Deep Well Vertical Pump Analysis using Electrical Signature Analysis (ESA)

Abstract

Deep well vertical pumps have been used in commercial and industrial plants to move fluids for many years. There are many advantages that vertical pumps provide over horizontally mounted pumps and when they operate properly they generally provide many years of reliable, maintenance free operation. However, in almost any plant with a large inventory of vertical pumps they will have one or more of these pumps that are considered “bad actors”. These pumps have a much shorter life than similar type pumps in the same application. Many times the plant will have a pump that does achieve expected life, yet a seemingly identical pump continually fails unexpectedly.

Machinery vibration analysis has proven to be a very effective technique for determining the condition of rotating equipment. However, when these techniques are applied to vertical pumps, they have proven to be ineffective.

This paper is going to examine these pumps, construction, fault modes and provide some ESA data showing some faults in vertical pumps.

Introduction

Vertical pumps have been in use in refineries, power stations, chemical plants, paper mills and almost any other industry or plant that has a requirement to move fluids. The advantages of vertical pumps range from space savings, ability to operate in both high and low temperature applications, operate at lower NPSHA, increased reliability and reduced maintenance costs. In many cases vertical pumps do provide increased reliability and reduced maintenance cost. However, in other cases, these pumps have proven to be very troublesome and have reoccurring and sometimes unexpected catastrophic failure.

Machinery vibration monitoring and analysis have been used successfully by these same plants to detect failures and plan maintenance on rotating equipment for over 70 years. These principles are well documented and have been applied throughout industry on most horizontally mounted machines. However, when these techniques have been applied to vertical pump applications these principles and techniques are inadequate. The reasons for this are numerous, but include:

1) Inability to determine the condition of the submerged portion of the pump.
2) Improper assembly guidelines and practices.
3) Gravitational forces applying pre-load to the bearings.
4) Changing conditions of the structural modes in the submerged portion of the pump by varying operating conditions.
5) Lack of literature and knowledge of these machines failures and diagnostics.
6) The vertical and horizontal pumps both have the same dynamic forces but the interaction with the support structure differs dramatically in the vertical pump, creating various different resonances and bending modes of both the rotating and the stationary components.

Vertical Pump Construction

These vertical pumps come in various configurations but all are normally placed in a sump or well below the level of the piping itself. The driver is mounted on the top of the pump in a dry location.

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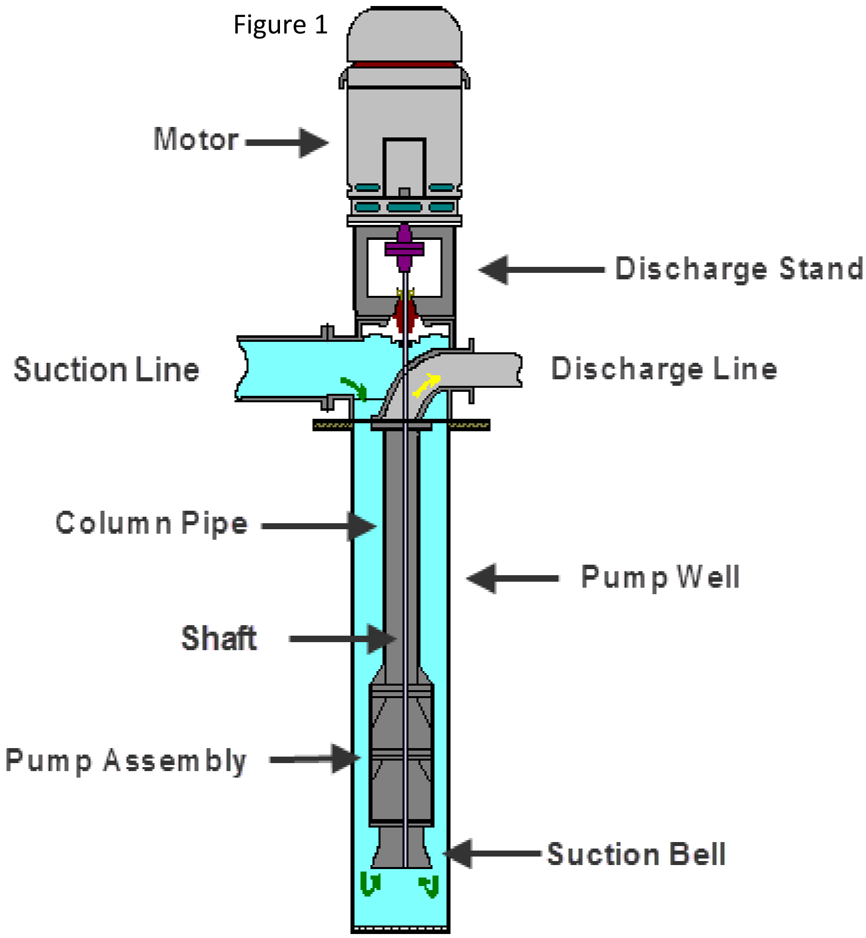
Figure 1



