DIAGNOSIS VIBRATION PROBLEMS OF PUMPING STATIONS: CASE STUDIES

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ABSTRACT

There are many pumping stations in Egypt with different types and installations including: centrifugal, axial, vertical, and inclined units operating under different conditions. These pumping stations are subjected to mechanical and hydraulic problems generating dynamic loads and stresses affecting performance and maintenance costs. Safe working of these stations verifies the operation function and efficient water management and distribution. Four different case studies are tested in the field and lab representing the common problems leading to failure and damage for some components of pumping stations. Failure and damage occur for these pumping stations lead to maintenance problems, environmental problems, and water management problems.

Vibration level generated from vertically mounted pumps is high, dangerous, ant not permissible. Maximum vibration level measured at no load is of level 29 mm/s; however, maximum vibration level increases 7% at full load. Pump loading creates other sources of high vibration of level 15 mm/s due to misalignment and pump bearings problems. Balancing of the unbalanced motor fan enhances dynamic performance greatly as vibration level decreased 92%. Envelope spectrum proves a good tool for diagnosing bearing problems. Replacing the damaged bearing reduces vibration level 95%. Vibration level increases with increasing speed of the variable speed motor and amplitude of vibration increases greatly with increasing excitation forces. At normal operating condition, vibration level increased 3.8% by increasing running speed 66%. However, Vibration level increased 58% by increasing speed 66% at minor unbalance, increased 98.0% for moderate unbalance, increased 200% at severe unbalance problem. Vibration analysis is necessary to detect and diagnose faults of the pumping stations and to avoid any failure or malfunction.

Keywords: Dynamic analysis, vibration analysis, fault diagnosis, pumps.

NOMENCLATURE

[I]	:	Identity matrix.
[K]	:	Global mass.

[M]	:	Stiffness matrices.
B_d	:	Ball diameter.
b	:	No. of balls.
f_r	:	Undamped natural frequency (Hz).
P_d	:	Pitch diameter.
R	:	Mode no.
R	:	Gear ratio.
R_v	:	Relative rev. per sec. between races.
Z_1	:	No. of driver gear teeth.
Z_2	:	No. of driven gear teeth.
β	:	Contact angle.
Ω	:	Motor rotation speed (rpm).
ω; f	:	Exciting frequency (Hz).
ω_r	:	Undamped natural circular freq. of the rth mode (rad/s) = 2π fr.

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 1500 large-scale irrigation and drainage pumping stations in Egypt operating under different conditions. The mission of these stations is to plan for serving irrigation and drainage operations. 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 operating is collected and analyzed to predict when the machine is about to fail. Many problems are encountered with pumping stations (**Nasser [1]**).

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 temperature, suction pressure, flow rate, etc., may yield significant information, but vibration still the most appropriate condition related parameter. 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.

2. PUMP VIBRATION

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 (**Smith,** et al **[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 (**Awasthi**, et al **[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 (**Hancock [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 (**Lifson**, et al **[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 (**Walter**, et al **[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 (**Lees**, et al **[7]**).

3. 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:

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

where:

[M]: Stiffness matrices.

[K]: Global mass.

Assuming harmonic motion:

$$\ddot{x}_r=-(2\Pi \mathrm{f_r})^2 x_r \qquad (2)$$

where:

$\mathrm{f_r}$: Undamped natural frequency (Hz), r: Mode no.

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

$$-(2\Pi f_r)^2\,[\mathrm{M}]\{x\}+[K\{x_r\}]=0.0 \qquad (3)$$

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

$$|\,[M]^{-1}[K]-(2\Pi f_r)^2[I]\,|=0.0 \qquad (4)$$

where:

[I]: Identity matrix.

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:

$$\omega;\mathrm{f}=\Omega/60 \quad \mathrm{Hz} \qquad (5)$$

where:

ω; f: Exciting frequency (Hz).

Ω: Motor rotation speed (rpm).

The exciting frequency of the pump is:

$$\omega;\mathrm{f}=\Omega/(60*\mathrm{R}) \quad \mathrm{Hz} \qquad (6)$$

where:

R: Gear ratio.

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 2nd harmonics of teeth-meshing frequency. The gear-mesh frequency can be calculated as:

$$\omega; \mathrm{f} = (\Omega * \mathrm{Z}_1)/60 \quad \mathrm{Hz} \qquad (7)$$

or

$$\omega; \mathrm{f} = (\Omega * \mathrm{Z}_2)/(60 * \mathrm{R}) \quad \mathrm{Hz} \qquad (8)$$

where:

Z_1: No. of driver gear teeth.
Z_2: No. of driven gear teeth.

