

Pump cavitation and how to avoid it

Cavitation is recognized as a phenomenon that can cause serious damage to water and wastewater pumps. As manufacturers strive to design smaller, more-efficient pumps while customers require more compact pump stations; small margins are left for avoiding cavitation. For those responsible for the planning and maintenance of pumping installations, it is therefore important to understand what causes cavitation and how it can be avoided.

Cavitation is the rapid process of formation and collapse of vapor bubbles in a liquid. In a pump, cavitation can damage a number of working parts, which can lead to results ranging from reduced performance to total failure. It can be said, therefore, to have serious effects on both the operation and lifespan of a pump. There are several reasons to why cavitation damage occurs. Some are a direct consequence of selected components in the pumping system; others are more difficult to control,

such as the contents in the pumped liquid. If the whole pumping system is insufficiently planned and designed, it may lead to cavitation and its problems: noise and vibration, material damage, reduced performance and even pump failure. This white paper explains how cavitation occurs. It is intended to serve as a reference in selecting correct pumps, planning pump stations and understanding cavitation-related complications in the field.

Typical indicators of pump cavitation

Noise and vibration

The onset of cavitation is often first detected by emitted noise rather than by visual observation of bubbles, material damage or decreased performance. The rattling sound that accompanies cavitation is one of the most evident and worrying characteristics of this phenomenon, with frequencies from 10 kHz to 100 kHz. The direct noise from cavitation in the pumped liquid is seldom the problem, but it can be transmitted through solid structures both in the pump and in the surrounding pipe system. For large dry-installed pumps the energy in cavitation-induced vibrations can occasionally be problematic and cause severe noise as well as damage to the pump and the piping system.

Surface damage

Perhaps the most obvious effect of cavitation on pumps is the material damage that cavitation bubbles can cause when they collapse in the vicinity of a solid surface. Eventually, this will lead to fatigue failure and the subsequent detachment or flaking off of pieces of material. The severity of the damage is dependent on the properties of both impeller material and the pumped liquid. Figure 1 shows the cavitation damage on the blade of a centrifugal pump impeller.

Corrosion will often increase the speed of deterioration. Since the oxides are less resilient to surface fatigue than the pure metal, cavitation continuously chips off pieces

of the oxide layer which protects the metal. This continuous exposure of a fresh metal surface accelerates the corrosion process. Hence, the cavitation and the corrosion each increase the damage done by the other mechanism.

Impact on performance

A large amount of cavitation will always reduce the head and the power of the pump as the vapor bubbles will reduce the active impeller passage. A small amount of cavitation will in many cases change the pump head and power in an often unpredictable way by altering the pressure distribution around the impeller blades. In rare cases this effect can even improve the pump efficiency; this can be confusing or misleading in trouble-shooting a situation which began with damage being observed.



Figure 1: Cavitation damage on the blade of a centrifugal pump impeller.

Physics of cavitation in liquid flows

Cavitation is the process of formation of vapor bubbles in a liquid flow caused by an initial reduction of pressure below vapor pressure, and subsequent collapse of the bubbles due to a pressure increase. Here pressure means absolute pressure referenced against a perfect vacuum (0 Pa).

Cavitation is a much faster phenomenon than boiling because the changes in pressure within a liquid can spread much faster than changes in temperature. While the vapor cavities formed by boiling emerge on the walls of the containing vessel, the vapor cavities formed by cavitation grow out of microscopic bubbles of dissolved gases, or on the surface of microscopic particles suspended in the liquid. These particles and bubbles are naturally occurring in any practical application and are always present to form the starting nuclei for cavitation bubbles.

When vapor cavities have started to form cavitation bubbles, they will be transported by the flowing liquid. While flowing through low pressure zones they will expand due to the pressure differences between the vapor inside the bubbles and the surrounding liquid. In zones with higher pressure the bubbles will contract. The rate of this expansion or contraction will be limited by the forces of surface tension and viscous forces.

When the bubbles contract towards an infinitely small radius, they will implode in a rather violent collapse; at the center of the bubble the pressure can occasionally be very high. The collapse is followed by a much localized pressure pulse that can give rise to small but powerful micro jets. These two effects can damage the surrounding surfaces. To avoid this state, we need to understand the parameters influencing the formation of cavitation and establish how close to the limit the conditions are in a particular application.

The vapor phase is created when the pressure level at some location in the liquid drops below the vapor pressure for the actual temperature in the liquid. In Figure 2 the vapor pressure curve indicates the boundary between the liquid phase and the vapor phase.

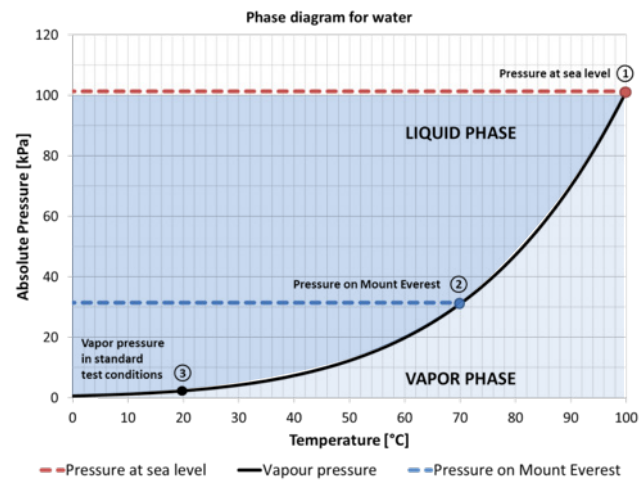


Figure 2: A Phase diagram for water illustrating the following conditions: 1) The boiling point at sea level, $T=100^{\circ}\text{C}$ and $P=101.3\text{kPa}$. 2) The boiling point at Mount Everest, $T=70^{\circ}\text{C}$ and $P=31.5\text{kPa}$. 3) Vapor pressure at "normal" testing conditions, $T=20^{\circ}\text{C}$ and $P=2.3\text{kPa}$.

From Figure 2 we can see that in normal testing conditions for pumps at a temperature of 20°C , a pressure reduction of 99kPa ($101.3-2.3$) is needed to reach the vapor pressure starting from the atmospheric pressure at sea level.