The pump portion is constructed of the pump shaft, pump assembly (impeller & pump casing or diffuser), and column pipe. The pump and shaft hang below the discharge stand, which is often connected to the suction and discharge piping. The suction and discharge lines are normally in line with each other. The discharge stand is mounted to the floor or a well flange.

The pump shaft and assembly may be connected to the driving machine's shaft employing a long, intermediate shaft. This shaft places the pump deeper in the well or sump. The pump shaft and the driving machine are normally connected using a solid coupling. Long intermediate shafts use bushings or fluid film bearings placed along the length of the shaft to supply some lateral support to the shafts themselves.

The driver for the vertical pump is commonly an electric motor but can be driven by other drivers, such as diesel/gas engines or steam turbines. The driver is positioned on the discharge stand with a machined or rabbet, spigot fit or positioned using pins or dowels**.**

The pump assembly may be a single or multiple stages (pump) depending on the application and head requirement. The pump assembly consists of one or more impellers and associated diffusers. The pump hydraulics is normally located at the very bottom of the pump, in the suction bell. Each impeller discharges upward to the impeller above it and into the intermediate column pipe and then into the discharge pipe. The suction bell has a spider, or steady bearing, to position the shaft and impeller in the center of the pump diffuser. Each additional pump stage normally has a bearing or bushing to center the shaft in each pump casing. The bearing positions the shaft and provides radial support for the rotating assembly. Normally these bushings or bearing are lubricated with the pumped product.

The pumped product is pumped into the intermediate column pipe through the discharge line and into the system.

Deep Well Vertical Pump Faults

Vertical pumps have all of the same faults as any other piece of rotating equipment and in particular centrifugal pumps. Additionally, because of their unique design, vertical pumps undergo resonant mode changes that affect the operation and mechanical condition of these pumps.

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Unbalance

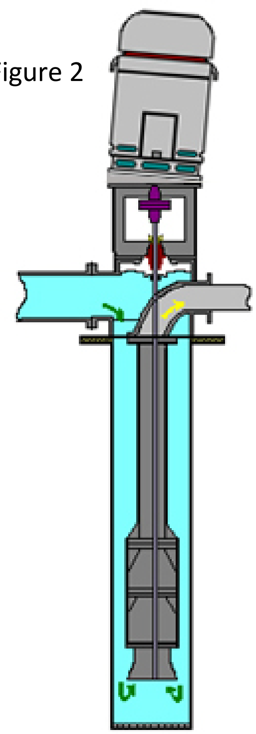
All rotating equipment exhibits some level of unbalance during operation. Vertical pumps in particular often times require special balancing considerations. Many vertical pumps are treated as rigid rotors and balanced as such using balancing tolerances for a rigid rotor. In almost all cases vertical pumps normally use flexible rotors and should be balanced as a flexible rotor, using stack balancing procedures. A second common problem is that the impellers are treated as narrow rotors by the manufacturer and often balanced in a single plane.

RIGID ROTOR - a rotor is considered rigid if its unbalance can be corrected in any of two corrections (planes). After the correction, the residual unbalance does not change significantly at any speed up to the maximum service speed.
FLEXIBLE ROTOR - a rotor that does not satisfy the rigid rotor definition because of elastic deflection.

Misalignment Driver and Driven Machine

It is a common misconception in industry that deep well vertical pumps do not require alignment, since they normally come from the factory assembled. The motor, or other driver, is positioned on the discharge stand using dowel pins or a machined fit in both the motor and the discharge stand. However, experience has shown that any eccentricities or improper sizing in either of the machined fit, on the motor or the discharge stand, will create misalignment between the motor and the pump shafts. Further complicating this fault is that these pumps normally use solid couplings.

Figure 2



Additionally, misalignment can come from a soft foot condition. A common cause of soft foot for these pumps are pulled threads caused by over torque of the hold down bolts between the motor and the discharge stand.

