A review of major centrifugal pump failure modes with application to the water supply and sewerage industries

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Summary: Centrifugal pumps are one of the world's most widely used type of pump, having an extensive range of applications, from food processing to water or sewage transportation. Problems that arise within these machines decrease the flow of the fluid within the pipelines, thus interrupting the production and transport of the fluid to its destination within the process. This may lead to other parts of the process system slowing down or behaving undesirably. As a result, it is imperative that these pumps be correctly monitored, diagnosed, maintained or replaced prior to the pump failing catastrophically to reduce downtime, material cost, and labour costs. This paper reviews the major fault modes that are found in centrifugal pumps, especially those in the water and sewage industry. Attention is given to the nature of the faults, symptoms shown within the pump that could be utilised for specific fault detection and diagnosis, and any mechanical corrective procedures that exist to help alleviate the problem. In addition, this paper contains a comparison and critique of previously published work that has attempted to diagnose the fault modes of centrifugal pumps.

Key Words: centrifugal pump, condition monitoring, vibration.

1 INTRODUCTION

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.

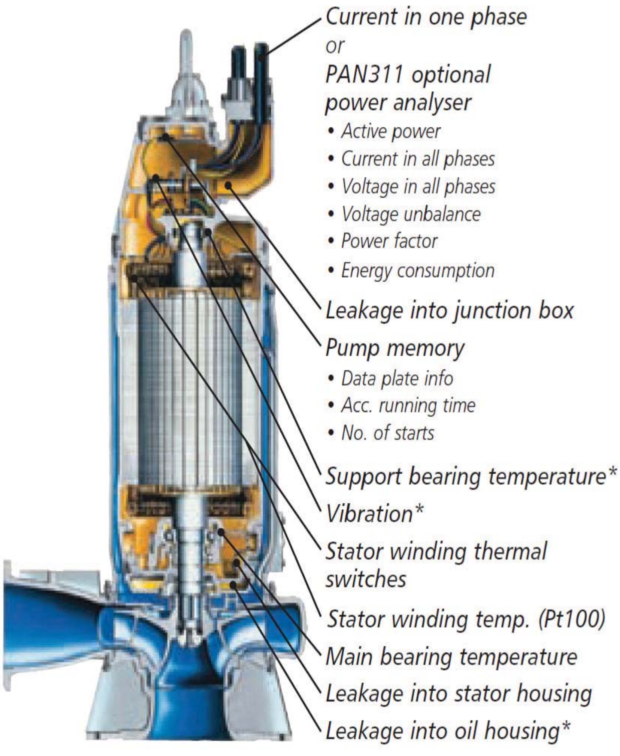

Figure 1: Cross section of a Flygt centrifugal pump with sensors (with permission from ITT Corporation) [2]

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.

2 HYDRAULIC FAILURES

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.

2.1 Cavitation

Cavitation is the formation of vapour bubbles in a moving fluid where the pressure of the fluid falls beneath its vapour pressure. In essence, cavitation results from a reduction in suction pressure, an increase in suction temperature, or an increase in the flow rate above that for which the pump has been designed [3]. Designers of pumps attempt to take into consideration the fact that pumps do not always run at peak efficiency, and try to take into account the operating range of the system. A centrifugal pump is usually operated comfortably within the range of 85% to 110% of its best efficiency point (BEP). However, many pumps are forced to operate outside of this range [4]. As a result, designers go to great lengths to ensure that cavitation bubbles do not collapse in the pump, but rather in the main piping system, far away from the impeller vanes.

There are several causes for cavitation in a pump and piping system, such as: high volumetric flow; a large decrease in the amount of fluid in the system which results in an abnormal increase in the temperature of the fluid; decrease in suction pressure due to changed conditions on the suction side of the pump; heating the fluid in the suction system, which leads to a higher fluid vapour pressure at the pump inlet; flow instability within the pump, which normally occurs at flow rates well below the pump's best efficiency point (BEP) flow rate; flow close to zero which results in rapid fluid heating in the pump casing, and quickly results in vapour-locking in the system; poor distribution of the fluid in pumps operating in parallel; oversized pumps operating at high capacity; pumping warm water with high vapour pressure; it is also hypothesised that cavities formed between the fluid and the vibrating parts of the pump that are in contact with the fluid; and a high percentage of leakage flows may lead to an increase in the temperature at the eye of the impeller, which would then possibly cause localized flashing [4-9].

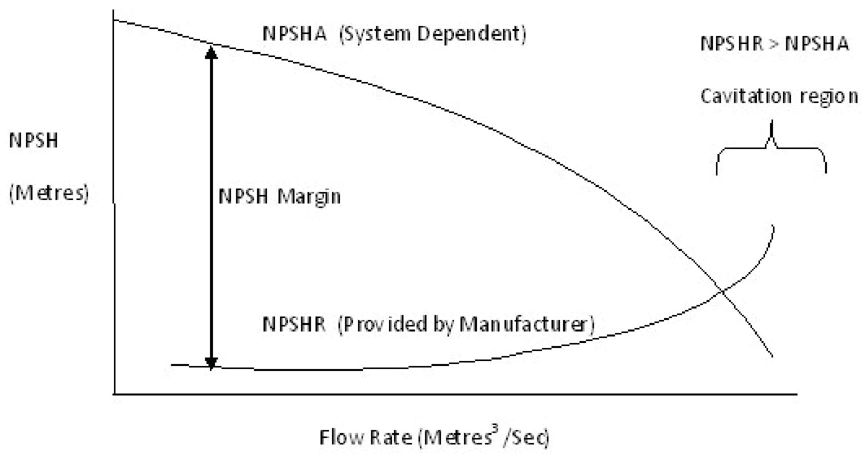

Figure 2: NPSH vs Flow Rate [1]

