Suction Side of Centrifugal Pumps

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Abstract

Many of the operational problems associated with centrifugal pumps can be traced to suction side of the pump. Inadequate attention to inlet conditions can lead to reduction or stoppage of flow, insufficient head generation, early failure of seals and bearings due to cavitation, heightened noise and vibration and an overall reduction in pump reliability.

The present paper attempts to summarise the areas of concern on the suction side of the pump and suggests some simple steps that a pump user can take, and, standards/codes that he can refer to while designing the intake, sump and pipe work leading to the pump suction.

This paper has been motivated by the authors' experience of analyzing several pumping installations with visibly flawed suction design.Distressingly, the same flawed designs are often repeated in multiple pumping installations across the same plant. In most cases, alteration of the suction layout in an existing plant is difficult and expensive. The right time to consider the suction parameters and configuration of the suction side of the pumping plant is the design phase of the project - a case study has been included to illustrate this point.

What do we really mean by "suction side" and what are the design elements that a pump user needs to consider for trouble-free suction performance of the pump?

In order to cover all critical areas, we will start from the suction branch of the pump itself and move upstream to include suction piping and support system, valves and fittings on the suction line, suction tank or sump and, finally, the fore-bay/piping leading to the sump. In the course of this journey, along the suction side of the pump, we will try to cover design issues relating to:

- Estimation of available NPSH
- Determination of pump operating speed based on available NPSH
- Margin between NPSHA and NPSHr
- Suction piping and velocities in the suction pipe
- Problems associated with air entrainment
- Guidelines relating to sump design

Estimation of available NPSH

By definition NPSHA is the net head available at the pump suction over the vapor pressure of the liquid at the pumping temperature. NPSHA is a pump system parameter and can be quite easily and accurately calculated by the pump users, if the location of the pump with respect to the suction sump,and the details of piping between the sump and the pump are known with sufficient accuracy.

It is important that the pump user tries to establish NPSHA based on the maximum and minimum levels of the liquid in the suction sump. It is preferable that NPSHA is calculated over the entire anticipated capacity range of the pump such that a NPSHA curve is available covering the duties over the entire operating region of the pump. Figure 1 provides an example of how NPSHA can be established for a simple system. While it is preferable to provide to an installation as much NPSHA as possible, there are often site and cost limitations restricting the location of the pump with respect to the water level in the suction sump. However, once NPSHA has been estimated with sufficient accuracy, the pump user can proceed to the next important step of determining the pump speed and often even the preferred pump type.

Determination of pump operating speed:

Both Europump and the Standards of the Hydraulic Institute provide guidelines for the estimation of maximum operating speed based on the available NPSH of a pumping system. Maximum operating speed is given as

$$N = \frac{8500 \times (NPSHA)^{3/4}}{Q^{1/2}}$$

Where,

N = Maximum operating speed in rpm

Q = Flow through the pump in USGPM (Consider 50% flow for a double suction impeller)

NPSHA = Net positive suction head in ft.

The underlying logic of this recommendation of a maximum operating speed is that most standard centrifugal pumps are designed for a suction specific speed varying between 9,000 and 11,000 US units, in order to achieve an acceptable suction performance coupled with good efficiency and operating range.

The maximum operating speed suggested above provides an adequate margin between the available and the required NPSH for these pumps. Suction specific speed is a type number signifying NPSH performance of the pump and is expressed as

$$N_{SS} = \frac{N \times Q^{1/2}}{(NPSHr)^{3/4}}$$

Where,

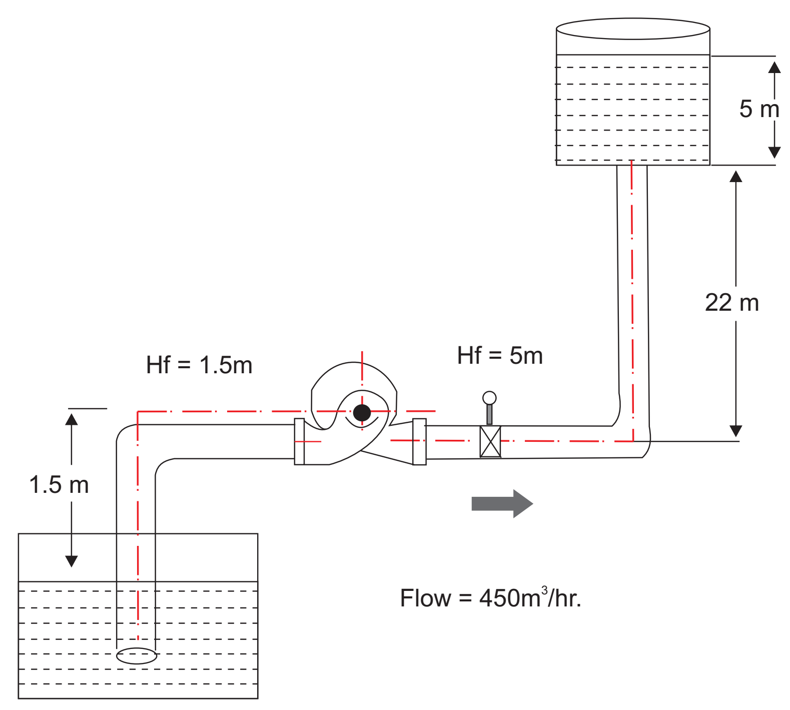
N = Operating speed in rpm

Q = Flow through pump in USGPM (consider 50% flow for double suction impellers)

NPSHr = Net positive suction head required in ft.

