are not already aware, please understand that the published NPSHr numbers for a pump (also referred to as NPSH3) can be slightly misleading to the uninitiated. In the test process for determining NPSHr, the operating pump is held at a constant flow rate and head and simultaneously forced to a point of cavitation by lowering the NPSHa. When the head drops by 3%, that NPSH point will be the published data point for NPSHr. That is, at conditions where NPSHa equals NPSHr the pump will cavitate. Or another way of stating it is the pump is already cavitating at the published point.

This is not some dirty little secret—the industry has been doing it this way since 1903 (formally adopted in 1932), but sometimes the OEMs need to be better at communicating to the end users. In defense of the pump manufacturers, there are no easy methods to accurately conduct these tests.

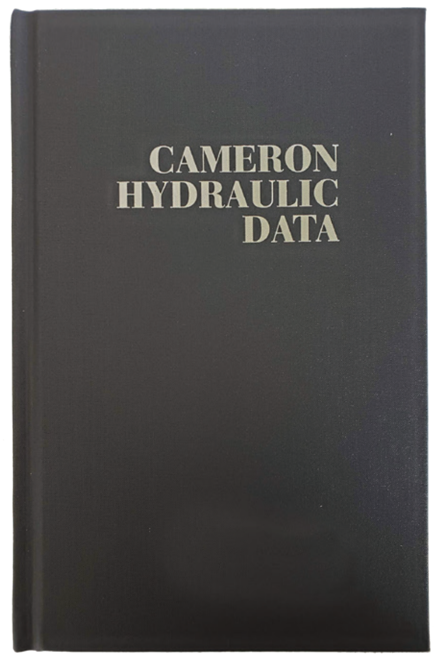


IMAGE 1: The "Cameron Hydraulic Data" book gives details on calculating NPSHa.

The Formula Is Your Friend

I learned this maxim initially in college and later it was tattooed into my brain during my U.S. Navy nuclear power training: "When in trouble…start with the formula to find the solution." And so, to this day I still tell my students that the "formula is your friend."

The easy formula for calculating NPSHa appears in Equation 2. One at a time, examine the factors in the formula. See my previous column in the October 2021 issue or consult "Cameron Hydraulic Data" book chapter 1 for detailed definitions and conditions for the individual terms. You should always calculate NPSHa for the worst conditions (think tank/sump levels and maximum flows at the highest temperatures). Also, look at and add for the

$$NPSHa = h_a - h_{vpa} +/- h_{st} - h_{fs}$$

Where:

- h_a = absolute pressure
- h_{vpa} = vapor pressure
- h_{st} = static head (positive if flooded; negative if a lift)
- h_{fs} = friction head
- Note: Not shown is velocity head (h_{vel}), which can be ignored in most cases. However, if you are measuring (empirical tests in the field) you should consider the factor and note it possesses a positive sign. If you are simply calculating NPSHa using the above formula, it is not necessary to include. If the suction piping system is undersized it will be a factor.

Equation 2

system age and fouling factors. All of the factors in the formula should be expressed as absolute.

An alternative method is to measure NPSHa in the field. I recommend using the formula specified in Hydraulic Institute 9.6.1. Note the difference between the calculation of NPSH and measurement of NPSH.

Absolute Pressure

This is the absolute pressure pushing down on the liquid source. If it is an open system, it is the atmospheric pressure (barometric) pushing down on the surface. Note that the atmospheric pressure changes with altitude above (or below) sea level, consequently the pressure must be adjusted for altitude and converted to absolute terms. If the system is closed, you will need to look at the pressure and convert to absolute head. If the system is in a vacuum, first realize there is still some pressure in a vacuum—it is just a pressure lower than atmospheric. Now, let's look at what we can do to change this first factor to our benefit in a situation with insufficient NPSHa.

In the event of an open system, chances are the owner is not going to change the weather or move the plant to an area of higher atmospheric pressure (lower elevation). There is not anything commercially feasible we can do about this situation. In the event of a closed system, there is a remote possibility that the pressure can be increased, but real-world issues and process requirements will normally preclude any changes. Nevertheless, ask the question and expect some negative critique. Closed hydronic systems can typically have the pressure increased slightly with little to no issues.

Vapor Pressure

Next up is vapor pressure. If you are not familiar, please review my April 2018 column, "Under Pressure: The Dangers of Vapor Pressure."