High flow cavitation occurs more frequently when the Net Positive Suction Head (NPSH) margin (actual NPSH – required NPSH) is small, possibly due to design constraints, and there is a reduction in the normal pump suction pressure as a result of an increase in the amount of suction piping pressure losses, or when the pump is operated well above its normal or rated flow. The Actual Net Positive Suction Head (NPSHA) is calculated using characteristics found at the pump's suction nozzle and is thus independent of the pump or the pump's characteristics. The Required Net Positive Suction Head (NPSHR) is the amount of net positive suction head that is required to avoid cavitation. It is independent of the system characteristics, and remains the same regardless of which system the pump is installed in. Excessive cavitation may lead to "vapour locking", a phenomenon where the fluid within the pump becomes mostly vapour, due to extreme vaporization of the fluid, or the pump being run for a long period of time at zero or near zero flow rate. Recovery from this phenomenon requires stopping the pump and allowing liquid to re-enter the pump [7].

Cavitation damage, which occurs on the low-pressure or visible surface of the impeller inlet vane, is accompanied by four symptoms:

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 [10]. It has been found that cavitation damage rates increase rapidly with the increase in the volume of the fluid. Erosion rates increase by a factor of 4 when the capacity is raised from 100% to 120% of the shockless flow. (The shockless flow point is a point on the head vs. capacity curve, which is generally 1.1 to 1.3 times the capacity of the BEP flow.) If the capacity of the fluid continues to increase, and the head decreases, past the shockless flow point, then the NPSHA will inevitably become less than the NPSHR, and cavitation will set in [9].

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 high-intensity 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 [11]. 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[12].

c) Vibration – Pump vibrations due to cavitation are characteristically high amplitude and low frequency, usually found in the 0 to 10 Hz range [9].

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 [13]. 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 [5].

Cavitation can be detected using a suction gauge or manometer to help determine if the NPSHA is equal to or less than the NPSHR by the manufacturer or detected by microphones capturing the acoustic radiation associated with the cavitation damage [14]. Since cavitation is a common phenomenon, corrective procedures have been devised in order to avoid or control the damage. Some of these corrective procedures are: replacing the impeller in the pump with one made of a more cavitation resistant material, such as stainless steel; reducing the flow of the pump so that the NPSHR will be equal to or less than the NPSHA; completely redesigning or altering the design of the impeller by reworking its geometry or surface finish to reduce losses, improve flow characteristics, or increase the flow inlet area, which would decrease the impeller inlet velocity; throttling the pump discharge so as to reduce pump flow and possibly restore pump operation back into the allowable range for NPSHA; placing an inducer on the suction side of the pump to increase the pressure in the pump before the fluid reaches the impeller; admitting small amounts of air into the pump suction will reduce the noise associated with cavitation, although it is not commonly used with centrifugal pumps; and providing low flow protection by placing a recycling line from the pump discharge back to the suction source [5, 6, 10, 11, 15].

2.2 Pressure Pulsations

Pressure pulsations are only of concern in high pressure pumps, where the head of water is more than around 300 m. For high-head pumps, suction and discharge pressure pulsations may cause instability of pump controls, vibration of suction and discharge piping, and high levels of pump noise. In addition any failure of internal pressure-containing members should be investigated with consideration given to the possibility that the fatigue failures are from internal pressure pulsations. If the fatigue failure shows microscopic striations, which are due to cyclic stress, then pressure pulsations may be a root cause of the problem [11].

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 [11]. 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 [16].

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 [16].

Research has shown that the pump acts like a non-moving valve for the pressure wave, producing a low resistance, which is very small in comparison to the wave's pressure amplitude. About 80 percent of the pressure wave passes through the pump, while the remaining 20 percent is dissipated by the pump. This number varies depending on whether an impeller's vane is found directly in front of the pump's outlet [16].

Since water has low acoustical absorption, pressure pulsations are easily sustained. Pipe lengths of odd multiples of one-quarter wave lengths and multiples of half wave lengths may resonate and amplify pressure pulsations by five or six times. Piping supports, particularly for feed water systems in power plants, have failed due to extreme amplification of pressure pulsations. In the suction system, resonant conditions in the intake structures or in the suction piping systems can aggravate sufficiently to cause incipient cavitation with vibration, erosion, and in some cases, failures. Resonance can be avoided by changing the number of vanes or by inserting expansion chambers or venturi sections in the piping system to break up the resonance [17].

Pressure pulsations are normally measured with piezoelectric pressure transducers and recorded as peak to peak pressure pulses over a broad frequency band. After the information is recorded, spectral analysis may then be performed on the data to retrieve information, and be used to devise a strategy to handle the situation. Three main problems arise from the pressure pulsations: vibration of suction or discharge piping, instability of pump controls, and fatigue failure of internal pressure-containing components of the pump from pressure pulsations [11].

If the pressure pulsation causes a vibration in the suction or discharge piping, vibration measurement on the piping structure could be more appropriate than the use of piezoelectric pressure transducers. The following strategy can be used to alleviate pressure pulsations in suction or discharge piping:: shift the resonant frequencies in the piping or supports that respond to any pressure pulsations; increase the output of the pump by either installing a bypass that would run from the discharge to the suction of the pump or change the mode of operation; replace impellers with ones containing more or less vanes if the piping responds to the vane passing frequency of the pump; increase the output of the pump by either installing a bypass that would run from the discharge to the suction of the pump or change the mode of operation; install acoustical filters to reduce the magnitude of the pressure pulsations [11].

