Severe vibration levels are observed on tested pumping stations in Egypt of different types: centrifugal, axial, vertical, and inclined units. Vibration severity has been evaluated on the bearings of the pumping units where unacceptable levels of vibration have been measured. Case studies are introduced from different irrigation and drainage pumping units in the laboratory and field at different operating conditions to detect faults early and design predictive maintenance programs for maintaining smooth and safe operation, and satisfying water requirements.

Cases include a fatigue failure problem in the field due to vibration of a centrifugal pump mounted horizontally, a bearing damage problem in the field of a vertically mounted axial flow pump, and excessive vibration level for a variable speed motor in the lab. The investigation has been carried out by recording vibration spectra at different operating conditions and doing modal testing to isolate sources of vibration, identify the exciting frequencies, and predict dynamic behavior at different operating conditions. The paper points out the importance of carrying out vibration analysis on pumping units in Egypt to control dynamic behavior and achieve optimum running conditions.

1. INTRODUCTION

The trend of using lightweight materials and high speed pumping units, resulting in increasing of the vibration excitation sources and leading to severe vibration problems, has increased the necessity of doing vibration analysis of pumps to detect faults early. There are many causes of vibration in the pumping units including hydraulic, mechanical, structural, etc., leading to energy losses, reduction in performance, and decrease of operating life.
There are more than 2000 large-scale irrigation and drainage pumping stations operating under different conditions. The mission of these stations is to plan for supplying irrigation and drainage water requirements. These stations must work in a good condition away from sudden faults and breakdown under controlled environmental impacts. Sudden faults can be avoided by applying predictive maintenance programs. Mechanical vibration or acoustic noise generated while the machine is deteriorating is collected and analyzed to predict when the machine is about to fail. Many problems are encountered with these pumping stations [1].

Periodic vibration monitoring is widely recognized as a reliable method of dynamically determining the health of pumping units. Analysis of the overall vibration levels and associated vibration frequency spectra can result into early detection and isolation of common pump problems. The early detection allows corrective actions to be scheduled in the suitable time resulting in increased pump productivity economically and efficiently.

Flow induced vibration in pumping system is mainly dependent on operating conditions, inlet distortion, cavitation, surge, etc. Such flow induced phenomena is more complex in nature and more difficult to single out than the mechanical causes which are directly related to the operational speed of the pump. Vibration measurements and analysis has proven to be the workhorse of machine condition assessment. Because of the intimate relationship between the shaft or casing vibration and the disturbing forces acting on the pumps' internal components, vibration is a sensitive indicator of changes in machine condition which influence the fluctuating loads on pumps. Other operational parameters such as temp., suction pressure, flow rate, etc., may yield significant information, but vibration still the most appropriate condition related parameter.

Experience dealing with vibration of pumps show that the most common problems are due to wrong installation and operation near the pump-motor system natural frequency resulting in excessive vibration levels on the motor and the pump impellers [2]. Increasing stiffness of the pump mountings by using rigid mounting resulted in lower vibration levels. Vibration analysis of large capacity pumps suggested that greater care is required in designing the foundation supports stool/frame & bracings and its proper fixing be ensured especially in case of tall motors [3]. Many factors are involved in causing a pump to vibrate. Smooth operation begins with proper selection and design for a specific service. Vibration measurement provides a sound basis for establishing the running condition of process pumps. The vibration data will provide an excellent foundation on which preventive/corrective maintenance programs can be designed [4]. Vibration in rotating machinery may be the result of several phenomena and may affect various machine parts. Most vibration failures can be classified to structural fracture caused by fatigue or dynamic overload; wear, fretting, or surface fatigue of bearings, gears, couplings, etc.; and performance loss due to internal machine clearance rubs such as seals, blades, and impellers [5].

Vertical pumps can exhibit high vibration levels than horizontal mounted pumps. These pumps often operate with signs of unstable operation, large misalignments, and other characteristics that would cause immediate shutdown in most machinery [6]. Disassembly and reassembly of long coupled vertical pumps should be precisely done, as it requires more attention than horizontal pumps. Dependence of the measured spectra on the rotating speed should be defined well before diagnosing faults. A fixed relationship between spectra and rotation speed is an indicator of the forcing terms of damage to gear, couplings, and shafts where a variable relation is an indicator of instability [7].

