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Condition-based monitoring of a small centrifugal pump by vibration analysis

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Abstract. Rotating machinery vibration problem is the most significant restriction which control the life time and may cause disastrous failure of the entire system. The vibration monitoring is the main element for any customized maintenance program to guarantee power plant availability and minimize danger of predictable failures. The present work is an experimental study on a small centrifugal pump running in healthy and intentionally defected to show the difference in the vibration spectral and time domain. Triaxial acceleration measurements were made at the shaft of the pump. The most dangerous case of vibration denoted by the highest amplitude as determined as well as its corresponding frequency. The most serious case was the pedestal looseness with full flow rate which cause a vibration of amplitude of (0.104g) at frequency (47Hz). The unacceptable vibration, as classified according to ISO 10816 to have amplitude of (0.5g) for small machines within this rang of acceptable vibration it was not difficult to predict the artificial defects mode in the pump and demonstrated the capability of condition monitoring.

Key words: small pump, vibration monitoring, rotating shaft, NI USB-6001.

1. Introduction

Vibration analysis of rotating equipment is capable to detect a great number of system errors. Unbalance, rotor crack, looseness, impeller defects, bearing defects, misalignment, high speed, rotor to stator rub, flow whirl/whip make up the main portion of the spotted vibrational frequency spectra of rotating equipment. Vibrational spectra can be used to define the kind of rotating system irregularity. Machine condition monitoring is a significant portion of condition-based maintenance (CBM), which is becoming recognized as the most efficient strategy for carrying out maintenance in an extensive range of industries. The first rejoinder was 'preventive maintenance', where maintenance is carried out at intermissions such that there is a very minor probability of catastrophe between repairs. However, these consequences in much bigger use of spare parts, as well as further maintenance effort than necessary.



In this paper, focus is on unbalance, pedestal looseness and impeller defect, because the vibration caused by unbalance may destroy critical parts of the machine, such as bearings, seals, gears and couplings. The looseness may cause other faults such as rub impact fault of the rotor-stator, even may lead to catastrophic coincidences. The impeller defect is considered to be the most dangerous fault as it makes perturbation in the amount of flow, consequently; the system will vibrate with huge amplitudes and the pump may be destroyed at any time.

The case study of this paper is a water pump that pumps the water in a closed loop system and the system will run continuously. Some faults will be made in order to detect the corresponding vibrational spectra.

Condition monitoring of centrifugal pumps were studied by many researchers in the previous work using different techniques. Atkins, et al. (1985) performed a critical speed and imbalance response analysis of an eight-stage centrifugal pump. They considered the variation of stiffness and damping coefficients with speed for both the oil film bearings and the fluid seals. Cerwinske, et al. (1986) presented the effects of high pressure oil seals on the dynamic response of a recycle gas compressor. This effect caused several serious wrecks due to unpredicted high vibration at synchronous speed. Nikolajsen and Gajan (1988) developed a new computer program for pump rotor analysis to improve the accuracy of critical speed, stability and response calculation for multistage centrifugal pump. Nelson and Dufour (1992) discussed some of the basic questions of where to start and how to conduct the problem analysis and then apply corrective measures to centrifugal pumps. Lienau and Lagas (2008) discussed rotordynamic criteria for static behavior, dynamic behavior, and stresses. Scheffer (2008) used the vibration measurement analysis to identify faults in pumps. He showed the advantage of using vibration as a condition monitoring tool as it does not disturb the normal operation of the equipment. Birajdar, et al. (2009) diagnosed the noise and vibration in centrifugal pump and its remedies can be worked out by using specific techniques to identify and rectify specific pump problems, such as unbalance, misalignment, turbulence, cavitation and many others. Das, et al. (2010) attempted in their work to control active vibration of an unbalanced rotor–shaft system on moving bases with electromagnetic control force provided by an actuator consisting of four electromagnetic exciters. Kumar, et al. (2012) performed an experimental studies on a rotor to predict its unbalance. The vibration velocities were measured at five different speeds using FFT (Fast Fourier Transform) at initial condition. Marscher (2014) presented pump rotordynamic problems, including the bearing and seal failure problems that they may occur. They are responsible for a significant amount of the maintenance budget at many refineries and electric utilities. Abdel Fatah, et al. (2018) used condition based maintenance technique for performing diagnostics, troubleshooting and determining the proper maintenance action. After the previous reviews this manuscript will investigate all the possible cases that cause fault of the pump, not previously investigated globally in the same study but studied individually, which decrease the system efficiency to highlight the most dangerous cases and its effect on the system.

