VIBRATION AND NOISE IN CENTRIFUGAL PUMPS -SOURCES AND DIAGNOSIS METHODS

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VIBRATION AND NOISE IN CENTRIFUGAL PUMPS - SOURCES AND DIAGNOSIS METHODS

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ABSTRACT

In every centrifugal pump, dynamic forces of mechanical and hydraulic origin are present and a certain vibration and noise is therefore inevitable. To ensure the safety of the pump and associated plant components, the vibration and noise must be kept within certain limits. If the mechanical state of the pump and its drive are good, the inflow conditions are in order and the duty point is admissible, these limits can be observed without difficulty. Higher vibrations ultimately results in decreased component life due to cyclic loads, lower bearing life, distortion to foundation, frequent seal failures etc. Similarly noise has got huge impact on working environment and comfort conditions of an individual. Exact diagnosis of vibration and noise sources is very difficult in centrifugal pumps as this may be generated due to system or the equipment itself. In this paper an attempt has been made to address some general causes of noise and vibrations, its diagnosis and remedies in centrifugal pumps.

1.0 INTRODUCTION

Vibrations basically are the displacement of a mass back and forth from its static position. A force will cause a vibration, and that vibration can be described in terms of acceleration, velocity or displacement. The force that will cause the vibration, must overcome the structure's mass, stiffness and damping properties. These properties are inherent to the structure and will depend on the materials and design of the machine. Whereas noise is one of the derivatives of vibrations. Both phenomenon affect the centrifugal pump performance and its service life adversely.

Sources of vibrations and noise are well known but the methods to trace the exact source are still in development stage. The major challenge in diagnosis of vibrations and noise in centrifugal pumps is service of the centrifugal pump itself. When we compare the machine tools or other utility equipments with centrifugal pumps, diagnosis of the sources of noise and vibrations in machine tools is simpler than pumps as all the components are mechanical and are visible. Whereas in centrifugal pumps, the root of vibrations and noise may lie in mechanical or hydraulic aspects. It is very easy to trace the mechanical causes but it becomes very difficult to trace hydraulic causes. This makes pumps vibration and noise diagnostic very complex.

2.0 VIBRATIONS IN CENTRIFUGAL PUMPS

It is necessary to be interested in vibration in centrifugal pumps because it has a major effect on the performance. Generally, increasing vibration levels indicate a premature failure, which always means that the equipment has started to destroy itself. It is so because excessive vibrations are the outcome of some system malfunction. It is expected that all pumps will vibrate due to response from excitation forces, such as residual rotor unbalance, turbulent liquid flow, pressure pulsations, cavitation, and/or pump wear. The magnitude of the vibration will be amplified if the vibration frequency approaches the resonant frequency of a major pump, foundation and/or piping component. Generally higher vibration levels (amplitudes) are indicative of faults developing in mechanical equipment.

2.1 Sources of vibrations in centrifugal pumps

The sources of vibration in centrifugal pumps can be categorized into three types Mechanical causes, Hydraulic causes & Peripheral causes.

2.1.1 Mechanical Causes of Vibrations

The mechanical causes of vibrations includes –

- Unbalanced rotating components,
- Damaged impellers and non concentric shaft sleeves
- Bent or warped shaft
- Pump and driver misalignment,
- Pipe strain (either by design or as a result of thermal growth),
- Inadequacy of foundations or poorly designed foundations
- Thermal growth of various components, especially shafts,
- Rubbing parts
- Worn or loose bearings, Loose parts,
- Loosely held holding down bolts
- Damaged parts.

2.1.2 Hydraulic Causes of Vibrations

The hydraulic causes of vibrations includes –

- Operating pump at other than best efficiency point (BEP)
- Vaporization of the product
- Impeller vane running too close to the pump cutwater
- Internal recirculation
- Air entrapment into the system through vortexing etc.
- Turbulence in the system (non laminar flow),
- Water hammer.

2.1.3 Peripheral Causes of Vibrations

The peripheral causes of vibrations includes –

- Harmonic vibration from nearby equipment or drivers.
- Operating the pump at a critical speed
- Temporary seizing of seal faces (this can occur if you are pumping a non lubricating fluid, a gas or a dry solid, a pump discharge recirculation line aimed at the seal faces.)