It will be seen from the above procedure for establishing maximum operating speed that when available NPSH is low, a pump user has the following options:

- Divide the total capacity required over a larger number of pumps. This will permit individual smaller pumping units to run at a higher and hence more economical speed –



Given vapour pressure of water at the pumping temperature is 0.5m and suction vessel is open to the atmospheric pressure. Loss of head due to friction in the pipe on the suction side, H_f is 1.5m.

We know,

NPSHA = Atmospheric pressure head (m) – Suction lift (m) – Frictional losses in the pipe on the suction side (m) – Vapour pressure (m)

= (10.3 – 1.5 – 1.5 – 0.5) m = 6.8 m

Fig. 1 : Calculation of NPSHA for simple systems

Example of Estimation of Maximum Operating Speed and Determination of Preferred Type of Pump Based on NPSHA

A ship-owner is considering installation of two diesel engine driven external fire pumps on the ship's deck. Pump duty and suction conditions are as follows:

Rated capacity of each pump is 600 m^3/hr and the rated head is 140 m. Static lift (minimum water level to pump centerline) = 3.0 m. Total losses in the pipeline (strainer, bend, straight pipe, etc.) = 0.5 m. Vapour pressure = 0.6 m. Atmospheric pressure = 10.3 m.

Available NPSH = 6.2 m (10.3 - 3.0 - 0.5 - 0.6 m)

The ship owner wants to maintain a safety ratio of 1.2 (NPSHA/NPSHr) to prevent cavitation. What is the maximum speed at which he can run a) an end suction pump b) a double suction pump, considering that pumps operate at BEP for rated duties and that they have been designed for N_{SS} = 9000 US Units.

Solution: Limiting NPSHr = NPSHA/1.2 = 6.2/1.2 = 5.17 m

End suction is a single suction design pump, so

$$N_{\max} = \frac{Nss \times NPSHr^{0.75}}{Q^{0.5}} = \frac{9000 \times (5.1667 \times 3.28)^{0.75}}{(600 \times 4.403)^{0.5}} = 1462\ rpm$$

For double suction design pump

$$N_{\max} = \frac{Nss \times NPSHr^{0.75}}{(Q \times 0.5)^{0.5}} = \frac{9000 \times (5.1667 \times 3.28)^{0.75}}{(600 \times 4.403 \times 0.5)^{0.5}} = 2068 rpm$$

The above example illustrates that when available NPSH is low, a double suction impeller permits operation at a higher speed.

Fig. 2

higher the speed, smaller is the pump size and lower, therefore, is the cost.

- Reduce the pump speed. For example, use a 1000 rpm pump in place of 1500 rpm in a 50 Hz supply system.
- Use a pump with a double entry impeller, such as an axially split case pump in place of a pump with a single entry impeller such as an end suction pump.

An example on the estimation of maximum operating speed and preferred type of pump based on NPSHA is shown in Figure 2.

Margin between NPSHA and NPSHr

NPSHr is a design parameter of the pump and represents the head drop that takes place between the pump suction flange and the impeller eye. The largest part of this head drop occurs when the liquid enters the impeller eye – there is a sudden increase in velocity leading to a consequent fall in the pressure head.

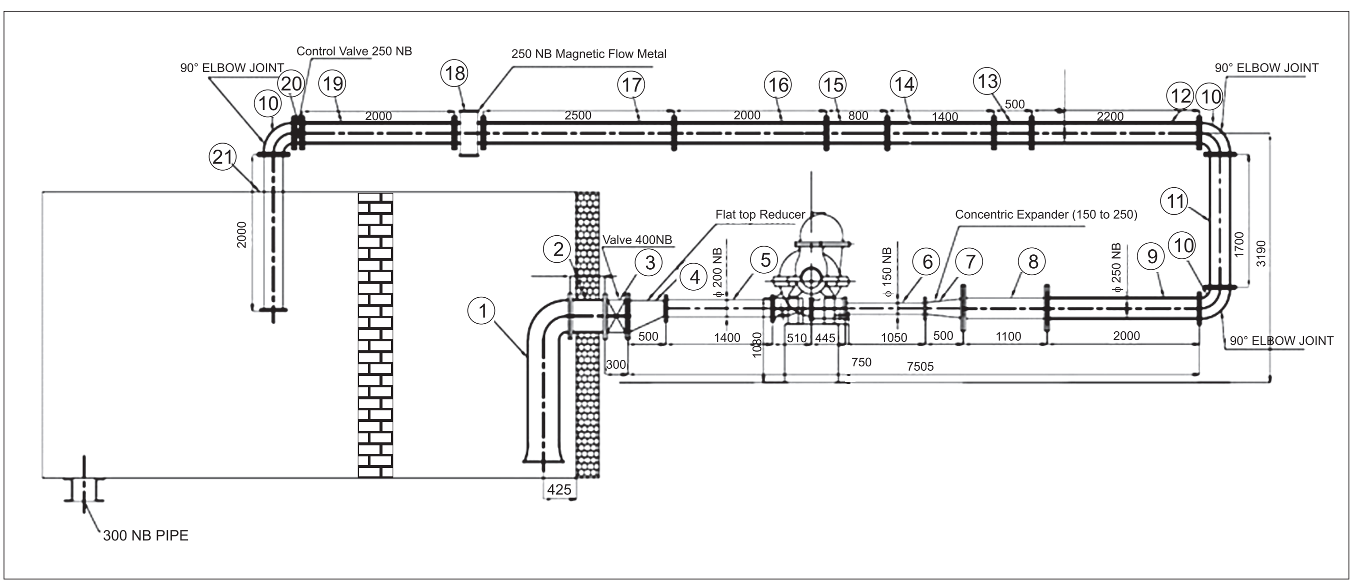


Fig. 3: A Typical Test Setup for NPSH test

If the available NPSH is not adequate, this pressure drop will bring the fluid stream below the vapor pressure leading to formation of vapour filled cavities and their subsequent implosion associated with "cavitation" phenomenon.