2.3 Radial Thrust

Radial thrust is thrust imposed on the pump rotor that is directed toward the centre of rotation. Forces on the pump's rotor are usually comprised of a dynamic cyclic component, which is superimposed onto a steady state load. The dynamic component increases rapidly when the pump is operating in its recirculation state, during low flow in the system. The static load increases with both low and high flow operation, having its minimum value at or near the best efficiency point (BEP) flow rate [11]. This effect is often found in pumps of a single volute design, less in pumps of a double volute design, and rarely in pumps with a diffuser design. The geometry of the volute may also change the amount of radial thrust found in the system [3].

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 [11].

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 [11].

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 [11].

2.4 Axial Thrust

Axial thrust is imposed along the shaft axis, either in the inboard or outboard direction. The thrust is usually a result of a dynamic cyclic component, which is superimposed on a steady-state load in either direction. Increasing the dynamic load on the shaft may impose excessive stresses, which could ultimately lead to metal fatigue. The static, steady-state load may impose an excessive load on the bearing, which may increase the temperature to an unacceptable state and lead to a short bearing life. The majority of thrust bearing failures are fatigue failures due to the dynamic cyclic axial loads on the bearing components [11].

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 [11].

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 [11].

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 [11].

2.5 Suction and Discharge Recirculation

Recirculation usually occurs during reduced flows, and is the flow of some fluid around the impeller to the suction side. If this is found in the inlet of the impeller, then it is known as suction recirculation. If this is found at the outlet of the impeller, then it is known as discharge recirculation [11]. Recirculation is inevitable in every impeller design. The discharge recirculation can be reduced in design, but this would result in a reduction in the rated efficiency of the pump. The suction recirculation can likewise be reduced, which would result in an increase in NPSHR [9].

In order to avoid recirculation, the recommendation is not to exceed certain suction specific speeds (a dimensionless ratio describing operating conditions in a pump). Although useful, this advice cannot be applied blindly to all cases. During recirculation, heat is added to the fluid being pumped due to the pump losses. As a result, if the pump operates in this mode for an extensive period of time, temperatures may increase leading to vaporization and potentially an explosive and dangerous condition may exist. This may result in a limiting the flow through the pump [9].

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 [11].

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 analysed 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 [11].

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 [11].

3 MECHANICAL FAILURE

Mechanical failures can arise from a number of different parts of the pump, including bearing, seals, lubrication in the pump, and other miscellaneous problems. This section will cover the main mechanical problems, reasons behind them, and solutions, if any.

3.1 Bearing Failure

Bearings fail, in general, either through contamination of the bearing oil by water, another liquid, or solid particles or because of high heat, which is often caused by an overload on the bearing or by excessive lubrication [9]. These faults can have various root causes.

Moisture contamination in the bearings comes from multiple sources: packing leakage flows back to the bearings, and seeps into the bearing housing; using a water hose to clean a packing leakage, which would allow water to splash into the bearing case and into the housing through the vent or grease seal area; replacing the heated air through the bearing casing vent with cooler atmospheric air when the bearing case cools down; or the leakage of steam, condensate or cooling water from a mechanical seal quench gland and directed at the radial bearing [18].

When the pump is operating at its BEP, the only bearing loads are due to the weight of rotating assembly, the stress caused by the interference fit on the shaft, and any bearing preload that has been specified by the manufacturer. However, most pumps do not work continuously at their BEP, and thus overloading is possible [19]. Overloading of bearings can be a result of many different conditions: an unbalanced rotating element; a bent rotating shaft ; blocked impeller balance holes; cavitation; excessive axial thrust; excessive radial load caused by low flow operation or some mechanical failure inside the pump; excessive heating of bearings due to improper dissipation of heat; improper cooling of the bearings such as cooling the bearing housing with a water hose or some other similar system; increased pump internal running clearances around the wear rings; increasing the speed of the bearing; misalignment between the pump and its driver; operating the pump away from its best efficiency point (BEP), which may create excessive radial forces on the impeller that would felt by the bearing; pipe work exerting strain; pulley driven pump drivers; pumping a high specific gravity fluid such as sulphuric acid, which can almost double the radial load on a bearing; pushing the bearing too far up a tapered sleeve; rusting of antifriction bearings because of water in the bearing housing; having the thermal expansion of the shaft greater than the thermal expansion of the bearing; causing the bearing housing to be out of round; having the impeller located too far away from the bearing; having the wrong interference fit between the bearing and the shaft (the shaft was out of tolerance causing the bearings to be too tight); having the vane passing frequency coincident with the resonance of the pump assembly; having the vibration of almost any form from other parts of the piping system or within the pump; or producing water hammer in the lines [3, 18, 20, 21] .

Causes of overheating may be due to a lubrication problem in the bearing. In this case, this may be due to having: the oil level too low or too high; the wrong grade of oil; moisture in the oil; too much grease inside the bearing; lack of lubrication, in general; or foreign matter inside the lubricant, often due to shaft seal leakage [9, 21].

Moisture in bearings can also cause failure due to such problems as pitting and corrosion of the bearing races and rolling elements that increases the fatigue of the metal components, hydrogen embrittlement due to the free atomic hydrogen in the water which leads to an acceleration of the fatigue in the bearing metal, and the lack of a good lubricating film due to the non-mixing of water and oil. [18] Ball bearings generate vibrations that cover a wide range of high frequencies that may not be a multiple of the shaft running speed. Vibration caused from bearing faults can often by relatively subtle in the early stages of the bearing degradation, and will not be noticed in the overall vibration signal and more sophisticated signal processing of the vibration signal is often needed. Stress waves and shock pulses on the pump bearing housing will show failure trends that prelude an increase in the level of mechanical vibrations. Monitoring these waves or pulses falls under the category of acoustic high-frequency monitoring or incipient-failure detection (IFD) [11], or narrow band envelope analysis. Definitive early detection of bearing faults would be unlikely to be detected outside of vibration or acoustic emission measurements.

