A Review of CFD Analysis for Cavitation Reduction and Impeller Design Enhancement of Centrifugal Pump

[1]Anil Kumar Yadav, [2]Prof. Shamir Daniel

[1]Anil Kumar Yadav, Department of Mechanical Engineering, Bhopal (M.P.)

[2]Prof. Shamir Daniel, Department of Mechanical Engineering, Bhopal (M.P.)

[1]anil.mpec@gmail.com, [2]shamirdaniel3004@gmail.com

* Corresponding Author: Anil Kumar Yadav

Abstract: This study focuses on improving the efficiency of centrifugal pumps by modifying the impeller with micro grooves. The smooth and grooved impellers were compared using computational fluid dynamics (CFD) calculations to identify their characteristics and evaluate the potential benefits of microgeometry in the impeller design. The objective was to reduce the different losses in the pump and increase its efficiency. The study also examined the major failure modes of centrifugal pumps and their application in water supply and sewerage industries. The results showed that the grooved impeller design had higher efficiency and lower cavitation compared to the smooth impeller.

Keywords: *Centrifugal pumps, Cavitation, impeller design, operating pressure, CFX etc.*

I. INTRODUCTION

Centrifugal pump is used to convert kinetic energy to hydrodynamic energy. The type of kinetic energy is rotational kinetic energy and the rotational energy typically comes from engine to electric motor. Already obtained efficiency is not too high so the efficiency of the pump should be increased. So, the different loses should be reduced. So, the impeller should be modified with micro grooves. The characteristics of smooth impeller and the grooved impeller should be identified and compared. To evaluate this CFD calculations should be done and the application of micro geometry in centrifugal pump impeller is done and calculations are made. The main objective is to increase the efficiency of the pump. [1]

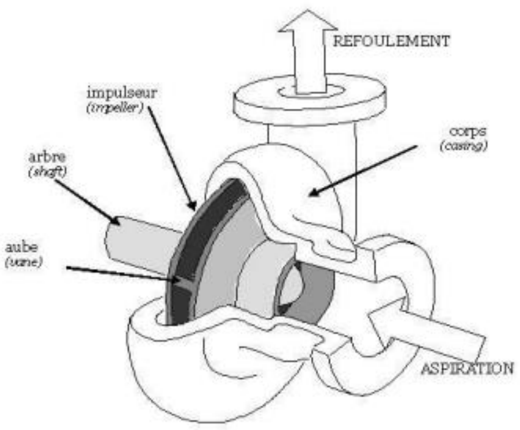

Fig 1. Components of centrifugal pump

A. CAVITATION PROCESS IN CENTRIFUGAL PUMP

The cavitation process consists of discrete events: the formation, development and collapses of cavitation bubbles. It must be avoided, or at least brought under control. The accompanying phenomena of cavitation, such as head and flow drops, damage to solid boundary surfaces, noise generation over a wide frequency spectrum, pressure pulsation, and vibration. In high energy process pumps, the forces generated by cavitation as well the internal recirculation can reduce the lives of bearings to few working days or hours. In recent days, the tendency to increase the rotational speeds of new pumps is desired to obtain required head and flow rate. Vibration analysis is used to determine the operating and mechanical condition of equipment. A major advantage is that vibration analysis can identify the problem before they become too serious and cause unscheduled downtime. This can be achieved by conducting regular monitoring of machine vibrations either on continuous basis or at scheduled intervals. Vibration analysis is used primarily on rotating equipment such as steam and gas turbines, pumps, motors, compressors, paper machines, rolling mills, machine tools, and gearboxes. In water pumps, the cavitation would have limited the impeller life due to cavitation erosion to a few hundred hours [2] .

Similar to advancement in condition monitoring systems,[3] computation fluid dynamics (CFD) helps to predict the onset of cavitation in marine propeller in simpler way than the experimental methods.[4] The maximum fluctuation can be found at the blade trailing edge near the blade pressure side for every flow channel under the part load condition, which results in vibration.[5] At high flow rates the structure vibration of cavitation phenomena in the centrifugal pump are different from those at low flow rates.

The vibration frequency in each monitoring point occurred mainly in the shaft, multiple shaft and the dominant frequencies.[6] The frequency spectra of vibration signals of cavitation are used to analyse cavitation phenomenon and to obtain a critical NPSH (Net Positive Suction Head) using the average magnitude in frequency band, predominant frequencies, and repeatable frequencies. Noise and vibration signals were better parameters to sense the occurrence of cavitation than the 3% head drop based on NPSH value obtained irrespective of all flow rates, speeds and the choice of leading edges.[7] It is seen from this study that cavitation occurs long before the performance characteristics is affected (i.e. much before 3% head drop) and likely to have peak of the cavitation vibration curve, and thus to the point where cavitation erosion is most likely.[8] Figure 3 shows the cavitation characteristics of centrifugal pump (low metric specific speed pump of 23) at 1500 rpm for three flow rates (80%, 100%, and 120%) along with the vibration curve for 992 Hz frequency (repeated frequency) with reference to NPSH (in X-axis). NPSH is defined as the difference between energy at the suction side of pump and the vapour energy corresponding to the temperature of water. It is inferred that the trend of vibration changes for reduction in NPSH (Net Positive Suction Head) with respect nominal flow rate. The head drop for 3% is drawn in the cavitation curve to get the idea of cavitation effects with reference to vibration signal.

