

Cavitation Effects in Centrifugal Pumps- A Review

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ABSTRACT

Cavitation is one of the most challenging fluid flow abnormalities leading to detrimental effects on both the centrifugal pump flow behaviors and physical characteristics. Centrifugal pumps' most low pressure zones are the first cavitation victims, where cavitation manifests itself in form of pitting on the pump internal solid walls, accompanied by noise and vibration, all leading to the pump hydraulic performance degradation. In the present article, a general description of centrifugal pump performance and related parameters is presented. Based on the literature survey, some light were shed on fundamental cavitation features; where different aspects relating to cavitation in centrifugal pumps were briefly discussed.

Keywords: Cavitation, Centrifugal pump, Performance,

I. INTRODUCTION

1.1. Centrifugal pumps

The Centrifugal pump, as any other dynamic pump, is used to move the fluid from one point to another in a system, by simply adding momentum to it. It is mainly composed of two important parts which are: 1) The impeller and 2) the volute casing. The fluid axially enters the impeller eye where it is spread radially towards inter-blades flow areas, and gets tangentially whirled by the effect of the rotating impellers vanes. The fluid momentum is increased while passing between the fastly rotating impeller blades, until it reaches the impeller exit zone whereby, by means of the in-place pump volute, the acquired fluid high velocity is converted to a pressure increase enough to overcome the required Head. The impeller blades rotating motion, continuously creates the vacuum at the impeller eye, which then results in a continuous suction of the fluid from the inlet pipe towards the impeller eye. In other words, centrifugal pumps add energy to the flowing fluid striving to get to the destination point. The added energy is a result of power conversion from the engine driven drive shaft's mechanical power to the fluid hydraulic power.

Because of different losses within the pump or outside during the energy transfer process, such as disc friction, shock losses, mixing, change in direction of fluid, separation, bearing losses, turbulence, and leakage losses, the fluid-acquired hydraulic energy (water horsepower) is always smaller than the shaft transmitted energy (Brake horsepower).

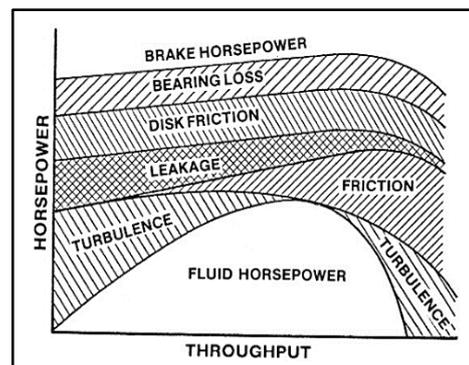


Figure1. Power losses in centrifugal pumps

This difference brings an idea of centrifugal pump performance in terms of Efficiency as shown by the here down shown 2D formulas.

$$\eta = \frac{P_w}{bhp} = \frac{\rho g Q H}{\omega T} \quad (1)$$

Where η , ρ , g , Q , H , ω , T , and P_w stand for pump efficiency, density, gravity acceleration, flow rate, head, rotational speed, and hydraulic Energy respectively. Considering the pump efficiency as a unity ($\eta=1$), the theoretical head H linearly varies with the pump discharge capacity Q as follows:

$$H \approx \frac{u_2^2}{g} - \frac{u_2 \cot \beta_2}{2\pi r_2 b_2 g} Q \quad (2)$$

Where u_2^2/g is the pump shutoff head value.

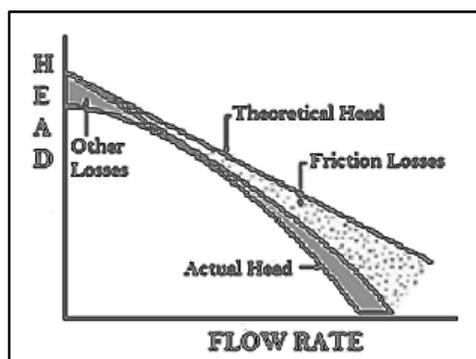


Figure 2. Head losses in centrifugal pump

In practice, the pump efficiency is never a unity; it will always fall below a unity. For instance the measured shutoff centrifugal pump head usually comes to 60% or below the theoretical value [1]. Due to the complexity of centrifugal pump fluid flow, it is hard to accurately predict its performance parameters on a theoretical basis. Designers have rather opted to experimentally measure these parameters through tests, where the resulting pump characteristics have been presented in form of graphs. The experimentally found actual developed head h_a , the fluid gained power P_f , and the overall efficiency are here down presented.

$$h_a \approx \frac{P_2 - P_1}{\gamma} \quad P_f = \gamma Q h_a \quad \eta = \frac{\gamma Q h_a}{bhp} \quad (3)$$

Pump performance parameters are usually given in form of graphs where the pump brake horsepower, the developed head, and the efficiency are plotted against the delivered capacity Q .