Over the course of the years of using bearings, procedures have been used to keep water, moisture and dirt out of the bearing housing, and overcoming bearing overheating and wear. There are many different methods used by pump companies to keep moisture and water out of the bearing housing.

3.2 Seal Failure

Mechanical seals may fail due to the pump running dry. Many applications are best protected by using dual pressurized mechanical seals, which remain lubricated even through periods of complete dry running. In non-hazardous applications, a pump sealed with packing that is lubricated from an external source will survive dry running better, given that the source is compatible with the fluid being pumped [22].

Mechanical seals, in general, fail for two reasons: the lapped faces open up, or one of the seal components becomes damaged. When a seal face opens, it allows solids to penetrate between the lapped surfaces. The solids embed themselves into the softer carbon / graphite face causing it to act like a grinding wheel, which then causes severe wear on the hard face. This type of failure accounts for the majority of mechanical seal failures [19].

Temperature gauges help assist in the diagnosis of a failed seal. Many fluids are affected by a change in their temperature, which may eventually lead to seal failure. This can happen, especially in the stuffing box, when one of the seal components (such as the elastomer, the seal faces or the metal parts) becomes destroyed, or the coated hard faces crack. This can be prevented by keeping the stuffing box temperature within the specified limits. In addition, by controlling the pressure in the stuffing box, one can control the temperature and prevent the fluid from vaporizing in the stuffing box or across the seal faces [19].

3.3 Lubrication Failure

The purpose of lubrication is to decrease the friction between two moving parts, thus decreasing their wear, and prolonging the life of the parts of the system. Non-contaminated oil cannot wear out and has a useful life of about thirty years at 30^{o} C. This life span is halved for every 10^{o}C rise in the temperature of the oil [18].

Overloading bearings can cause excessive heat to be generated within the bearings. The temperature rise will result in a decrease in the viscosity of the lubricant, which then leads to generation of more heat as it loses its ability to support the load. A varnish residue forms which then cokes at the elevated temperature. This coking will destroy the ability of the oil or grease to lubricate the bearing, as well as introduce solid particles into the lubricant [18]. Infrared thermography can be used to determine whether or not the bearing is overloaded by the amount of heat that is produced [23].

3.4 Excessive Vibrations

Excessive vibrations, or otherwise unsatisfactory or unacceptable vibrations, are classified, according to ISO 10816, to have amplitudes larger than 2.80 mm/s for small machines, 4.50 mm/s for medium machines, 7.10 mm/s for large machines with rigid foundations, and 11.2 mm/s for large machines with soft foundations. Vibrations result from unbalanced moving parts found within the pump system, interactions of the fluid and its particles with the pump and the connecting pipes, and movements of the pipelines themselves. There are numerous reasons that can lead to unwanted vibrations in a pump system, such as impeller unbalance, hydraulic imbalance, problems in the bearings, movement in the baseplate, component run-out, cavitation, air or vapour lock, and hydraulic excitation.

Impeller imbalance usually appears in the vibration signature as a 1x running speed frequency vibration, and may result from either a mechanical problem, such as a mechanical seal or bearing failure, or a hydraulic problem. Mostly, inspections to determine if an impeller is balanced are not performed until heavy pitting is found on the impeller. The degree of etching or pitting is usually used as an indicator that the impeller needs balancing. Unbalanced forces which result from loose fit impellers could results in large amplitude vibrations. Impellers may shift due to the decrease in residual stresses that are created when the impeller is cooled and contracted around the shaft. Shaft vibrations and flexing tend to result in the impeller cocking or bowing the shaft, which removes it from its original balance along its centreline. Hence, when balancing an impeller, it is imperative that it must be balanced at the operating speed so as to evaluate the importance of shaft deflection due to the previously mentioned factors and due to possible modal resonance components [11].

Hydraulic imbalance is mainly due to an uneven flow distribution within all the vane passages, which causes a 1x running speed frequency vibration. Unsteady flow into the impeller may result in axial thrust and high axial vibrations. In double suction impellers, the nonsymmetrical positioning of the impeller or the offset of the upper or lower case half will result in a 1x unbalance due to nonsymmetrical flow. In addition, recirculation forces and pulsation recirculation, which are present in a pump when the flow is less than that for which the pump and the piping system were designed, may result in noise and/or vibration with random frequencies, and an increase in pressure. Increasing the NPSHA has been found to help alleviate this problem by using a bypass [11].

Baseplates have also been known to produce unwanted vibrations. Trends in pump and pipeline construction have moved from low rotating speeds and cast iron baseplates to higher rotating speeds with less rigid fabricated steel baseplates. These higher rotating speeds, with higher operating temperatures, coupled with larger impellers have greatly increased the probability of baseplate vibrations, distortions, and decreased stiffness for the rotor dynamics occurring within the operating range for the system. As a result, baseplate vibration problems must be resolved at the design stage or during construction of the system. Alterations to the design to alleviate these problems may include levelling screws, grout filling holes, venting holes, and corrosion protection [11].

Insufficiently stiff bearing housings, or improperly designed baseplates, may cause bearing housings or casings to vibrate at resonance. Even though the shaft vibration amplitude may not be excessive, and even almost constant with speed, the bearing housing may show distinct resonance peaks with significant amplitudes. This can be determined by impact or modal testing. By changing masses or the stiffness of components in the system, the natural frequencies can be moved outside of the operating frequency range [24].