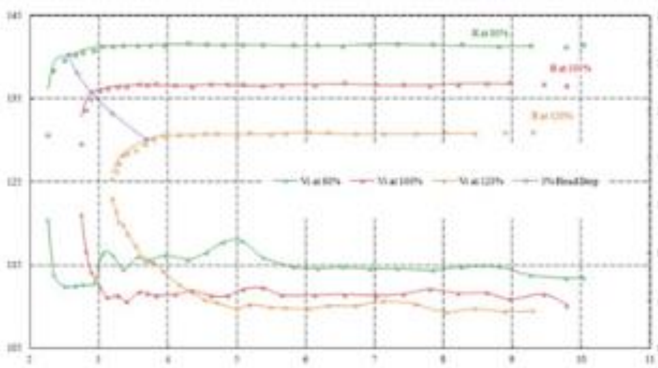

Figure 2 Cavitation characteristics of centrifugal pump at 1500 rpm for three flow rates.

The performance decrement of centrifugal pumps occurs in the form of head loss owing to recirculation of fluid at the inlet, disk friction, and skin friction losses. The skin friction and disk friction losses are among the major losses concerning pumping viscous fluids [9]. However, minimal attempts have been made to improve the pump performance, especially concerning the volute design or manipulation of the impeller.

B. IMPELLER DESIGN

Impeller design parameters, such as the wrap angle and angles of the inlet and outlet blades, affect the pump performance significantly; a large wrap angle increases the head and hydraulic efficiency of the pump, with a notable increment of input power [10]. The effects of variation in inlet profile and blade angle become significant and are generally observed as the inlet recirculation and cavitation performance changes [11]. The increase in outlet blade angle enhances hydraulic losses in the impeller and volute [12]. Ideally, an infinite number of blades can lower the slip factor greatly; however, it causes the flow pattern to distort [13]. An optimum number of blades are preferred when pumping a fluid for high head and low cavitation. Blade loading is an effective parameter that dictates the performance and maintenance frequency of a centrifugal pump [14].

C. CENTRIFUGAL PUMP DESIGN [28]

Centrifugal pump is a machine that imparts energy to a fluid. This energy can cause a liquid to flow or rise to a higher level. Centrifugal pump is an extremely simple machine which consists of two basic parts: The rotary element or impeller and the stationary element or casing. The centrifugal pumps are widely used in the world because the pump is robust, effective and inexpensive to produce. Centrifugal pumps are more economical to own, operate and maintain than other types of pumps. Pumps operate via many energy sources, including manual operation, electricity, engines, or wind power, come in many sizes, from microscopic for use in medical applications to large industrial pumps. Mechanical pumps serve in a wide range of applications such as pumping water from wells, aquarium filtering, pond filtering and aeration, in the car industry for water-cooling and fuel Injection etc.

II. Major centrifugal pump failure modes with application to the water supply and sewerage industries [29]

Pumps and their associated systems are essential in a wide variety of industrial applications for the efficient transportation of fluids, from clean water to sewage. Centrifugal pumps, which are a common pump used in industry, are known to fail as a result of problems that arise within the fluid, such as cavitation, and mechanical faults, such as found in bearings and seals. Vibration monitoring has been found to be suitable in determining faults within pumps. Permanently fixed condition monitoring sensors are well suited for applications where the pump is submerged in inaccessible environments as commonly occurs in water supply and sewerage industries. It is becoming increasing common for pump manufacturers to provide onboard sensors on their equipment; however end user interpretation and analysis of this data is not being used to its full potential. In the following sections it will be outlined that the use of multiple sensor readings and the synthesis of these measurements with the information of the fault and failure modes within centrifugal pump use still needs to be utilised for condition monitoring to create a more efficient maintenance strategy in the water supply and sewerage industries. Pump failures result in operational changes that reduce efficiency or result in a breakdown of the pump. There are 13 main problems that afflict centrifugal pumps when in use. These problems, which include both mechanical and hydraulic problems, have been discussed in the literature over a number of years in a wide variety of industries. The problems that will be addressed here will be hydraulic failures (cavitation, pressure pulsations, radial thrust, axial thrust, suction and discharge recirculation), mechanical failures (bearing failure, seal failure, lubrication, excessive vibrations, fatigue), and other types of failure (erosion, corrosion, excessive power consumption). Each problem will be outlined including its cause and effect, symptoms, and pertinent mechanical corrective procedures.

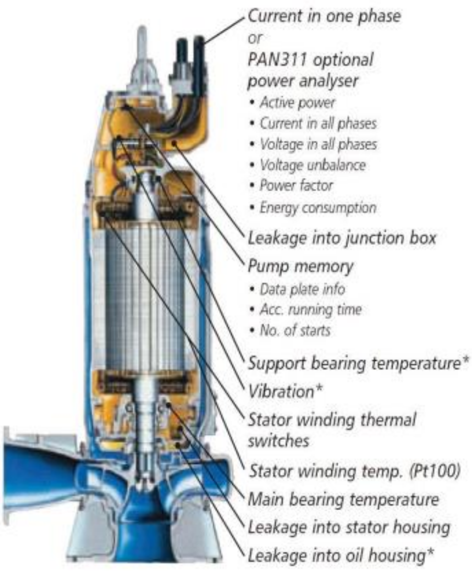

Figure 3 : Cross section of a Flygt centrifugal pump with sensors (with permission from ITT Corporation) [30]