3.0 NOISE IN CENTRIFUGAL PUMPS

Noise is an important quality and comfort feature of any centrifugal pump. Abrupt noise in pumping system for most of the times is an outcome of vibrations set in pumping system. Although a certain amount of noise is expected from centrifugal pumps and their drives, usually high noise levels (in excess of 100db) is an indication of potential mechanical failure or vibration problems. The occurrence of significant noise levels indicates that sufficient energy exists to be a potential cause of mechanical failure of the pump and piping. Noise in centrifugal pumps and systems can be generated by mechanical motion of pump components and by liquid motion of the pumps and system.

3.1 Mechanical sources of noise

Common mechanical sources of noise include vibrating pump components or surfaces because of pressure variations that are generated in the liquid or air. Impeller or seal rubs, defective or worn bearings, vibrating pipe walls and unbalanced rotors are the examples of mechanical sources of noise in pumping systems.

Improper installation of couplings in centrifugal pumps often causes noise at twice pumping speed (Misalignment). If the pump speed is near or passes through lateral critical speed, noise can be generated by high vibrations resulting from imbalance or by rubbing of bearings, seals or impellers. If rubbing occurs, it is characterized by a high-pitched squeal.

3.2 Hydraulic sources of noise

Hydraulic sources of noises includes following –

3.2.1 Transients

Starting and stopping of pumps with the attendant opening and closing of valves is a major cause of severe transition in pumping systems. The resulting pressure surges, known as water hammer can apply a sudden impact force on pump. The discrete energy produced during the course is then dissipated as noise and is way above the audible noise intensities. Rapid closure of conventional valves used in feed water lines can cause severe water hammer. Analytical methods are available to evaluate severity of water hammer in a particular piping configuration for different closure rates.

3.2.2 Instationarity of flow

The instationarity of the flow is caused by secondary flows in the impeller due to rotation, the finite number of blades and finite blade thickness, but also by effects of turbulence. The finite number of blades leads to a secondary flow, caused by the asymmetric outgoing flow of the impeller. The finite blade thickness causes a notch in the wake flow. Both effects result in a time-dependent incident flow on the resting parts (guide vanes of the diffuser, volute tongue of the guide casing) and consequently excite vibration of these parts. Instationarity of flow is combined effect of pressure pulsations and turbulence.

3.2.2.1 Pressure pulsations: Pressure pulsations are detected at discrete frequencies that are multiples of the rotating frequency and the number of blades; these frequencies are also called blade passing frequencies (BPF):

Blade Pass Frequency (BPF) = number of blades (or vanes) × rpm

The amplitude of these pressure pulsations depends on a number of design parameters of impeller and diffuser and operating parameters. One of the most important parameters is the distance between impeller and the volute tongue. Smaller distances typically result in much higher amplitude of the BPF. However, this distance also affects the efficiency of the pump. Therefore, in industrial practice often a compromise has to be found between hydraulic and noise specification.

3.2.2.2 Turbulence In pumps, flow turbulence induces vortices and wakes in the clearance space between the impeller vane tips and the diffuser or volute lips. Dynamic pressure fluctuations or pulsations produced in this way can result in shaft vibrations because the pressure pulses impinge on the impeller. Flow past a restriction in pipe can produce turbulence or flow induced vibrations.

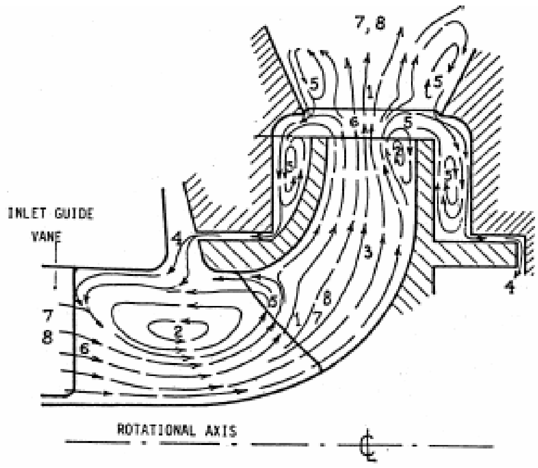

Fig1. Turbulence and vortices formation in flow through impeller

The pulsation could produce noise and vibration over a wide frequency range. The frequencies are related to the flow velocity and geometry of the obstruction. These in turn excite resonant frequencies in other pipe components. The shearing action produces vortices that are converted to pressure disturbances at the pipe wall, which may cause localized vibration excitation of the pipe or its components. It has been observed that vortex flow is even higher when a system's acoustic resonance coincides with the generated frequency from the source. The vortices produce broadband turbulent energy centered around the frequency determined by the following formula:

$$f = Sn * V / D$$

Where f = vortex frequency (Hz), Sn = Strouhl number (dimensionless, between 0.2 and0.5), D = characteristic dimension of the obstruction.