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:

$$\omega; \mathrm{f} = \left(\frac{\mathrm{b}}{2}\right) R_v \left(1 - \frac{B_d \cos\beta}{\mathrm{P_d}}\right) \quad \mathrm{Hz} \qquad (9)$$

where:

b: No. of balls.
R_v: Relative rev. per sec. between races.
B_d: Ball diameter.
β: Contact angle.
P_d : Pitch diameter.

For inner race defect, the exciting frequency is:

$$\omega; \mathrm{f} = \left(\frac{\mathrm{b}}{2}\right) R_v \left(1 + \frac{B_d \cos\beta}{\mathrm{P_d}}\right) \quad \mathrm{Hz} \qquad (10)$$

And for ball defects, the exciting frequency is:

$$\omega; \mathrm{f} = \left(\frac{\mathrm{P_d}}{B_d}\right) R_v \left(1 - \left(\frac{B_d \cos\beta}{\mathrm{P_d}}\right)^2\right) \quad \mathrm{Hz} \qquad (11)$$

4. PUMP FAULT DETECTION AND DIAGNOIS

There are many problems affecting dynamic performance of pumping stations. These problems include misalignment, unbalance, bearing and hydraulic problems. These problems generate vibrations of high levels which may damage the pump components. The most common problems that can be found in any pumping station are unbalance, and misalignment.

4.1 Unbalance Problem

Unbalanced rotors are a very common cause of machinery malfunction. An improperly balanced machine has many hidden costs in downtime and parts due to accelerated wear and performance issues (**Spectra Quest.com [8]**). In fact, if the amount of unbalance weight was known, along with the distance from the shaft centerline, the actual amount of unbalance force generated could be easily calculated. A 2-ounce weight on the shaft will generate an applied force of 826 lbs. at a speed of 3,600 rpm, doubling that weight to 4 ounces will double the force to 1,652 lbs. again at 3,600 rpm**.** Because the force is proportional to the square of the speed, increasing the speed from 3,600 rpm to 7,200 rpm increases the unbalance force by a factor of four to nearly 3,304 lbs. Although actually calculating the force generated by an unbalance is not typically done for routine vibration measurement and analysis work, this example illustrates how a relatively small unbalanced weight can produce a significant amount of force at machine operating speeds (**Coutney [9]**). Unbalance is the result of the coupling bore not being perfectly centered or being off angularly. If the coupling mass is located only a few mils off-center, it creates static unbalance. If the bore angle is not square with the coupling faces within a few mils, it causes couple unbalance. Most often the coupling will increase the total assembly's unbalance and, therefore, the amplitude at 1 x rpm (**update-intel.com [10]**).

4.2 Misalignment Problem

Misalignment between two mating shafts is the most common cause of machinery deterioration. A properly aligned machine can save replacement parts, inventory, and energy consumption (**Spectra Quest.com [8]**). As with unbalance, misalignment is a fact of life in machinery installations. There are many ways to non-intrusively detect and monitor changes in alignment, the two most effective are vibration and temperature monitoring. Examples of these methods are as follows (**Coutney [9]**). The first way is a change in amplitude in a specific direction may indicate misalignment. Vibration is normally measured in a radial direction on all bearing housings, and at least one reading is taken per shaft in the axial direction. The golden rule applied by vibration analysts when looking for misalignment: "When the amplitude in the axial direction is more than 50 percent of the highest radial amplitude, then misalignment is suspected." Misalignment will also be detected by looking for the 2X and 3X components of the vibration spectrum. The second way, confirmation of misalignment is best performed by using phase analysis. The amplitude and phase of the 1X vibration can be trended. When a phase change is detected, alarms can be set-up to indicate a developing problem.

Detecting shaft misalignment using vibration analysis can be difficult and somewhat deceptive unless understanding the true mechanism of misalignment. Shaft misalignment will be indicated by higher running speed and/or twice running speed vibration frequency components with high axial vibration and a 180 degree phase shift across the coupling. Several controlled tests by several individuals over the past ten years have indicated that vibration spectral patterns can be different under similar misalignment conditions depending on the type of flexible coupling installed on the machinery and under certain conditions, virtually no vibration can be detected even under moderate to severe misalignment (**maintenance resources.com [11]**).