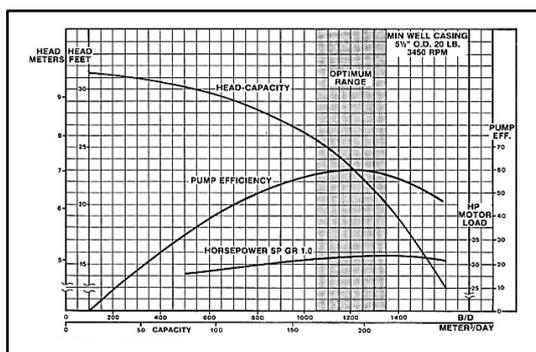


Figure 3. Pump performance curve

These graphs are generally called Pump performance curves. The points from other curves but corresponding to the maximum efficiency value Q_{max} , are generally called the Best Efficiency points BEP, and it would be better for a pump to work at the vicinity of the BEP. Performance curves may also be presented in a different form depicting the pump behaviors at different impeller diameters or rotational speeds as most of centrifugal pumps can both

accommodate different impeller sizes and work under different speeds as shown in figure.

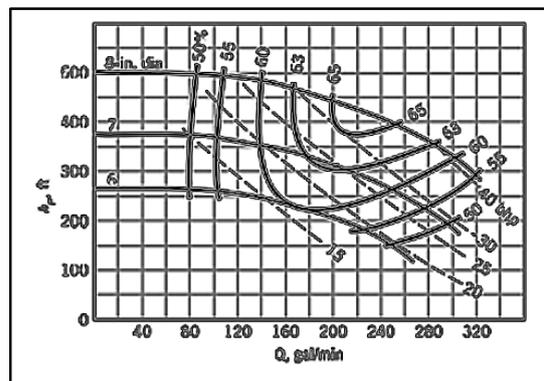


Figure 4. Pump performance curve (2nd form)

However, the centrifugal pump performance also depends on some design parameters such as the Blade number, blade geometry and the casing shape and size; and operational parameters like the impeller rotational speed and the pump flow rate. For instance, the impeller inlet design is mostly of crucial importance. Different researchers, Qing Zhang et Al¹⁰ [2014] and Xian Luo et Al¹¹ [2008] among others, through numerical and experimental methods, carried out studies on the impact of impeller inlet geometry on pump performance, where different design parameters such as the impeller inlet diameter, blade inlet angle, and inlet blade thickness, showed a great importance in terms of pump performance improvement.

Another currently used method for centrifugal pump performance prediction is the Computational Fluid Dynamics usually denoted as CFD. Computational fluid dynamics (CFD) is the use of applied mathematics, physics and computational software to visualize how a liquid flows as well as how the liquid affects objects as it flows past. Computational fluid dynamics is based on the Navier-Stokes equations. These equations describe how the velocity, pressure, temperature, and density of a moving fluid are related [2]. Computers are used to perform the calculations required to simulate the interaction of liquids with surfaces defined by the pump boundary conditions [3]. Researchers currently prefer this method due to its different advantages, the high accuracy, time and resources saving and fluid flow visual ability among others. Due to the development of CFD code, one can get the efficiency value as well as observing the equipment actual flow conditions.

Sujoy C.⁴ [2012] carried out the centrifugal pump numerical investigation of the effect of blade number and impeller speed on the pump performance. Using the CFD code of the commercial software Ansys Fluent 6.3, pumps of different impeller blades numbers from 4 to 12 at different speeds (2900, 3300,

and 3700 rpm) were simulated, only to finally find that the pump head had been increasing with the increase in both the blades number and pump speed, whereas the efficiency was a bit complex with the 10 bladed impeller as the one with higher efficiency.

Pandey K.M.⁵ [2012] studied the influence of blades number variation on the pump performance keeping the impeller diameter and speed constant for all models. Using the standard k-ε turbulence model and SIMPLC algorithm to solve the RANS equations in Ansys Fluent software; the flow through impellers of six, eight, and nine blades at the speed of 2500 rpm was studied. Both the pump head and static pressure increased with the blade number but the efficiency variation was complicated with specific optimum values for each model.

Houlin L.⁶ [2010] also studied the influence of the impeller blades number on the pump performance through numerical simulation with Ansys Fluent, and experiments. Through a comparative scheme between the two methods, the flow through the model pumps with 4, 5, 6, and 7 blades at a specific speed of 92.7 was studied. The maximum deviation in the prediction results for the head, net positive suction head, and efficiency were 4.83%, 0.36m, and 3.9% respectively. The developed head increased with the blades number while the efficiency was complicated.

Ahmed A.⁷ [2014] carried out a numerical simulation using CFD to investigate the effect of volute geometry on the pump performance. Simulation results were compared to the experimental data where finally the agreement between both methods results was noticed. The pump head was found to increase with pump volute cross-sectional area. Patel M.G.⁸ [2013] used three pumps models at different specific speeds to investigate the effect of impeller blade exit angle on centrifugal pump performance. The Gulich mathematical model was first validated with the manufacturer's H-Q curve before being used in their research. After validation, this method helped find out that both the head and pump efficiency increased with the impeller exit angle.

1.2. Cavitation

Cavitation is defined as the process of formation of the vapor phase in a liquid when it is subjected to reduced pressures at constant ambient temperature [9]. When the fluid vapor drops below its vapor pressure (P_v), there is formation of gas bubbles which, with the continuous pressure drop grow progressively and finally collapse when they get to higher pressure zones, the phenomena which is generally called "cavitation".