3.2.2.3 Cavitation: Cavitation normally generates random, high frequency broadband energy, which is sometimes superimposed with the blade pass frequency harmonics. Gases under pressure can dissolve in a liquid. When the pressure is reduced, they bubble out of the liquid. In a similar way, when liquid is sucked into a pump, the liquid's pressure drops. Under conditions when the reduced pressure approaches the vapour pressure of the liquid (even at low temperatures), it causes the liquid to vaporise. As these vapour bubbles travel further into the impeller, the pressure rises again causing the bubbles to collapse or implode. Thisimplosion has the potential to disturb the pump performance and cause damage to the pump's internal components. This phenomenon is called cavitation. Each implosion of a bubble generates a kind of impact, which tends to generate high-frequency random vibrations, as depicted in Figure 2. Cavitation can be quite destructive to internal pump components if left uncorrected. It is often responsible for the erosion of impeller vanes

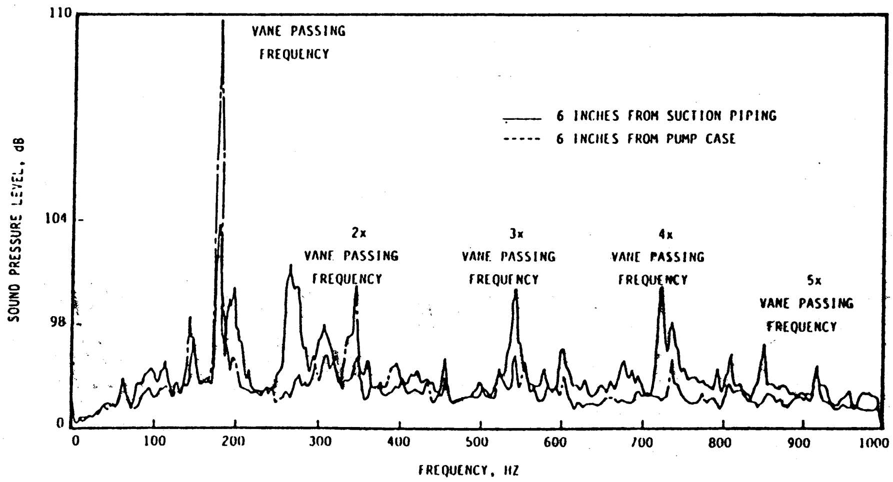

Fig 2 - Noise spectrum of cavitation

4.0 DIAGNOSIS OF NOISE AND VIBRATIONS IN CENTRIFUGAL PUMPS

The diagnosis of noise and vibrations in centrifugal pumps is divided into two steps –

- Noise / Vibration measurement.
- Analysis of measured vibration values.

4.1 MEASUREMENT

4.1.1 Noise measurement

Noise heard everyday is not a pure sound of only one frequency, but it is often a combined sound that includes various frequency components. Suppose two types of sources were measured and it was found that their sound pressure levels were the same. If the noise is composed of differing frequencies, the countermeasure of soundproofing to be taken would naturally be different. To know the characteristics of a noise and propose preventative measures, it is important to identify the frequency components of that noise. The sound pressure level per Hz is called the "spectrum pressure level". A sound in which all of the spectrum pressure levels are equal (whatever the pitch, the loudness is the same) is called "white noise". The preferred series of octave bands for acoustic measurements divides the audible range into ten bands. The centre frequencies of these bands are 31.5, 63, 125, 250, 500, 1000, 2000, 4000, 8000 and 16000 Hz. The actual nominal frequency range of any one of these bands is 2:1. This means that the effective band for the 1000-Hz octave extends from 707 to 1414 Hz. In a frequency analysis, a graph is made of the results of octave-band sound pressure level measurements. The frequency scale is commonly divided into equal intervals, between the position designated for each band and the position for the band adjacent to it in frequency. The pressure level in each band is normally presented as horizontal lines centered on the band at the measured level. Below is an example of octave band frequency analysis [fig 3]

4.1.2 Vibration measurement

Mechanical vibrations are most often measured using accelerometers, but displacement probes and velocity sensors are also used. Generally, a portable vibration analyzer is preferred. The analyzer provides the amplification of the sensor signal, it does the analogue to digital conversion, filtering, and conditioning of the signal. Many analyzers also offer advanced processing of the collected signals, as well as storage and display of the data.