2. Experimental setup

Experiments were done on a system shown in (Fig.1) composed of centrifugal pump (QB-60). The centrifugal pump vibration is detected by 3D accelerometer sensors kit, the TE Connectivity Model 4030^[12], the range and accuracy of the accelerometer is shown at table1. The kit is fixed on shaft case with position and directions of measuring shown in (Fig.2). The accelerometer kit is connected to National Instrument data acquisition^[13] (NI USB-6001) (Fig.3), the data sampling rate and accuracy of the data acquisition is shown in table1. The centrifugal pump has single stage with maximum flow rate 25 liter/min., maximum head 25 meter and rotating speed 2850 rpm.

Electric motor (0.5 HP, 220V 50 Hz) is used to drive the pump to study the effect of the defects and the flow rate. The working fluid is water, firstly; stored in the tank shown in (Fig.1). The system has two gate valves one at tank exit and the other at the pump outlet. The system is considered a closed system. The system is designed with a moving frame to be easily transport. LabVIEW is used in signal

conditioning. The LabVIEW block diagram for this experiment is shown in (Fig.4). Vibration measurements were made for the cases shown in table2.

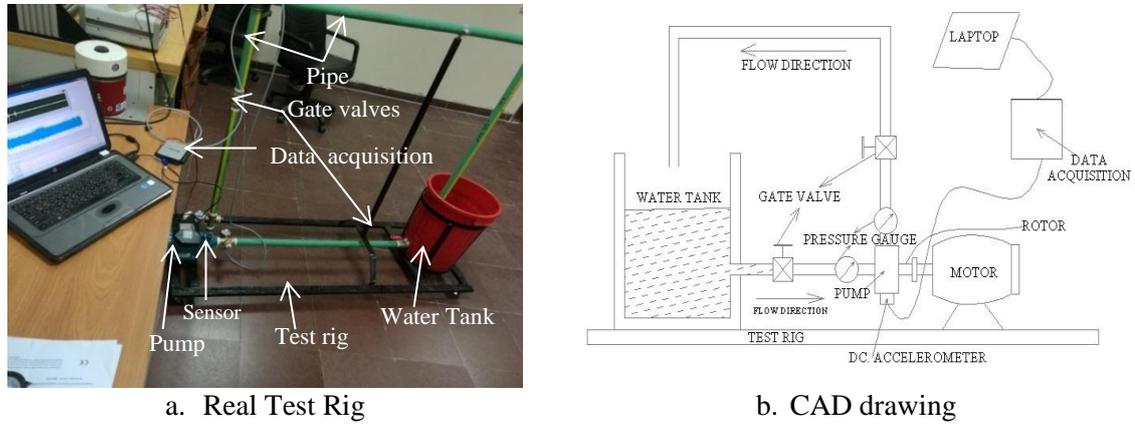


Fig. 1. Experimental test rig

Table 1. Uncertainty and sampling

	Range	Accuracy
Accelerometer model 4030	±2g	±10%
NI data acquisition (USB-6001)	Maximum sample rate = 20KS/s AI FIFO = 2047 samples AO FIFO = 2047 samples	Absolute accuracy : Typical full scale = 6mV Maximum over temperature = 26 mV System noise = 0.7 mVrms Absolute accuracy (no load) : Typical at full scale = 9.1 mV Maximum over temperature, full scale = 34 mV

Table 2. Design of the experiments

Defect type	Flow rate Liter/min
1 Balance mode	Zero flow
	8
	12.5
	16
	25
2 Unbalance mode (defected fan)	25
3 Pedestal looseness	Zero flow
	25
4 Unbalance and pedestal looseness	25

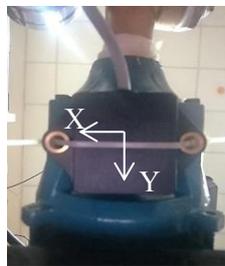
3. Results of measurements

For each case the spectral domain, time domain and waveform are studied in vertical, axial and radial (x, y and z) directions respectively. The spectral domain is the most significant indicator that indicates at which frequency the highest amplitude will take place. Time domain can be also investigated for every critical case to show the time corresponding to the highly vibration frequency. From the time domain and spectral domain the high amplitude can be determined at certain frequency and time.

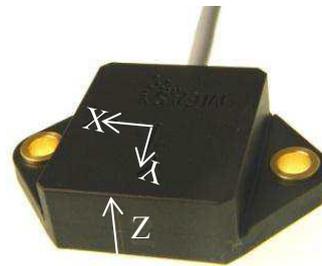
Since the time is independently available parameter so at any time the amplitude can be estimated, whenever, the critical time, corresponding to highly amplitude, can be avoided. The most efficacious and important results will be displayed and taken care of. By examination, the rotating speed of the pump should be less than 2850 rpm so that the most vital frequency to take care of should be less than (47.5Hz) and the results that showed below will be selected based on the most dangerous and influential cases.