The pump assembly hangs below the discharge stand and motor. Therefore, if the discharge stand is not perfectly level and the two shaft centerlines are aligned (using an alignment tool that aligns the shaft centerlines), the pump and motor may actually be misaligned.

Resonance

Resonance is a condition in mechanical systems, which occur when a forcing function applies a cyclic force at a frequency that is near a structure's natural frequency. When this condition occurs the system amplifies its response to the applied force and can create motion that is 15 to 20 times larger than when in a *non-resonant condition.*

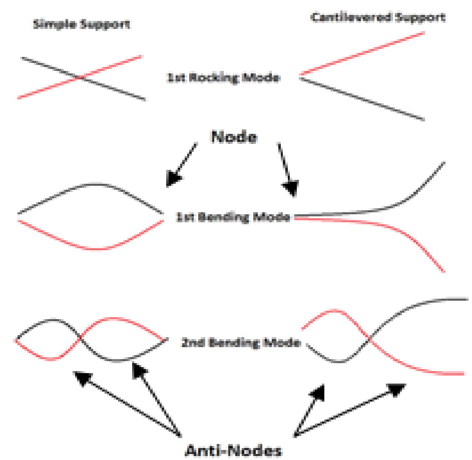
Natural frequencies

All spring systems have natural frequencies. A natural frequency is the frequency or frequencies that a spring system will oscillate or vibrate at when plucked, struck or externally excited by an outside force. The system properties that determine the system's natural frequencies, are the mass and stiffness of the spring system. Each mass stiffness combination has multiple natural frequencies. A component that is vibrating at its natural frequency takes on a mode shape depending on which natural frequency is excited.

Mode Shapes

The characteristic of a component, which is in significant resonance mode shape, is the node and the anti-node. The point of maximum deflection is referred to as the anti-node and has the minimum stress whereas the point of minimum deflection is called the node and has the maximum stress.

Figure 3



The mode shapes vary slightly depending on the mounting design. The two mounting designs are the simple support mode which has the spring system mounted between two supports. The second mounting support is the cantilever support, which is supported only on one end with the spring component free at the opposite end.

The 1st natural frequency bending mode on a component that is a simple support system has one anti-node and two nodes. The first bending mode of a cantilevered mounted spring system has one anti-node and one node. As a component's higher order of natural frequencies is progressively excited their mode shapes take on an additional node and anti-node. The 2nd natural frequency in a simple support system will have three nodes and two anti-nodes. The cantilevered systems 2nd bending mode will have two nodes and two anti-nodes.

Critical speeds (Resonant Whirl)

Normally critical speed is present when the rotor passes through its own natural frequency. However, more commonly, a critical speed is any speed at which the rotor speed creates a resonant condition. Early definition of resonant whirl was explained as when the rotor excited its own natural frequency. When this occurs the rotor actually bends and takes on the mode shape of its own natural frequency. For example, a rotor at its 1st natural frequency will bend in a "U" shape (see Figure 3). A rotor operating at its 2nd natural frequency will take on an "S" shape. Following the same rules for any resonance, as the rotor passes through higher order natural frequencies, the rotor will take on that bending shape.

Experience has shown that generally a rotor with a high 1st critical speed has a large amount of static unbalance, whereas, a rotor with a high second critical speed has a large amount of coupling unbalance.

Changing natural frequencies

Spring systems' natural frequencies are determined by the mass/stiffness combination. The natural frequency will increase with an increase in stiffness and decrease with an increase in mass. However, it is also important that the most effective way of changing natural frequencies is to add mass or stiffness at the anti-nodes. Experience has shown that attempting to change the natural frequency at a node has no effect.

Eliminating resonance

There are several methods that are effective for eliminating the effects of resonance. However, the most common is to detune the system. This involves moving the natural frequency away from the frequency of the vibrating force. This is often accomplished by adding stiffness to increase the natural frequency or adding mass to reduce the natural frequency. There are 2 distinct disadvantages to this approach. The first one is obvious in that by detuning the system it is possible to move the natural frequency of one of the components to a different natural frequency. The 2nd disadvantage is that if the exciting force driving the resonance is large enough, it could cause another system or component to resonate.