2. VIBRATION ANALYSIS

The objective of the analysis is to determine the sources of high vibration. Knowing dynamic characteristics (natural frequencies & damping) of the pumping system is the primary step to
solve any structural weakness leading to resonance problems. Each faulted element has its exciting frequencies to the pump system. It is very important to define all the exciting frequencies for the rotors, bearings, couplings, gears, etc., in the beginning of doing vibration analysis in addition to modal analysis to easily relate each exciting frequency and high vibration level to its source. The governing equation of the system can be written as:

\[ [M] \ddot{\{x\}} + [K] \{x\} = 0.0 \]  
(1)

Assuming harmonic motion

\[ \ddot{x}_r = -(2\pi f_r)^2 x_r \]  
(2)

Substitute from Eq. (2) into Eq. (1) to get:

\[ -(2\pi f_r)^2 [M] \{x\} + [K] \{x\} = 0.0 \]  
(3)

The following formula can be used to find the natural frequencies \( f_i \) of the pumping units and building structure:

\[ | [M]^{-1}[K] - (2\pi f_r)^2[I] | = 0.0 \]  
(4)

The exciting frequencies are related to the rotating frequencies of the motor and the pump, their harmonics and sub-harmonics. Before analyzing measured vibration data, the exciting frequencies must be calculated first. The rotating frequency of the motor is:

\[ \omega; f = \Omega / 60 \text{ Hz} \]  
(5)

The exciting frequency of the pump is:

\[ \omega; f = \Omega / (60 \ast R) \text{ Hz} \]  
(6)

For the complete pumping system, other exciting frequencies are obtained for the gearbox failures. These faults are function of the rotating speed, gear ratio, number of teeth, and the faulted tooth number. In the new health gear, exciting frequency will be function of rotation speed and number of teeth on the gear.

More peaks may be found due to wear in the gearbox. Some harmonics will be revealed in the faulted gear where the maximum peak due to wear is the 2\textsuperscript{nd} harmonics of teeth-meshing frequency. The gear-mesh frequency can be calculated as:

\[ \omega; f = \left( \Omega \ast Z_1 \right) / 60 \text{ Hz} \]  
(7)

or

\[ \omega; f = \left( \Omega \ast Z_2 \right) / (60 \ast R) \text{ Hz} \]  
(8)

For ball bearings, the fault may be in the outer race, inner race or due to the ball defects. For outer race defect, the exciting frequency is:

\[ \omega; f = \left( \frac{b}{2} \right) R \left( 1 - \frac{B_d \cos \beta}{P_d} \right) \text{ Hz} \]  
(9)

For inner race defect, the exciting frequency is:

\[ \omega; f = \left( \frac{b}{2} \right) R \left( 1 + \frac{R_d \cos \beta}{P_d} \right) \text{ Hz} \]  
(10)

And for ball defects, the exciting frequency is:

\[ \omega; f = \left( \frac{P_d}{D_d} \right) R \left( 1 - \left( \frac{B_d \cos \beta}{P_d} \right)^2 \right) \text{ Hz} \]  
(11)

3. FATIGUE FAILURE PROBLEM

A repeated shaft break problem has been observed in some irrigation units, El-Shorouk pumping Station, Bohira. These problems are the repeated fatigue damage of the pump shafts, wear of the shafts in the stuffing box region and some damage in the ball bearings \[8\]. This pumping station worked safely for 2 years with average 10 hours per day, then the first shaft break of the pump occurred and repeated after this many times for periods 3-4 months apart at the same operating conditions. It was noticed that a big clearance between the shaft end and the impeller where all the shafts break occurred at the keyway area and at the shaft end next to the impeller. Repeated self unlock of the impeller fasten nut with the shaft was also noticed. From the visual inspection of the
defected shafts, two areas of defects were determined; repeated break at the free end of the shaft under the impeller hub where break takes many patterns, and wear of the shaft at the stuffing box of the pump due to friction between the stuffing box material and the shaft, as shown in Fig. 1. Due to fatigue and wear of the shafts at the areas of impeller and stuffing box, these shafts were subjected to failure as a result of mechanical vibration. A trial was made by the station staff to eliminate or reduce the repeated damage in the pump shafts by manufacturing them from steel alloys (steel 35). This trial was rapidly failed under the corrosion and wear actions. The fatigue phenomena motivated to measure and analyze the mechanical vibrations in the pumping systems to diagnose the causes of vibration, which has instigated the fatigue.

Overall vibration level measured on the components of the pumping system show that vibration level is very high and greater than the allowable level. Frequency analysis was done on the pump components as shown in Fig. 2. Misalignment of the motor and the pump shafts, unbalance, water hammer and cavitation generate cyclic bending, torsional and axial tension at the end of the pump shaft. These faults make the pump seems to be under actual fatigue test of several frequencies, every one of them has its own stress level dependent on the severity of the associated fault. These loading conditions make a combination of opening, tearing, and sliding modes of fatigue crack opening. The combination of opening and tearing modes of cracks are developed at the tip of a circumferential crack instigating the failure of the end which was separated with the jamming unit. The same combination of crack modes is developed at the middle region of the bottom of the keyway causing fatigue damage. Another combination of the opening and sliding modes take place in the free end direction of the pump shaft causing damage.