Vapor pressure is temperature-dependent for the liquid you are pumping. The higher the temperature, the higher the vapor pressure. Because the factor is always negative, vapor pressure is never your friend in the race for more NPSHa. To improve NPSHa, the mitigation step is to lower the liquid temperature if you can. Again, the system owner will fight you on this because in almost every case there is a good system design reason why the temperature is what it is. To lower the temperature in a process typically means the temperature will need to be raised (heat added) somewhere else in the system and the overall cycle efficiency will be reduced. In my 50 years in the business, I have only one example of a user acquiescing to lower the temperature of the process.

Static Head

As explained earlier, the static head factor will be positive if the suction is flooded and negative if the pump is on a lift.

In industrial and process plants, there is good reason you will see so many pumps with the suction source elevated and/or the pump location in a lower level or basement. The reason is to gain/maximize static head. In countless NPSHa calculations I have made, the static head is the only factor that allows the system to work correctly. Go to any plant that has a steam system and look for the condensate pumps—with the possible exception of packaged boilers, the condensate pumps will always be on the lowest level. The plant owners don't want the initial construction expense (elevated towers and excavations), but it is a smart compromise to increase equipment reliability and reduce the cost of operations and maintenance over the life of the plant.

On your mission to increase NPSHa on a system with flooded suction, you can increase the height of the supply source and/or lower the pump. Both options are expensive and disruptive to plant operations. In my career experience, I've had two instances where the pump owner accepted one of these changes.

Note: For suction lift situations, if possible/feasible you can set the sump levels higher to reduce the lift. My experience with this option usually reveals it is either not feasible or the gains are small.

Friction Head

The last factor in the NPSHa formula is friction. This is a factor that can sometimes be manipulated. In recent years I have witnessed clients making pipe changes on several occasions. Pipe changes, like the other factors, can be expensive and disruptive. The added gremlin that often surfaces in these cases is artificial financial restraints on the size of equipment and plant footprint. One sector of upper management will demand a small plant footprint to reduce cost, but often these decisions are made in the absence of a reliability champion with a long-term perspective on total cost of ownership (TCO).

Calculate the friction head and look at the suction velocity. Look for methods and means to reduce the friction and velocity. I have seen systems so poorly designed and constructed that we were able to easily correct the NPSHa issue by simply installing unobstructed straight pipe of the proper size. Yes, there was some downtime and cost associated with the project but in the overall picture, the return on investment (ROI) was realized in less than a year.

In industrial and process plants, there is good reason you will see many pumps with the suction source elevated and/or the pump location in a lower level or basement. The reason is to gain/maximize static head.

The Formula Was Fun, But My Mission Failed

As I predicted at the beginning of the column, most of the time the required corrections will be either cost prohibitive or not feasible. Now the question becomes, what else can I do? I offer the following list of possible mitigation methods.

1. Operate the pump in a region that requires less NPSHr. Not always possible, but I have also been surprised by the number of times the client was not aware the pump was operating too far right or left on the curve.

2. Very small amounts of air or gas introduced into the suction side of the system will mitigate the negative effects of the cavitation damage. The pump performance will be slightly diminished.

3. Be aware that 1% air entrainment is normally the maximum limit and amounts exceeding 1% will have a negative effect.

4. Investigate a different type of impeller. Many manufacturers will have alternate impeller offerings for a given pump model and size. Ask the manufacturer if they have an impeller that requires less NPSHr. Note that the optional impeller will most likely be of a higher suction specific speed (Nss) and may present recirculation issues in the operating areas between shutoff and best efficiency point (BEP). Some choices will and some will not, but you need to ask.

5. Investigate another pump model or different manufacturer. Either decision will at the very least involve piping changes and perhaps foundations, baseplates and electrical supply changes.

6. You can improve NPSHa and/or reduce NPSHr by eliminating items that protrude into the piping such as instrumentation and even the penetrations themselves. Also, look at changing the impeller leading edge vane profile and the surface finish of the pump, especially the impeller.

7. Change the speed of the pump to be slower, which will commonly play out as a change to a different pump. NPSHr varies approximately as the speed ratio squared. For example, if your current pump is operating at 3,550 rotations per minute (rpm), you can choose a pump that is operating at 1,750 rpm and expect the NPSHr to reduce by somewhere in the range of 50%.

The caveat will be that the new pump will be about twice the physical size of the original pump. The good news is that the added cost will eventually play out as reduced maintenance and longer mean time between failures (MTBF).