3.1. Balance mode without flow in x-direction

The system will run in balance mode without any additional flow or faults. When the pump runs dry without water, air is the working fluid which does not act as a coolant. So the pump will be overheated and damage the bearings, shaft and impeller. It is shown in (Fig.5) that there are small peaks at (47Hz and 100Hz) but the highest amplitude is (0.054g) at frequency (200Hz) because the pump started to collapse.



a. Sensor position on the pump



b. Sensor directions

Fig. 2. Vibration sensors location on pump



Fig. 3. Accelerometer kit (NI USB-6001)

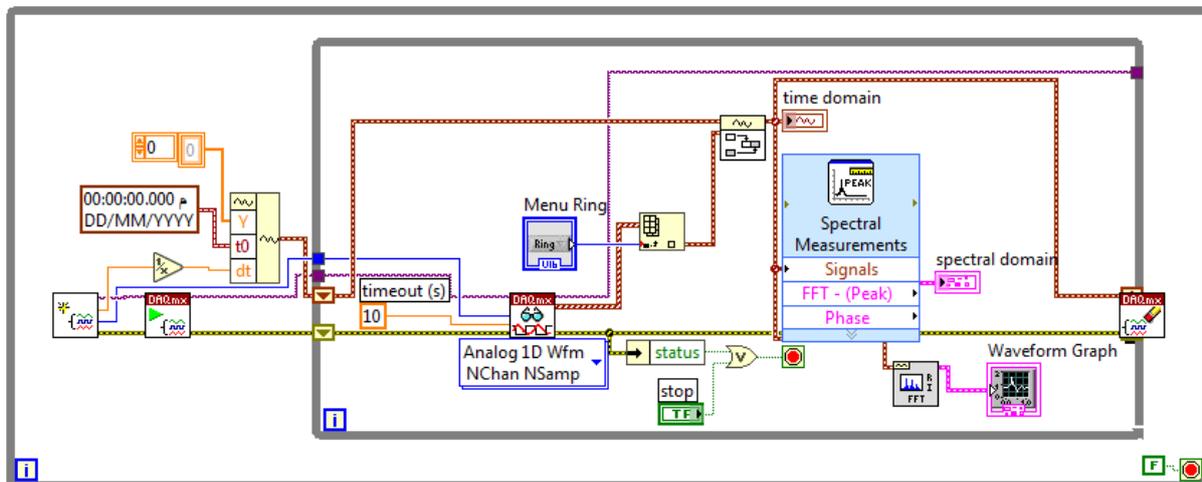
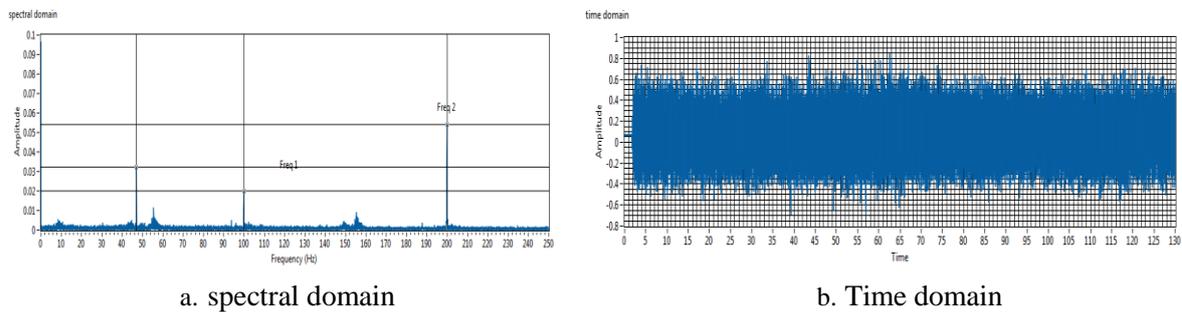