Shafts also produce unwanted vibrations due to a mass imbalance, or a natural bow in the shaft. Mass imbalances and bows produce a 1x running speed frequency, depending on whether there is only one defect in the shaft. Multiple defects produce more complex vibrations. Component run out, or a misalignment between components, especially annular seals, produces unwanted vibration. Shafts that produce a component run-out, which is separate from the shaft bow, produce a radial force [24]. Other sources of excessive vibrations may be due to: air or vapour lock in suction; inlet of suction pipe insufficiently submerged; pump and driving unit incorrectly aligned; worn or loose bearings; impeller choked or damaged; foundation not rigid; coupling damaged; or pipework exerting strain on the pump [9].

An area of controversy in the topic of vibrations for a centrifugal pump is the cause of a subsynchronous vibration at 0.7 – 0.9 of the running speed. One theory says that it is a result of destabilizing forces developed by the angular rotation of liquid in the internal running clearances. Another theory is that the excitation is due to the rotating stall at the inlet of the impeller, which is a phenomenon that is widely discussed in the literature. Field experience shows that in variable speed pumps, a subsynchronous vibration is seen to exist over a part of the pump's operating range, that ceases as the pump approached the design flow [21] .

Several authors [25] have published a list of vibration frequencies that can be found in a centrifugal pump, and the possible causes of each vibration. In addition, [25] published a table that contains the stages of bearing degradation and the vibrations associated with each stage. Various testing performed on a centrifugal pump can also provide useful information concerning the state of the pump. Impact testing of casing/ bearing housing may detect the proximity of natural frequencies to synchronous frequency and diagnose structural resonances. Tests using speed variations may detect structural resonance excitation and/or rotor critical speeds. Tests using flow variations at fixed speed may detect hydraulic unbalance. Tests using a change of temperature may detect misalignment due to thermal growth; and a change in casing natural frequency due to jamming / loosening of keyways. And finally, tests using trim balancing may also determine a coupling unbalance [24].

3.5 Fatigue

Fatigue is mainly categorized into one or more of the following: cyclical loading, material fatigue, or environmentally assisted fatigue. Pumps are basically machines that have a fluid, with or without solids, inducing a cyclic load on their components. Although centrifugal pumps are mainly steady-state rotational equipment, pulsations or fluctuating applied stresses are encountered. The source of these cyclic stresses can be from fluid interaction between impeller exit vanes and diffuser vanes, or, in the case of a volute pump, the impeller vanes and the casing at the cutwater. Mechanically induced forces result from the bending moments that act on the pump shaft or a component imbalance in the rotor assembly [11].

When cyclic forces are applied to parts found in the pump, a crack may appear over a period of time, in a component's surface. Fractures can occur on a component, even though the loading produces stresses that are far less than the tensile strength of the material. After the crack has been introduced into the component, the crack may grow with each cyclic loading until the component finally fractures [11].

Corrosion assisted fatigue occurs when corrosion damage changes the surface texture and significantly increases the local stresses acting on a pump component due to notch sensitivity. When this happens, fatigue cracking of the component is likely. In some cases, crack propagation can be influenced by oxidation, which can mask the features of the fatigue mechanism. Corrosion oxides, which form along the face of the crack, can produce a wedging effect, which mechanically is able to increase the local tensile forces acting on the crack tip, and thus, increase the rate of crack propagation. Fretting or wear can also produce sites where fatigue cracking can initiate. In addition, sharp radii and defects on the material surface, such as in the case of porosity and poor machining, can act as stress concentrations in susceptible materials [11].

Fatigue can be seen in its three stage progression from (1) crack initiation, which is sometimes associated with pre-existing defects, (2) crack propagation, and (3) final failure. The applied stress level, geometry, flaw size, and mechanical properties determine the existence and extent of these stages of fatigue. These three stages of fatigue cracking can be observed on the fracture surface, provided that there is no secondary damage that masks the characteristic appearance. The bands that result from the fatigue are often referred to as "clamshell markings", "crack arrest lines", or "beach marks" and reflect different periods of the crack growth. "Ratchet lines", which are the joining of two different crack fronts on different planes, are observed in cracks originating from multiple origins. These types of fatigue fractures are normally associated with rotating components. Once the cause of the fatigue cracking has been identified, corrective actions can be taken to remedy the problem. These corrective actions may include: using fatigue resistant materials, modifying the design of the pump, treating the affected surface, or using more highly corrosion resistant materials [11].

4 OTHER MODES OF FAILURE

There are some modes of failure that fall neither under the categories of hydraulic or mechanical problems. These modes of failure are sometimes structural, such as in the case of erosion and corrosion, or are a result of a multitude of different sources, such as in the case of excessive power consumption. This section will cover the causes and solutions, if any, of these types of problems.

4.1 Erosion

Erosion on pumps take on one of five different forms: cavitation erosion (see 2.1), adhesive wear, fretting, abrasive wear, or erosion by solid particle impingement [11].

Adhesive wear is a result of material to material contact. This is the primary cause of material loss when handling fluids that contain no solids in the stream. Surfaces of parts inside the pump are able to have material-to-material contact, producing surface disruptions, material grooving, transferral of material, and possibly, galling. Two important characteristics that should be considered for materials that come into contact are their adhesive wear traits and their galling threshold. Adhesive wear is the only theory of sliding wear that is able to offer a general wear equation that quantifies the prediction of wear [11].