5. RESEARCH PROCEDURES AND FACILITIES

Four different case studies are tested in the field and lab representing the common problems leading to failure and damage for some components of pumping stations. Failure and damage occur for these pumping stations lead to maintenance problems, environmental problems, and water management problems. A new axial flow vertical pump, showed high vibration and noise in the early stage of operation, was measured with and without load. The second case for a pump system had unbalance problem; the third case was a bearing problem for a pump system. Finally, a case for a laboratory variable speed pump model, where modal testing and forced vibration analysis, were done at different speeds with simulating unbalance problem of different degrees.

Vibration measurements and dynamic analysis were done for the four cases of pumping system by measuring overall vibration levels and vibration spectra. Overall vibration levels indicate severity of vibration and compared with **ISO 10816-1 [12]**. Also, Vibration spectra, is a relation of vibration amplitude with frequency, are measured to determine the excitation frequencies and the source of high vibration. According to ISO 10816-1, class II was used as a guide limit for the pumping systems. The good vibration limit is up to **1.12 mm/s** rms vibration velocity, acceptable limit is up to **2.8 mm/s**, just tolerable limit is up to **7.1 mm/s**, and not permissible higher than **7.1 mm/s**. Vibration signals were recorded along the different components of the pump systems axially and radially.

Two Vibration systems were used to measure and analyze data on the four case studies. The first system (Data Collector/Analyzer) consists of two analyzers: type 2526 B&K with accelerometers type 4391 B&K, personal computer, and software package type 7107. The second system (pulse lab shop) of seven channel pulse system and software package for vibration analysis and modal testing.

6. VIBRATION RESULTS AND ANALYSIS

6.1. Vibration Problems of a New Pumping System

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 **(Abdel-Rahman et al., [13])**. 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. 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. (1)**.

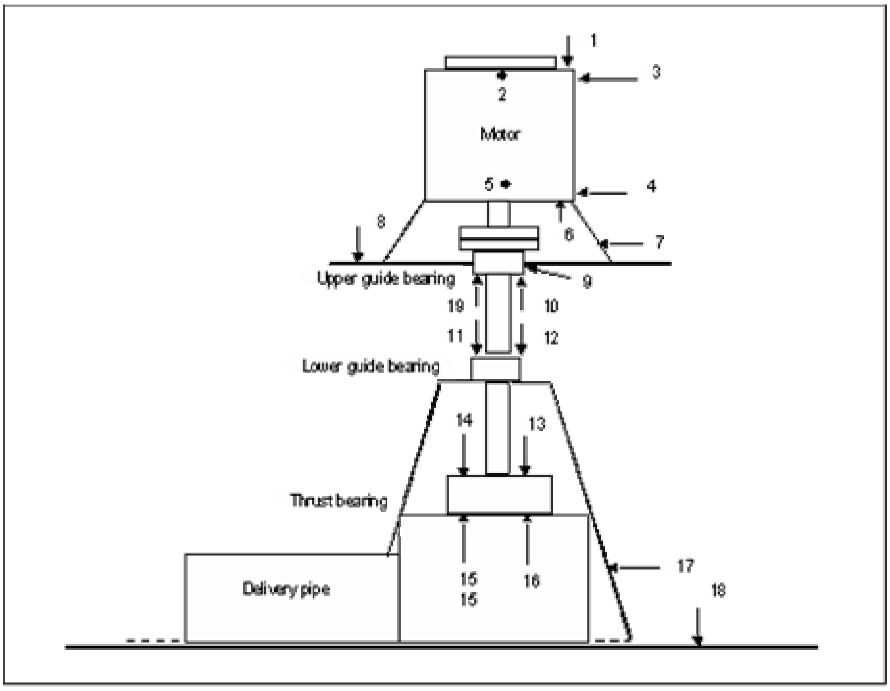


Fig. 1 Measurement locations for vibration of Awlad-Tuke (2) Pumping Station

Overall vibration measurements done on the pumping units during the normal operating conditions showed 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. (2)** for the six motors of the pumping units. It is apparent that the maximum vibration level during no load conditions occurred 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. However, maximum vibration level measured at point (3) for pump unit (2) is of level 29 mm/s.

Overall vibration levels measured at full load condition is shown in **Fig. (3)**. It is apparent that vibration levels measured during full load are slightly larger than that during no load conditions at the corresponding locations. However, maximum vibration level measured at point (3) for pump unit (2) increases 7% and of level 31 mm/s. Moreover, full load condition creates other sources of high vibration of level 15 mm/s due to misalignment and pump bearings problems for some pump units. 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. 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.