A. **HYDRAULIC FAILURES [29]** Hydraulic failures arise from changes in pressure either in the volute or the pipes leading to the pump due to changes in factors such as temperature, velocity of the fluid flow, and volumetric flow rate of the fluid. This section will cover the main hydraulic problems, reasons behind them, and solutions, if any.R

1. **Cavitation**

a) Erosion – the collapse of the cavities in areas of higher pressure can exert enormous local stresses on the surfaces against which they are collapsing, causing damage to the pump surfaces. Signs of erosion will appear as pitting due to the water hammering action of the collapsing vapour bubbles. Damage occurs because when the cavities collapse, the jet of liquid that is released hits the surface of the pump at the local speed of sound, which results in a local high surface pressure that can be higher than the ultimate strength of the material . [29]

b) Noise – The sound of the cavities collapsing under higher pressure is a sharp crackling sound. Some refer to it as if it was pumping stones. The level of the noise that results from cavitation is a measure of the severity of the cavitation. The noise can be found in and around the pump suction. If the crackling noise seems to be random, and is accompanied with highintensity knocks, then this indicates cavitation in the suction recirculation. However, this does not indicate a reduction in the performance of the pump if the NPSHA is greater than the NPSHR . There are actually three main types of cavitation, classified based on the location of the cavities inception and the location of implosion of the vapour bubbles, and each is accompanied with its own range of acoustic radiation. Sheet cavitation, which is the first type, forms cavities across the vane surface when pumps operate close to their design flow with low suction pressure. It creates a broad band noise, with low amplitude, in the range of 2 kHz to 40 kHz. Cloud cavitation, the second type, forms cavities downstream of the cavity sheet when the pump operates away from its design flow at low suction pressure. This is the loudest of the three types of cavitation. It generally appears at high frequencies, such as 20 kHz to 40 kHz, and gives the familiar sound of "pumping gravel". Vortex cavitation, the third type, is a highly unstable form of cavitation when pumps operate at very low flows and in the inlet backflow regime. Although it is a bubble collapse phenomena, like the previous two, it is less damaging because the collapse of the vapour bubbles occur well away from solid surfaces. This type of cavitation is characterized by random bursts of noise accompanied by the typical cavitation sound. When NPSHA gets very close to the 3% head decay line and the pump operates in the back flow regimes simultaneously, vortex cavitation generates a low frequency beat in the region of 1Hz to 4 Hz. This is known as cavitating surge. [29]

c) Vibration – Pump vibrations due to cavitation are characteristically high amplitude and low frequency, usually found in the 0 to 10 Hz range .[29]

d) Reduction in pumping efficiency – Vapour bubbles created in the passages around the impeller impede the flow of the fluid being pumped, thus resulting in a reduction in output . A drop in efficiency of the pump is a more reliable sign of cavitation occurring, since noise is not prominent until cavitation has progressed to the point where the efficiency of the pump is poor. On some occasions it has been found that the pump's efficiency may slightly increase moments before cavitation begins. This may be due to a reduction of friction at the beginning of the separation in the flow, just before the cavities start to implode .[29]

2. Pressure Pulsations

As important fluid machinery, pumps are widely used in both industry and daily life, but they also consume large amounts of electric power [15]. Taking China, for instance, Wang et al. estimated that power consumption by pumps accounts for almost 20% of the total generated electricity [16]. Therefore, pumps are a crucial consideration in building an energy-saving society, considering their significant energy consumption. Fu et al. note that to address the increasing demand for energy saving, the stability and long-term operation of the pump are very important in many fields [17]. For example, in power stations, if an unexpected shutdown of the main feed pump occurs, significant losses will result. This is more significant in the nuclear power plant, where the nuclear reactor coolant pump is required to run safely for almost 60 years without any repair of the main hydraulic components [18,19]. As for the safe operation of the pump, two main factors are discussed in this paper, namely the mechanical and fluid dynamic effects. The problem of unsteady fluid dynamics is attracting more attention. It is a phenomenon that leads to alternating forces acting on the impeller and the volute, threating pump safety [20–22]. Furthermore, compared with the mechanical factor, the phenomenon of unsteady fluid dynamics is complex and more difficult to predict and analyze. Therefore, it should be emphasized to guarantee the safe operation of the pump.

As a typical unsteady flow phenomenon, rotor–stator-interaction- (RSI) induced high pressure pulsation is detrimental to the stable operation of the pump [23]. Rotor–stator interaction is considered the main cause of intense pressure pulsations in pumps, and is a focus of research. With the blade sweeping the tongue or diffuser, the flow shedding from the blade trailing edge interacts with the stator, and the corresponding non-uniform flow distribution is generated [24,25]. Such a phenomenon is characterized by intense fluctuations in pressure and velocity. On some occasions, if the resonant frequency of the structure coincides with the induced pressure frequency, severe vibration phenomenon is caused [26,27].