Up to this point pressure has been referred to as the absolute pressure at sea level or on the top of Mount Everest. It has to be understood that in a moving liquid the *total pressure* or energy in the water can be transferred between the *static pressure* (measurable absolute pressure) and the flow-derived *dynamic pressure*. If no external energy is added to a flowing liquid and internal energy losses are neglected, an acceleration of the fluid will increase the dynamic part and reduce the static part while the total pressure remains constant.

Occurrence of cavitation in pumps

A centrifugal pump adds pressure energy to the liquid by increasing the kinetic energy which is converted to static pressure. It does this by diffusing the flow inside the impeller and in the guide vanes and volute. Although the primary purpose of a pump is to increase the total pressure of the liquid, the acceleration into the impeller will initially cause a drop in the static pressure. Depending on the inlet static pressure, vapor bubbles can start to grow if the local static pressure drops below the vapor pressure of the liquid. Occurrence of cavitation inside the pump is therefore highly dependent on the static inlet pressure. In Figure 3 the passage through a centrifugal pump is depicted together with the static pressure in positions 1 to 4.

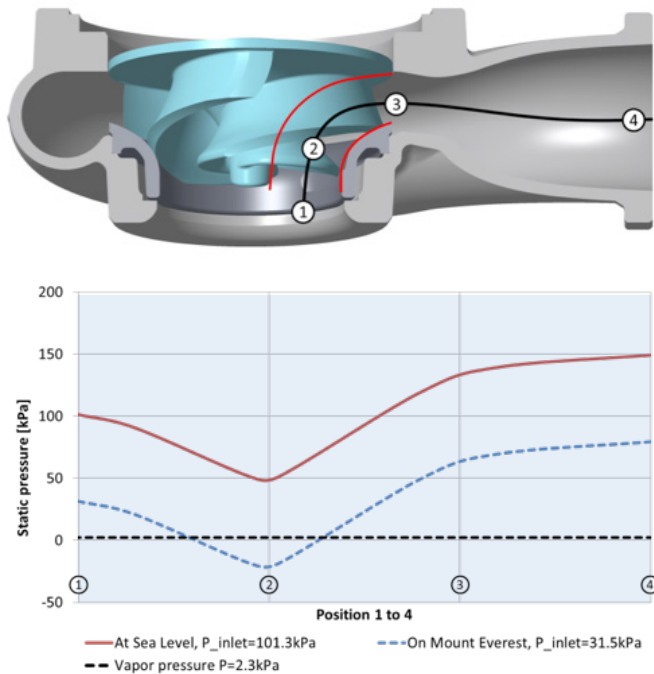


Figure 3: Schematic figure of the static pressure variation in the pump at a given flow rate. The red and blue dotted lines represent the influence of the inlet pressure given at Sea level and on top of Mount Everest. With an inlet pressure corresponding to the pressure on top of Mount Everest the minimum pressure is clearly lower than the vapor pressure (20°C) near position 2, which means that cavitation will be induced. Note that in reality when the static pressure reaches vapor pressure it is not reduced further.

Pump performance curves and flow regions

The pressure energy delivered by a pump is usually presented as total dynamic head (H) given as the height of a liquid column in meters or feet. This gives a relation between head and pressure, $p = \rho g H$, where p is pressure, g is gravity and ρ is the liquid density. In figure 4 the pump head curve (H), power curve (P) and efficiency curve (η) are plotted against the flow rate (Q). In this case flow values have been normalized with the flow at the best efficiency point (BEP), Q_{BEP} . This allows a generalized comparison of performance between pumps of different types and sizes as each flow rate has a reference to the design flow, the optimum operational flow of the pump.

In Figure 4 the pump head curve is divided into three flow regions; *recirculation*, *part load* and *over load*. Furthermore the best efficiency point, Q_{BEP} , and the approximate starting point for recirculation, Q_{RS} , are marked. These operating flow regions and flow rates will be used to schematically describe the influence of the flow rate on cavitation performance.

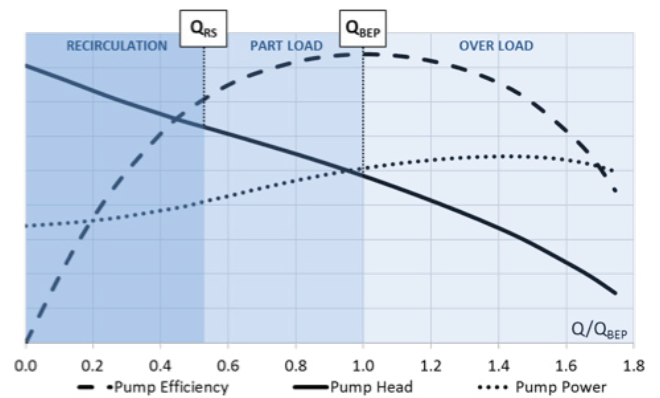


Figure 4: Pump head, power and efficiency curves with three defined flow regions: recirculation, part load and over load.

Flow direction inside the impeller

When a centrifugal pump operates at Q_{BEP} , the flow inside the impeller is aligned with the impeller blades since it is specifically designed for this flow rate. In recirculation and part load ($Q/Q_{BEP} < 1$) or over load ($Q/Q_{BEP} > 1$) the static pressure distribution on the impeller blades is distorted, and areas of low static pressure are formed. The location of low-pressure areas largely depends on the difference between the direction of flow and the geometrical direction of the blade at the impeller's leading edge.

Figure 5 depicts the top view of a typical centrifugal pump impeller. A blade section in the impeller is used to describe the deviation between the blade direction and the flow direction in part load.