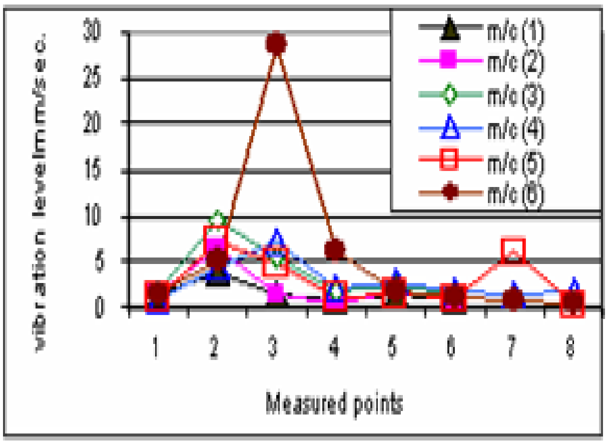


Fig. 2 Vibration levels (mm/sec) at no load

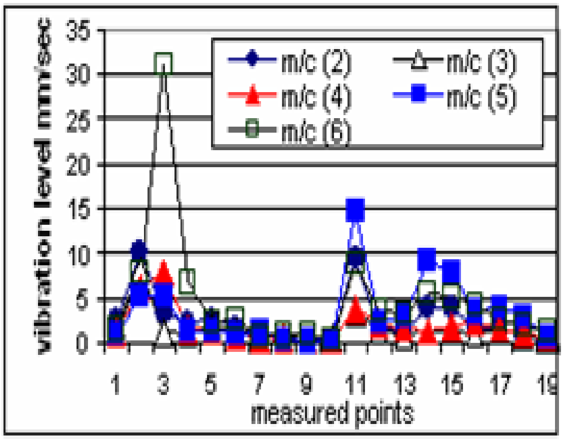


Fig. 3 Vibration levels at full load

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 the 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. (4-7)**. 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.

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, faulted components must be replaced completely and the pumping station should be operated under inspection and regular maintenance to bring back the pumps to a good condition capable of performing their duty in safe operation and minimum maintenance costs.