Fretting is a special case of adhesive wear. It occurs when two parts of the pump experience repeated, small amplitude relative motion between close-fitting surfaces such as between an impeller and a shaft. In a pump, there is the potential for fretting in the small amplitude motion of loose fitting impellers, beneath loose bearings, and between impeller wear rings and the impeller hub. Although not placed into the design by the engineer intentionally, it can still happen and lead to catastrophic results. And since the amplitudes are small, it makes fretting almost impossible to detect. Fretting can be recognized on parts in the pump when a red powdery oxide forms along the fretted surface. In a pump, however, this red-coloured debris is often washed away, although a distinct damaged surface appearance will develop on the fretted surfaces. This type of damage is often described as having coloured spots or blotches in its appearance, along with an eroded surface with what appears to be random damage. And even though the red-coloured debris may be washed away with the fluid, some staining of the adjacent components may be observed after the pump is disassembled. Fretting can be avoided with tighter clearances between two components, or shrink fitting the components to prevent small, unnecessary movement. If fretting is unavoidable, then various methods, such as coating or lubricating the contact surfaces, are advised. Coatings may include flame-sprayed high-nickel alloys, silver plating, or possibly adding a thin, dense chrome plating to one or both of the faces in contact [11].

Abrasive wear is characterized by solids interacting with internal components, as either two-body or three-body wear. Three body abrasive wear is the primary mechanism of damage in centrifugal pumps. This occurs when hard solid particles, which are found in the pumped fluid, enter between the ring fit areas or the impeller keyway faces. To minimize this effect, a few variables must be taken into consideration. The wear ring clearance, which is the clearance between the casing ring and the impeller ring, influences the amount of damage on the pump. If the particles in the fluid are too large or too small to be trapped within the clearance, then the damage will not occur. However, if there are particles that barely fit the wear ring clearance, and become lodged between the two rings, then damage can occur. Since fluids usually contain particles of different sizes, various conditions will exist [11].

Finally, the last category of erosion is due to solid particle impingement. Many fluid-handling applications require pumps to move fluids that are not clear liquids. Solid particles found in these fluids may be removed with the use of a costly filtration system, with impact on fluid velocity, and pressure heads. In addition to the particles in these fluids from their source, either naturally such as from river or sea water or man-made, fabricated piping systems may also introduce solids in the fluid from weld slag and pipe burn. There are several factors that should be taken into consideration when pumping fluids with particles found in the flow: the hardness of the particle, the concentration of the particles, size distribution, geometry, velocity of the fluid flow, and the angle of the fluid impingement. Particle velocity is an important component in the degree of damage that occurs in a pump handling slurries. The pump gives the entrained particle kinetic energy that, when striking a hard surface, erodes the material. In addition, the pump's ability to absorb the kinetic energy of the particle, based on the material's hardness and/or resilience, also plays a role in the amount of material loss upon impact. The characteristic feature of erosion damage due to solid particles striking the pump is usually recognizable; however, sometimes it is mistaken as corrosion-erosion or vice versa. In cases like this, improper diagnosis will lead to an improper solution to the problem [11].

4.2 Corrosion

Corrosion is the chemical alteration of a material. Corrosion comes in 7 different forms (general, dealloying, galvanic, stress corrosion cracking, hydrogen embrittlement, microbiologically induced corrosion, intergranular corrosion), some of which are more common than others. General corrosion occurs without any localization of attack. This type of corrosion occurs on metals or alloys that do not develop an effective passive film on the surface. Corrosion usually occurs as a result of oxidation, and metal oxide is produced. Most often, pumps made with carbon steels, cast iron and copper base alloys experience this type of corrosion. Carbon steel, which is sometimes used in pump applications, does not develop a protective oxide film and will corrode at a rate that is dependent upon factors such as temperature, oxygen content, pH, fluid chemistry, and the velocity of the fluid. For most pump applications, since the corrosion rate is too high for this material to provide a useful life, a form of protective coating is placed over the steel to prevent corrosion [11].

Dealloying, the second type of corrosion, is a result of the removal of one phase from a multi-phase alloy or one element from a material. One of the most common cases of dealloying found in pumps is the graphitic corrosion of gray cast iron. Due to its low cost, easiness to machine, and its versatility, it is used in a wide variety of applications in the waterworks industry and in the pump industry. Although this type of corrosion is found in both fresh water and salt water, the high conductivity of salt water increases the corrosion rate. However, the corrosion rate can be slowed down if the water has a high mineral content, since the minerals tend to plug up the apertures left behind from the de-graphitization, sealing off the base metal from exposure to the passing fluid, and thus reducing the corrosion rate [11].

Galvanic Corrosion is the third type of corrosion, and occurs when one alloy is electrically coupled to another and is exposed in a conductive liquid. Several factors affect the rate of galvanic corrosion, such as the conductivity of the fluid, where seawater has less conductivity than fresh water; the ratio of the area of the coupled metals; the negative potentials of the metals; and the use of coatings to hinder the galvanic corrosion process [11].

The fourth type of corrosion is called Stress Corrosion Cracking (SCC). This form of corrosion is dangerous since it is not usually detected until it has advanced to a stage that can cause catastrophic failure. Although uncommon in most pumps, this type of failure can occur in several classes of material and thus the pump designer should be aware of its potential. The factors aiding SCC include tensile stress (residual or applied), a susceptible material, an environment capable of causing stress corrosion, and time [11].

Hydrogen embrittlement is the next type of corrosion that results from the combination of hydrogen and a residual or applied tensile stress. This type of damage results in cracking, blistering, hydriding, or a loss of ductility. Damage done by hydrogen embrittlement is occasionally found in pumps as a result of plating processes, such as chrome plating, that is used to rebuild pump shafts [11].