Pressure pulsations are found in both the suction and discharge of centrifugal pumps. The magnitude and frequencies of these pulsations are dependent on several factors: the design of the pump, the total head produced by the pump, the response of the suction and discharge piping, and the point of operation on the pump's characteristic curve. The frequency of the pulsation may come from known sources, such as the running frequency or the vane passing frequency or multiples of each, or it may seem random since it may come from sources such as the system's resonance, acoustic behaviour, eddies from valves or poor upstream piping. Regardless of the source, the pulsations should not be discarded as being irrelevant, since the pulsations carry information about the system. The observed frequencies in the pump suction are much lower than in the discharge. Typical frequencies in the pump suction are in the order of 5 to 25 cycles/s, and do not appear to have any direct relation to the rotational speed of the pump or the vane passing frequency . Pressure pulsations may be amplified acoustically in a piping system or its elements, which may lead to alternating stresses and excessive vibration beyond the endurance limit of the system. The wake flow found at the impeller's outlet is one of the strongest sources of pulsation. This is caused by the interaction between the fluid flow at the impeller vanes and the volute, which results in a pressure pulsation at the blade passing frequency and its harmonics .

Water hammer is another major cause of pressure pulsations in the piping system. Water hammer is essentially caused by the rapid closing of a valve or by a pump failure with a subsequent abrupt closure of the check valve or by a sudden switch over and the pump start procedure. This sudden flow variation in the pipe produces a pressure surge that travels at the speed of sound through the system. To prevent water hammer happening in the piping system, a storage tank can be placed in the system to absorb the pressure pulsation, rather than other parts of the system. In this case, water hammer generated at the check valve/failed pump would travel with the speed of sound through the piping system. At the storage tank, the pressure wave is absorbed, then reflected back to the check valve/failed pump. Once at the check valve/failed pump, the wave is reflected again back to the storage tank. This procedure repeats until the energy of the wave is dissipated due to dynamic friction and fluid-structure interaction. The damping of the reflected water hammer is controlled by the speed of the pump. Low pump speeds have no influence, but higher pump speeds can reduce the pressure pulsation. Pumps are also able to create pressure pulsations. The maximum of a pulsation is generated when the pump is running part load, and the minimum occurs when the pump is running at its design point. Pulsations also change when the pump speed shifts, where in this case, high pump speeds generate high pulsations and low pump speeds are a minor source for pressure pulsations. [29]

3. Radial Thrust

High radial thrust that results in excessive shaft deflections may lead to persistent packing or mechanical seal problems, and possibly, shaft failure. Shaft failures usually occur in the middle of the shaft span in the double-suction or multi-stage pump. End suction pumps usually have shaft failures at the shoulder of the shaft, where the impeller joins the shaft sleeve, or at the location of the highest stress concentration. High radial loads may also produce high temperatures in the bearings, which may also reduce the life of the bearing. Sleeve bearings will have bearing metal worn only in one direction and the journal will be worn uniformly. However, if the reverse is present, which is the bearing is worn uniformly and the journal is worn excessively in one direction, then the cause is not excessive bearing loads but instead most likely an unbalance or a bent shaft . It is difficult to detect high radial thrusts in a pump. Temperature rises in the bearing may or may not be a symptom of excessive radial loading. High bearing temperatures may be a result of misalignment, lack of lubrication, or excessive axial loading of the thrust bearing. As a result, further investigation should be done to eliminate other causes before concluding that the radial loads are excessive. Most failures that are a result of excessive radial thrust occur when the pump is operating at low flow rates. As a result, radial loads can be reduced by operating the pump at higher capacities or by installing a bypass from the pump discharge back to the pump suction or suction source .[29]

4. Axial Thrust

Bearing damage is caused by both static and dynamic axial thrust. Heavy static thrust will cause cracking in the balls or rollers, and in the race of the rolling element bearings, and in the metal scoring of the shoes in the tilting-pad bearings. Excessive dynamic loads which surpass the bearing ratings will result in fatigue failures of the balls or rollers, and raceways in the rolling element bearings. In order to determine which of the two types of loads caused the failure, one would have a close examination under the microscope. Fatigue failure from dynamic loading will show a hammering effect caused by the points of impact. Fatigue failure from static loading will show metal fatigue without the hammering effect of the impact loading. Rolling element bearing failures can be addressed in large, between bearing pumps by substituting a tilting-shoe type of thrust bearing. The high cyclic axial forces are better absorbed in the oil film of the tilting-shoe bearing than in the rolling element bearing . Shaft failure is mainly due to the high cyclic loading induced on the shaft when the pump is partially recirculating its output. In this case, axial cyclic stresses can be reduced by increasing the pump output, or by installing a recirculation line to bypass sufficient flow to move the pump total flow rate beyond the critical point. If this is not possible, then the shaft failures can be reduced by substituting a shaft material of higher endurance limit . Depending on the location of the axial thrust, different instrumentation would be used to determine its magnitude. Proximity-type sensors should be used to determine the axial movement of the shaft relative to the bearing housing. Deflection of the thrust-bearing housing can be obtained using seismic instruments. Finally, axial loading of the tilting-shoe type of thrust bearing can be monitored by a load cell permanently installed in the levelling plate .[29]