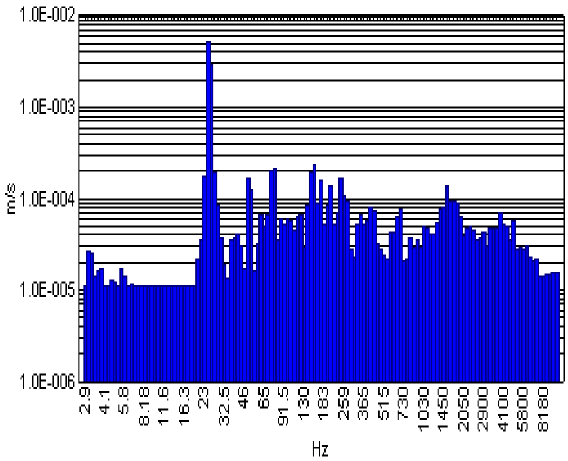


Fig. 4 Spectrum at pt. 2 of m/c 2 at no load

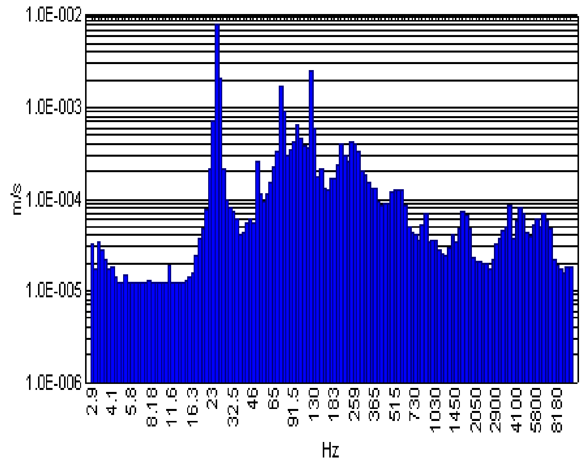


Fig. 5 Spectrum at pt. 2 of m/c 6 at no load

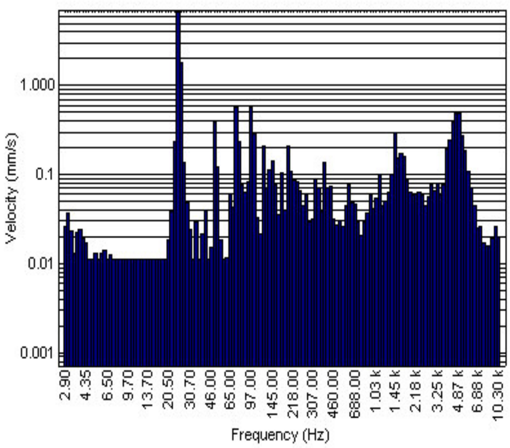


Fig. 6 Spectrum at pt. 2 of m/c 2 at full load

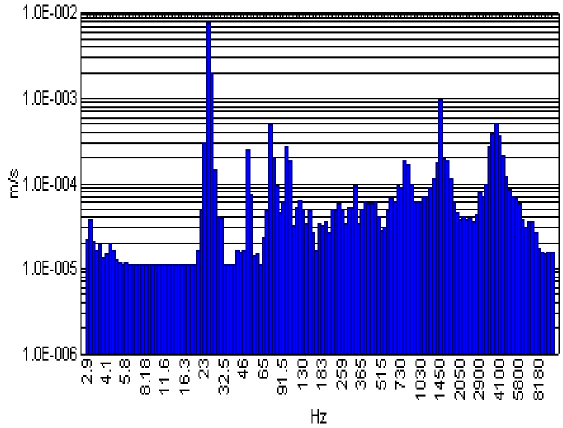


Fig. 7 Spectrum at pt. 2 of m/c 6 at full load

6.2 Unbalance Problem of a Pumping System

In this case a pump system consists of a motor, gearbox, and pump was measured and analyzed, where the motor speed is 1492 rpm and electrical power is 110 kilowatts. Overall vibration levels measured and analyzed and compared with (ISO 10816-1) showed vibration peaks of level 24 mm/s measured at the motor drive end next to the gear box. This high vibration in the not permissible zone of ISO 10816-1 indicates a great problem for the pump system. It is not safe and dangerous and not allowed to operate the pumping system at this running condition. The pump system was shut down to diagnose the problem and fix it.

Dynamic analysis was done in the axial and radial direction to determine exciting frequencies, and evaluate sources of high vibration. Vibration spectrum, as shown in **Fig. (8)**, showed high vibration levels in the axial and radial directions of maximum amplitude 23.3 mm/s at the motor speed (25 Hz). This situation indicated severe unbalance problem for the motor fan. Balancing was done to the motors shaft-fan-coupling and vibration level was measured and analyzed. Vibration amplitude decreased greatly to 1.8 mm/s in the safe and allowable zone according to the ISO 10816-1 and dynamic performance enhanced greatly as vibration level decreased 92%.

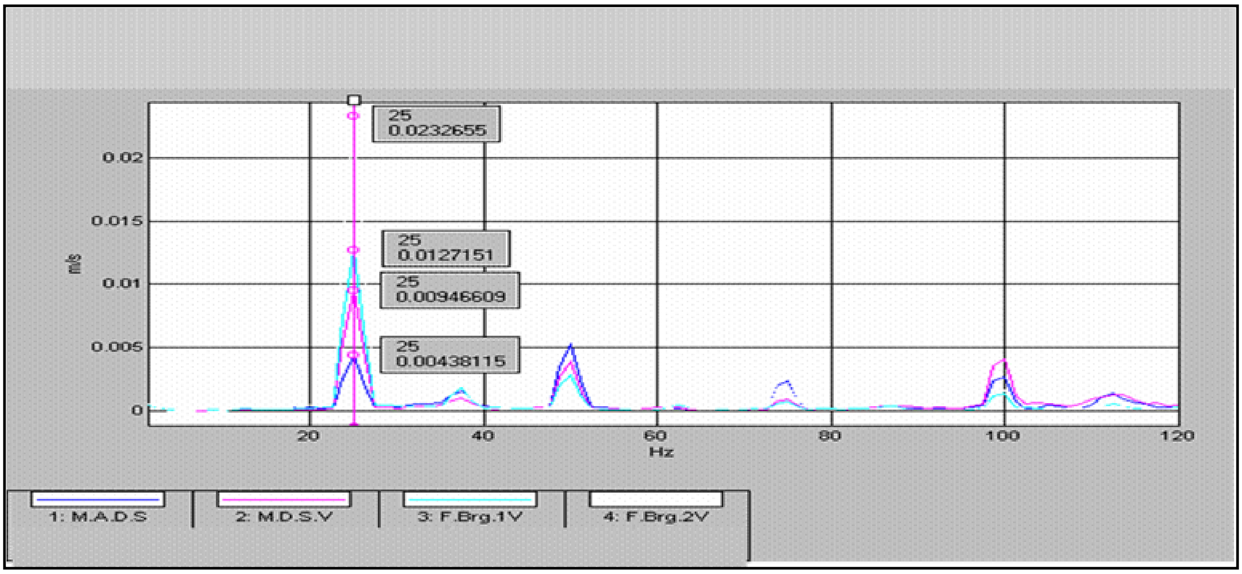


Fig. (8) Spectrums in the Vertical direction Shows highest peak at fundamental speed