It is also important to know the location to mount the vibration mounts. We know that a force cause vibration. If we know what types of forces are generating the vibration, we will have a good idea how they will be transmitted through the physical structure of the machine and where they will cause vibrations. With rotating machines, this point is almost always directly on the machine's bearings. The reason for this is that the various dynamic forces from a rotating machine must be transmitted to the foundation through the bearings. As a rule of thumb, vibration readings on rotating machines must be taken in the horizontal, vertical and axial direction on each bearing as shown in fig 4 and 5.

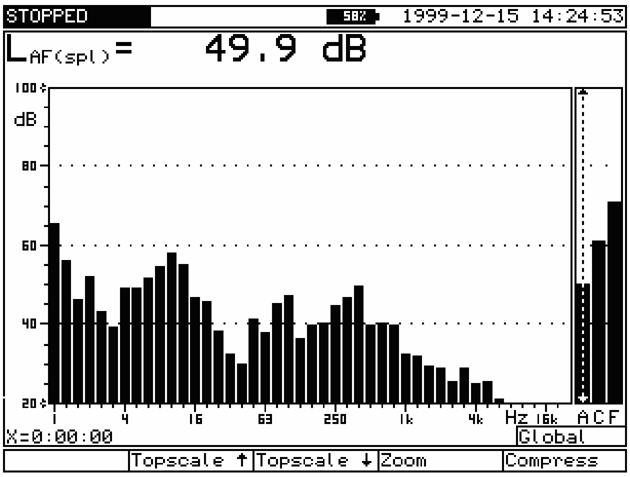

Fig 3 – Octave band frequency analysis

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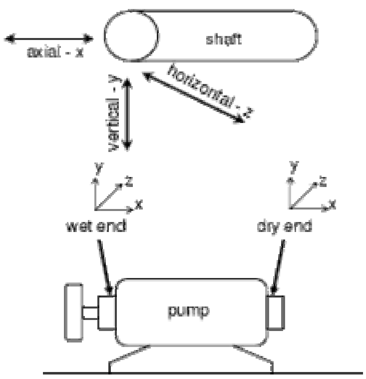

Fig 4 – Radial locations of vibration mounts

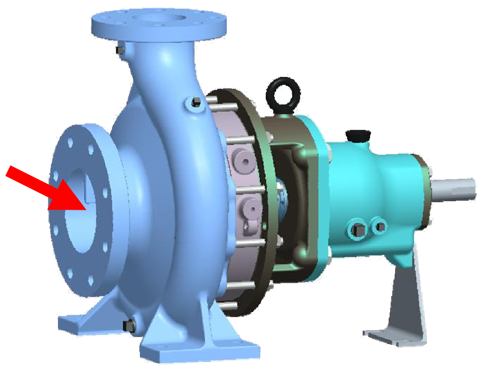

Fig 5 – Axial locations of vibration mounts

4.2 Analysis of Vibration measurements data.

There are many different methods available for analyzing vibration data. The most basic method involves displaying the vibration data in the frequency domain, also called the vibration spectrum. The frequency of the vibration is the number of vibration cycles per time unit. The vibration spectrum is fundamental to vibration monitoring, because it yields the information that is effectively "hidden" in the vibration waveform. Vibration spectra can be represented in various different ways, of which the Fast Fourier Transform (FFT) and the Power Spectral Density (PSD) are the most popular. The concept of the vibration spectrum can be simply explained by means of an example. Consider the time waveform in Figure 6, which has a frequency of 10 Hz (we can count ten complete cycles during one second) and amplitude of 5 mm (the units of the amplitude could be any unit related to vibration, e.g. displacement, velocity or acceleration).