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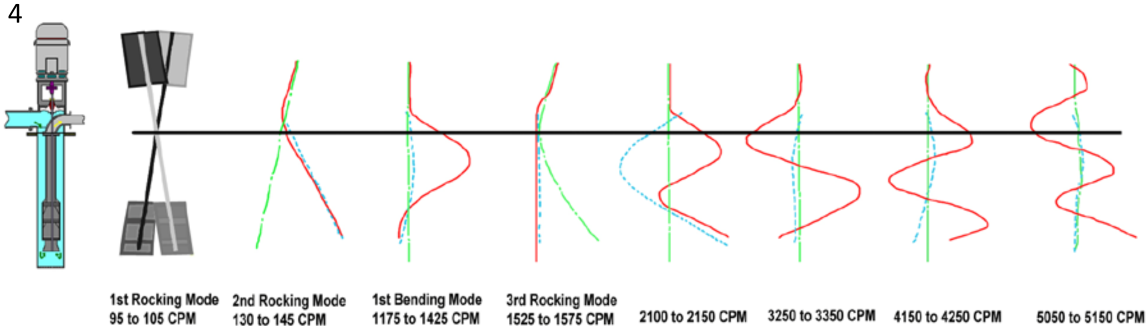
Perhaps the most effective way, but least attempted, is to eliminate the source. This holds true particularly in vertical pumps. Figure 4 below shows the mode shapes of the three major components of a vertical pump. The most significant mode shape is that of the shaft itself. At different frequencies the pump shaft will take on a different shape. These may or may not be an issue as it depends on the location and condition of the steady bearings that are mounted in the intermediate tube. The other condition that determines the rotor mode shape is the balance condition.

Figure 4



Flow related faults

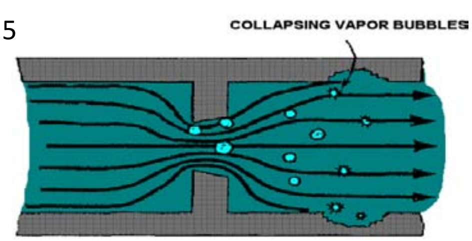
Deep Well Vertical Pumps have all of the same flow related faults as any other centrifugal pump. Problems such as cavitations', flow turbulence and even flow pulsations caused by the impeller vane interaction with the diffuser, can cause severe damage to the pump impellers or the pump assembly.

Cavitation

Cavitations can occur if the pressure of a fluid drops below the vaporization pressure for that fluid. When this occurs some of the fluid will change state from a liquid to a gas and form small vapor bubbles in the fluid itself. If the pressure of this vapor entrained fluid now increases above its vaporization point the vapor bubbles formed in the low pressure region will collapse. These collapsing vapor bubbles release high energy micro-jets that impinge on the surface of the vessel containing the fluid.

Figure 5 shows the formation of vapor bubbles as a fluid passes through an orifice. As the velocity of the fluid increases through the orifice and causes the pressure to decrease below its vaporization pressure, vapor bubbles are formed in the low pressure region.

Figure 5



When the velocity of the fluid decreases on the other side of the orifice the pressure increases. When the pressure exceeds the vaporization pressure these vapor bubbles collapse. They are formed in the low pressure region and releases high energy micro jets that impinge on the surface of the pipe. These high energy micro-jets erode the piping walls.

The release of the micro-jets creates random bursts of energy and broadband excitation within the vessel or piping containing the fluid. Cavitations can occur on the suction side of the pump if there is insufficient Net Positive Suction Head to keep the pumped product in a liquid state. It can also occur on the discharge side of the pump and usually caused by low discharge flow; a result of increasing the pump's internal recirculation by throttling the discharge.

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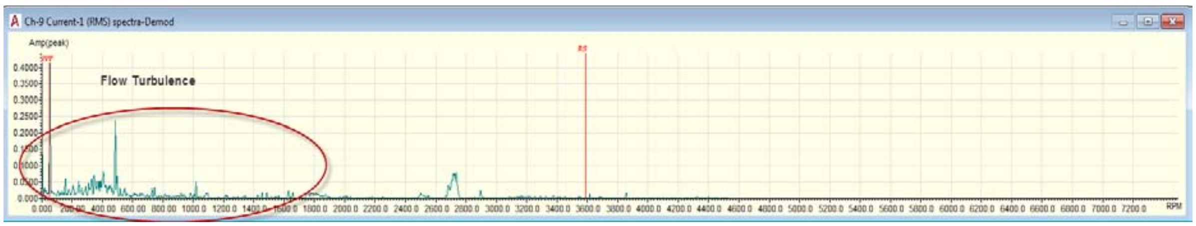
Oil whirl

A common problem associated with fluid film bearings is a fault known as oil whirl. Oil whirl occurs on horizontally mounted fluid film bearings if the upward lifting force of the oil exceeds the load on the bearing. This allows the oil wedge to whirl around the bearing and can create forces that are normally at 42% to 43% of rotor speed.