6.3 Motor Bearing Problem of a Pumping System

In this case, a centrifugal pump system was tested where, the motor speed is 1146 rpm and of 120 kW power. The motor non drive end has a bearing type SKF 2324, where the bearing of the motor drive end is SKF NU 324. Frequency of the different components of each bearing was calculated and is shown in **Table (1)**. Overall vibration level measured is 4.5 mm/s which is beyond the allowable limit according to ISO 10816-1. Dynamic analysis was done and auto spectrum showed different vibration peaks of level 4.1 mm/s at 655 Hz, level 1.6 mm/s at 425 Hz, as shown in **Fig. (9),** implying either a hydraulic problem or a bearing problem. Also, constant percentage bandwidth (CPB) confirmed the same trend of high acceleration vibration levels of maximum amplitude 13 m/s^2 at 655 Hz, as shown in **Fig. (10-a).**

Table [1] Frequencies for the different components of the motor bearings

Element **Rotational Speed : 1146 RPM**	**Frequency (Hz)**	
	Bearing SKF 2324	**Bearings SKF NU324**
Inner ring defect frequency	92.99	148.72
Outer ring defect frequency	59.81	99.58
Rolling element defect frequency	83.78	92.73
Inner ring rotational speed	19.10	19.10
Cage rotational speed	7.48	7.66
Rolling element rotational speed	41.89	46.36

Envelope spectrum was done and showed vibration peaks at 100 Hz and its harmonics, as shown in **Fig. (11-a)**, implying a defect of the outer ring of the motor drive end bearing according to the calculated exciting frequencies in Table (1). The damaged bearing was replaced and vibration measurements were done showing improvement in the dynamic performance, where vibration peaks at the excitations frequencies were almost insignificant, as shown in **Figures (10-b)** and **(11-b)**.