From the continuum mechanics view angle, cavitation can be defined as the breaking of a fluid medium under excessive stress or tension [12]. As

shown in the Thermodynamic diagram of Phase change, there is a big difference in driving mechanism between cavitation and boiling.

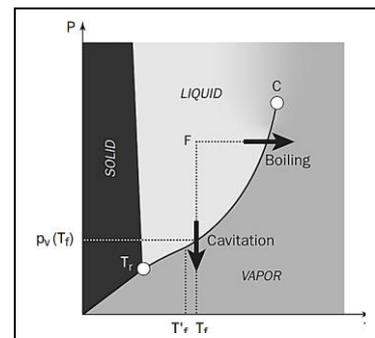


Figure5. Thermodynamic diagram of Phase change

Boiling is the bubble formation in the liquid as a result of the increase of the fluid temperature to specific values at a constant pressure whereas cavitation is now a result of the fluid pressure decrease under its vapor pressure at an approximately constant temperature.

1.2.1. Typical cavitation occurrence situations

Here down are some of situations where cavitation phenomenon is more likely to occur in the fluid flow.

- A. The wall geometry at the fluid flow zones imposes high velocities and a subsequent pressure drop due gradual decrease in flow area. This is the case of flow constriction in fluid ducts and pump impellers.
- B. The shearing between two neighboring flows having very different velocities entails large turbulent fluctuations of the pressure: that is the case for jets and wakes. [12]
- C. The high unsteadiness character of certain flows which as a result increase the fluid flow temporal acceleration terms, thus pressure drop in the fluid flow.
- D. The roughness at the fluid flow wall boundary, which results in formation of wakes and subsequent attached cavities.
- E. Presence of water in interstices due the inaccurate joining between two or more pieces in a mechanical system. There is high probability of cavitation occurrence when the interstice walls start to move.
- F. The influence of moving solid bodies having sharp edges immersed in the liquid. The acquired fluid acceleration at these edges produces a pressure drop, thus high chances of cavitation occurrence.

1.2.2. Cavitation types

The cavitation phenomenon is generally characterized by the formation and growth of vapor bubbles in the fluid flow. The fluid-vapor interface can be under different shapes on which cavitation

classification is based. Different cavitation patterns are here down presented.

A. *Traveling cavitation*

In this type of cavitation, the micro bubbles otherwise called cavitation nuclei are carried along the flow field until they get to the flow's lower pressure zones, where they become macroscopic cavitation bubbles before collapsing at pressure recovery zones. The developed bubbles are usually in complex shapes mainly from their interactions with neighboring walls or other bubbles

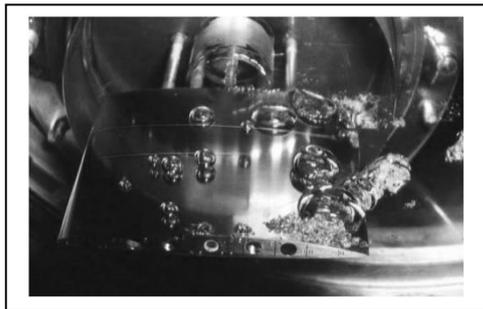


Figure6. Traveling Bubble Cavitation on the Hydrofoil Suction Side

B. *Attached cavitation*

Contrary to the above presented traveling cavitation, the attached cavitation stays at the same location attached on a wall. This does not mean that the flow is steady. Actually, cavitation is almost always the source of unsteadiness which may be quite strong. In the down presented case of attached cavitation, the main source of unsteadiness originated in the rear part of the cavity [13].



Figure7. Leading Edge Cavity on a Foil Suction Side

C. *Vortex Cavitation*

This type of cavitation is the mostly found in marine propellers. It is found at the vortex core generated by the secondary flow at the blade tip. The blade tip second flow is a result of the pressure difference between the vane pressure and suction side. The pressure at the generated tip vortex is much lower than pressure far away in the same fluid making it vulnerable to cavitation phenomenon.

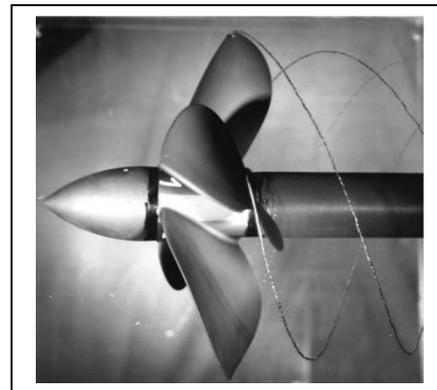


Figure8. Vortex Cavitation Generated by a Propeller (Courtesy of DGA/BEC)

D. *Shear Cavitation*

Such a cavitation is observed in the wake of bluff bodies or in submerged liquid jets; Figure below presents a typical image of cavitation in the wake of a wedge.