5. Suction and Discharge Recirculation

During suction recirculation, a loud crackling noise is produced around the suction of the pump, for discharge recirculation, at the discharge volute or diffuser. Noise produced by recirculation has a greater intensity than that produced by cavitation, and is normally characterized by a random, knocking sound . Suction or discharge recirculation can be determined by monitoring the pressure pulsations found in the suction and discharge of the pump. Piezoelectric transducers are normally placed close to the impeller on either the suction or discharge side of the pump. Data obtained may be analyzed using a spectrum analyser to generate a plot of the pressure pulsations versus the frequency of selected flows. On this plot, a sudden increase in the magnitude of the pressure pulsations would represent the beginnings of recirculation. Pitot tubes installed at the eye of the impeller can also help determine the onset of suction recirculation. With the pitot tube directed into the impeller eye, suction recirculation will occur when the flow reversal from the eye impinges on the pitot tube with a rapid rise in the gauge reading . In order to correct recirculation in the system, the following steps are suggested: increase the output flow of the pump; install a bypass between the discharge and the suction of the pump; substitute an improved material for the impeller that is more resistant to cavitation damage; increase the output capacity of the pump; or modify the impeller design .[29]

II. LITERATURE REVIEW

(K. Sathish et al., 2021) [1] The main objective of this review is to investigate the centrifugal pump and to increase the efficiency of the pump. The average efficiency of centrifugal pump is about 65–70 percent. The significance of efficiency in pump is selecting a proper pumping system will conserve fuel or electricity and decrease the annual pumping costs. Inefficient and poorly chosen pumping systems can increase annual costs dramatically. Obtained efficiency of centrifugal pumps with low specific speed is not high. The scope of the work is to reduce hydraulic losses. In this paper a literature review is identified by providing a modified impeller channel with design changes that are capable of improving the efficiency of the centrifugal pump

(Stephen C., 2018) [2] When pumps are operating at flow rates considerably lower or higher than normal, intensity of cavitation is likely to be greater than that at the best efficiency point. In particular for high specific speed pumps such as mixed flow and axial flow pumps, various kind of cavitation has been seen and similar cases was reported for low specific speed pump. Cavitation is a local vaporization of liquid induced by a hydrodynamic pressure reduction; it usually disturbs the normal flow field in pumps with various effects. This article will glimpse about the effects of cavitation in centrifugal pump with the help of vibration monitoring system and the technique is suggested to predict the onset of cavitation by means of average magnitude in frequency band, predominant frequencies, and repeatable frequencies.

(Helal MM et al., 2018)[3] Determining and understanding the performance characteristics of marine propellers by experiments is quite a complex and costly task. Numerical predictions using computational fluid dynamics simulations could be a valuable alternative provided that the laminar-to-turbulent transition flow effects are fundamentally understood with the suitable numerical models developed. Experience suggests that the use of classical turbulent flow models may lead to high discrepancies especially at low rotational speeds where the effects of fluid flow transition from the laminar to the turbulent state may influence the predicted propeller's performance. This article proposes a complete and detailed procedure for the computational fluid dynamics simulation of non-cavitating flow over marine propellers using the "k–kl–ω" transition-sensitive turbulence model. Results are evaluated by "ANSYS FLUENT 16" for the "INSEAN E779A" propeller. Comparisons against the fully turbulent standard "k–ε" model and against experiments show improved agreement in way of flow transition zones at lower rotational speeds, that is, at low Reynolds numbers.

(Pei Ji et al., 2014) [4] Numerical simulation and 3-D periodic flow unsteadiness analysis for a centrifugal pump with volute are carried out in whole flow passage, including the impeller with twisted blades, the volute and the side chamber channels under a part-load condition. The pressure fluctuation intensity coefficient (PFIC) based on the standard deviation method, the time-averaged velocity unsteadi- ness intensity coefficient (VUIC) and the time-averaged turbulence intensity coefficient (TIC) are defined by averaging the results at each grid node for an entire impeller revolution period. Therefore, the strength distributions of the periodic flow unsteadiness based on the unsteady Reynolds-averaged Navier-Stokes (URANS) equations can be analyzed directly and in detail. It is shown that under the 0.6Qdes. condition, the pressure fluctuation intensity is larger near the blade pressure side than near the suction side, and a high fluctuation intensity can be observed at the beginning section of the spiral of the volute. The flow velocity unsteadiness intensity is larger near the blade suction side than near the pressure side. A strong turbulence intensity can be found near the blade suction side, the impeller shroud side as well as in the side chamber. The leakage flow has a significant effect on the inflow of the impeller, and can increase both the flow velocity unsteadiness intensity and the turbulence intensity near the wall. The accumulative flow unstea- diness results of an impeller revolution can be an important aspect to be considered in the centrifugal pump optimum design for ob- taining a more stable inner flow of the pump and reducing the flow-induced vibration and noise in certain components.

(Yanxia Fu et al., 2015) [5] The characteristics of flow instabilities as well as the cavitation phenomenon in a centrifugal pump operating at low flow rates were studied by experimental and numerical means, respectively. Specially, a three-dimensional (3D) numerical model of cavitation was applied to simulate the internal flow through the pump and suitably long portions of the inlet and outlet ducts. As expected, cavitation proved to occur over a wide range of low flow rates, producing a characteristic creeping shape of the head-drop curve and developing in the form of nonaxisymmetric cavities. As expected, the occurrence of these cavities, attached to the blade suction sides, was found to depend on the pump's flow coefficient and cavitation number. The experiments focused on the flow visualization of the internal flow patterns by means of high-speed digital movies and in the analysis of the inlet pressure pulsations near the impeller eye by means of fast response pressure transducers. The experimental results showed that the unsteady behavior of the internal flow in the centrifugal pump operating at low flow rates has the characteristics of a peculiar low-frequency oscillation. Meanwhile, under certain conditions, the low-frequency pressure fluctuations were closely correlated to the flow instabilities induced by the occurrence of cavitation phenomena at low flow rates. Finally, the hydraulic performances of the centrifugal pump predicted by numerical simulations were in good agreement with the corresponding experimental data.