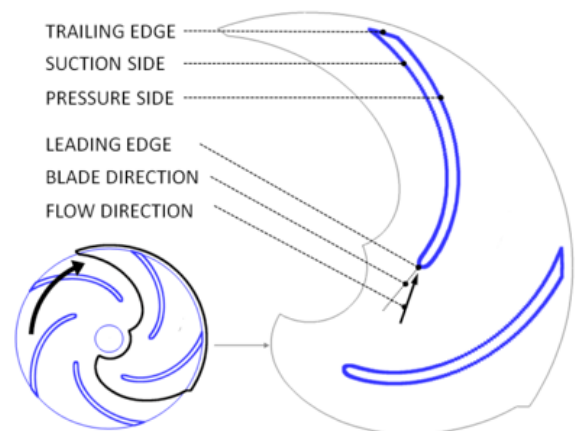


Figure 5: A section, including two blades, of a centrifugal impeller is used to describe several important impeller denominations such as the geometrical blade direction and the flow direction.

The blade section pictured in Figure 5 will be used throughout the paper to describe various flow characteristics.

Influence of flow rate on the blade pressure

In Figure 6 the regions where cavitation typically starts are marked with red color. Streamlines describing the flow path are plotted with thin black lines. The lower adjacent blade in each figure presents the static pressure distribution around the blade where blue color represents low static pressure and orange color high static pressure.

As can be seen the red regions are in the vicinity of the leading edge and will be the starting point for formation of cavitation. Further down the blade the static pressure increases, leading to the collapse of vapor bubbles and potential damage to the impeller surfaces.

The size and location of these vapor cavities will depend on the inlet static pressure and actual flow rate. The images in Figure 6 reflect a condition where the inlet pressure is below the normal test conditions for pumps. If the inlet pressure is increased then the cavitating region shrinks back towards the leading edge.

Low pressure location at Q_{BEP}

Starting with the best efficiency point, Q_{BEP} , Figure 6-a shows a flow direction that is aligned with the blade. However, it can be seen that the presence of the blade causes a blockage effect on the incoming flow which leads to low static pressure regions on both sides of the blade. If the inlet static pressure is reduced significantly, then cavitation can be present even when the flow direction is perfectly aligned with the blade.

Cavitation patterns in over load

In over load, Figure 6-b shows that the flow direction at the impeller inlet is turned towards the suction side of the blade. As a consequence, part of the liquid has to accelerate around the leading edge and a region of low static pressure is formed on the pressure side of the blade, again introducing a region prone to cavitate. If the flow is increased further then the misalignment between the flow direction and the geometrical blade direction will increase, and consequently the region of low static pressure enlarges.

Cavitation patterns in part load

Figure 6-c examines the other part of the performance curve, in part load. Figure 6-c reveals that in contrast to overload conditions, the flow direction is turned towards the pressure side. This causes a low pressure region at the suction side of the blade.

Figure 6-a: The static pressure at Q_{BEP} where the flow direction is aligned with the blade. The enlarged circle visualizes the local decrease in static pressure due to the acceleration around the leading edge of the blade.

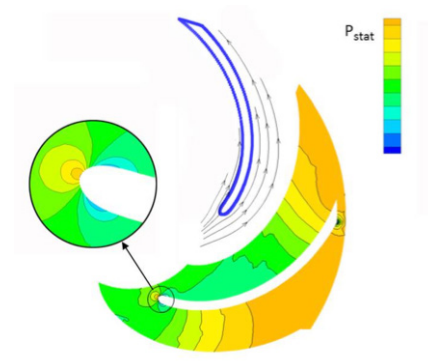


Figure 6-b: The blade pressure in over load. Cavitation in over load starts as thin sheets of vapor near the leading edge on the blade pressure side.

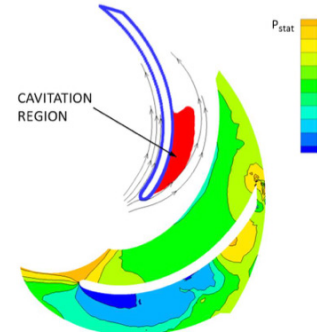
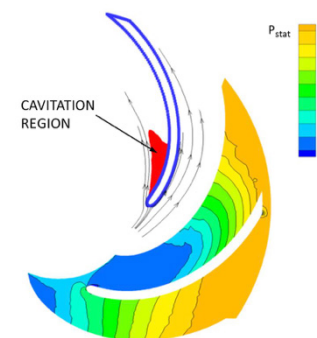


Figure 6-c: The blade pressure in part load. Cavitation in part load starts as thin sheets of vapor on the suction side by the leading edge.



Recirculation

When the flow rate decreases below the Q_{RS} the pump is operating in the recirculation region of the performance curve. Recirculation usually begins at the impeller discharge, where flow starts to recirculate back from the pump volute into the impeller. This is called discharge recirculation.

When the flow rate is decreased further, the deviation between the flow direction and the geometric blade direction at the leading edge increases to the extent that the liquid starts to separate from the blade suction side and recirculate back into the inlet of the pump. This is called suction recirculation under which a vortex starts to form between the impeller blades. A vortex is an isolated swirling fluid structure that comprises a core of low pressure at which cavitation can be induced.

Recirculation is a very complex and highly three-dimensional flow regime that is not stationary and difficult to describe in general terms. The onset point of recirculation is highly dependent on the type of pump as well as on the geometrical details of the impeller and pump suction.

Evaluation of cavitation risks

In order to make it convenient for pump manufacturers and users to evaluate the risk for cavitation to take place inside a pump, the terminology **Net Positive Suction Head (NPSH)** has been introduced.

Definition of cavitation performance for pumps

NPSH is the difference between the absolute total pressure at the eye of the pump impeller and the liquid vapor pressure converted into head of the liquid. Here **Net** refers to what is remaining after all deductions, from the atmospheric condition to the impeller eye in the present installation. **Positive** means it is absolute pressure and **Suction Head** refers to the pressure at the pump suction elevation. NPSH is defined in the following way:

$$\text{NPSH} = H_{\text{tot}} - H_{\text{vap}} = H_{\text{stat}} + H_{\text{dyn}} - H_{\text{vap}}$$

Cavitation performance of pumps can be tested with several different methods described in the international standard ISO9906:2012. All methods include running the pump at different levels of NPSH, while measuring head, flow and power. From the test data the **NPSH3**-performance of the pump is calculated. NPSH3 is the NPSH-level at which the head of the pump at a certain flow rate has dropped 3% due to influence from cavitation. By repeating the test procedure for several flow rates an NPSH3 curve can be established.