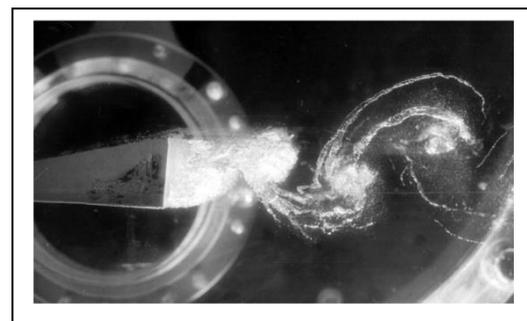


Figure9. An Example of Cavitation in the Wake of a Bluff Body

1.2.4. Cavitation mechanism

A. *Fluid nucleation*

Nucleation is a physical process in which a change of state for example, liquid to solid, occurs in a substance around certain focal weak points, known as nuclei [14].

Some of conditions which cause the fluid nucleation are: 1) Formation of gaps between the liquid molecules which is called "Homogeneous nucleation". 2) Formation of gaps between container and the liquid's molecules resulting in "Heterogeneous nucleation". 3) Presence of micro bubbles in the liquid, which eventually grow to macro sizes with the increase of the applied tensions. 4) Development of bubbles in the fluid from the outside radiations.

B. *Cavitation inception*

Cavitation inception is all about how close is the fluid pressure to its vapor pressure. This closeness is mostly measured through a constant called

“cavitation number”. When the cavitation number is reduced to certain specific values, bubbles start to form and the related cavitation number is termed as inception cavitation number (σ_i). Further decrease below this value results in the increase in bubble number and size.

C. Bubble growth

The single bubble growth to its maximum radius (size) is governed by the Rayleigh-Plesset equation or its variations. This equation gives out the relationship between the bubble radius variation and the surrounding fluid pressure.

$$\frac{P_B(t) - P_\infty(t)}{\rho_L} = R \frac{d^2R}{dt^2} + \frac{3}{2} \left(\frac{dR}{dt} \right)^2 + \frac{4V_L}{R} \frac{dR}{dt} + \frac{2S}{\rho_L R} \quad (4)$$

This equation is based on assumptions like spherical bubble symmetry and the absence of all thermal effects.

D. Bubble collapse

At the last stage of bubble collapse, the bubble is generally more unstable; which results in a development of a re-entrant jet.

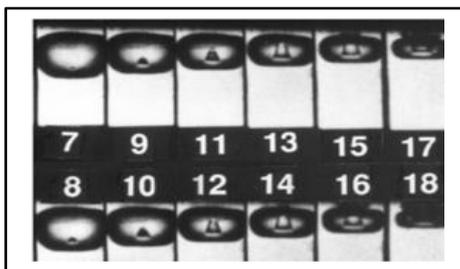


Figure10. Steps of the re-entrant jet development in a collapsing bubble

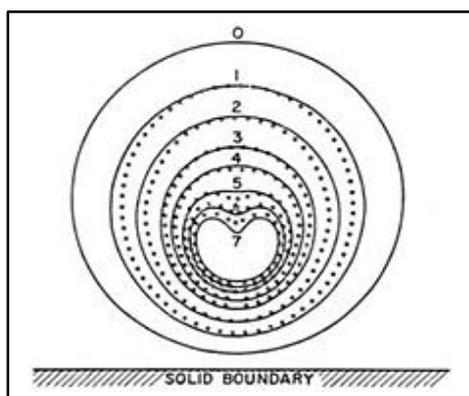


Figure11. Collapsing bubble shape by Plesset (solid lines) and Lauterborn (points)

When it is situated closer to the solid wall, the re-entrant jet is formed from the bubble’s furthest side from the wall, oriented inwards, towards the solid boundary and hit the wall with very high speeds enough to remove the material. Figure11 compares re-entrant jets as measured by Lauterborn and Bolle³⁸

[1975] and Plesset and Chapman³⁹ [1971]. On the other side, when the bubble is situated near the fluid free surface, the generated re-entrant jet is oriented away from the free surface.

1.2.5. Cavitation damage and its basic principles

Cavitation damage is the most recognized cavitation detrimental effect. It is known to remove materials from the flow boundary surfaces however hard or tough the material can be.



Figure12. Cavitation erosion in a centrifugal pump impeller

Solid materials like metals, glass, rubber, quartz, and plastic, can all be damaged by cavitation. Cavitation damage is mostly manifested from four basic principles [15, 16]: 1. The smaller the molecular size of the liquid is and the lower the viscosity, the easier it is for the liquid to penetrate into the surface pores of the metal, thus the penetration of water into the surface of the metal is deeper than in the case of oils. 2. The greater the pressure, the deeper and quicker the liquid penetrates into the pores of the material. 3. The smaller the area of the pore the greater the pressure produced when the vapor bubbles collapse. 4. The higher the frequency of vibration, the more intensive is the destruction of the surface layer of the metal.