8. Switching to a different style pump/ impeller may help. Horizontal split case pumps have dual suction impellers that will reduce the NPSHr by approximately 50%. Some OEMs offer vertical pumps with dual suction first-stage impellers—a real winner on condensate systems because the pump can be extended (further down) into a pit/can to gain more static head and at the same time offer a 50% reduction in NPSHr with the dual eye impeller.

9. An often-overlooked remedy to fix NPSHr issues is to add a booster pump in series. Instead of trying to accomplish all of the hydraulic head requirements in one step, look at doing it in two steps. The booster pump could also be a slower speed to reduce NPHSr requirements even more. If space requirements preclude adding a booster pump, look at changing the base pump to a two-stage, which is in essence doing the same thing, just in one casing/volute.

10. An alternative subset of adding a booster and/or two-stage pump is to change to a pump that can incorporate an inducer. I warn end users not to add inducers on their own. Inducers must be matched to the impeller.

11. My next-to-last suggestion is to live with the cavitation but treat the symptom by changing the impeller material. I don't have room for details, but your pump manufacturer will be able to inform you of materials that resist cavitation damage better than others.

12. I know of at least a dozen clients that simply live with the results of insufficient NPSH. Their cost analysis and business decisions are that it is just simpler to live with the issue and replace impellers, mechanical seals and bearings on some regular maintenance schedule.

In the end, it is always simpler and less costly to first calculate and allow for a proper NPSH margin and then select the correct pump in the first place. ■

BEP: Parallel Pumps & Head Requirements

How best efficiency point affects sizing and operation.

By **Amin Almasi** | Principal Machinery/Mechanical Consultant

There are many reasons for such overwhelming interest on best efficiency point (BEP) and performance curves. BEP and desired operation at or around BEP is closely related to energy savings. Also, the operation at/around BEP is logically related to the pump selection and it links to some serious points such as the pump sizing, bid tabulation, manufacturer selection and tense discussions often happening between pump vendors, purchasers, consumers and operation teams at key stages of projects. BEP is also closely related to the operation, reliability and possible failures, repairs, maintenance, etc. BEP, particularly plausible deviation from BEP during parallel operation, is one of the main considerations for the performance and reliability related to parallel operation of pumps. Nearly all parties involved in a project deal with BEP and performance curves.

Operation Far From BEP

The internal flows of pumps are complex, and it is difficult to note what will happen for pumps when they are operated far from BEP. This article covers some general guidelines and approximate figures about BEP and its effect on the selection, sizing, operation and reliability of pumps. This is to help better assess dangers and risky operation.

Many engineers and experts agreed that pumps should be sized and selected to operate most of their life at or near BEP. There is no question about this key requirement. But, how can this be achieved? What factors should be considered to achieve this important goal? What might prevent or impair that ideal long-term operation at or near BEP? How will each party involved (engineers, consumers, purchasers, vendors, operators, etc) affect this requirement, and how each will be affected?

Possible Overestimation of Head

Users often overstate head requirements when they specify pumps. Too often, there are multiple engineers involved in the overall design and sizing process. Each one may add 5 to 10 percent factor to the head or capacity, and as a result, added margins to the head would be remarkable.

Overstating the head with a pump selected to operate at BEP finds the actual duty point moving right on the curve. In other words, as different safety factors and margins are added to the head by process, operation and mechanical engineers during the design, the selected pump might be bigger than needed and, in the operation, the pump might operate most of the time to the right of BEP.

In fact, the above-mentioned cases of the overestimation of head and the selection of the pump with more power ratings and head capability actually happened. However, there are two other factors that prevent such cases of gross overestimation of the head.

The first and most important limiting factor is tense competition of pump vendors and manufacturers over the cost reduction particularly at bid tabulation and selection stages. Although vendors and manufacturers use state-of-the-art software and programs to size/select pumps accurately, those software and methods seldom oversize the pumps or their drivers due to the pressure to keep costs down. In other words, the actual performance of pumps is usually at the predicted performance or slightly below it.

The author has been invited to different shop/site performance tests and at nearly all of them pumps were performed at theoretical predicted curves or below the curves. In most cases, the actual performance at the performance tests were just slightly below the predicted performance curves. This shows the accuracy of those so-called predictions or theoretical curves by reputable pump manufacturers. However, cases of over-performance were rare.