(Kai Wang et al., 2016) [6] To study the pressure fluctuation and vibration in mixed flow pumps, we chose a mixed flow pump with specific speed of 436.1 to measure. The time domains and frequency domain at each monitoring point on diffuser and outlet elbow were analyzed, as well as the vibration frequency domain characteristics at the impeller outlet and near the motor. The results show that the peak value of pressure fluctuation peak decreased gradually with the increase of flow rate. The pressure fluctuation of each monitoring point had periodicity, and the frequency domain dominated by blade passing frequency and multiple shaft frequency. The vibration frequency of each monitoring point occurred at shaft frequency and its multiple shaft frequency. The dominant frequency and the second frequency were distributed in shaft frequency and double shaft frequency.

(Christopher S et al., 2013) [7] Experimental investigations concerning cavitation in radial flow pump for three different leading edge profiles of the vane were carried out in an open circuit system. The operating condition of the

radial flow pump under cavitating case was understood by measurement of noise and vibration along with the pump parameters for various speeds and flow rates. The outcome of the experimental results revealed that the noise and vibration were better predictors of inception and development of cavitation. Further observation inferred from critical net positive suction head (NPSH) curve of 3% head drop and critical NPSH value of noise and vibration are presented.
(Stephen, C et al., 2019) [8] An experimental investigation of the cavitation behaviour of a radial flow pump of metric specific speed 23.62 rpm having different leading edge profiles of the vane is presented. The pump was operated for flow rates from 80 to 120% of the best efficiency point. The measurement included noise and vibration signals apart from the hydraulic parameters. The results exhibited the trends of noise and vibration with respect to percentage of head drops for all operating conditions. It was concluded that the trends were totally different for various flow rates. Hence it is suggested that the criteria to be used for detecting the early cavitation in pump based on noise and vibration signals should be a function of the flow rate. Further, it was found that the range of frequency band for noise and vibration was within 5 kHz with reference to the magnitude of fluctuation. The repeatable predominant frequency of vibration for prediction of cavitation behaviour of this particular pump was established as 0.992 kHz.

III. CHARACTERISTICS OF IMPELLLER DESIGN

A. FLOATING IMPELLER DESIGN

Stages with floating impellers are the original and simplest constructions where the impellers are free to “float” axially (move up and down relative to the pump shaft) because they are not fixed to the pump shaft in the axial direction. As shown in Fig. 4, impeller hubs are not stacked so there is a vertical distance between the successive hubs. Furthermore, each stage contains down thrust washers to absorb the axial forces occurring during operation as well as to seal and minimize recirculation of fluids within the stage. Although called “floater” impellers, its washers are normally in contact with the diffuser pad because of the axial load on them. Because of this condition, the unit's main thrust bearing, situated in the protector section, has to carry only the downward thrust acting on the pump shaft.

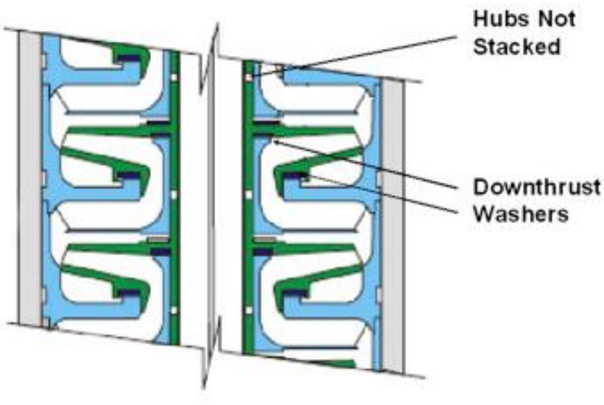

Figure 4. Construction details of floating impeller stages.

The downthrust on the shaft is due to its cross-sectional area being exposed to a large pressure differential that equals the difference between the pump's discharge and suction pressures. This load is directly transferred to the protector and has to be taken into account when selecting the right protector.

Benefits of floating impeller design include

•the elimination of having to fix the impellers axially, a time-consuming work requiring high precision,

•the building of pumps with several hundreds of stages is possible,

•smaller-capacity thrust bearings are needed in the protector section because most of the hydraulic thrust is absorbed inside the pump

•lower investment cost, as compared to fixed impeller pumps.

Limitations are related to the load-bearing capacity of available thrust bearings, which, in turn, are restricted by the annular space available in different casing sizes:

•such pumps are usually manufactured in smaller diameters, up to a size of about 6″,

•the recommended operating range is somewhat narrower than that for the same pump with fixed impellers.

B. FIXED IMPELLER DESIGN

Fixed impellers are locked on the pump shaft in the axial direction, their hubs being in contact, see Fig. 3.8. As the stages are not equipped with down thrust washers, the axial thrust developed on them must be fully carried by the unit's main thrust bearing in the protector section. Because of this, these pumps may be operated outside their normal operating ranges without much damage to the stages. Pumps with such stages are often called "**compression pumps**" and are commonly used in larger-sized ESP units (greater than 6″ in diameter) capable of producing large volumes of liquids. As already mentioned, they may have a wider operating range than pumps of the same type with floating impellers. Generally, they tolerate pumping of fluids containing abrasives much better than floating impeller type pumps.