Fig. 4. LabVIEW block diagram



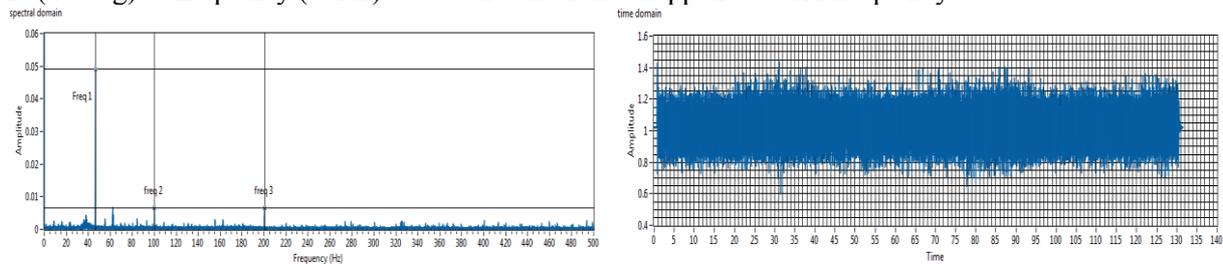
a. spectral domain

b. Time domain

Fig. 5. Balance mode without flow in x-direction

3.2. Balance mode with flow rate of 8 liter/min. in z-direction

When the pump runs at a very low flow rate, the power input is converted to thermal energy, causing a rapid temperature rise and the liquid will vaporize and cause thermal expansion of the pump internal parts like that shown in (Fig.6). There are small peaks at (100Hz and 200Hz) but the highest amplitude is (0.049g) at frequency (47Hz) due to the unbalance happened at 1X frequency.



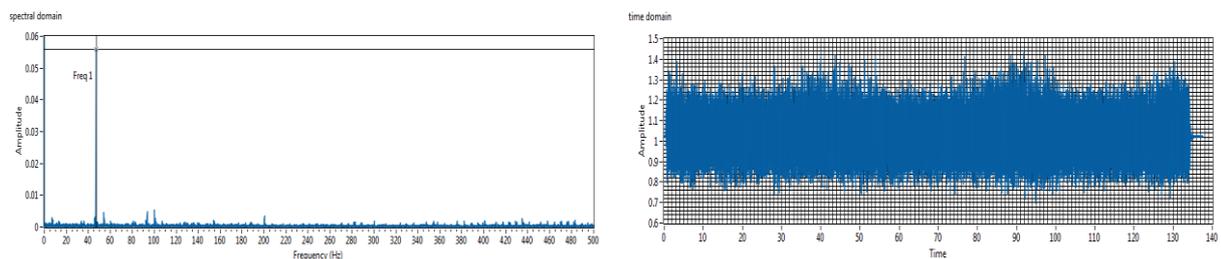
a. spectral domain

b. Time domain

Fig. 6. Balance mode with flow rate of 8 liter/min. in z-direction

3.3. Balance mode with flow rate of 12.5 liter/min. in z-direction

Rotor instability that occurs at low flow operation can lead to shaft failures, premature packing wear and mechanical seal failures. There are small peaks at (100Hz and 200Hz) but the highest amplitude is (0.056g) at frequency (47Hz) due to the unbalance happened at 1X frequency see (Fig.7)



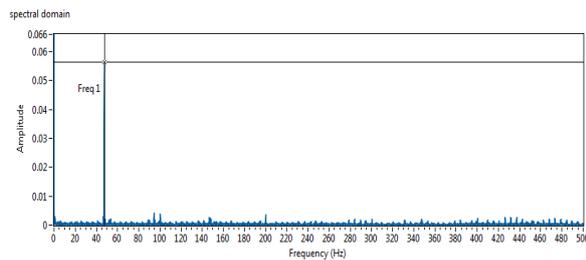
a. spectral domain

b. Time domain

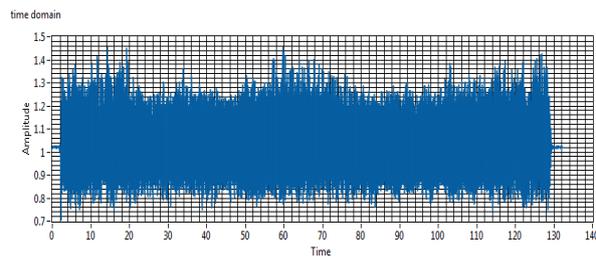
Fig. 7. Balance mode with flow rate of 12.5 liter/min. in z-direction

3.4. Balance mode with flow rate of 16 liter/min. in z-direction

The subsynchronous vibration also takes the same shape that appears at 8 liter/min. and 12.5 liter/min. spectral domain graphs. The highest amplitude is (0.056g) at frequency (47Hz) as shown in (Fig.8) because suction and discharge recirculation, which happens when the fluid does not flow through the pump properly. This can lead to catastrophic failure of the pump when portions of the impeller inlet or discharge vanes fatigue and fail by breaking off.