As pumps are selected based on lowest costs, pump manufacturers tune the overall sizing, selection and design process to bare minimum margins and factors on their side. Obviously pump manufacturers should not be blamed for this practice, because cost-based selection is imposed by end users. However, everybody should be aware of this process, the interaction and their consequences.

Another consideration is the actual system including piping and equipment in the process can impose more friction and head loss than what were initially predicted. In nearly all cases, the final/actual length and complexity of the piping are more than predicted during pumps selection. This leads to the fact that in real plants and facilities, more head is usually needed than the initial head estimate. This is not just for commissioning and startup, but virtually any degradation or deterioration in the system including pump itself, piping, downstream, etc., means there will be more friction and head loss as time passes and more power will be wasted. In other words, nearly all unpredicted events during different stages of detail design, installation, commissioning and operation leads to more head being needed in the system. Most often, those independent factors added by different people turn out to be required.

It is true that independent safety factors and margins added to the head by different engineers and representatives might theoretically lead to overestimation of the head and power, and purchasing of a larger pump than needed. However, there are other limiting factors that have prevented such an overestimation in many cases. There has been a limited number of pumps that were actually oversized. In real world cases, where cost-cutting and cost-based selection dominate, most pumps were slightly undersized, and there are more complaints about low power ratings and low heads in actual operation.

BEP, NPSH & Temporary Operation at Left/Right Sides of BEP

Every pump should be sized to work most of the time at or near BEP. However, it is wrong to assume every sizing parameters or factors, such as net positive suction head (NPSH) margin, should be checked and verified only at or near BEP. Just the opposite: Pumps might be pushed to operate temporarily at left or right side of BEP and everything should be ready, safe and reliable for such plausible operation.

Typically the NPSHr (required) curve begins to swing up quickly at BEP (or just left of BEP). If the only NPSH margin (NSPHa – NPSHr [NPSH available – NPSH required]) at BEP is considered and verified, there is a possibility the pump operation at the right side of BEP, even slightly right of BEP, can lead to cavitation. A selected pump should be able to operate temporarily at any point on its performance curve. NPSH margin should be checked and verified for the entire performance curve. As NPSHr is at highest at right side of the BEP, it should be verified that there is enough NPSH margin at right side of the performance curve as cavitation is not acceptable even if the pump is pushed to operate temporarily at high capacity points.

So, here is the general rule: A pump should be sized to operate most of the time at or near BEP. At the same time, temporary operation at the left or right side of BEP is possible and plausible. Such an operation might not be desired but may happen during transients or predicted/unpredicted events. Therefore, it should be safely and reliably managed. Every sizing factor and parameter including NPSH margin should be checked and verified accordingly.

BEP & Parallel Operation

Parallel operation of pumps is one of the most oversimplified concepts in the pump industry. As a result, there have been many operational problems, complaints, failures, shutdowns and financial losses associated with mistakes and wrongdoing in the parallel operation of pumps.

One risk of parallel operation is that one pump is pushed to operate at points far from BEP. In long-term operation, this can mean energy waste, low reliability and risks of failure. There can be many reasons for such an undesired operation where a pump is pushed to operate far from its BEP. The complete list of reasons and root causes of such faulty operation are outside the scope of this article.

One reason for operational problems could be differences, tolerances, deviations and degradations in so-called identical pumps in parallel operation where a pump is pushed to operate at points far from its BEP to catch up with the head generated by other pumps. This prolongs operation far from BEP and can lead to operational problems or failures. In fact, parallel operation of pumps should be avoided where possible.

However, all the above-mentioned risks and precautions do not mean that the parallel operation of pumps is impossible. There are cases where the only feasible pump options are pumps operating in parallel. If pumps are selected properly for parallel operation, and if all operational cases were carefully simulated and verified, the parallel operation would be safe and reliable. To start, the performance curve of pumps in parallel should not be flat or relatively flat as any small change in head would lead to a shift of the operating point away from BEP. There should be other precautions and requirements depending on each specific case of pumps in parallel operation.

A more complicated and riskier situation is the parallel operation of dissimilar pumps. In fact, this topic is an overlooked and neglected one in the pump industry. In many cases, different pumps have been installed to operate in parallel without proper assessment and the result would be problems, failure and breakdowns.

An interesting configuration is when multiple pumps are operated in parallel where a couple of pumps are operating at fixed-speed and one (or two) pump(s) is controlled by a variable speed drive (VSD). Such a configuration is used with the promise of fine flow management, where the fixed speed pumps provide the base-flow and the VSD pump(s) operates at variable-flow for the flow adjustment.