NPSH required by a pump, by common agreement, is determined by a 3% head drop NPSH test. There are multiple methods of conducting this test as per ISO 9906 and the Hydraulic Institute Standards.

In the most commonly used method, for a constant capacity, the available NPSH is reduced by suction suppression (throttling, level/pressure variation in the suction tank) till the total head breaks down by 3% due to cavitation.

This value of NPSHA corresponding to a 3% drop in total head is considered to be NPSHr of the pump (sometimes, called NPSH3). The Fig. 3 shows a typical NPSH test setup.

The curve in fig. 4 shows 3% head drop NPSH test at a constant flow rate of 250 m^3/hr with total head of 36.74m. Suction valve was throttled to reduce NPSHA and ten observations were taken till total head broke down due to cavitation.

It can be seen from the above that if the manufacturer does not add any margin to this NPSHr figure for publishing his data (called NPSHr curve) a pump will cavitate when NPSHA = NPSHr. It is, therefore, necessary to ensure that a margin between NPSHA and NPSHr is maintained in order that the pump operates without any cavitation damage.

The best guide in this regard is the Standards of the Hydraulic Institute HI/ANSI 9-6-1. The user may refer to this document to establish the level of NPSHr for a specific installation NPSHA.

NPSHr test results of 5"×6"×13" split case pump at 1500 rpm

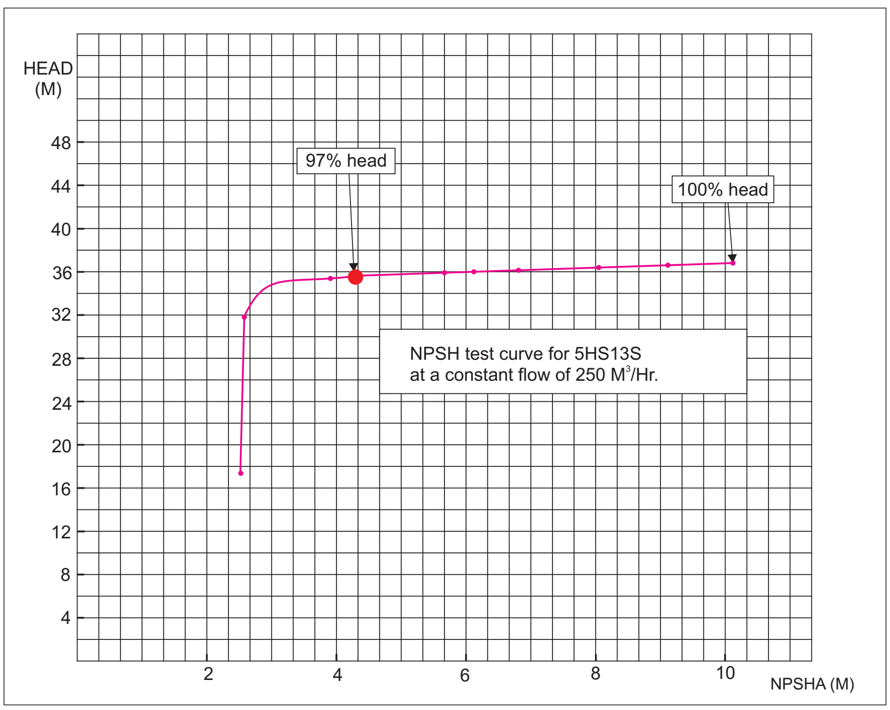


Fig.4

It is important for the pump users to consider the following additional points regarding NPSHA, NPSHr and the margin between the two:

(1) Pump user specifies a low acceptable NPSHr: If the pump user decides to specify a low acceptable NPSHr, the result may be counterproductive. This may lead to the selection of a pump with relatively large eye diameter.

These pumps are prone to recirculation and cavitation during part flow operation. This demand of low NPSHr may also force the pump manufacturer to select a larger pump or a slower speed pump with cost and reliability implications. A pump with low NPSHr does not always signify greater reliability.

(2) NPSHr margin and pump parameters: Several pump parameters are affected by NPSH margin. If the pump is required to operate over a wide range of duties on its H-Q curve (refer to Figure 5 on next page), the margin should be higher to suppress cavitation.

Similarly, if the margin between NPSHA and NPSHr is small, a more cavitation resistant impeller material such as NiAlBronze or stainless steel should be used in preferences to cast iron or bronze.

In general, larger the pump capacity, the larger is the margin needed between NPSHA and NPSHr.

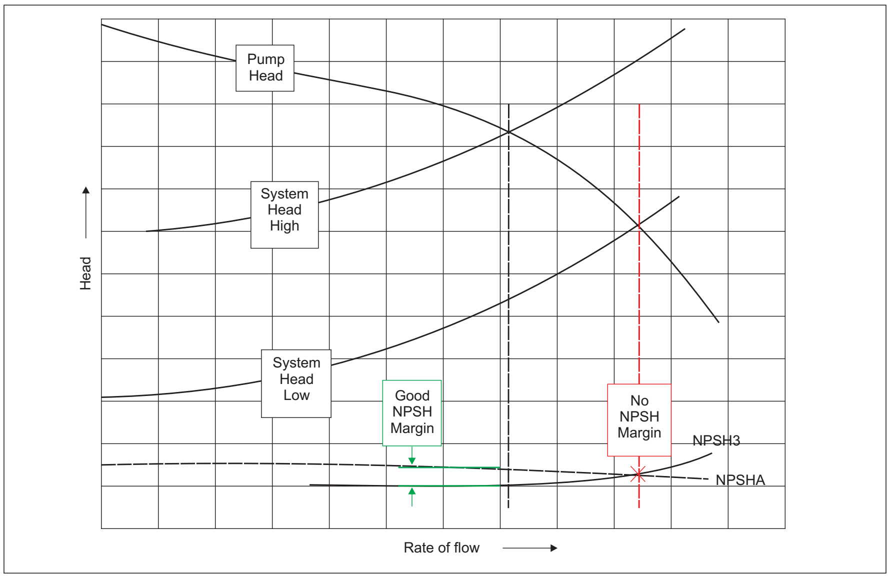


Fig. 5 Wide operating zone requires consideration of NPSH margin across the range