Additional cavitation criteria for pumps

To recognize the complexity of cavitation in pumps, several other NPSH criteria are often used to describe different states of cavitation in a pump. In this paper **NPSHi** denotes the incipient cavitation where the first vapor bubbles occur, **NPSH0** is the level where the head first starts to drop and **NPSHR** indicates the manufacturer specified level above which the pump will operate without risk for major cavitation-induced damage. It is important to understand that all the mentioned NPSH criteria are pump characteristics considered to be independent of the system.

NPSHav is the available suction head in the system which must be compared with the pump characteristics to determine under which condition the pump is operating. Reliable operation is generally guaranteed by the supplier if $\text{NPSHav} > \text{NPSHR}$ at the given flow rate, but it must be understood that to avoid cavitation completely the NPSHav level needs to be higher than NPSHi. Curves representing the different types of NPSH-criteria are plotted schematically in Figure 7. The understanding of the different NPSH-criteria will be discussed further below.

Inconsistent definitions of NPSHR

Be aware that, in contrary to the presented definitions some manufacturers use the term NPSHR with the same meaning as NPSH3. NPSH3 signals that the pump is already severely affected by cavitation; operating a pump at these conditions will most certainly damage it. A margin towards the NPSH3 level must always be used and recommended ratios $\text{NPSHav}/\text{NPSH3}$ can be found in e.g. the guideline on NPSH-margin issued by Hydraulic Institute.

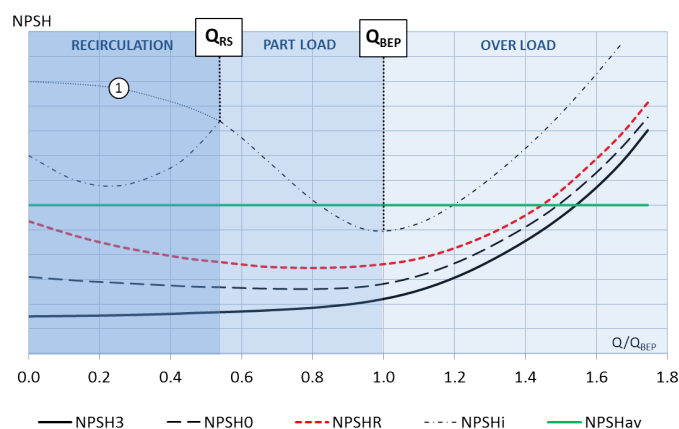


Figure 7: Schematic NPSH-curves for a centrifugal pump. Recalling the definitions of NPSHav and NPSHR, it is clear that for this example, reliable operation would be guaranteed in the flow range $0 < Q/Q_{\text{BEP}} < 1.45$; also that a completely cavitation-free operation only exists in the flow range $0.83 < Q/Q_{\text{BEP}} < 1.19$.

Incipient cavitation

The formation of the first vapor bubbles are induced by local pressure drops caused by discontinuous surfaces or in the gaps between the impeller and the pump casing. Methods to capture these effects are far more complicated than those used to evaluate NPSH3 and the effects on pump reliability are in most cases negligible. Therefore information about NPSHi levels is generally not of interest for those using the pump, and is usually not available from the manufacturers. However, the interpretation of incipient cavitation is important for the general understanding of cavitation in pumps.

The NPSHi curve has its lowest value when the flow direction at the impeller inlet coincides with the blade direction at the leading edge. This is often referred to as the *shockless* flow condition and usually occurs at a flow rate slightly higher than Q_{BEP} . The more Q/Q_{BEP} deviates from 1, the more the NPSHi level increases to fully suppress the starting point for cavitation. In over load the NPSHi curve follows the principal shape of the NPSH3 curve while in part load it increases till it reaches the starting flow for recirculation Q_{RS} .

Below Q_{RS} the pump operates under recirculation, a flow regime with partially reversed flow in the impeller. In recirculation NPSHi levels are very dependent on the design of the pump suction. The curve denoted ① in Figure 7 shows the typical effect on NPSHi from pre-rotation reducing inlet ribs or a bend on the inlet. Operation in recirculation should be avoided especially for larger pumps in dry installation where the pressure pulsations may cause vibrations in piping and fundamentals as well as high loads on the pump shaft and bearings.

Damaging cavitation conditions

When NPSHav falls below NPSHi cavitation starts to develop inside the impeller. In part load thin sheets of cavitating flow form near the leading edge surface on the suction side of the impeller. In over load the same pattern occurs on the pressure side of the impeller. If NPSHav is decreased further, then the area of cavitation enlarges and starts to cause noise; this is followed by cavitation damage, head loss and eventually total breakdown of head.

It is important to understand that the NPSH-level that causes most damage to the impeller or other hydraulic parts is not necessarily when the size and concentration of vapor cavities is the largest. This might seem contradictory, but it can be explained by a dampening effect which the vapor cavity itself has on the pressure pulsations caused by imploding bubbles. In general, a definite NPSH-level for material damage is difficult to specify as the impact of the implosions is affected by several parameters like bubble dimension, type of vapor cavity, liquid properties, temperature, blade geometry and material. Nevertheless, a few general points can be discussed about the location of typical damage.

As vapor bubbles are transported with the flow in the blade direction they will eventually reach a region where the static pressure exceeds the vapor pressure. At that point the vapor bubble will implode, and if the cavity is located near a surface then it can be damaged. If the operating flow rate remains stable then the flow field and consequently the region of cavitation on the blade surface will remain stable as well. Thus, the damage to the surface will be found on more or less the same location.

Studies to determine the NPSHi level for different pump types show large variations of the ratio NPSHi/NPSH3. Depending on pump type and relative flow rate, values between 1.5 and 10 have been observed. The required level of NPSHav to fully suppress all cavitation, $NPSH_{av} > NPSH_i$, is not practically attainable or required to guarantee reliability in most applications. Therefore NPSHR has been defined by the manufacturers as the recommended minimum NPSH-level.