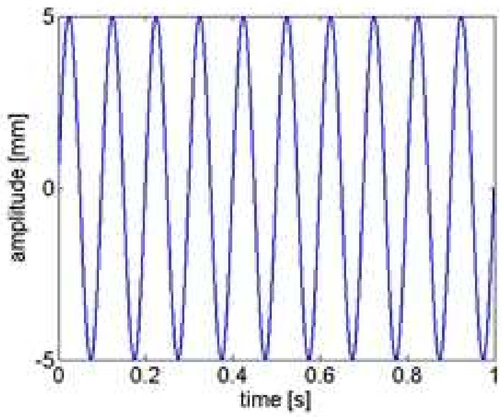

Fig6. Vibration spectrum

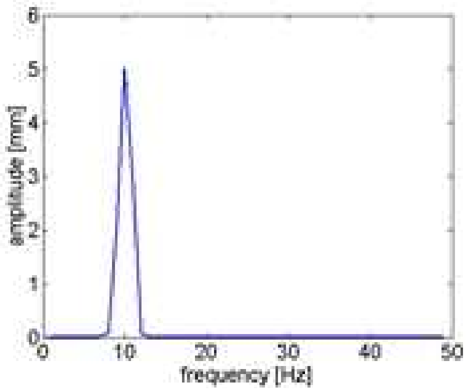

Fig7. FFT time waveform

The time waveform is a plot of time vs. amplitude, and is referred to as the time domain. The time domain signal can be converted into a frequency domain representation, which is in fact the spectrum. The spectrum is a plot of frequency vs. amplitude. The FFT for the time waveform from Figure 6 is plotted in Figure7. We can clearly read from this plot that the frequency content of the signal is 10 Hz, and that its amplitude at 10 Hz is 5 mm.

What now, if our time waveform has more than one frequency present? Let's take a look at another example. Consider the waveforms in Figure 8. In the top graph we have our 10 Hz waveform, called S1. The second waveform is a 25 Hz waveform with amplitude 2 mm, which we can call S2. The third waveform, plotted at the bottom, is S1+S2, which yields a much more complex waveform.

Let us examine the FFT for S1+S2, depicted in Figure 9. We can clearly observe two peaks in spectrum, namely a 5 mm peak at 10 Hz, and a 2 mm peak at 25 Hz. Thus, by only looking at the spectrum, we can characterize our S1+S2 waveform much better than by examining the waveform from Figure 8. The frequency spectrum is hence much easier to interpret and gives us information that is often impossible to observe by just looking at the time waveform.

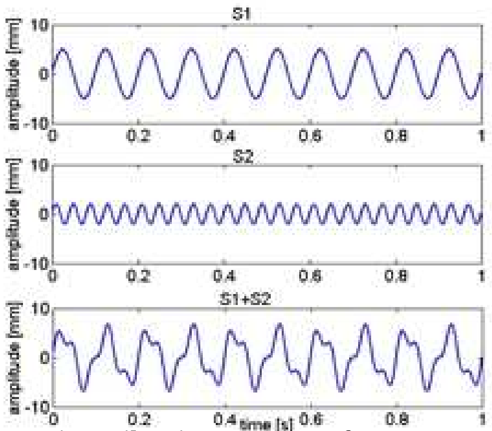

Fig8. Vibration spectrum of S1+S2

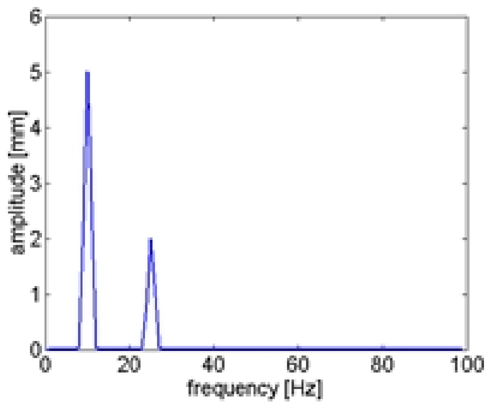

Fig9. FFT time waveform of S1+S2

From the trends in FFT time waveform, we can diagnose the cause behind vibrations. Some of the common vibration causes and its diagnosis are given below –

4.2.1. Unbalance

For all types of unbalance, the FFT spectrum will show a predominant peak at the 1× rpm frequency of vibration, and the vibration amplitude at the 1× rpm frequency will vary proportional to the square of the rotational speed. If the problem is unbalance, this peak usually dominates the vibration spectrum [fig 10 & 11].

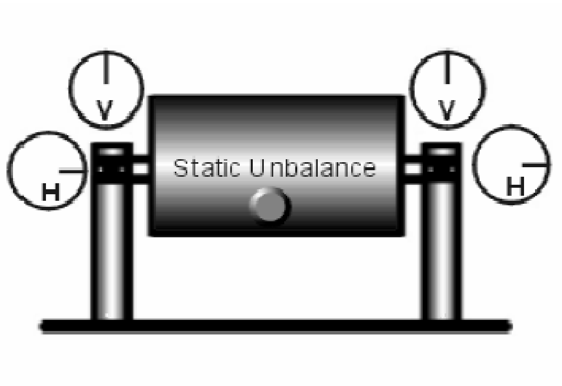

Fig10. Vibration measurement locations

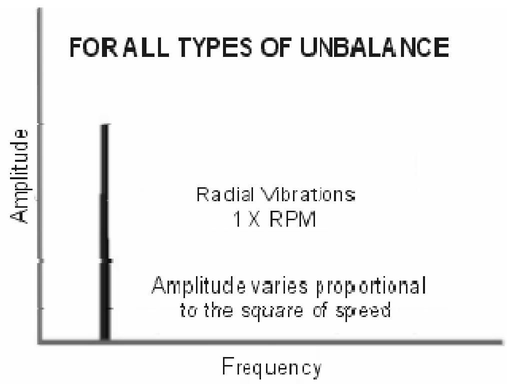

Fig11. FFT readings showing vibration peak due to unbalance.