1.2.6. Cavitation detection

Cavitation is a very high speed phenomenon at a point of not being easily perceived by a human eye; this has been complicating the adequate understanding about its mechanism. The individual process happens very fast that none of its stages (bubble formation, growth, and collapse) can be detected unless sophisticated devices and methods are used. Moreover, cavitation occurs in mostly hidden places where it is very difficult to find a way in, thus it’s practically inaccessible unless special technics are used. As a result, most of the early cavitation studies have been focusing on its theoretical and numerical details rather than observational ones. Presently, various cavitation detection methods have been developed where a big number of them are mainly founded on cavitation indirect observation; some of them are: Indirect observation by determining the effect of cavitation on the performance of a piece of equipment, Indirect observation by measuring the

effect of cavitation on the distribution of pressure over the boundary at which cavitation occurs, Indirect observation by sensing the noise emitted by cavitation[17], Indirect observation by allowing cavitation to scatter laser-beam light into a photocell[15], Cavitation susceptibility meters[18][19][20], and Direct observation by visual and photographic means[21][22]. Each of the above mentioned cavitation detection methods, have their advantages and their drawbacks, depending on how effective they can be during cavitation measurement.

II. CAVITATION IN CENTRIFUGAL PUMPS

2.1. Theory

In the suction zone of a centrifugal pump the rotational effect of the blades is increasingly imposed on the liquid as it nears the impeller. This gives rise to the tangential velocity component which together with the axial velocity component results in increasing the absolute velocity, and decreasing static pressure [23][34-35][27].

The liquid from the impeller eye goes to the leading edge and then to the pressure or suction side of the blade where pressure seems to decrease again on the blade suction side until the blade differential pressure will be established. If the suction pressure falls below the fluid vapor pressure, there will be formation of vapor bubbles which will be eventually carried within the flow to only explode at comparatively high pressure zones. This happens at a very high speed; and is mostly a function of the flow velocity change, pressure gradient, liquid physical properties, and the closeness to the pump walls. The flow complexity in the impeller inter-blade passages is not only caused by the induced centripetal forces but also by its three-dimensionality. Tests, [23-29], have clearly demonstrated the turbulence influence on cavitation phenomenon through its pressure pulsations, particularly, at cavitation inception. Recently, cavitation bubbles have been proved to not concentrically collapse. There is an indentation which comes in for bubbles closer to the wall, before the implosion. For bubbles in the main stream, the indentation happens at high pressure zones. As this indentation keeps on growing, there is formation of the micro-jet which then divides the bubble into two or more parts. If the bubble was closer or attached to pump wall, this micro-jet hits the wall surface with very high velocities enough to remove the material. The cavitation damage to the wall surface results is a sponge-like structure. Cavitation doesn't take place only in the impeller passages, its damages can also be found at the diffuser blades and in the volute passage as well. At the inception stage, bubbles implode at the impeller passages. At the fully developed stage, the rest of bubbles which are carried by the flow, go on

to explode at the outer regions like diffuser and volute passages.

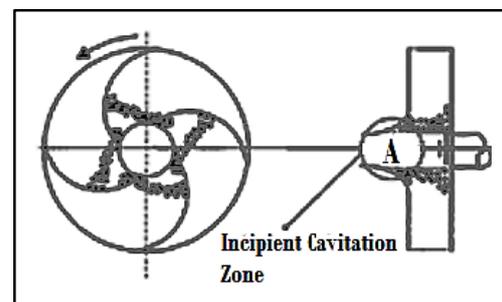


Figure 12. Impeller most low pressure zones

Low pressure zones are the most vulnerable in the pump; for instance, the back face of the blade inlet is mostly attacked due to its sudden flow velocity increase resulting in pressure decrease and thus bubbles formation.

2.2. Cavitation parameters

From different already published literature, it is commonly known that cavitation occurs when the fluid pressure drops to its Vapor pressure P_V . However, due to so many factors, this hypothesis is not always correct. The flow static pressure in a non dimensionalized form is given by the pressure coefficient C_p .

$$C_p = \frac{P - P_1}{\rho U^2 / 2} \quad (5)$$

Where P_1 and U are the reference static pressure and tip velocity at the inlet, with $U = \Omega R_{T1}$. The value of the pressure coefficient at the zones of lowest pressure in the flow is given by:

$$C_{P_{MIN}} = \frac{P_{MIN} - P_1}{\rho U^2 / 2} \quad (6)$$

Where P_{MIN} is the flow minimum pressure, $C_{P_{MIN}}$ is a negative value depending on the pump geometry and the Reynolds Number $Re = 2\Omega R_{T1}^2 / \rho g$. The value of the static pressure P_1 at which cavitation will occur ($P_{MIN} = P_V$) is given by:

$$P_{ICAV} = P_V + \frac{1}{2} \rho U^2 (-C_{P_{min}}) \quad (7)$$

Now the most commonly used cavitation parameters is the cavitation number which is given by:

$$\sigma = \frac{P_1 - P_V}{\frac{1}{2} \rho U^2} \quad (8)$$

The cavitation coefficient value corresponding to the first cavitation inception is defined as:

$$\sigma_i = \frac{P_{ICAV} - P_V}{\frac{1}{2} \rho U^2} \quad (9)$$

Combining the above presented equations, it's clear that $\sigma_i = -C_{P_{MIN}}$. The second parameter which is generally used is the pump suction specific speed. To finally come to it, let's pass through some specially used terminologies.