Suspension Piping and Velocities in the Suction Pipe

The underlying principles for all recommendations relating to suction pipes for centrifugal pumps are the following:

- Optimizing NPSHA–Suction pipe should be as short and as direct as possible and should ensure highest possible available NPSH. This influences suction pipe velocity, pipe length, pipe layout and types of fittings used.

- Excluding potential for air/gas accumulation – This requires elimination of all high spots in the suction line (refer to Fig. 6), a suction pipe which rises constantly to the suction flange.

- Elimination of swirls –If bends in the suction pipeline cannot be avoided, the bends should be in one plane only.

  Bends in 3D plane create defined swirls due to superimposition of secondary flows created in the individual bends. Suction pipes branching off a main pipe at right angle create vortices and require special consideration.

Hydraulic Institute Standard HI/ANSI9-6-6 provides comprehensive guidelines for suction piping and by following its recommendations the user can avoid problems arising out of the hydraulic issues mentioned above.

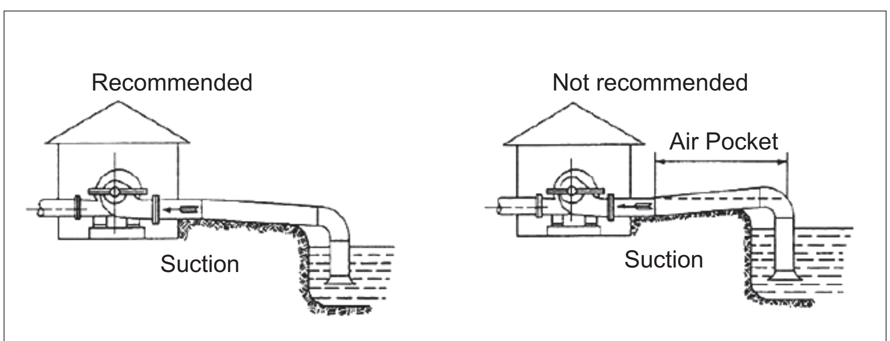


Fig. 6

On the mechanical side it is important to ensure that the inlet pipes do not impose any excessive loads on the pump flange due to piping misalignment and inadequately supported valves or suction fittings.

A few important recommendations of the Hydraulic Institute document are as follows:

- The suction piping should be as short and as direct as possible. If the pipe sizes are changing, the transition should be gradual to eliminate chances of vortex formation. The velocity in the suction piping should be constant or increasing as the flow approaches the pump.

- The suction pipe shall be at least as large as the pump suction nozzle. Valves and other flow-disturbing fittings located in the pump inlet piping should be at least one pipe size larger than the pump inlet nozzle.

- The maximum velocity at any point in the inlet piping is 2.4 m/s according to the Hydraulic Institute Standards. This velocity limit does not apply to the straight run of pipe immediately upstream of the pump suction flange provided this pipe is the same diameter as the pump suction flange and directly connected to the pump. Other references including those from highly regarded pump makers put this velocity limit to 4 m/s. It is, therefore, recommended that suction pipe velocities above 2.4 m/s should be evaluated with respect to flow distribution, erosion, NPSH, noise, water hammer, and the manufacturer's recommendations.

Before concluding this segment, it is important to point out a fallacy one notices in some purchase specifications of pumps. These specifications put a limiting value on the nominal suction bore size with a view to restricting the velocity at the suction flange to 4 m/sec. This is counterproductive and often results in sub-optimal pump selection.

In all centrifugal pumps suction volutes are designed such that the liquid accelerates near the impeller eye. Nominal bore of the suction is, therefore, quite intimately connected with the impeller eye diameter and the velocity at the suction flange can easily exceed 4 m/sec limit imposed by these specifications.

Air Entrainment

Centrifugal pumps suffer from performance degradation when handling liquid with entrained air. In the low-pressure zone at the pump impeller eye, air bubbles get trapped while the heavier liquid is thrown out. These air bubbles block the fluid flow, and the pump performance declines. This reduction in flow rate is accompanied by a drop in the total head and pump efficiency.

Air entrainment even at values as low as 2 to 4 percent causes increased pump vibration, which leads directly to premature bearing and seal failure. Partial air blockage of the impeller causes unbalanced hydraulic loads resulting in increased vibration. Air entrainment is one of the major contributors to broken pump shaft because of hydraulic surging & axial shuttling.

The unvented air also collects in the seal chamber to create air pockets that cause dry running of the mechanical seals. Dry or non-lubricated seal face operation contributes to shortened life and premature failure. Finally, air entrainment also introduces undesirable free oxygen into the system and this is a common contributor to general and stress corrosion.

While air is sometimes introduced into the pump suction deliberately to act as a cushion for collapsing vapour bubbles when a pump is cavitating, this measure is at best a temporary solution to mitigate the damaging effect of cavitation, since any introduction of air causes the pump performance to fall, often unpredictably.