4.2.2. Eccentricity

Eccentricity occurs when the centre of rotation is at an offset from the geometric centerline, and this may happen if the pump impeller is eccentric due to a manufacturing or assembly error. In the vibration spectrum, the maximum amplitude occurs at 1× rpm of the eccentric component, and will vary with the load even at constant speeds. In the horizontal direction, a phase shift of 90° will be observed.

4.2.3. Bent Shaft

When a bent shaft is encountered with a pump, the vibrations in the radial as well as in the axial direction will be high. Axial vibrations may be higher than the radial vibrations. The spectrum will normally have 1× and 2× components, as shown in Figure 12. If the:

- Amplitude of 1× rpm is dominant, then the bend is near the shaft centre

- Amplitude of 2× rpm is dominant, then the bend is near the shaft end.

The phase will be 180° apart in the axial direction and in the radial direction. This means that when the probe is moved from vertical plane to horizontal plane, there will be no change in the phase reading.

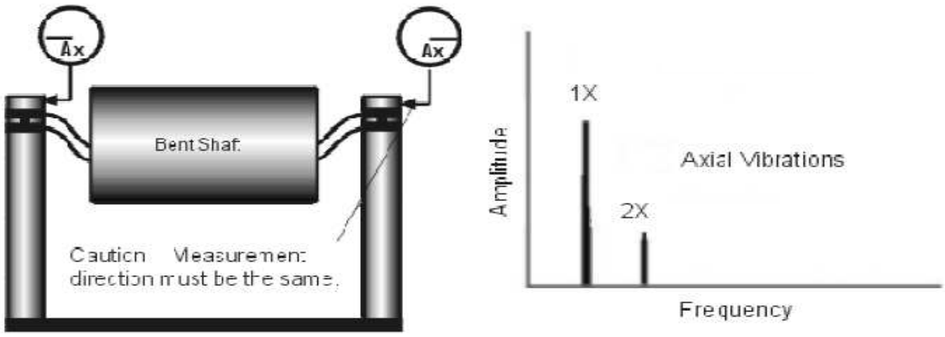

Fig12. Bent Shaft

4.2.4. Pump and motor misalignment

There are basically two types of misalignment that can occur between the motor and the pump:

- Angular misalignment – the shaft centerlines of the two shafts meet at angle
- Parallel misalignment – the shaft centerlines of the two machines are parallel

As shown in Figure 13, angular misalignment primarily subjects the motor and pump shafts to axial vibrations at the 1× rpm frequency. A pure angular misalignment is rare, and thus, misalignment is rarely seen just as 1× rpm peak. Typically, there will be high axial vibrations with both 1× and 2× rpm. However, it is not unusual for 1×, 2× or 3× to dominate. These symptoms may also indicate coupling problems (e.g. looseness) as well. A 180° phase difference will be observed when measuring the axial phase on the bearings across the coupling, as shown in Figure 13.

Parallel misalignment [fig 14] results in two hits per rotation; and, therefore a 2× rpm vibration in the radial direction. Parallel misalignment has similar vibration symptoms compared to angular misalignment, but shows high radial vibration that approaches a 180° phase difference across the coupling. As before, a pure parallel misalignment is rare and is commonly observed to be in conjunction with angular misalignment. Thus, both the 1× and 2× peaks will typically be observed. When the parallel misalignment is predominant, 2× is often larger than 1×, but its amplitude relative to 1× may often be dictated by the coupling type. When either angular or parallel misalignment becomes severe, it can generate high amplitude peaks at much higher harmonics (3× to 8×) or even a whole series of high frequency harmonics.