Microbiologically induced corrosion is a result of microbiological activity, which is mostly found in stagnant water. This type of corrosion occurs most frequently when stagnant water remains in the pump when shut down for an extended period of time. Sulphate producing bacteria, which are found in many waters, form tubercles (slimy, reddish hemispherical shaped mounds or colonies) on carbon steel or cast iron. Upon scraping off, they leave behind a saucer-shaped pit, inside of which will be a wet, black deposit. The pitting is a result of sulphuric acid excreted by the bacteria; however, this will not result in premature failure. In cases such as this, biocides have been found on occasion to help alleviate the problem. Finally, the decay of biological organisms can also generate hydrogen sulphide, which affects the protective oxide film on copper base alloys. This biological activity can impair the corrosion resistance of bronzes [11].

The final type of corrosion is called intergranular corrosion. This infrequent type of corrosion is usually caused by local chemical differences, such as the chrome-depleted regions of an austenitic stainless steel plate. Bronze alloys susceptible to this type of corrosion include aluminium brasses, silicon bronzes, muntz metal, and admirality metal. This type of corrosion often leads to corrosion-assisted fatigue cracks when cyclic loading is applied to the metal [11].

4.3 Excessive Power Consumption

Excessive pump power is an indication that there may be one of a number of different problems going on with the pump. One of the main problems that results in excessive power consumption, and possibly ultimately a motor trip, is the existence of small or high concentrations of particles in the fluid. Even though the fluid looks clean, it may have very small concentration of particles that can lead to long-term performance problems as running clearances gradually increase from the wearing down of parts. This can be found in any centrifugal pump application, regardless of how clear or clean the fluid looks like. This wearing away produces a loss of hydraulic performance, which will cause the control valve to gradually open wider without being noticed. The wider it becomes, the more power is required from the motor, which may result in a motor trip [26].

Other main causes of faults in a pump that could cause excessive power consumption are: the speed of the impeller is too high; the shaft packing is too tight; the liquid that is being pumped has a viscosity that is more than what the pump was designed for; there is a misalignment in the pump; the impeller is touching the case; the pumped liquid is denser than specified; the impeller is rotating in the wrong direction; the impeller is installed in the wrong direction [27]; the pump has trapped air inside; there is a serious leak in the delivery line to the pump; the pump is delivering more than its rated quantity; the neck rings on the impeller are worn excessively; the impeller is damaged; there is a mechanical tightness at the pump's internal components; or the pipework is exerting a strain on the pump[9].

Corrective procedures that may be taken in order to eliminate the causes of the excessive power consumption include: periodic tracking of process control valve position versus flow rate; periodic recording of motor current and voltage if the pump is operating on the head/flow curve where the head flow curve is not steep; periodic measurement of pump shut-off head; renew defective gaskets; repair existing leaks; reduce the speed of the impeller; dismantle pump and restore clearances to original dimensions; stop pump, close delivery valve to relieve internal pressure on packing, slacken back the gland nuts and retighten to finger tightness; dismantle pump and renew impeller; dismantle pump, check internal clearances, and adjust as necessary; or stop pump and reprime [9, 26].

Due to the number of faults which can cause excessive power consumption, in order to have a correct diagnosis of the underlying fault a variety of condition based measurement data is needed, from process information to temperature and vibration measurements

4.4 Blockages

From anecdotal data in logs obtained from companies in the water supply and sewerage industry, the authors conclude that blockages can also be a major fault mode of centrifugal pumps, however little evidence is able to be found in literature on this failure mode. The pumped fluid contains materials, such as rags or bricks that either wrap around the impeller to hinder and possibly stop the impeller's rotation, or clog the passages between impeller vanes to decrease the flow rate. Corrective procedures involve stopping the pump, removing the objects responsible for the blockage, possibly repriming the pump, and then returning it back online.

5 DISCUSSION

Older pumps in the field contain no sensors for measuring data such as temperature, displacement, acceleration, pressure, or rate of fluid flow. Newer pumps can be provided with temperature, vibration, and leakage sensors as well as power analysers to determine the state of the electric motor and associated electronic equipment on the pump. However, despite the provided information, there is no onboard interpretation and assessment of the collected data. Most pump suppliers would provide the raw data from sensors and information of normal operating ranges, leaving it up to the end user to determine if the information is to be used in the diagnostics and the prognostics of the pump. In most cases, very little of this information is utilized by the end user, and pumps are still left in either a run-to-fail mode or scheduled maintenance cycle. Without utilizing data obtained from multiple types and location of sensors to diagnose problems, the state of centrifugal pump diagnosis will remain stagnant in its current practice.

There has been a shift in the focus of research papers over the past two decades from having operator interpretation of mechanical and hydraulic signs of failure to creating expert systems that utilize neural networks, genetic algorithms and fuzzy logic to interpret the gathered information and present the end user with a diagnosis of the current state of the pump. These types of diagnosis include stating the existence and severity of the fault, and attempting to forecast the remaining life of the pump. Faster processing speeds on computers, smaller and faster computer hand held devices, smaller sensors, and better methods of transmitting data wirelessly or by wired connections, provides better means today of obtaining the raw data, interpreting the data either on site or at a local computer, and providing the end user in a distant location the interpreted results of the obtained signals. Most of the current types of diagnostic systems attempt to diagnose a few failure modes, focusing on hydraulic and mechanical failures, while not focusing on others due to the difficulty of detection. For example, failure modes, such as corrosion, may not be easily detected except through visual inspection which causes problems if the expert system is relying on vibration or hydraulic sensors only.