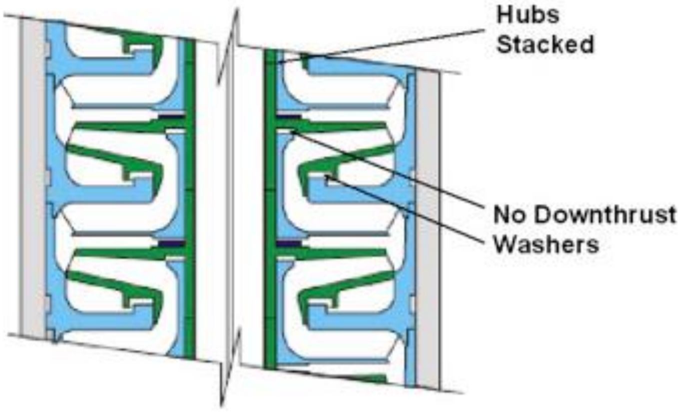

Figure 5. Construction details of fixed impeller stages.

The thrust to be carried by the main thrust bearing of the ESP unit is much greater than that for floater pumps. The total axial load has two components: (1) the shaft load due to the differential pressure acting on the shaft cross-sectional area and (2) the sum of the axial loads occurring in the pump stages. Clearly, thrust bearings of much greater capacity than that for floater pumps are required for proper operation.

Limitations of fixed impeller "compression" pumps include the following:

• They are more difficult to manufacture because impellers must be fitted very precisely along the pump shaft,

• Investment costs are higher because of manufacturing requirements,

• The maximum number of stages in one pump is limited to about 80–100,

• Protectors with high-capacity thrust bearings must be used.

IV CONCLUSION

In conclusion, the study highlights the importance of impeller design in improving the efficiency of centrifugal pumps. The micro grooves added to the impeller surface reduced the different losses and improved the pump efficiency. The study also identified the major failure modes of centrifugal pumps and their application in water supply and sewerage industries. The results showed that the grooved impeller design had higher efficiency and lower cavitation compared to the smooth impeller. The findings of this study can be applied to the design and optimization of centrifugal pumps for various applications in the industry.

REFERENCES

[1] K. Sathish1* , S. Abilash2 , S. Balaji3 , S. Aroon Daniel4 , V. Dhatchinamoorthi5 ,Department of Mechanical Engineering, Sri Eshwar College of Engineering, Coimbatore, India, A Review On Performance Analysis of Centrifugal Pump Impeller, International Journal of Research in Engineering, Science and Management Volume 4, Issue 6, June 2021, ISSN (Online): 2581-5792

[2] Stephen C. A review of cavitation effects on vibration in centrifugal pump. Fluid Mech Res Int. 2018;2(3):128-129. DOI: 10.15406/fmrij.2018.02.00028

[3]Helal MM, Ahmed TM, Banawan AA, et al. Numerical prediction of the performance of marine propellers using computational fluid dynamics simulation with transition-sensitive turbulence model. Proc. of the Institution of Mech Engineers, Part M: J of Engg for the Maritime Environment. 2018.

[4]Pei Ji, Yuan Shouqi, Li XiaoJun, et al. Numerical prediction of 3-D periodic flow unsteadiness in a centrifugal pump under part-load condition. J of Hydrodynamics. 2014;26(2):257−63.

[5]Yanxia Fu, Jianping Yuan, Shouqi Yuan, et al. Numerical and experimental analysis of flow phenomena in a centrifugal pump operating under low flow rates. J of Fluids Engg. 2015;137(1):011102−12.

[6] Kai Wang, Houlin Liu, Xiaohua Zhou, et al. Experimental research on pressure fluctuation and vibration in a mixed flow pump. J of Mech Sci and Tech. 2016;30(1):179−84.

[7] Christopher S, Kumaraswamy S. Identification of critical NPSH from noise and vibration in a radial flow pump for different leading edge profiles of the vane. J of Fluids Engg. 2013;135(12):121301−15.

[8] Stephen, C., Kumaraswamy, S. Experimental Determination of Cavitation Characteristics of Low Specific Speed Pump using Noise and Vibration. J. Inst. Eng. India Ser. C 100, 65–74 (2019).

[9] Shao, C.; Zhao, Y. Numerical study of the dimensionless characteristics and modeling experiment of a molten salt pump that transports viscous fluids. Int. J. Numer. Methods Heat Fluid Flow 2017, 27, 2131–2153. [CrossRef]

[10]Tan, L.; Zhu, B.; Cao, S.; Bing, H.; Wang, Y. Influence of blade wrap angle on centrifugal pump performance by numerical and experimental study. Chin. J. Mech. Eng. 2014, 27, 171–177. [CrossRef]

[11] Westra, R.W.; Broersma, L.; Van Andel, K.; Kruyt, N.P. PIV Measurements and CFD Computations of Secondary Flow in a Centrifugal Pump Impeller. J. Fluids Eng. 2010, 132, 061104. [CrossRef]

[12] Liu, J.; Li, Z.; Wang, L.; Jiao, L. Numerical Simulation of the Transient Flow in a Radial Flow Pump during Stopping Period. J. Fluids Eng. 2011, 133, 111101. [CrossRef]

[13] Stepanoff, A.J. Centrifugal and Axial Flow Pumps: Theory, Design and Application, 2nd ed.; Chapman and Hall: London, UK, 1957; Available online: https://cir.nii.ac.jp/crid/1573387449864191104 (accessed on 17 August 2022).