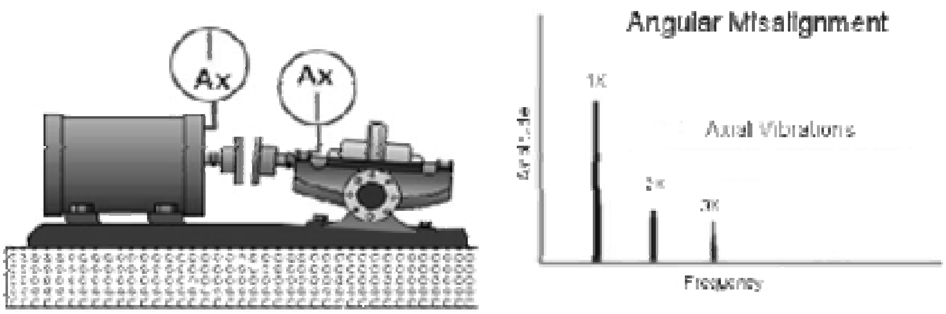

Fig13. Angular Misalignment

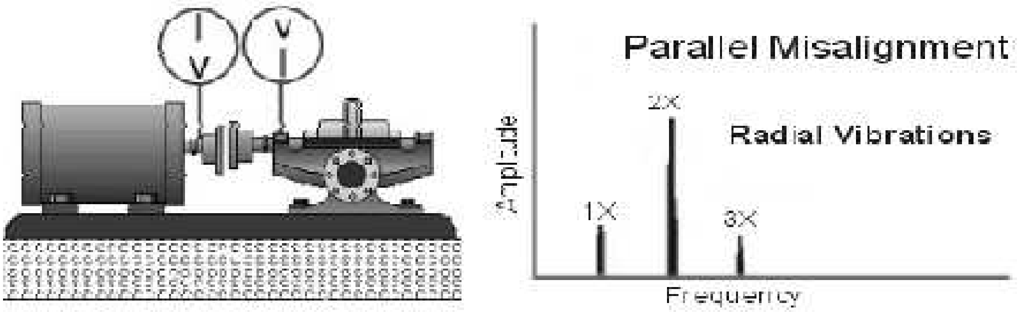

Fig14. Parallel Misalignment

4.2.5 Blade pass and vane pass vibrations

Blade pass or vane pass frequencies are vibrations inherent to pumps and fans. In itself, it usually not problematic or destructive, but can generate a lot of noise and vibration that can be the source of bearing failure and wear of machine components.

Blade Pass Frequency (BPF) = number of blades (or vanes) × rpm

This frequency is generated mainly due to the gap problems between the rotor and the stator. Large amplitude BPF (and its harmonics) can be generated in a pump, if the gap between the rotating vanes and the stationary diffusers is not kept equal all the way around. In centrifugal pumps, the gap between the impeller tip and the volute tongue or the diffuser inlet is in the region of 4-6% of the impeller diameter, depending on the speed of the pump. If the gap is less than the recommended value, it can generate a noise that resembles cavitation. However, a vibration reading will immediately reveal the vane pass frequency of the impeller (Figure 15). Also, the BPF (or its harmonics) sometimes coincides with a system natural frequency, causing high vibrations.

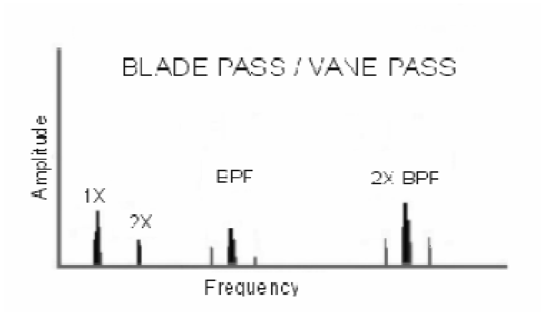

Fig 15. FFT waveform pattern for Vane pass pulsations

A high BPF can be generated if the wear ring seizes on the shaft or if the welds that fasten the diffusers fail. In addition, a high BPF can be caused by abrupt bends in line work (or duct), obstructions which disturb the flow path, or if the pump impeller is positioned eccentrically within the housing.

4.2.6 Flow turbulence

In pumps, flow turbulence induces vortices and wakes in the clearance space between the impeller vane tips and the diffuser or volute lips. Dynamic pressure fluctuations or pulsations produced in this way can result in shaft vibrations because the pressure pulses impinge on the impeller. Flow past a restriction in pipe can produce turbulence or flow induced vibrations. The pulsation could produce noise and vibration over a wide frequency range. The frequencies are related to the flow velocity and geometry of the obstruction. These in turn excite resonant frequencies in other pipe components. The shearing action produces vortices that are converted to pressure disturbances at the pipe wall, which may cause localized vibration excitation of the pipe or its components. It has been observed that vortex flow is even higher when a system's acoustic resonance coincides with the generated frequency from the source. The vortices produce broadband turbulent energy centered around the frequency determined by the following formula:

$$f = \frac{S_n \times V}{D}$$

Where f = vortex frequency (Hz), S_n = Strouhl number (dimensionless, between 0.2 and 0.5), D = characteristic dimension of the obstruction. An example of a vibration spectrum due to turbulence is shown in Figure 16.