Flow turbulence

Other flow turbulence problems can create destructive forces within the pump assembly. These will usually present themselves as low frequency broad band excitations.

Figure 6



Vane Passing

Flow pulsations within the pump assembly can occur when the forces caused by the impeller vanes interact with stationary parts of the diffuser, normally the cutwater. These forces are normally the result of insufficient clearance between the outside diameter of the impeller and the inside diameter of the diffuser. These forces occur at vane passing frequency which equals the number of vanes, times rotational speed.

Figure 7

Deep Well Vertical Pump Monitoring

In the early days it was assumed that faults in the submerged section of these pumps could be seen by measuring the vibration at the top of the pump. However, experience and many studies by EPRI and other organizations have well documented, that in order to determine faults occurring in the submerged portion of the pumps, it is necessary to mount the vibration transducer on the pump itself below the mounting surface.

The vibration as measured on the motor or discharge stand would not identify any of the mechanical or flow related faults in the pump. Generally any motion measured on the motor was normally a fault above the discharge stand or the attachment plate. When changes in the vibration measured on the motor do actually occur, as the result of a pump failure, the pump assembly is normally completely destroyed.

Most vibration signatures taken on the motor were only at 1 and or 2 times rotating speed. Following standard vibration analysis theory this would indicate unbalance or misalignment as the mechanism causing the unwanted vibration. Other frequencies were not present so it was assumed they did not exist.

In the EPRI study of vertical pump vibration monitoring EPRI NP-5704M; Mar 88; Prepared by Mechanical Technology Inc, stated that in order to detect the early on-set of pump problems it is necessary to mount sensors in the submerged portion of the pumps. And then an extraordinary number of failures to the sensors occur, and that depending on the placement of the sensor or the type of sensor used some of these faults will be either missed or misdiagnosed.

Electrical Signature Analysis (ESA)

Preliminary studies have shown that all of the faults that occur in the submerged portion of the pumps can be detected and identified using ESA.

ESA uses three current clamps around each of the three phases of the motor leads to measure the current supplied to the motor. It also attaches three voltage probes to the motor leads to measure the voltage to all three phases of the motor.

The ESA handheld instrument then performs a simultaneous capture of all three phases of voltage and current to determine the quality of the power supplied and consumed by the motor itself. Additionally, it captures one phase of voltage and one phase of current and digitizes the waveform of each. These digitized waveforms are then uploaded into the host computer allowing the associated software to perform a number of FFT’s on these captured waveforms.

Comparing the FFT’s, from the current waveforms, to vibration data, ESA will reveal faults in the submerged portion of the pump. If high forces are present at 1 and or 2 times the running speed and a signal is present in both the vibration signature and the current signature, then this would indicate that the fault/faults are above the surface. However, if these forces are only present in the current spectrum, then the fault is in the submerged portion of the pump.

Summary

Preliminary research has shown that ESA can detect early degradation of the pump assembly in Deep Well Vertical Pumps. On-going additional testing and research will enable a better understanding of the test data and the pumps condition.

BIOGRAPHY: William Kruger has been involved in predictive maintenance for over 40 years. He is a graduate of the US Navy Nuclear Power School and earned his BS degree from San Diego State University. His first introduction to predictive maintenance came from operating the vibration program on-board a SSBN submarine. He spent 10 years at San Diego Gas & Electric where he started their predictive maintenance program. He then worked as an applications engineer for the DYMAC division of Spectral Dynamics.

For the past 25 years Mr. Kruger has focused on training in the field of Predictive Maintenance. He was a senior instructor at Update International. Mr. Kruger has conducted training courses on 6 continents and is known worldwide for his practical approach to machinery analysis and his ability to present complex technical material in easily understood principles and demonstrations.

Mr. Kruger is currently the Training Technical Manger for ALL-TEST *Pro*. He has authored several technical papers and made presentations at various meetings, including the Vibration Institute, EPRI, Canadian Pulp & Paper Expo, & International Maintenance Conference. He has held memberships in ASME, Vibration Institute, ANST, IEEE & many other professional organizations.