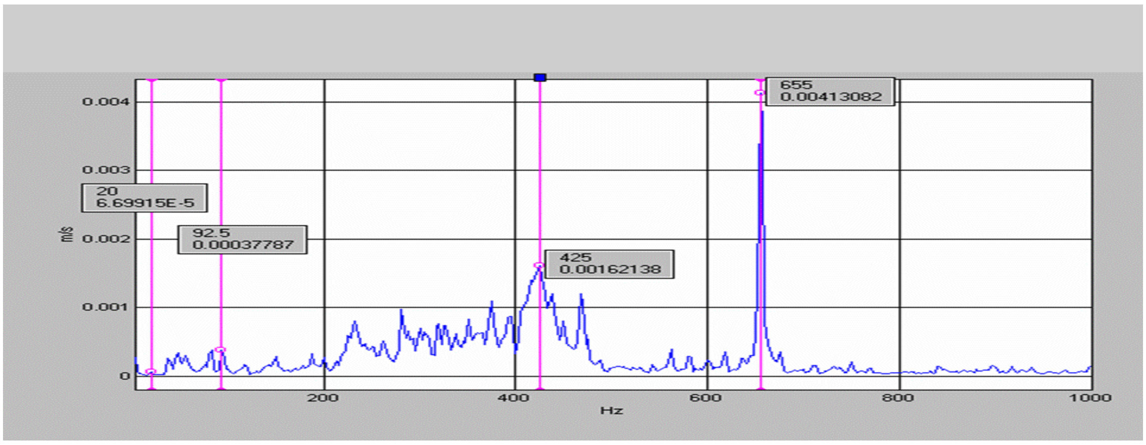
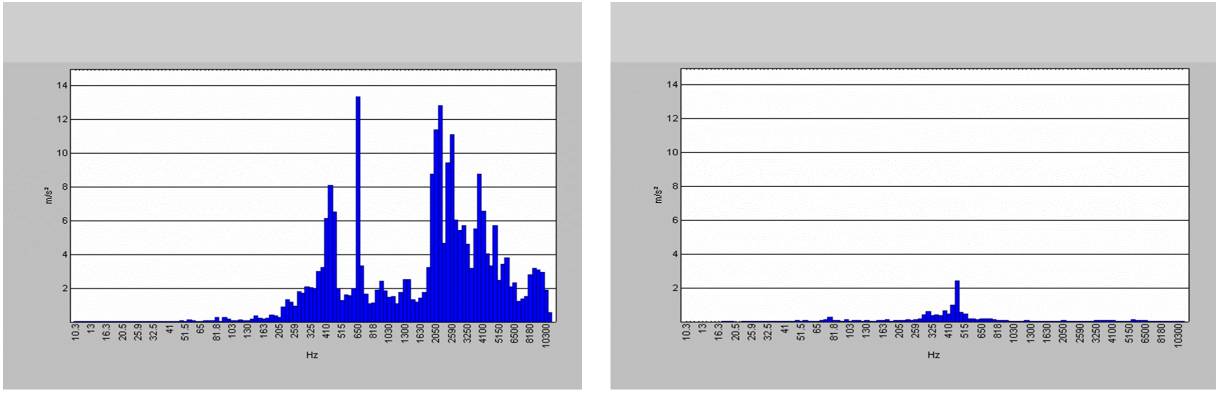
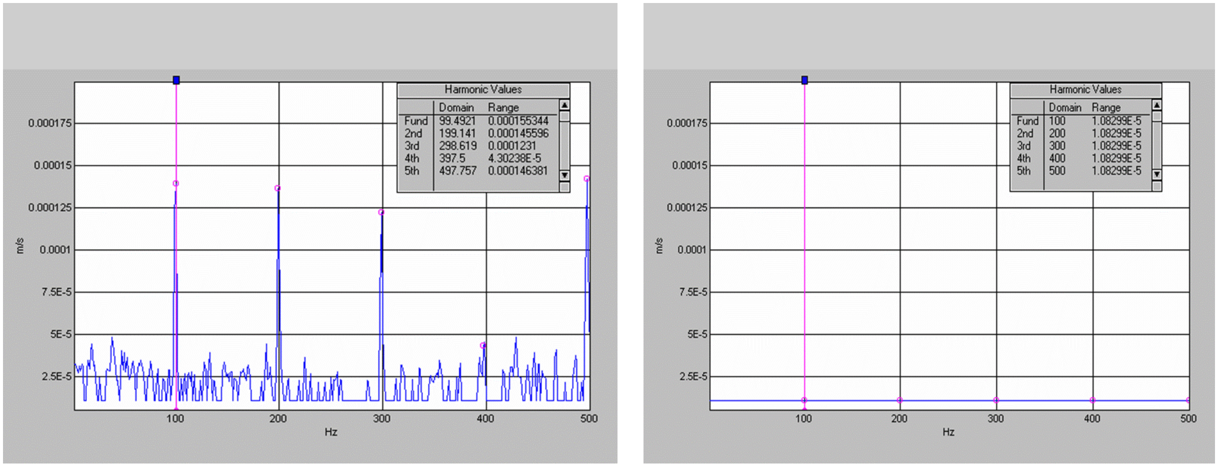


Fig. (9) Auto spectrum of the Motor Inboard Bearing Radial (Before replacement)



(a) Before Bearing replacement **(b) After Bearing replacement**

Fig. (10) CPB Spectrum of the Motor Drive end Bearing Radial



(a) Before Bearing replacement **(b) After Bearing replacement**

Fig. (11) Envelope Spectrum of the Motor Drive end Bearing Radial

6.4 Vibration Analysis of a Laboratory Variable Speed Pump System

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 pump systems at different operating conditions. A variable speed pump system model of 15 kW power was dynamically tested and analyzed in the laboratory, where unbalance problems of different degrees were simulated. Experimental modal testing was done on the pump system to determine its dynamic characteristics. **Table (2)** summarizes the modal parameters of the pump system 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 at 3360 rpm is dangerous; 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. (12)**, where sensitive nodes are defined to implement any modifications that enhance its structural properties.

Forced vibration tests were done at different speeds of the motor at 1500, 2000, and 2500 rpm where vibration levels were measured at normal operating condition. Unbalance problem of different grades from minor to moderate to severe were simulated, by adding different weights in different planes, to determine their effect on vibration level during operation at different speeds. Results of vibration measured at different speeds and different operating conditions are summarized in **Table (3)**. Vibration levels measured at normal condition are within allowable standard limit in the order of 2.65 mm/s and did not change much with varying speed. At normal operating condition, vibration level increased 3.8% by increasing running speed 66%. On the other hand, vibration level measured at minor unbalance problem is of level 5.0 mm/s (outside allowable standard limit) at 1500 rpm. However, vibration level increases 58% by increasing speed 66% at minor unbalance problem. For moderate unbalance problem, vibration level increased 98.0% by increasing speed 66%. Moreover, vibration level increased dramatically 200% by increasing speed 66% at severe unbalance problem. At 1500 rpm speed, vibration level increased 90% than normal operating condition due to minor unbalance, 210% due to moderate unbalance, and 400% due to severe unbalance. However, at 2500 rpm speed, vibration level increased 190% due to minor unbalance, 450% due to moderate unbalance, and 1400% due to severe unbalance.