In Figures 8 and 9 the black blobs mark the locations on the blade where traces of imploding vapor bubbles could be expected in the different flow regions. Here, for visibility reasons the NPSHav level used to simulate the flow was selected such that the cavitation regions have expanded into large clouds. If NPSHav is increased then the cavitation region would shrink back towards the leading edge, and the potential damage would follow the same pattern.

Damage patterns in over load

When the pump operates in over load, damage can be expected on the blade pressure side near the cessation of the cavitation region, usually near the leading edge. A typical location is depicted in Figure 8.

In high over load ($Q \gg Q_{BEP}$) cavitation can also be induced on the shroud surface between the blades, due to the curvature-related acceleration in the impeller channel.

If the pump is operated in high overload, then damage can also appear on the pressure side of the volute cut water. A typical location for this type of damage is marked in Figure 8. As the ability to inspect the volute is much more limited than for the impeller, these traces are seldom discovered.

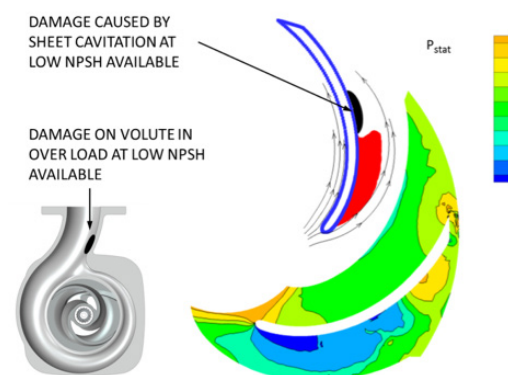


Figure 8: Typical cavitation damage in over load is located on the pressure side.

Damage patterns in part load

Figure 9 shows that in part load a damage location can be expected on the blade suction side. Figure 10 shows cavitation pitting caused by sheet cavitation close to the blade leading edge on the suction side.

Upon inspection of pumps, it is common to see how cavitation damage has been exacerbated by the pumped liquid chemical composition or content of abrasive particles. On the semi-open impeller in Figure 11, cavitation has been induced by operation in part load, and due to heavy leakage between the insert ring and impeller tip. The damage has clearly been worsened by a corrosive and erosive environment.

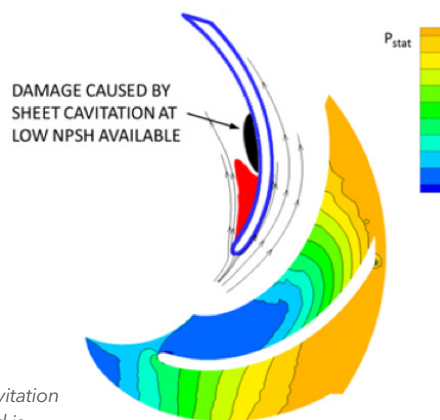


Figure 9: Typical cavitation damage in part load is located on the suction side.



Figure 10: Damage from sheet cavitation on the suction side near the leading edge of an impeller blade.



Figure 11: Cavitation damage on the suction side of the blade that has been worsened in a corrosive and erosive environment.

Damage patterns in recirculation

When the flow rate has been reduced below the starting point for recirculation, the flow field becomes very complex and particular events that lead to surface damage become difficult to describe in a distinct and clear way.

Damage from recirculation can appear even though the customer has followed guidelines for minimum NPSH_{av}.

This can be attributed to the local maximum level in the NPSH_i curve which is often overlooked when the NPSH_R level is determined.

Recirculation usually starts at the impeller discharge where liquid starts to flow back from the volute into the impeller on the upper part of the trailing edge. Discharge recirculation can cause damage on both impeller and volute, especially with small distances between the impeller trailing edge and the cutwater in the volute. Figure 12 shows a typical damage location on the pressure side of the impeller at the trailing edge and the corresponding location on the suction side of the volute cut water.

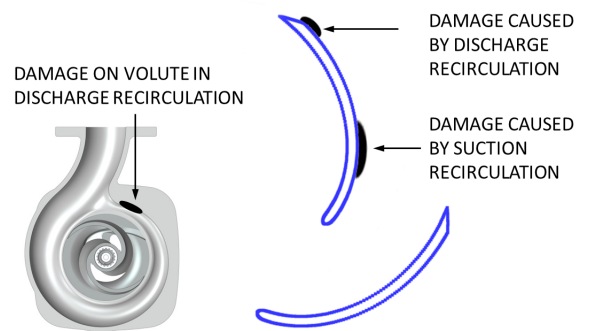


Figure 12: Typical damage induced in the recirculation flow region. In discharge recirculation damage is found at the trailing edge of the impeller or at the suction side of the cut water. In suction recirculation damage is often found on the pressure side a bit down the blade from the leading edge.

If the flow rate is reduced further, then the fluid at the pump suction starts to recirculate as well. Typical damage from suction recirculation is caused by the vortex between the blades and is usually located on the pressure side, slightly down the blade from the leading edge. This vortex cavitation is contrary in location to damage from sheet cavitation in part load that would appear on the suction side near the leading edge. This damage location is marked in Figure 12. Figure 13 shows a sewage pump impeller where damage has been attributed to suction recirculation.

Independent of the relative flow rate, cavitation can also be induced by vortices located in the intersection between the blade and the hub or shroud discs.

Determination of damage cause

Cavitation damage reveals some details about the operation of the pump; however, inspection of the surface damage, as a sole method, is not sufficient for determining the reason for the damage or probable running conditions.



Figure 13: On this impeller, cavitation damage is located near the hub on the pressure side of the blade. In this case the pump has been operated in the recirculation region; matters have been worsened with speed variation and a flat system curve.

Establishment of the required NPSH, NPSHR

A standardized test for pumps to determine the resistance to cavitation-induced damage does not exist. Therefore manufacturers have to rely on indirect methods to establish the minimum required NPSH-level, NPSHR. These methods are rather complicated and can be based on knowledge of NPSH3, sound and vibration levels, field testing or further analysis of the performance parameters such as flow, head and torque during the standardized NPSH-test.