More than 1.5 to 2 percent air in the pumped liquid leads to both hydraulic performance degradation and, on a longer term basis, to mechanical problems.

Some studies point out that air entrainment at 2 percent reduces the pump performance by 12 percent. At 4 percent, it will reduce the pump performance by 40 percent and at 10 percent, it is likely to stall the pump altogether. Pumps with open impellers are known to perform better when handling liquid with entrained air.

Effects of Air or Gas in the Liquid: Why Pump Performance Falls?

Centrifugal pumps operate on the principle of forced vortex. In achieving this, a pressure differential is created across each impeller blade. Pump designers attempt to increase this differential to a moderately high value in order to achieve an optimally sized pump.

Often this results in significant pressure gradients across the passages of an impeller, at right angles to the flow passage.

The lowest pressures usually occur on the suction (concave or visible) surface of traditional designs. Here, pressures may actually become low enough for

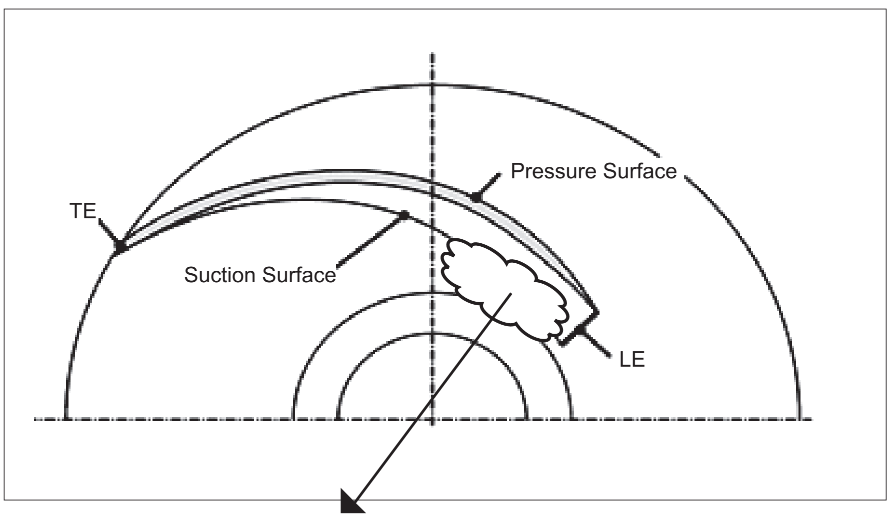


Fig. 7: Gas tends to accumulate on to the low pressure (or suction) surface of the impeller blade

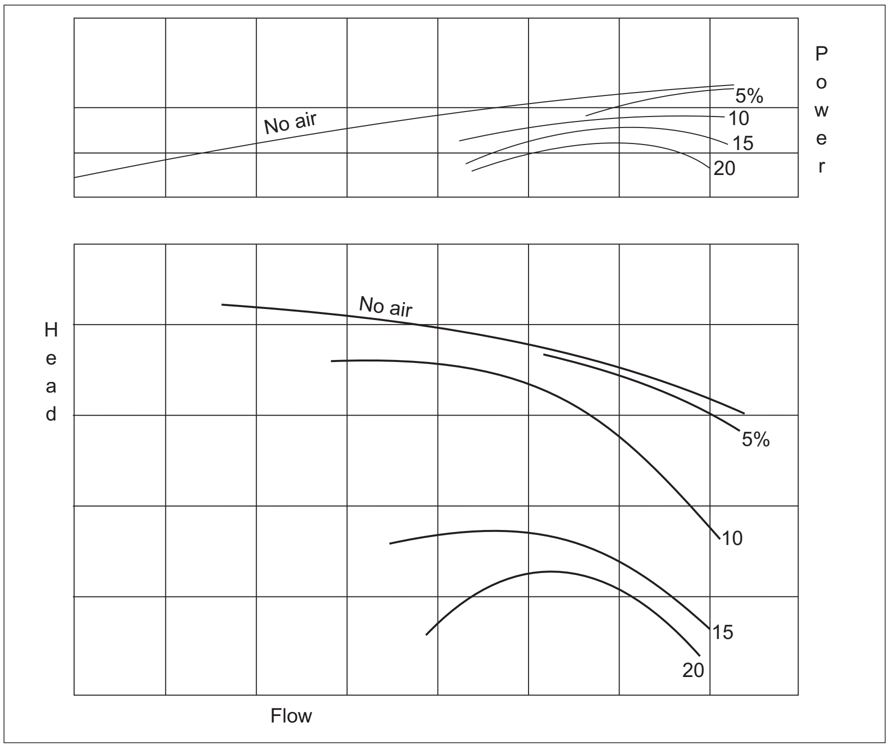


Fig. 8: Effect of air entrainment on the pump characteristics

dissolved gas to come out and collect. Additionally, any free gas will also collect in these low pressure areas.

Gas accumulation causes the passage areas to become partially blocked. This restricts the flow capability of the pump relative to its single-phase liquid performance.

Also, since free gas modifies the passage velocity distribution, the head generated by the pump is reduced- Figure 7.

The decline in head becomes more pronounced when the pump is operated at part capacity. At part flow condition, there are several areas of high pressure gradients across the blade passages.

The resulting high gas concentration seriously inhibits the impeller's ability to generate and sustain the forced vortex, and so the generated head is significantly reduced. Fig. 8 illustrates the behavior of the pumps at part flow condition when the air entrainment is greater than 10%.