The incorrect assignment of tolerances between the pumping unit elements leads to a bad assembly during the units production leading to misalignment problem causing mechanical vibration, which causes some leakage problem around the pump shaft. To avoid leakage process, more stuffing material is pressurized in the stuffing box instigating more wear rate, causing rapid failure of the shafts. All fatigue failures were not at the shaft cross sections. The operational or the working stresses affecting the shaft were smaller than the fatigue limit. Design, production, and operation errors causing failure must be avoided or reduced to eliminate the fatigue and wear of the pump shafts.

4. VIBRATION ANALYSIS OF A VARIABLE SPEED MOTOR

The trend of using variable speed drives in pumping station to increase pump efficiency and reduce energy losses, has motivated the authors to study dynamic behavior of such variable speed drives at different operating conditions. A variable speed motor without load was used in the study where mechanical problems such as misalignment and unbalance were simulated in the lab. Experimental modal testing was done on the motor to determine its dynamic characteristics. Table 1 summarizes the modal parameters of the motor where the fundamental natural frequency is 56 Hz (3360 rpm) and damping 61%. It is very important to define the natural frequencies of the system to determine the optimum operational speed that satisfies water requirements and works away from the resonance problem. It is apparent that the running speed up to 3000 rpm is safe. Operation far from 3360 rpm is required; however, damping coefficient at this speed is high (60%) and capable of suppressing any excessive vibration level. So, the only resonance problem that may arise from varying speed of that motor is running at 3360 rpm and structural modification is required to improve dynamic behavior of the motor if operation at this speed is a must. First bending mode shape of the motor is shown in Fig. 3, where sensitive nodes are defined to implement any modifications that enhance its structural properties.

Forced vibration tests are done at two different operating speeds of the motor at 1725 rpm and 1950 rpm to determine the vibration levels associated with varying speed. Mechanical
problems (unbalance and misalignment of the shaft) were simulated to determine their effect on vibration level during operation at different speeds. Vibration levels associated with such mechanical problems at two different speeds are summarized in Table 2. Vibration level measured on the variable speed (v.s.) motor at 1725 rpm is allowable according to the standards and in the order of 2.5 mm/sec. However, by simulating such mechanical problems, vibration level at this speed increases to non-permissible value in the order of 14 mm/sec and reaches to 22 mm/sec. By increasing operating speed of the motor to 1950 rpm, vibration level increased to 3 mm/sec without simulating such mechanical problems. However, vibration level increased up to danger level (50 mm/sec) by simulating the mechanical problems at 1950 rpm. It is apparent that increasing running speed has increased vibration level slightly; however, vibration level increased twice when simulating mechanical problems by increasing running speed about 15%. A spectrum measured on the motor at speed 1950 rpm is shown in Fig. 4 showing that a single peak of high vibration amplitude (50 mm/sec) at the exciting frequency and indicating misalignment and looseness problems that generated the high vibration level. It is obvious that vibration level increases with increasing speed and the vibration level increases greatly with increasing excitation sources in the system. Periodic maintenance of variable speed drives is important to avoid sudden breakdown and failure.

5. VIBRATION ANALYSIS OF A VERTICALLY MOUNTED PUMP

Vibration measurements were done on a vertically mounted pump of axial flow type in the field at Awlad-Tuke Pumping station no. 2. The pump station was in the early stage of operation where high level of vibration and noise was observed besides bearing failure problem occurs frequently during the final submission of the plant [9]. Measurements were done on six pump units at no load conditions where the motor was disconnected completely from the pump via the coupling, and at full load condition where the pump is rotating at the design point. For no load condition, vibration measurements were done on 8 locations on the motor where for full load condition, vibration data were recorded on 19 locations on the motor, pump, bearings, and foundation in the axial and radial directions as shown in Fig. 5.

Overall vibration measurements done on the pumping units during the normal operating conditions show that vibration level is not permissible on some locations at the units. To determine the exciting sources of vibration and to reveal whether the high vibration level is due to excitation forces or due to resonance of the system, the motor was disconnected from the pump system and elevated short distance above to isolate the motor from any resonance source of the pump or structure. The motor was operated at this condition and vibration level was measured. Vibration levels at no load conditions are shown in Fig. 6 for the six motors of the pumping units. It is apparent that the maximum vibration level during no load conditions occurs at the motor non-drive end in the radial direction on both sides of the motor at locations 2 & 3. Maximum vibration level for no load conditions occurs at point 2: for unit 1 is 4 mm/sec, for unit 2 is 6.5 mm/sec, for unit 3 is 9.5 mm/sec, for unit 4 is 7.3 mm/sec, for unit 5 is 7 mm/sec, and for unit 6 are 5.2 mm/sec at point 2 and 28.4 mm/sec at point 3.