Determination of NPSHR for Flygt pumps
 To facilitate the demand from pump users to be able to select a pump with proper cavitation performance, Flygt has defined NPSHR to be NPSH3 plus a margin that accounts for cavitation-induced problems such as noise, material damage and reduced efficiency. The NPSHR level is usually set at the threshold value where the pump efficiency begins to be affected due to cavitation. Upon that an additional margin can be used at $Q < Q_{BEP}$ to account for noise and potential material damage.

Ensuring trouble-free installations

NPSH_{av} is the available amount of NPSH at the pump suction elevation in a specific installation, and is the difference between the pressure at the eye of the pump impeller and the vapor pressure of the liquid.

NPSH_{av} is determined by the design, configuration, elevation and relative water levels on the suction side of a pumping system. It can be calculated as the atmospheric pressure at present elevation plus the submergence depth, minus the vapor pressure head of the liquid and minus the friction head losses in the suction piping. If it is a submersible installation, the suction friction head loss is usually zero. Figure 14 shows the components involved in calculating NPSH_{av}.

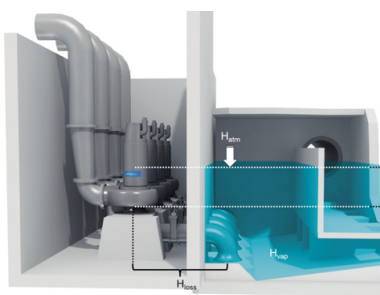


Figure 14: Calculation of NPSH_{av} in a dry installation.

$$NPSH_{av} = H_{atm} + H_{sub} - H_{vap} - H_{loss}$$

H_{atm} = Atmospheric pressure

H_{sub} = Submergence depth

H_{vap} = Vapor pressure of the liquid at a certain temperature

H_{loss} = Dynamic head loss in the suction piping in a dry installation

NPSH_{av} must always be greater than NPSHR (NPSH_{av} > NPSHR) in order to achieve a reliable and long-term operation for a pumping system.

Suction piping design in dry-installations

A dry-installed pump has greater risk of cavitation than submerged pumps. Low head losses in the suction piping and a uniform inflow pattern to the pump are always preferred to avoid cavitation issues. To ensure these conditions the choice of components and their locations require careful consideration:

- Enlarging the diameter of the inlet pipe and reducing the number of fittings can reduce the head losses.
- Avoiding sharp angles or sudden area changes along the inlet pipe can reduce the head losses and ensure uniform inflow.
- If a valve is required, a gate valve that reduces the impact on the flow field is preferred, and should be installed more than five pipe diameters away from the inlet of the pump.
- If elbows have to be used, it is preferable to maintain a distance between them to reduce the flow contraction and avoid a swirling flow in the pipes.
- An inlet bell creates both a better inflow distribution to the pump and decreases the head loss.
- Make sure that the submergence of the suction inlet is sufficient. Designing a deeper sump or increasing the water level is a robust solution to increase the NPSH_{av}.

Figure 15 shows a correctly configured dry-installed pump with a bell-shaped suction inlet where two bends are kept in a plane and a gate valve is placed at least five diameters upstream of the pump.



Figure 15: A dry-installation including a bell-shaped suction inlet, a gate valve, a suction elbow (180°) and a pump.

Adverse hydraulic phenomena

Adverse hydraulic phenomena are unfavorable conditions in the pumping station that can reduce NPSH_{av}, affect the NPSHR of the pump, and ultimately increase the risk of cavitation. Typically these involve air entrainment, vortex formation, excessive pre-swirl or uneven velocity distribution at the pump intake.

Air entrainment

Air can be entrained by the cascading of the water as it enters the sump from a weir or an inlet pipe located above the water surface. As it falls, from the station inlet down to the water surface the water accelerates, and at impact will draw air into the water.

Figure 16 shows how entrained air bubbles are transported with the flow into the pump. If the air is more than approximately 3% of the water volume, then the presence of air bubbles will lead to a clear reduction in pump performance and loss of efficiency. The air bubbles will expand in low pressure zones such as the impeller eye and cause a blockage. This will then increase the water velocity and reduce the pressure, and therefore increase the cavitation risk. By contrast, a small amount of air might reduce the damaging effects of cavitation due to the dampening of the pressure waves induced by imploding vapor cavities.

Air entrainment can be avoided by eliminating the cascade or by sufficiently reducing its height. If the cascading is unavoidable, then the problem can be prevented by increasing the distance between the station inlet and the pump or by separating the two by an inlet baffle; this will allow air bubbles to rise to the water surface instead of entering the pump.

Vortex formation

Vortex formation in the pumping station can also cause problems in terms of cavitation damage to pumps. Due to its swirling flow field the core of the vortex has a relatively low static pressure that will slightly reduce the NPSH_{av} if the vortex enters the pump. At the same time the swirling motion will disturb the inflow to the pump and can affect NPSHR.

Figure 17-a shows a surface vortex that enters the pump. A surface vortex can form when the submergence depth is low or when obstacles exist in the flow channel. In case of a surface vortex with air core that extends into the pump suction, the atmospheric pressure will prevail at the impeller inlet, so the submergence of the inlet cannot be counted towards the NPSH_{av}.

Figure 17-b shows a sub-surface vortex that enters the pump. A sub-surface vortex can form from the sump bottom, walls or between two pumps, and can extend to the pump inlet. This type of vortex can achieve high rotational speed with reduced pressure inside the core of the vortex, and therefore it can increase the cavitation risk.

Vortex problems should be prevented by a proper design of the flow approaching the pumps. If the design is not fully satisfactory in this respect, different anti-vortex devices such as curtain walls and splitters can be used.

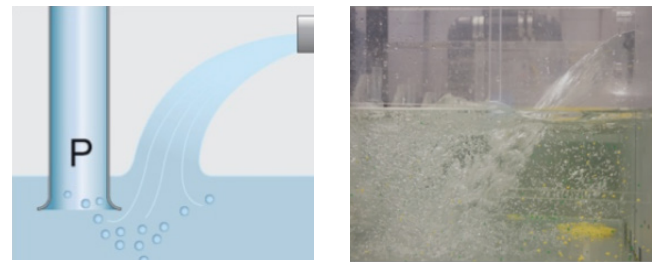


Figure 16: Air entrainment due to high elevation of the station inlet.