The Most Common Reasons of Air Entrainment in Pump Systems are as Follows:

- Inadequate submergence of the suction pipe in the sump causes air entraining vortices to be drawn into the pump. Vortexing is the most common source of air entrainment in the pump.
- The pipe & tanks were full of air to begin with and not properly filled and/or vented during the startup process.
- Liquids being directed to the suction supply tank/sump are "free falling" onto the fluid surface as shown in Figure 9. This action drags air into the pump.
- Suction systems are frequently operated at pressures below atmospheric pressure which can cause air to leak at several places. Most common examples are packed gland pumps, loose or corroded joints on the suction side, improper gaskets and pumps working on suction lift services.
- Air or non-condensable gases are purposely injected into the fluid as a part of the process – paper industry is a common example.

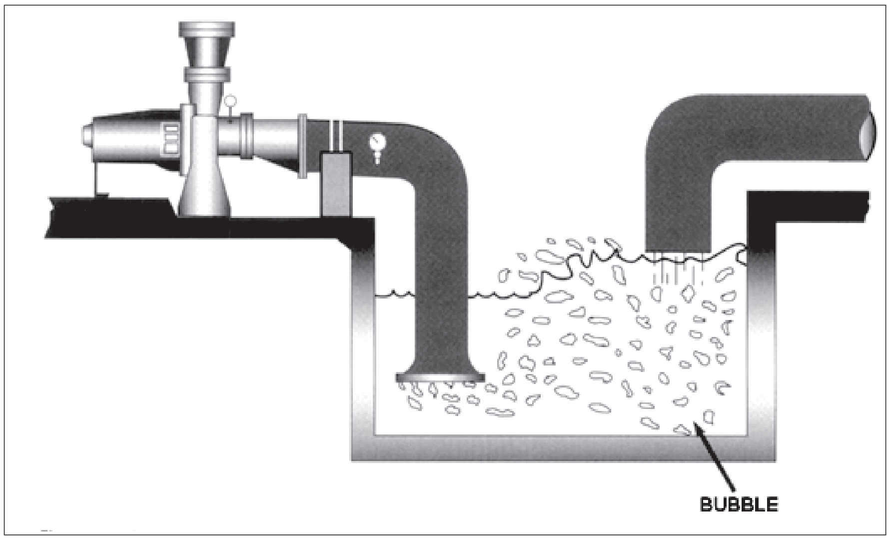


Fig. 9: Inadequate sump design leads to entrained air bubbles and turbulence

Example of the Effect of Air Entrainment During Test of a Pump Working on Suction Lift

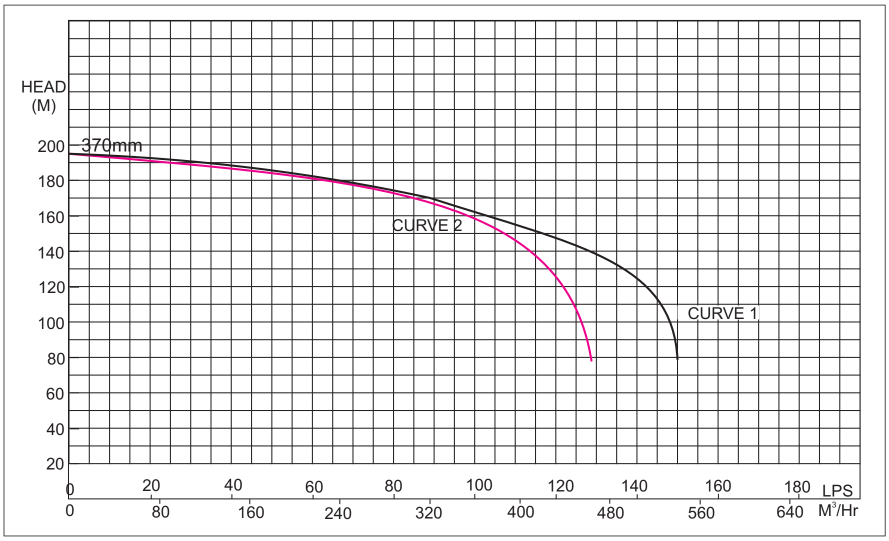


Fig. 10: Curve 1 shows the performance of the 1000 GPM pump with positive head on the suction side at all flows; Curve 2 shows the performance curve when a constant 15 ft. suction lift is maintained on the suction side throughout the entire test.

- The process itself creates non-condensable gases (Chemical reaction).
- Water has some natural air entrainment.
- Improper design of suction pipe geometry/arrangement with unvented high points that will not dissipate or reduce the air & gases present.
- Temperature changes also affect the solubility of air and gases in the fluid.
- Pressure at the suction eye of the pump impeller is the lowest pressure in the system that the pumped fluid experiences. The air bubbles in the solution increases in size in proportion to this pressure drop.

Suction Sump

The last element in the suction arrangement is the suction sump. The design of approach flow to the suction sump, proportions of the sump itself and the submergence of the pump or the suction pipe in the sump, all play major roles in ensuring reliable pump operation.

The importance of the sump design in ensuring present and future reliability of the pumping plant must be fully recognized by the pump users. It is not an overstatement to stress that as the pump size becomes larger the sump increasingly becomes a part of the pump itself – any disturbance in the sump is carried into the pump with detrimental effects on performance and reliability.

As explained in the earlier sections, the fluid flow into the pump suction should be steady, uniform, devoid of swirls and entrained air. Non-uniform flow into the pump can force a pump to operate at off-design condition and with low operating efficiency. When the flow is unsteady, the impeller is subjected to a fluctuating load and this leads to vibration, and consequent seal and bearing problems. Vortices in the pump intake can cause significant change in the hydraulic performances of the pump including its head-capacity characteristics and efficiency.