The pump Net Positive Suction Pressure (NPSP) is given by $P_1^T - P_v$ where P_1^T is the total pressure at the pump inlet and is given by:

$$P_1^T = P_1 + \frac{1}{2} \rho v^2 \quad (10)$$

The pump Net positive Suction Energy (NPSE) is defined as $P_1^T - P_v / \rho$ and the pump Net Positive Suction Head (NPSH) is given as $P_1^T - P_v / \rho g$ and finally the suction specific speed S given as:

$$S = \frac{\Omega Q^{\frac{1}{3}}}{NPSE^{\frac{3}{4}}} \quad (11)$$

The third mostly used parameter is the Thoma cavitation number σ_{TH} which is given as:

$$\sigma_{TH} = \frac{P_1^T - P_v}{P_2^T - P_1^T} \quad (12)$$

Where $P_2^T - P_1^T$ is the total pump pressure rise. The lastly presented parameter in this literature, is the blade blockage factor or coefficient, i.e. B. It is a particularly important parameter in pump flows. It is defined as:

$$B = \frac{2\pi r_1}{2\pi r_1 - z \delta_u} \quad (13)$$

Where r_1 , z, and δ_u stand for blade inlet radius, blades number, and the blade thickness respectively

2.3. Types of impeller cavitation

Cavitation in pumps can take different forms depending mainly on the pump flow rate and the flow inlet velocity angle at the blade leading edge, which strongly affects the pressure distribution on the blades at the inlet [36]. It is worth to note that the here-down presented pump cavitation classification is arbitrary, therefore, there may occur some cases which may not readily fall within this classification's reach. When the suction pressure is continually decreased (NPSH decreased), the first cavitation inception occurs at the intersection of the blade leading edge and the tip thus leading to a "tip vortex cavitation" occurrence. Lowering the cavitation number a bit farther, there is formation of traveling bubble cavitation at the blades suction sides. This type is otherwise referred to as "bubble cavitation". This type of cavity corresponds to a low incidence angle of the flow and depending on the design of the impeller, the minimum of pressure at the impeller throat [36].

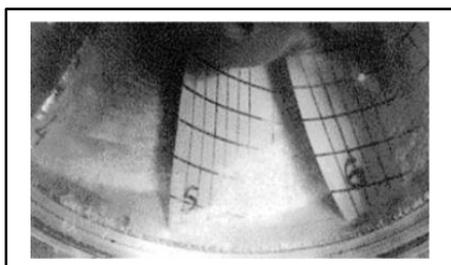


Figure13. Travelling cavitation

Further reduction results in the bubbles combination which gives rise to the attached cavitation at the blades suction side. This type is often referred to as "blade cavitation" in pumps.

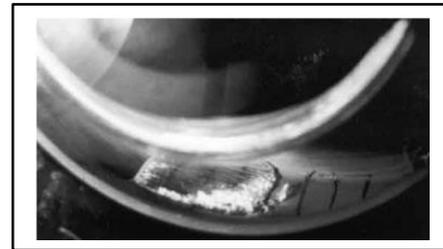


Figure14. Blade cavitation on the centrifugal impeller's suction side

Finally, it is valuable to create the catch-all term "backflow cavitation" to refer to the cavitating bubbles and vortices that occur in the annular region of backflow upstream of the inlet plane when the pump is required to operate in a loaded condition below the design flow rate. To the observer, this can often represent the most visible form of cavitation. [37].

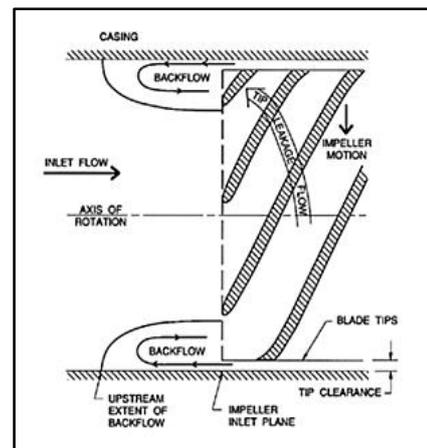


Figure15. Lateral view of impeller tip leakage and backflow

2.4. Cavitation and pump performance

To avoid the cavitation occurrence in the pump, a pressure reserve compared to the fluid vapor pressure is required at the impeller suction zone. This pressure difference between the inlet suction pressure and the fluid vapor pressure is otherwise called the Net Positive Suction Head or NPSH, which is generally measured in meters (m).

$$NPSH = \frac{P_1 - P_v}{\rho g} \quad (14)$$

Subscripts, 1 and v stand for suction side and the vapor phase, respectively. The NPSH has two aspects; the Net Positive Suction Head Available (NPSHa) and the Net Positive Suction Head Required (NPSHr). The later must be smaller than the first.