As a result of the lack of any single system that diagnoses all the major failure modes, such as those presented in this article, a rotating machine health management (RMHM) system or asset manager is needed in industry to shift the current method of diagnostics to a more advanced predictive maintenance program utilizing both condition monitoring and process data. This type of monitoring approach would utilize a variety of sensors to obtain a more informed view of the state of the pump to provide a diagnosis of the nature and severity of the pump problem. Predictive maintenance approaches will provide industry end users with savings in both time and money. This type of holistic approach to pump maintenance is still in its infancy, and thus is not available on pumps in the field as of yet.

6 CONCLUSION

A review of 13 main problems that afflict centrifugal pumps in the water supply industry has been presented. Various issues dealing with each problem, such as their cause and effect, symptoms, and mechanical corrective procedures was included. Detail was given of symptoms that an engineer could use in order to diagnose the problem, whether by visual inspection, audio inspection, or vibration. Commentary was provided detailing the gaps in industry where this type of information, which is sometimes widely known, is not used to the extent that it should be via sensors or other instrumentation attached to the pumps, and expert systems to interpret the information. The future vision is of an integrated condition monitoring and asset health manager system for pumps and pump stations capable of alerting the operators of impending problems with enough warning so as to reduce maintenance costs, improve equipment availability and prevent significant infrastructure damage by removing catastrophic failure.

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8 REFERENCES

[1] B. Hart, "NPSH Curve. (Image)," http://www.pumpingmachinery.com/pump_magazine/pump_articles/article_15/article_15.htm, 2005.

[2] 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.

[3] S. Shiels, "How centrifugal pump hydraulics affect rolling element bearing life," *World pumps,* vol. 1998, pp. 32-35, 1998.

[4] S. Shiels, "Optimizing centrifugal pump operation," *World pumps,* vol. 2001, pp. 35-39, 2001.

[5] A. J. Stepanoff, *Centrifugal and axial flow pumps : theory, design and application*. New York: Wiley, 1957.

[6] I. Karassik, *Centrifugal pump clinic*, Second ed. New York: Dekker, 1989.

[7] S. Shiels, "Centrifugal Pump Academy : Causes of intermittent and chronic cavitation," *World pumps,* vol. 1998, pp. 57-60, 1998.

[8] S. Shiels, "Centrifugal pump troubleshooting Part two: a retrospective approach," *World pumps,* vol. 2001, pp. 38-42, 2001.

[9] R. Rayner, *Pump Users Handbook*. Oxford: Elsevier Advanced Technology 1995.

[10] R. K. Turton, *An introductory guide to pumps and pumping systems*. London: Mechanical Engineering Publications 1993.

[11] Karassik, Igor J., J. P. Messina, P. Cooper, and C. C. Heald, *Pump Handbook (3rd Edition)*. New York: McGrawHill, 2001.

[12] R. Palgrave, "Diagnosing Pump Problems From Their Noise Emissions Signature," in *New Challenges – Where Next? – 11th International Conference of the British Pump Manufacturers' Association* UK: BHRA, 1989, pp. 9 - 28.

[13] T. Hicks and T. W. Edwards, *Pump application engineering* New York: McGraw-Hill 1971.

[14] J. Tuzson, *Centrifugal pump design* New York: Chichester: Wiley, 2000.

[15] S. Shiels, "Protecting centrifugal pumps at low flow," *World pumps,* vol. 1999, pp. 25-27, 1999.

[16] A. Ismaier and E. Schlücker, "Fluid dynamic interaction between water hammer and centrifugal pumps," *Nuclear Engineering and Design,* vol. 239, pp. 3151-3154, 2009.

[17] W. S. Everett, "Predicting Centrifugal Pump Pressure Excursions (Noise)," in *25th annual International Gas Turbine Conference and Exhibit and the 22nd annual Fluids Engineering Conference*, New Orleans, 1979 pp. 287 – 289.

[18] W. Mc Nally, "Troubleshooting the ball bearings in a centrifugal pump," *World pumps,* vol. 2004, pp. 28-29, July 2004 2004.

[19] T. Sahoo, "Making centrifugal pumps more reliable," *World pumps,* vol. 2009, pp. 32-36, 2009.

[20] S. Shiels, "Troubleshooting centrifugal pumps: rolling element bearing failures," *World pumps,* vol. 2001, pp. 28 - 30, 2001.

[21] I. Karassik and J. T. McGuire, *Centrifugal pumps*. New York: Chapman & Hall, 1998.

[22] S. Shiels, "Successful centrifugal pump commissioning and start-up," *World pumps,* vol. 2000, pp. 24-26, 2000.

[23] R. A. Epperly, G. E. Heberlein, and L. G. Eads, "A tool for reliability and safety: predict and prevent equipment failures with thermography," in *Petroleum and Chemical Industry Conference, 1997. Record of Conference Papers. The Institute of Electrical and Electronics Engineers Incorporated Industry Applications Society 44th Annual*, 1997, pp. 59-68.

[24] U. Bolleter and A. Frei, "Courses and control of synchronous vibrations of multi-stage pumps," in *The Fluid Machinery Committee of the Power Industries Division of the Institution of Mechanical Engineers* Institution of Mechanical Engineers Mechanical Engineering Publications for the Institution of Mechanical Engineers, 1990, pp. 1- 6.

[25] R. S. Beebe, *Predictive Maintenance of Pumps Using Condition Monitoring*. Oxford: Elsevier Advanced Technology, 2004.

[26] S. Shiels, "Centrifugal pump academy : Motor trip! Predicting the unforeseen disaster," *World pumps,* vol. 1999, pp. 29-31, 1999.

[27] F. A. Holland and F. S. Chapman, *Pumping of liquids*. New York: Reinhold Publishing Corporation, 1966.