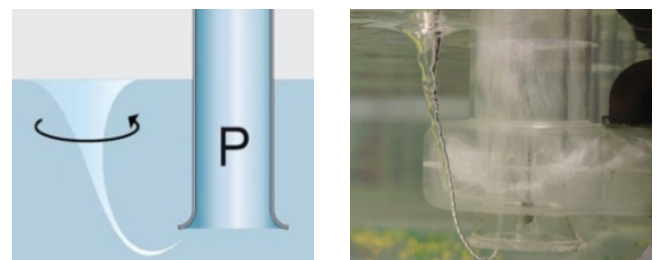


Figure 17-a: A surface vortex that enters the pump suction.

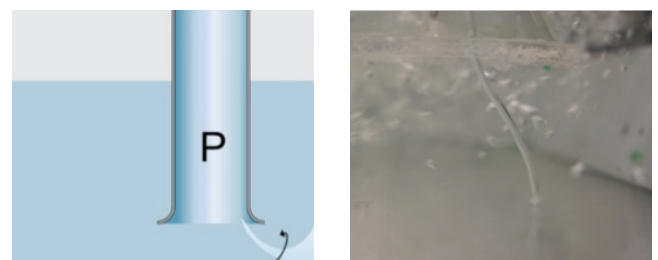


Figure 17-b: A sub-surface vortex that enters the pump suction.

Excessive pre-swirl

Excessive pre-swirl can change the flow condition at the pump inlet in a detrimental way. As a result of pre-swirl the flow direction is not well aligned with the blade direction, which can result in reduced pump performance and a higher cavitation risk. Pre-swirl usually originates from a non-uniform velocity distribution in the approaching channel, which evolves into a pre-swirl at the pump inlet, as shown in Figure 18. Baffle walls and splitters are often used to straighten the inflow stream and prevent excessive pre-swirl.

Uneven velocity distribution at pump intake

Uneven velocity distribution can result from any type of disturbances, such as sudden sectional area changes in the flow channel, obstacles or a lack of proper suction design for the pump. Because of the uneven velocity distribution, the velocity in some region of the inlet is larger than the average velocity for which the pump is designed, the NPSH_{av} which the pump actually gets is lower than the calculated value. Therefore, the cavitation risk is increased under such conditions.

Figure 19 shows a skewed velocity profile at the pump intake that could be caused by the positioning of the suction too close to the bottom and wall in the sump. An enclosed intake can be used to improve velocity distribution at the pump intake.

General consideration in the design phase

The recommendations described above should be considered in both submerged and dry installations. At early stages of a project the design recommendations provided by pump suppliers or other credible sources should be closely followed to reduce any hydraulic risks. If the pump sump design differs from the standard and adverse hydraulic phenomena are difficult to predict, then a physical model or computational fluid dynamics (CFD) technology is strongly recommended to verify the station design.

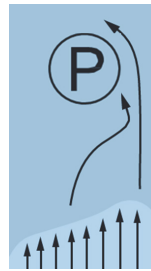


Figure 18: A top view of a pumping station with a pre-swirl into the pump.

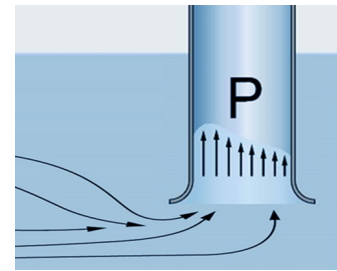


Figure 19: A non-ideal positioning of the pump too close to the bottom and wall of the sump causes an uneven velocity at pump intake.

Common field problems and typical solutions

Cavitation is affected by several factors related to the type of system curve, station design, pump selection, suction piping, solids contents and air dissolved in the water. In the design stage of the pumping station all these parameters need to be taken into account to avoid unnecessary and costly problem-solving when the pumping station is in service

Several pumps in parallel duty

In many systems, multiple pumps are in a parallel configuration used to vary the flow rate. By controlling the number of pumps that are running simultaneously by a switch on and off sequence in the control unit, the required flow can be maintained. In this type of pump configuration each duty point will have different relative flow rates Q/Q_{BEP} and thus different NPSHR-levels. It is not uncommon to have a large variation in the required

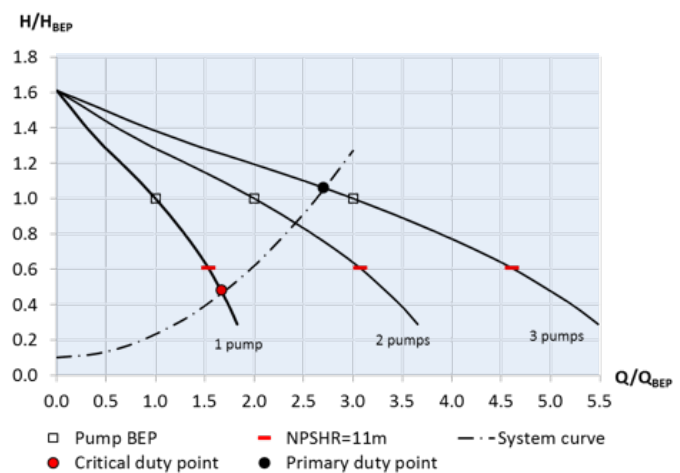


Figure 20: Duty points for multiple pumps in parallel operation.

flow rate. It is important to verify the condition $NPSH_{av} > NPSH_R$ for each duty point separately, especially in a friction system. The example in Figure 20 shows a three-pump configuration where the primary duty point is located close to Q_{BEP} when three pumps are running. When the flow is reduced to single pump operation $Q/Q_{BEP} \gg 1$ and a critical duty point occurs if $NPSH_{av} < NPSH_R$.

In a field issue situation, the problem can in many cases be solved by equipping one of the pumps with a Variable Speed Drive, VSD. The head will thereby also be reduced when the flow is adjusted and Q/Q_{BEP} would remain within reasonable values.