A concern is that this VSD pump would operate against so-called fixed headed pumps and, therefore, can potentially operate far away from BEP. For instance, it might operate at lowest possible speed with a low flow, whereas the operating point is at the far-left side of BEP. Obviously, such an operating case in the long run would cause problems.

The configuration of fixed-speed pumps and a VSD pump has been widely used due to low initial costs. However, it should be recognized that this is a complicated configuration and it is theoretically categorized as the parallel operation of dissimilar pumps. Although the pump casing of the fixed-speed pumps and VSD pump may be identical, their operational details are different (fixed-speed drivers versus VSD driver), and they are actually dissimilar pumps in parallel operation.

Within cases of the parallel operation of pumps, the parallel operation of identical pumps is considered less risky and the parallel operation of dissimilar pumps should be avoided.

However, there have been still cases where the parallel operation of dissimilar pumps should be selected as the only available option and it should be managed properly. For such a risky operation, pumps should be selected carefully, and everything including the control system, operating envelope, operational procedures, etc., should be managed to avoid dangers and risks.

Obviously, a significant risk is the situation where a pump can be pushed to work far from its BEP over a prolonged period of operation. Simulations of all possible operating cases and considering all possible factors and parameters such as plausible deviations, degradation and others are important for proper risk assessment. In the next step, corrections and improvements are needed to prevent such problems and risks. ■

Suction Piping, NPSH & Cavitation

Practical notes and useful guidelines for dealing with these aspects of centrifugal pumps.

By **Amin Almasi** | Principal Machinery/Mechanical Consultant

Most pumps used in different industries have been centrifugal pumps. This is because of their flexibility, reliability, favored head-flow curves and reasonable prices. This article discusses key aspects of centrifugal pumps such as suction piping, net positive suction head (NPSH) and cavitation.

Suction piping systems of pumps have always been challenging due to the sensitivity of the pump to the flow pressure drop on the suction and the relatively larger size of the piping used in the suction. On one hand, the suction piping should be straight, simple and short, which is unique to suction piping, to mitigate issues associated with NPSH, and a mandatory minimum pressure should be maintained at the pump entrance to ensure the proper operation of the pump. On the other hand, the piping is relatively larger in diameter, so nozzle load limits are more challenging.

NPSH

NPSH is particularly important to dynamic pumps such as centrifugal pumps. These pumps are vulnerable to cavitation. If cavitation occurs, the drag and friction in impeller vanes increase drastically, seriously restricting the flow and interrupting the operation. Cavitation has many adverse effects on impellers and generally on pumps. Prolonged exposure to cavitation can damage the impellers. NPSH refers to two quantities:

- NPSH available (NPSHa): a measure of how close the liquid at a given point is to boiling, and so to cavitation. NPSHa is usually calculated at the suction flange of the pump.
- NPSH required (NPSHr): the head value at a specific point (usually the inlet of a pump) required to keep the liquid from cavitation in a pump.

An appropriate NPSH margin (NPSHa minus NPSHr) should always be provided for the entire operating range. In other words, the NPSH margin is the NPSH that is available more than the pump's NPSHr.

Cavitation & NPSH

When a liquid enters the eye of a pump impeller, it accelerates as it is drawn into the impeller. This acceleration creates a pressure drop in the liquid at the impeller eye. If the liquid is close to its boiling point (bubble point), the pressure drop may be great enough to cause some of the liquid to boil. The bubbles that are formed by the boiling liquid enter the pump impeller along with the liquid. As the liquid (and bubbles) flow toward the tip of the impeller, the pressure rises and the bubbles collapse or implode. When these bubbles collapse, a large amount of energy is transferred from the fluid to the impeller at a very small point on the impeller. This energy is sometimes great enough to damage the impeller and is frequently enough to cause vibration and noisy pump operation. This process is called cavitation, and it must be avoided in the operation of any pump.

The damage caused by cavitation depends on factors such as pump speed, impeller material, amount of cavitation, type of liquid, etc. The type of liquid is particularly important. For example, cavitation in water pumps is usually more serious than in hydrocarbon pumps. This is because water has a much higher latent heat of vaporization than hydrocarbons (say, three to eight times higher). As a result, when the bubbles collapse, far more energy is released causing more damage to the impeller.