Overall vibration levels measured at full load condition is shown in Fig. 7. It is apparent that vibration levels measured during full load are slightly larger than that during no load conditions at the corresponding locations. This means that connecting the motor to the pump system has little effect on vibration level measured on the motor. So, the vibration source is from the motor itself whether connecting to a load or not. There was a problem of storing the motor for more than 5 years in the horizontal direction, as it should have been stored in the vertical direction according to its installation and operation in the field. Also, there was another problem of fixing the motors to the foundation through weak ribs. Such vertical big pumps should be connected to the foundation.
through strong supports; otherwise, they work as cantilevers fixed at the bottom to the suction basin foundation and free at the upper end. High vibration levels were measured on the motor non-drive end in the radial direction, and also on the upper and lower guide bearings showing vibration forces at these locations. Vibration level monitored for pump no. 2 at three different dates show that vibration level change at some locations and nearly constant at others as shown in Fig. 8. Due to excitation sources at the motor non-drive end and at the pump bearings, vibration levels at these locations are usually changing due to the presence of faults and problems at these locations and operation during such faults increases vibration levels and may lead to damage and failure.

Frequency analyses were done on the six machines at no load and full load conditions to define the causes of high level of vibration. As was seen before from overall vibration measure that maximum vibration levels were found on the motor non-drive end at points 2 & 3 and on the bearings of the pump. Vibration spectra measured at point 2 for two pumping units 2 & 6 at both no load and full load conditions are shown in Figs 9-12. Each machine has its distinct spectra at no load and full load conditions. For load conditions, high non-harmonic frequencies are found due to hydraulic disturbance in the pumps, which were not found in no-load conditions. Spectra of no-load conditions show peaks at the running speed of the motor and its harmonic, where spectra of load conditions show peaks at the running speed of the motor in addition to vane passing frequency and pump bearings frequencies. No resonance problems were found and all the high vibration levels were due to the running condition of the pumping components. Spectra taken at point 11 on the lower guide bearings for pumps 2 & 6 at full load show similar spectra for that pumps measured at different locations as shown in Figs. 13-14. Summary of the results show that there are problems at upper and lower bearings of the motors, unbalance of the rotors, upper and lower guide bearings of the pumps, and misalignment of the shafts. These problems were found in different levels in the different pumps. For the operation requirement, some components of some pumps had been replaced completely where others had to be operated under inspection and periodic measurement of the vibration. Dynamic unbalancing of the rotors, alignment of the shafts, and replacement of the faulted items were done according to the scheduled maintenance program which could bring back the pumps to a good condition capable of performing their duty in safe operation and minimum maintenance costs.

6. CONCLUSIONS

1. Design, manufacturing, assembly, and operation errors that cause structural and mechanical damage can be reduced through quality assurance in the large scale pumping station.
2. The trend of using variable speed drives in the pumping stations increases the necessity of applying predictive maintenance and periodic inspection as vibration level increases with increasing speed and amplitude of vibration increases greatly with increasing excitation forces.
3. Vibration level generated from vertically mounted pumps is greater than that for horizontally mounted pumps. Correct aligning of shafts, good balancing of rotors, and strong supporting of the components to the foundation are necessary to decrease the produced vibration levels.
4. Increasing load of the pumps has little effect on the generated vibration levels.
5. Special care should be done to monitor operational health of vertically mounted pumps. Disassembly and reassembly for long coupled vertical pumps should be done precisely.
REFERENCES


Table 1 Modal parameters of the variable speed motor

<table>
<thead>
<tr>
<th>Mode</th>
<th>Nat. freq. (Hz)</th>
<th>Damping (%)</th>
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<tr>
<td>1</td>
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<tr>
<td>2</td>
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<td>9</td>
<td>1544</td>
<td>2.0</td>
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Table 2 Vibration levels (mm/sec.) for the v.s. motor

<table>
<thead>
<tr>
<th>Measured points</th>
<th>At 1725 rpm</th>
<th>At 1950 rpm</th>
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<tbody>
<tr>
<td>1</td>
<td>13.8</td>
<td>27.0</td>
</tr>
<tr>
<td>2</td>
<td>14.8</td>
<td>30.9</td>
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<tr>
<td>3</td>
<td>14.0</td>
<td>50.5</td>
</tr>
<tr>
<td>4</td>
<td>22.0</td>
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<tr>
<td>5</td>
<td>14.6</td>
<td>22.3</td>
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<td>6</td>
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</table>

Fig. 3 Mode shape no. 1 for the v.s. motor.

Fig. 4 Spectrum at pt. (3) on the v.s. motor at 1950 rpm
Fig. 5 Measurement locations for vibration measurements of Awlad-Tuke (2) P.S.

Fig. 6 Vibration levels (mm/sec.) at no load conditions.

Fig. 7 Vibration levels at full load conditions.

Fig. 8 Variation of vibration levels for m/c (2) during different periods of monitoring vibration of Awlad-Tuke P.S.