Variation of flow rate by the use of a VSD

When a VSD is used to alter the flow rate and the system curve is dominated by geodetic head the situation might become critical at low flow rates, $Q/Q_{BEP} < 1$. Figure 21 shows a single pump operation where a primary duty point is located slightly below Q_{BEP} . When the speed is reduced to decrease the flow rate, there is a risk of entering the recirculation region of the pump. In this part of the performance curve the NPSH_i is high and cavitation damage can occur even though $NPSH_{av} > NPSH_R$. If the pumps are of the low-head type (mixed or axial flow pumps) the consequences of drifting into the recirculation region can be costly with cavitation damage and vibration issues.

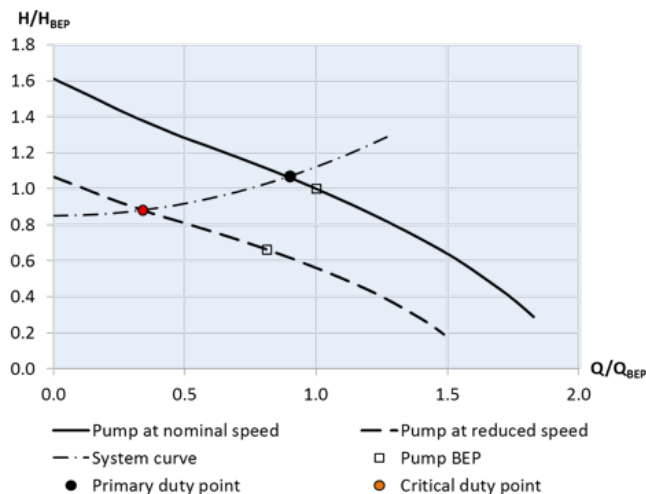


Figure 21: Duty points for a single pump with frequency variation utilizing a VSD.

A minimum speed limit can solve the problem without restricting the flow range of the installation to an unacceptable extent.

Clogging

NPSH_R is established from testing in clean water; these idealized conditions are designed in order to compare NPSH-performance between pumps. In unscreened sewage applications, objects such as rags and flushable wipes in the water can cause clogging. For improperly designed impellers these objects can stick onto the leading-edge surface of the impeller without tripping the pump. With time this build-up will alter the shape of the impeller and result in a much thicker blade. Since this reduces the area between the blades, the increased velocity will lead to a low-pressure zone where cavitation bubbles can generate. Damage from clogging-induced cavitation can be located at various positions. In Figure 22 the damage is located near the impeller discharge diameter on the pressure side.

The benefit of selecting a clog-resistant pump is not only the sustained pump efficiency; it also reduces the cavitation risk and increases the overall reliability of the pumping installation.



Figure 22: Cavitation damage on the pressure side caused by clogging

Incorrect estimation of the system losses

It is a rather common issue that the estimated system head loss on site deviates significantly from intended or given values. In old pipe systems, this can be attributed to such factors as leakage, eroded and corroded piping, blockage in the piping, or malfunctioning valves. Even in new designs accurate calculation of the head loss can be difficult if the system is complex or component data deviates from catalog values. Regardless of the reason for the discrepancy, it means that the actual flow is different from the assumed value, which may require an adjustment in place to avoid cavitation-induced problems in the current situation. A slight alteration of the flow rate leading to a reduction of NPSH_R can be achieved by:

- Trimming the impeller diameter
- Using a VSD to adjust the speed
- Throttling the flow on the discharge side of the pump

Conclusion

In this paper, the conditions are described under which cavitation is created in rotodynamic pumps. The flow field and pressure distribution around the impeller blades at different flow rates are discussed and linked to the typical locations of vapor zones to aid the understanding of the mechanisms creating cavitation. Detailed pictures of the impeller working conditions are used to describe the position of the resulting surface damage based on the vapor zones. Furthermore, the definition and usage of different NPSH values and the establishment of NPSHR is presented. The following conclusions can be drawn:

- Close to Q_{BEP} the NPSHR value is normally the lowest since the flow direction is aligned with the blade direction and no vapor zones are present. The demand for NPSHav is the lowest.
- When a pump operates in the overload region ($Q > Q_{BEP}$), damage can be expected on the pressure side close to the leading edge.
- When operating in the part load region ($Q < Q_{BEP}$), damage can be expected on the suction side close to the leading edge.
- Recirculation is a flow regime with high complexity where the specific details in the hydraulic component design will affect the cavitation behavior of the pump.
- In discharge recirculation ($Q < Q_{RS}$), damage is normally located at the pressure side close to the trailing edge and occasionally in the volute on the suction side of the cut-water.
- In suction recirculation ($Q < Q_{RS}$), damage is normally located at the pressure side further down the blade from the leading edge.

The cavitation characteristics of a pump as well as the location of damage under certain operational conditions are very much related to the specific design details of the hydraulic components. Therefore, it is not recommended to make a determination of the reason for cavitation based solely on inspection of the cavitation damage. Instead the pump control settings and details of the overall system design should be included as well.

Given this theoretical background and description of mechanisms causing cavitation, the following conclusions about the application are given:

- The condition $NPSH_{av} > NPSHR$ should be verified for all possible duty points.
- If the pump is dry-installed, then the suction piping should be given careful consideration.
- Increasing submergence depth will always be an easy solution to increase NPSHav and decrease cavitation risk, but might be expensive if the depth of the sump must be increased.
- The pump sump should be free of any hydraulic adverse phenomenon. Design recommendations provided by pump suppliers or applicable standards should be followed.
- In an unscreened sewage pumping system, a clog-resistant pump is always the best selection.
- If the duty point onsite deviates significantly from the designed value, then trimming the impeller, using a VSD or throttling the flow can be used to change the duty point and solve the problem.

If a cavitation issue becomes a troubleshooting case, it can be difficult, expensive and time-consuming to solve. The entire pumping system should be checked until the root cause of the cavitation problem is discovered. The solutions can be suggested, but might be very difficult to execute because of the limits onsite. It is much easier and less costly to make changes at an early design stage rather than after construction and installation.