Localised vortices cause pressure reduction and generation of air core at their centre. When these air entraining vortices enter the pump, they cause reduction in pump capacity and fluctuations of impeller load. Consequent vibration may damage the pump components and compromise operating reliability of the system upstream of the pump.

In view of the importance of the pump intake there have been extensive theoretical and experimental work carried out in this field by institutions such as British Hydrodynamic Research Association. A contemporary guideline is provided by the Standards of the Hydraulic Institute (ANSI/HI 9.8-2012) and the pump users are advised to refer to this document for design guideline for their pump intake.

Depending on the size and the complexity of the pumping installation, a pump user can usually adapt one of the following three approaches while designing the suction sump:

- Design by convention: This can be done for smaller pumps up to 150 mm discharge branch size using design conventions well established by the past practices of the user organization.

- Design by codes or standards: For larger installation with pump flows in excess of 5,000 gpm (1135 m^3/hr approx.), it is suggested that the recommendation of the Hydraulic Institute are followed as closely as possible.

- Design by modeling and/or analysis: Finally, for larger pumps (vertical wet pit power station circulating water pumps, for example) the user may commission a model study by a hydraulic laboratory with adequate experiences of studying intakes and suction sumps by hydraulic modeling. In a typical model study a scale model of the forebay and sump are fabricated with Reynolds number and Froude number identical to the full scale sump, to study the flow pattern and to recommend positions and orientations of flow passages, walls, etc. In the recent times the use of computational fluid dynamics (CFD) has increased quite considerably to provide analytical support for validating designs of pump sumps. While CFD codes have become quite powerful as predictive tools, their use is still supplemented by a full sump model study in case of large pumping stations.

Conclusions

Design of the suction sump and suction piping greatly affect the reliability of a centrifugal pump installation as do the selection of pump speed and pump type based on the suction configuration. Irrespective of the pump size, elements on the suction side of the pump should, therefore, be carefully considered by the pump users, taking help of he established guidelines.

Major objectives of the sump and suction piping design should be to keep the hydraulic losses on the suction side as low as possible, to exclude possibilities of air entrainment and to ensure smooth, uniform and swirl-free flow into the pump suction. Vertical wet pit pumps of the mixed flow type and double suction split case pumps are more sensitive to sump configuration compared to smaller centrifugal pumps of other types. When large pumps of these types are used for duties such as urban water supply or cooling tower applications in thermal power plants, sump design should be validated both by hydraulic model study and by CFD analysis.

Fig. 11: Suction Layout

Case Study : High Vibration and Cavitation Noise in Cooling Water Pumps in a Steel Plant

Introduction

The following case is based on an investigation of premature failure of cooling water pumps in a steel plant.

Pump Size : discharge : 350 mm, Suction : 450 mm, Impeller Diameter : 475 mm

Design duty : 1800 m^3/hr, 60m @1480 rpm
Rated duty : 1500 m^3/hr, 60m
Actual Flow at present - 1160 m^3/hr

Main Problems: High vibration and cavitation noise

Observations

Sharp bend at the pump suction parallel to the pump shaft. Pump operates at part capacity.

Effects

- Axial Thrust
- Unequal flow and flow disturbances
- Increased vibration
- Part flow operation leading to recirculation

Explanation

When a bend in the suction line is located asymmetrically relative to the suction of the pump