The rotation of the impeller, during the pump operation creates a vacuum at the eye, sucking the liquid from the inlet pipe to let it enter the eye. The NPSHa is then defined as the energy measure with which the liquid enters the pump from the inlet pipe. For every flow rate and rotational speed, every pump must be having a specifically predefined Head at the pump suction zone to prevent cavitation occurrence at the pump inlet. This Head measure is referred to as Net Positive Suction Head Required, NPSHr. Its empirical formula is as follows:

$$NPSHr = K_a \frac{V_{R1}^2}{2g} + K_b \frac{U_1^2}{2g} \quad (15)$$

Where V_R stands for the radial velocity and respective values of K_a and K_b are 1.0 to 1.2 and 0.1 to 0.3. Pearsall Theoretical work, [30], showed how to get an optimum cavitation pump performance; he emphasized that the choice of pump speed, correct size and geometry are the main factors to consider when it comes to pump cavitation performance analysis, rather than any other details. He by the way suggested a new method of calculating the optimum pump inlet diameter. He also proposed another NPSHr formula where the blade cavitation coefficient, had to be taken into account. Both his formulas are here-down presented.

$$D_{in} = 1.37 \left[\left(\frac{2(1+\sigma b)}{\sigma b} \right)^2 \frac{Q}{\omega} \frac{1}{1-\lambda^2} \right]^{\frac{1}{5}} \quad (16)$$

$$NPSHr = \frac{V_{R1}^2}{2g} (\sigma b + 1.04) + \sigma b \frac{U_1^2}{2g}$$

Where σb stands for blade cavitation coefficient, D_{in} , Q , λ , and ω , are the inlet tip diameter, the flow rate, the hub to tip ration, and the rotational speed; respectively. The Pearsall's Optimum Inlet Diameter formula shows that large eye diameters are required for low suction pressure pumps as the inlet diameter increase may be influenced by the increase of the blade cavitation coefficient.

In the past years, different methods have been used to clearly define centrifugal pump cavitation mechanism and its limits. Two technics have been under usage until now: 1) the pump capacity upper limit for given NPSH and speed, and 2) the NPSH lower limit for given pump capacity and speed. The second statement is the one widely adopted whereby cavitation occurrence is mostly diagnosed through the drop in head and pump efficiency.

Because of the difficulty of determining the exact condition when this change takes place, it is often the practice to define required NPSH as that value where a drop of 3 percent in head will have taken place; NPSH3 at given capacity and speed is NPSH3 which is that NPSH at which the head will have been reduced by 3 percent [31]. Cavitation in centrifugal pumps is mostly detected through its

detrimental effects such as pump hydraulic performance drop and pump physical structure damages. Figure 14.A shows the gradual increase in pump capacity at constant NPSH and Speed, and the usual subsequent Head curve, where at a certain pump capacity value, the head curve deviates from normal and drops straight downwards, marking the performance breakdown starting point. Any farther increase in pump capacity may result in a fully developed cavitation leading to pump walls damages and other effects.

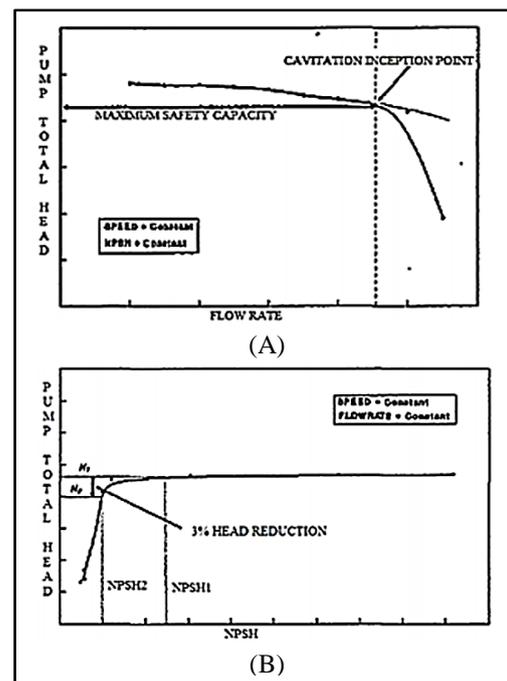


Figure 14. Cavitation effect on pump H-Q curve (A) and H-NPSH curve (B)

Also, as shown in the Figure 15, from different experimental observations [24, 25, 32], cavitation inception was shown to start at point A. The figure shows a continuously decreasing NPSH at constant flow and speed where from point A; the cavitation grows big until point B where the performance starts being affected, and proceeds to C where there is a complete performance breakdown.

This critical performance breakdown is generally considered as a point where there will be a 3% head drop (Fig. 14.B), which is not always correct as the exact values depend very much on the shape of the characteristic curve, which can also be influenced by experimental errors.[33,30]. Moreover, the position of the point by which the pump performance will start being affected is of a questionable truth. However for safety reasons, point B and points in its vicinity have been considered, which actually marches the cavitation noise characteristic curve as presented in figure 15. This curve shows a sudden

frequency increase at point A marking the inception, and from thereon, it gradually increases to finally attend its peak at point D marking the complete pump performance breakdown.

The reason for the variation of noise level is not immediately evident. [17, 30]. But a suggested interpretation of this variation is given.