Cavitation can be prevented by making sure that the pressure at the suction of the pump is sufficiently above the bubble-point of the liquid to prevent the liquid from boiling as it enters the impeller eye. Pump manufacturers publish NPSHr values for their pumps. It should be ensured that NPSHa is greater than NPSHr at all times.

NPSH Margin

It is necessary to have an operating NPSH margin that is sufficient at all possible flows—from the minimum continuous stable flow to the maximum expected operating flow—to protect the pump from damages caused by cavitation. A key concern is to provide a suitable margin for the maximum expected flow at the right side of the curve where NPSHr is higher compared to its value at the rated flow.

It is difficult to give general advice for required NPSH margin. As a very rough example, an NPSH margin of 2 meters (m) or 2.5 m might be used for ordinary pumps. As another rough guideline, the formula NPSHA = 1.2 × NPSHr + 2 m can be used for NPSH margins and the relation between NPSHa and NPSHr. NPSH margins of 2 m, 2.5 m or 3 m have widely been accepted for small/medium pumps or low-/medium-energy pumps. For high-energy pumps, higher factors and margins should be used. For example, for some high-pressure and high-energy pumps NPSHa = 1.5 × NPSHr + 3 m might be used.

Difficulties & Challenges

NPSH calculations should be done with great care. During the early stages of the development of a plant or facility, the layout is not yet firm. Hence, NPSHa for the pump(s) cannot yet be calculated with confidence. However, preliminary NPSH can be estimated using information from a preliminary layout and elevations. NPSH margins can be changed by later modifications to the layout and particularly the elevations. NPSH plays an important role during the pump selection and could significantly impact the overall cost of the pump if a lower NPSHr pump is specified, since pumps with a lower NPSHr tend to be more expensive.

At this stage, the goal usually is to calculate a preliminary NPSH value and provide it to pump manufacturers to get feedback, proposals and values of NPSHr. This allows all involved parties to determine whether a pump with the specified NPSHr can be selected or not. It can be achieved with some modifications to the pumping system layout (higher NPSHa), or it might be achieved by selecting a pump with lower NPSHr. Based on the manufacturer's feedback, the layout can be modified to have a suitable NPSH margin.

A Booster Pump: Last Solution

For high-speed pumps, such as boiler feed water pumps or high-pressure pumps, NPSHr can be high. High values for NPSHr, as high as 30 m or more, are not unusual for some applications. In these cases, there might not be a way to provide enough NPSH margin and a booster pump might be used to provide NPSHr. All other options should be considered before this expensive solution is employed. This might be the case in revamp, renovation or upgrading developments where elevations are fixed and cannot be increased. Booster pumps are typically low-speed centrifugal pumps with a low NPSHr. They are typically installed to provide 40 to 80 m of head to the liquid.

Considerations for Suction Piping

For pumps, the suction piping is almost always more critical and challenging than the discharge piping, even though the discharge piping operates at much higher pressure and temperature differences than the suction. The diameter of suction piping is more than the diameter of discharge piping. Also, the suction piping is relatively shorter and stiffer than the discharge piping. A discharge piping of a pump can be provided with different loops and flexibility provisions. However, this is not the case for the suction piping.

Often, a simple layout of preliminary suction piping is not adequate to reduce the piping load at the pump suction nozzle and bring them below allowable limits. In many cases, some flexibilities should be included in the suction piping, keeping an eye on pressure loss and NPSH margin. Flexibilities should be accounted for and the required piping length and added bends, loops, etc., should be minimized to keep pressure drop under control.

A Stop at Suction Piping

In some cases, the piping-imposed load on the pump can be reduced by placing stops at strategic locations. The exact location of the stop is determined by the configurations of the piping and nozzle load/piping stress analysis. There have been cases where a stop for the vertical direction was included to limit the reaction load on the vertical direction. This is effective as many pumps have suction piping with a relatively long vertical run.

A Loop in Suction Piping

In some cases, a loop might be needed in the suction piping to deal with extreme temperature differences. Such a loop on suction piping is only acceptable as the last resort. The location of the loop is very important. One may think to place a loop at a higher elevation due to better space and support structure availability. However, when handling near saturated liquid, the high elevation loop can be the cause of many operational problems.

The loop, as the last resort, should be placed at the low elevation section (bottom portion) of the suction piping. Although this portion of piping is generally more congested, the loop should still be placed at the bottom portion in combination with the original bends. Such a combination reduces the number of elbows required. This reduces the pressure loss of the loop. ■