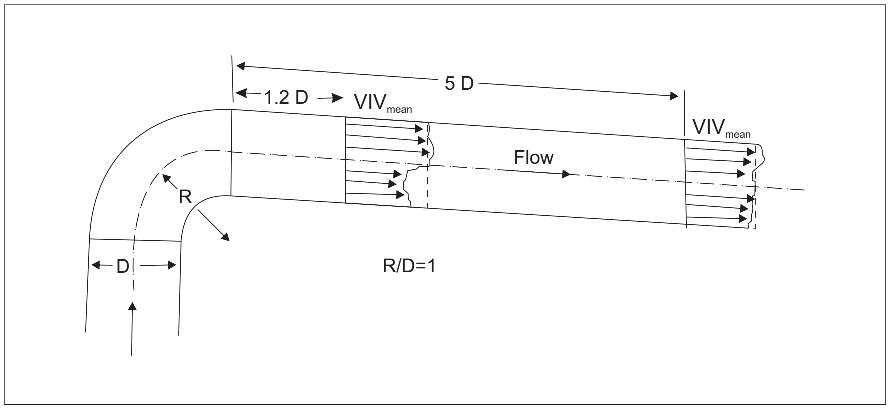


Fig. 12

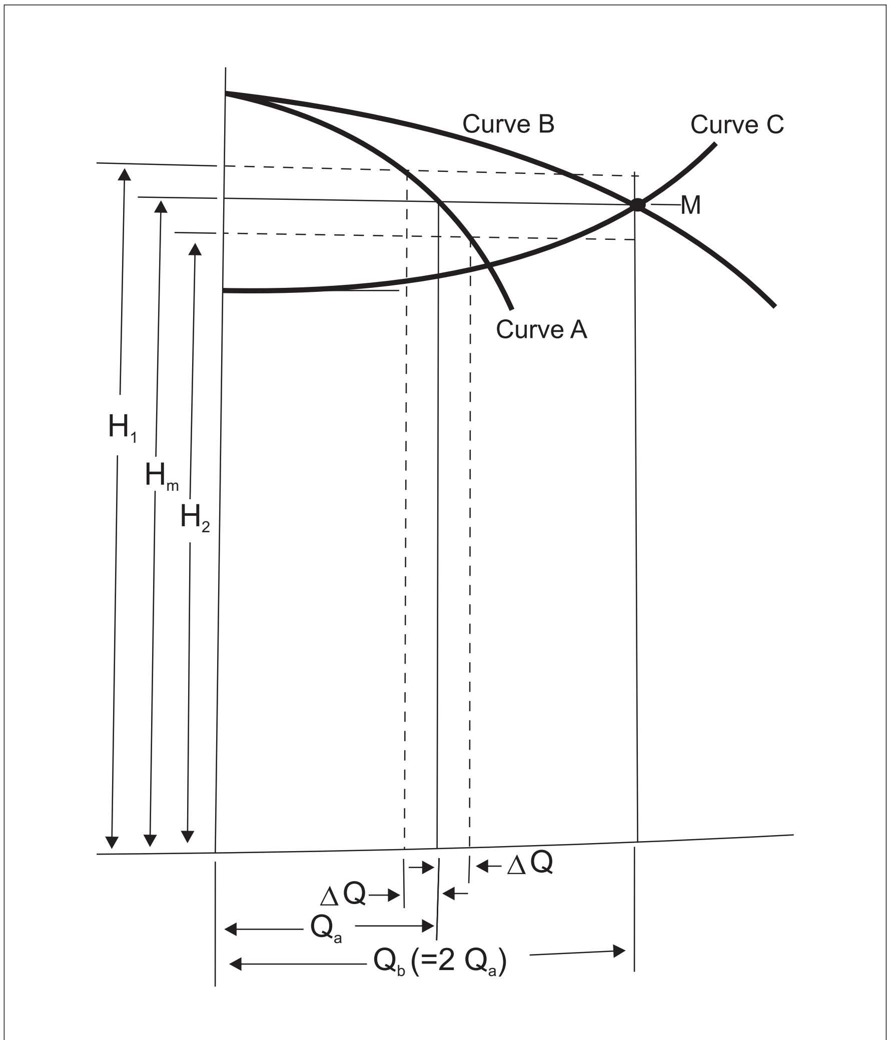


Fig. 13

distribution of the flow between the two halves of a double entry impeller will be uneven. This occurs when a uniform flow passes through a single bend, because it exits with a non-uniform velocity distribution as shown in the Figure 12. As a result of this distortion of the uniformity of flow, each suction inlet of the impeller receives a different flow rate.

A double-suction pump can be considered to consist of a combination of two identical, single-suction impellers operating in parallel and discharging into a common casing. Let curve A in Figure 13 represent the Q-H characteristics of each of the single suction impellers.

Because of their parallel operation, the pump delivers a resultant flow rate equal to the sum of all the flow rates that each of these impellers delivers against the given head. The actual Q-H curve of the pump shows a flow rate equal to twice the flow rate delivered by each of the given head. This is shown as curve B.

We have also shown a curve C which represents the resistance of the pumping system to a given flow rate. The actual flow against which the pump operates is at the intersection M of curve B & C. It is equal to the flow Qb = 2Qa.

This means that a bend immediately upstream of the suction inlet and parallel to the shaft of a double-suction pump causes an uneven distribution of the flows entering each side of the impeller. One side of the partial impeller delivers a flow rate equal to Qa-δQa against head H1.

The other side of the impeller delivers a flow rate Qa+δQa against a head H2. At the impeller outlet both flows encounter the same resistance Hm which has been determined by the intersection of curves B & C.At the impeller outlet, the flow rate Qa-δQa encounters a sudden drop in pressure from H1 to Hm. This effect is similar to that of a sudden breakdown in the discharge line. A sudden acceleration of the flow occurs, resulting in cavitation

Typical Problems Related to Suction Side of Centrifugal Pumps

Symptoms of Pump Distress	Pump "Suction" Related Issues
Pump does not discharge or discharge is not up to expectation; Irregular discharge	The pump suction line is not completely filled with liquid
	Air/gas pockets in the suction due to high points in the suction piping
	Faulty foot valve
	Clogged strainer
	Air leakage into the suction line or through stuffing box
	Insufficient submergence of the suction pipe in the suction sump, Vortexing
Insufficient discharge pressure	Air/gas entrainment in the liquid pumped
	Air leakage into the pump suction
Short mechanical seal life	Inadequate NPSHa/NPSHr margin leading to cavitation
	Unequal flow into two eyes of a double suction impeller due to faulty suction piping creates axial imbalance and affects seal life
	Casing distortion due to piping strain / inadequately supported suction pipe
Noise and vibration	Pump cavitates due to inadequate NPSHa/NPSHr margin
	Air/gas entrainment
	Suction valve partially closed
Pump loses liquid after starting	Suction line not fully primed; vapor lock in the suction line
	Inlet of suction pipe insufficiently submerged
	Air leaks into the suction line or through stuffing box
	Seal water line to stuffing box is clogged

upstream of the corresponding partial impeller. The flow rate Qa+δQa, encounters a sudden increase in the resistance to flow, from H2 to Hm. This has an effect equivalent to a sudden shut off of a discharge valve, and strong water hammer may result and the pump may operate with high vibration.

Recirculation

The pump is operating at part flow i.e. 1160m³/hr which is 64% of the BEP flow. According to ANSI HI 9.6.3-1997, if the specific speed is below 4500 US units the preferred operating region lies between 70% and 120% of the BEP. Otherwise flow reversal takes place leading to cavitation and vibration.

Recommendations

Following solutions were recommended:

- Providing flow guides in the suction bend to ensure equal flow to both the impeller eyes.
- Operating the pump closer to its rated duty and best efficiency point to avoid recirculation flow and cavitation.
- If recommendation 2 could not be implemented because of system constraints, it would be beneficial to use a low capacity impeller with BEP close to the current operating flow of 1160m³/hr.

The customers implemented the first and third recommendations with successful results.

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