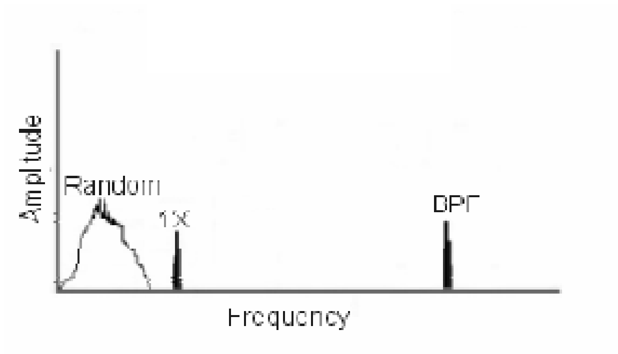

Fig 16 - FFT waveform pattern for Flow turbulence

4.2.7 Cavitation

Cavitation normally generates random, high frequency broadband energy, which is sometimes superimposed with the blade pass frequency harmonics. Gases under pressure can dissolve in a liquid. When the pressure is reduced, they bubble out of the liquid. In a similar way, when liquid is sucked into a pump, the liquid's pressure drops. Under conditions when the reduced pressure approaches the vapour pressure of the liquid (even at low temperatures), it causes the liquid to vaporise. As these vapour bubbles travel further into the impeller, the pressure rises again causing the bubbles to collapse or implode. This implosion has the potential to disturb the pump performance and cause damage to the pump's internal components. This phenomenon is called cavitation. Each implosion of a bubble generates a kind of impact, which tends to generate high-frequency random vibrations, as depicted in Figure 17. Cavitation can be quite destructive to internal pump components if left uncorrected. It is often responsible for the erosion of impeller vanes. Measurements to detect cavitation are usually not taken on bearing housings, but rather on the suction piping or pump casing.

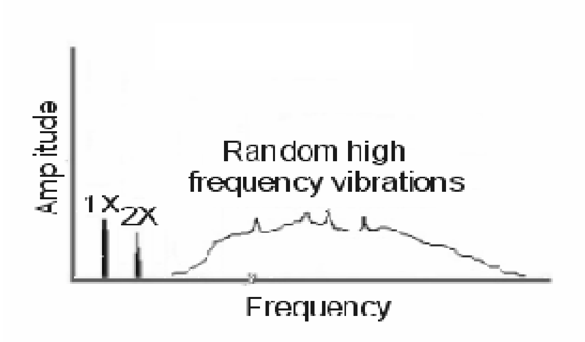

Fig 17 - FFT waveform pattern for Cavitation

5.0 CONCLUSION

In this paper we have tried to emphasize the vibrations and noise in centrifugal pumps, its causes / sources and the diagnosis methods. It is shown how noise and vibration in centrifugal pump could be diagnosed and its remedies can be worked out on the basis of diagnosis. To ensure proper functioning and safety of the pump and associated plant components it is very essential to know the sources and remedies. Specific techniques can be used to identify and rectify specific pump problems, such as unbalance, misalignment, turbulence, cavitation and many others. By knowing the vibration causes and using proper diagnosis tools, one can easily save lot of manpower, plant downtimes and spare parts. With the appropriate implementation of vibration and noise diagnosis techniques, pumps can operate with higher reliability and efficiency.

References

1.0 Cornelius Scheffer *"Pump Condition Monitoring Through Vibration Analysis" Pumps: Maintenance, Design, and Reliability Conference, 2008.*

2.0 Bernd Dürrer, Frank-Hendrik Wurm *"Noise sources in centrifugal pumps" Proceedings of the 2nd WSEAS Int. Conference on Applied and Theoretical Mechanics, Venice, Italy, November 20-22, 2006*

3.0 Entek IRD International *"Introduction to vibration technology" Edition I, Jul 1984, Tata McGraw hills.*

4.0 Val S. Lobanoff & Robert R. Ross, *"Centrifugal Pumps design and application", Jaico publishing house – 2nd edition 1995.*

5.0 Igor J Karassik & Ray Cartor, *"Centrifugal pumps", Tata McGraw Hill Book company- 2nd Edition, 1960.*

6.0 Paul R. Donavin, J. J. Patel, Herbert L. Miller & Curtis G. Sterud *"Feed pump discharge line vibration solved at Byron station unit 2" presented at the American Power Conference, Chicago, Illinois; April 13–15, 1993.*