A spectrum measured on the motor at speed 2500 rpm is shown in **Fig. (13),** showing simulated severe unbalance at 2500 rpm, where a single peak of high vibration amplitude (45 mm/sec) is measured at the exciting frequency. It is obvious that vibration level increases with increasing speed and the vibration level increases greatly with increasing excitation sources in the system. Vibration level increased greatly with increasing grade of unbalance. Also, at the same grade of unbalance, vibration level increased with increasing running speed. Periodic maintenance and inspection of variable speed pump systems is important to avoid sudden breakdown and failure.

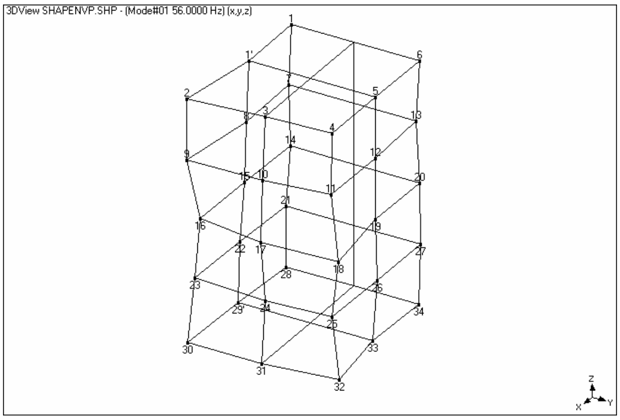


Fig. (12) Mode shape no.1 for the variable speed pump system

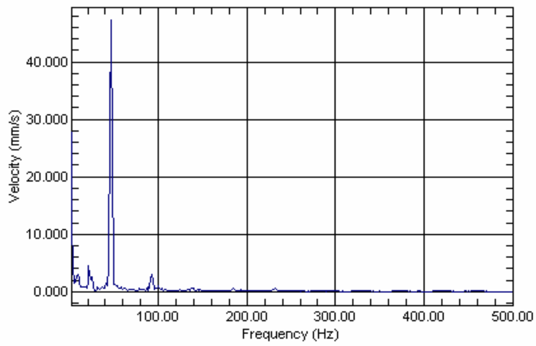


Fig. (13) Spectrum measured at severe unbalance at 2500 rpm

Table (2) Modal parameters of the variable speed pump system

Mode	Nat. freq. (Hz)	Damping (%)
1	56	61.0
2	316	7.0
3	520	9.0
4	646	0.1
5	866	0.2
6	1024	3.0
7	1186	0.6

Table (3) Vibration levels (mm/sec.) measured at different speeds and different operating conditions

Operating conditions	1500 rpm	2000 rpm	2500 rpm
Normal operation	2.60	2.65	2.70
Minor unbalance	5.00	6.80	**7.90**
Moderate unbalance	**8.20**	**10.50**	**14.80**
Severe unbalance	**14.00**	**25.00**	**42.00**

7. CONCLUSIONS

- Vibration level generated from vertically mounted pumps is high, dangerous, ant not permissible according to the standards (ISO 10816-1). Maximum vibration level measured at no load is of level 29 mm/s at the motor non drive end. However, maximum vibration level increases 7% at full load at the motor non drive end. Pump loading creates other sources of high vibration of level 15 mm/s due to misalignment and pump bearings problems.
- Unbalance problem of a pumping system produces high vibration of level 4 mm/s; however, balancing of the unbalanced motor fan enhances dynamic performance greatly as vibration level decreased 92%.
- Envelope spectrum proves a good tool for diagnosing bearing problems. A damaged bearing produces high vibration of level 4.1 mm/s vibration velocity and 13 m/s^2 acceleration vibration. Replacing the damaged bearing enhances dynamic behavior and reduces vibration level 95%.
- 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. At normal operating condition, vibration level increased 3.8% by increasing running speed 66%. However, Vibration level increased 58% by increasing speed 66% at minor unbalance, increased 98.0% for moderate unbalance, increased 200% at severe unbalance problem.

- Vibration analysis should be done regularly to bring back the pumps to a good condition capable of performing their duty in safe operation and minimum maintenance costs. 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.

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