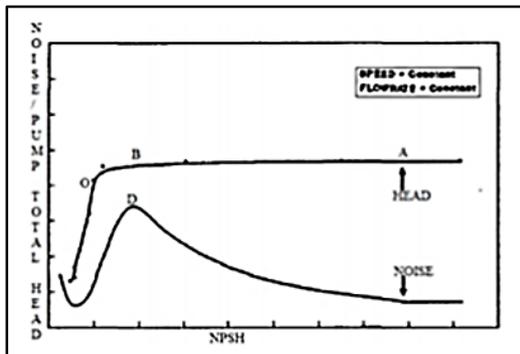


Figure15. Cavitation development and noise characteristic

When cavitation commences in a pump from a change in NPSH, the high frequency levels rise as a result of the growth and collapse of the corresponding range of small vapor bubbles. As the pressure is reduced the range of bubble radii and the number of bubbles increases and the noise levels continue to rise in this frequency band. The high frequency noise levels then decrease as the bubbles tend to grow bigger and absorb sound and radiate less noise in the higher frequencies. [31]

2.5. Cavitation mitigation

Because of the complex nature of cavitation, it's always not very straight forward to know the exact cause of its occurrence. But one can at least mention about some of the most seen cavitation cases in pumps. Cavitation in centrifugal pumps may have different reasons, some may be related to the pump design, others to the operating conditions such as the insufficient head at the inlet from different losses in the inlet pipe or faulty liquid depth estimation above the inlet, faulty system configuration leading to higher pump speeds, and pump developed head overestimation beyond the design values. Depending on the system conditions, each of the above mentioned faults can be arbitrarily corrected in a way or another. For instance, the first would get resolved through raising the fluid's free surface level in the supply tank or lowering the pump inlet some distance downwards. For the second case, if the speed can't be readjusted, replacing the pump with a suitable one would be worth considering. The remedy to the last would be correcting the pump head by different ways like speed reduction, throttling and impeller trimming. Recently pump designers have also opted to the use

of inducers to efficiently improve centrifugal pumps suction performance by decreasing its NPSHr. Another method to mention about is the pressurized air injection at the pump inlet to decrease the cavitation damage and subsequent noise. This method has to be carefully controlled as only a percentage of 1% of free air can drop the pump H-Q curve. However, research by Karassik⁴⁰ [1989] showed that the air volume of 0.25 to 0.5% can decrease on cavitation damage. For instance, Budris⁴¹ [1998] showed that the air content of about 0.89 % can decrease about 82% of cavitation damage. While researchers like Peter H.⁴⁴ [1996] and Masao O.⁴⁵ [1982] suggested other mitigation technics such as the increase in the impeller inlet diameter and blade incidence angle, decreasing the blades number, and decreasing the inlet inception cavitation number by sharpening the blade leading edge; there is still a big literature on the influence of pumps' internal design geometry on cavitation performance, some covering the same. Wei W.⁴² [2012] investigated the effect of inlet flow incidence angle on cavitation performance in centrifugal pumps. Based on SST $k-\omega$ turbulence model and the Mixture cavitation model, numerical simulation was carried out on four different impellers, namely R1, R2, R3, and R4; some with different, others with same inlet incidence angles at the blade hub, mid-span, and blade tip. The study results showed that, for a better cavitation performance, impellers should be having a large incidence angle, and that the same used incidence angle should be kept constant all along from the hub to the blade tip. An almost similar research was conducted by BaotangZ.⁴³ [2011] where, using the mixture cavitation model and the SST $k-\omega$ turbulent model, they carried out a numerical study on a double suction pump with three different impellers, namely IB1, IB2, and IB3. IB2's blade angle was smaller than IB1's by 10 degrees, whereas IB3 had 12 radial blades. The Radial impeller was found to produce higher heads but poor cavitation performance. Double suction pumps were found to produce uniform flow pattern at the upstream which could prove a better cavitation performance.

Xianwu L.⁴⁶ [2008], by experimentation and numerical simulation, studied the effect of the blade profile on pump cavitation performance in a miniature pump. Two semi-open impellers, the first with leaned blades the second with two-dimensional blades, were studied; where $k-\epsilon$ turbulence model and VOF cavitation model were used for the simulation process. The leaned blades were found preferential to both the hydraulic and cavitation performance; however, the increase in axial tip clearance was found to make the pump cavitation performance even worse.

Xianwu L.⁴⁷ [2008] again carried out an experimental and numerical research on the influence of impeller inlet geometry on pump cavitation

performance. Five different impellers were studied where the blade inlet angle and blade leading edge were being gradually extended. The numerical simulation was conducted based on RNG k- ϵ turbulence model and the VOF cavitation model. The pump inlet geometry proved to be influential when it comes to centrifugal pump performance, where larger blade leading edge and blade inlet angle positively affected the pump hydraulic performance and cavitation performance respectively.

III. CONCLUSION

Cavitation phenomenon is globally understood like formation of vapor bubbles in the fluid flow from a pressure drop below its vapor pressure. Due to its speedy and complex nature, cavitation detection requires sophisticated methods; otherwise it can only be noticed by its effects on the equipment like unusual noise, vibrations and material damage. In fluid machinery, based on the system physical and working conditions, Cavitation can appear under different forms, which after getting to its full development, present almost similar effects on the system characteristics. In centrifugal pumps, cavitation performance mostly depends on the impeller geometrical design such that, any geometry modification can result in a totally different performance. Therefore, the design process requires a more careful control, such that, through experimental and numerical methods, the centrifugal pump's performance can be well predicted where cavitation can be decreased to acceptable levels if not completely eliminated.

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