[14] Li, X.; Gao, P.; Zhu, Z.; Li, Y. Effect of the blade loading distribution on hydrodynamic performance of a centrifugal pump with cylindrical blades. J. Mech. Sci. Technol. 2018, 32, 1161–1170. [CrossRef]

[15] Arun Shankar, V.K.; Umashankar, S.; Paramasivam, S.; Hanigovszki, N. A comprehensive review on energy efficiency enhancement initiatives in centrifugal pumping system. Appl. Energy 2016, 181, 495–513. [CrossRef]

[16] Wang, C.; Shi, W.; Wang, X.; Jiang, X.; Yang, Y.; Li, W.; Zhou, L. Optimal design of multistage centrifugal pump based on the combined energy loss model and computational fluid dynamics. Appl. Energy 2017, 187, 10–26. [CrossRef]

[17] Fu, Y.; Yuan, J.; Yuan, S.; Pace, G.; D'Agostino, L.; Li, X. Numerical and experimental analysis of flow phenomena in a centrifugal pump operating under low flow rates. ASME J. Fluids Eng. 2015, 137, 011102. [CrossRef]

[18] Lu, Y.G.; Zhu, R.S.; Wang, X.L.; Wang, Y.; Fu, Q.; Ye, D.X. Study on the complete rotational characteristic of coolant pump in the gas-liquid two-phase operating condition. Ann. Nucl. Energy 2019, 123, 180–189.

[19] Lu, Y.G.; Zhu, R.S.; Fu, Q.; Wang, X.L.; An, C.; Chen, J. Research on the structure design of the LBE reactor coolant pump in the lead base heap. Nucl. Eng. Technol. 2019, 51, 546–555. [CrossRef]

[20] Gao, Z.X.; Zhu, W.R.; Lu, L.; Deng, J.; Zhang, J.G.; Wang, F.J. Numerical and experimental study of unsteady flow in a large centrifugal pump with stay vanes. ASME J. Fluids Eng. 2014, 136, 071101. [CrossRef]

[21] Yao, Z.F.; Wang, F.J.; Qu, L.X.; Xiao, R.F.; He, C.L.; Wang, M. Experimental investigation of time-frequency characteristics of pressure fluctuations in a double suction centrifugal pump. ASME J. Fluids Eng. 2011, 133, 101303. [CrossRef]

[22] Pei, J.; Yuan, S.Q.; Yuan, J.P. Fluid-structure coupling effects on periodically transient flow of a single-blade sewage centrifugal pump. J. Mech. Sci. Technol. 2013, 27, 2015–2023. [CrossRef]

[23] Feng, J.J.; Benra, F.K.; Dohmen, H.J. Investigation of periodically unsteady flow in a radial pump by CFD simulations and LDV measurements. J. Turbomach. 2011, 133, 011004. [CrossRef]

[24] Stel, H.; Amaral, G.D.L.; Negrão, C.O.R.; Chiva, S.; Estevam, V.; Morales, R.E.M. Numerical analysis of the fluid flow in the first stage of a two-stage centrifugal pump with a vaned diffuser. ASME J. Fluids Eng. 2013, 135, 071104. [CrossRef]

[25] Pei, J.; Yuan, S.Q.; Wang, W.J. Numerical analysis of three-dimensional unsteady turbulent flow in circular casing of a high power centrifugal diffuser pump. Adv. Mech. Eng. 2013, 5, 204521. [CrossRef]

[26] Li, S.Y.; Chu, N.; Yan, P.; Wu, D.Z.; Antoni, J. Cyclostationary approach to detect flow-induced effects on vibration signals from centrifugal pumps. Mech. Syst. Signal Process. 2019, 114, 275–289. [CrossRef]

[27] Zhou, W.J.; Qiu, N.; Wang, L.Q.; Gao, B.; Liu, D. Dynamic analysis of a planar multi-stage centrifugal pump rotor system based on a novel coupled model. J. Sound Vib. 2018, 434, 237–260. [CrossRef]

[28] Sahil, Shubham Singhmar, Dr. Thakur Sanjay Kumar, 2015, CFD Analysis of Centrifugal Pump using ANSYS, INTERNATIONAL JOURNAL OF ENGINEERING RESEARCH & TECHNOLOGY (IJERT) RACEE – 2015 (Volume 4 – Issue 03)

[29] Kristoffer K. McKee, Gareth Forbes, Ilyas Mazhar, Rodney Entwistle and Ian Howard Curtin University, Western Australia, A review of major centrifugal pump failure modes with application to the water supply and sewerage industries

[30] ITT, "Cross section of a Flygt N pump with sensors labeled," W. W. F. N.-p. ITT, 3306, 3312, 3356, 3400 [Brochure]. , Ed. White Plains, NY: ITT, 2010.

A PEER REVIEWED RJETM REFEREED JOURNAL

inprotected.com