a. spectral domain

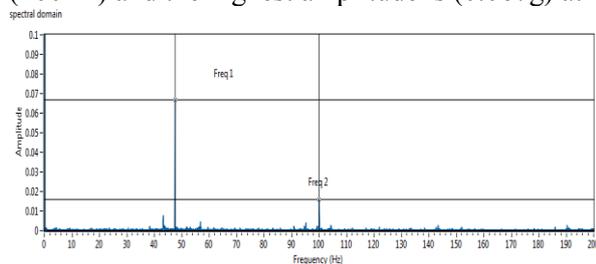


b. Time domain

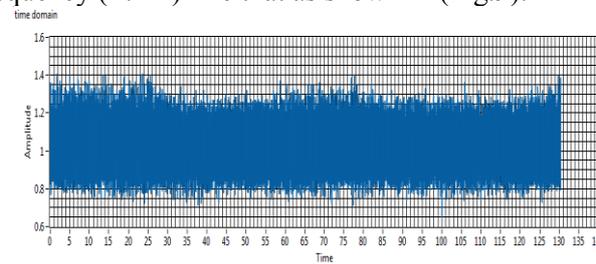
Fig. 8. Balance mode with flow rate of 16 liter/min. in z-direction

3.5. Balance mode with flow rate of 25 liter/min. in z-direction

The system will run at full load (full flow) 25 liter/min. The smallest peak appeared at frequency (100Hz) and the highest amplitude is (0.067g) at frequency (47Hz) like that as shown in (Fig.9).



a. spectral domain

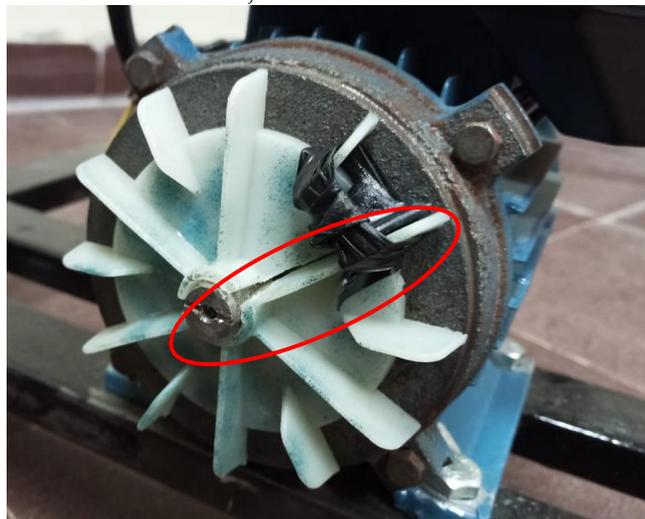


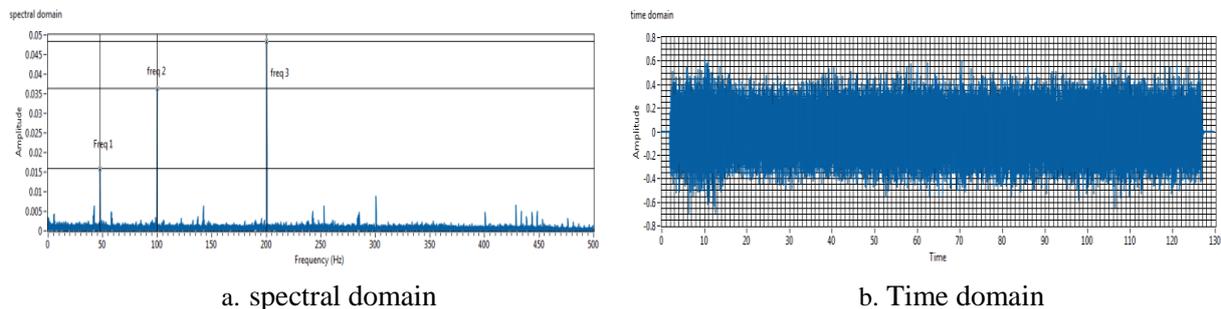
b. Time domain

Fig. 9. Balance mode with flow rate of 25 liter/min. in z-direction

3.6. Unbalance mode with flow rate of 25 liter/min. in x-direction

Unbalance is the most common cause of machine vibration, an unbalanced rotor always causes more vibration and generates excessive force in the bearing area and reduces the life of the machine. Most of the rotating machinery problem can be solved by using the rotor balancing and misalignment adjustment. The vibration caused by unbalance may destroy critical parts of the machine, such as bearings, seals, gears and couplings. There are many sources of unbalance. In the current study, unbalance is simulated by making a defect in the cooling fan showed in (Fig.10). As shown in (Fig.11) there are some peaks take place at (47Hz and 100 Hz), also the subsynchronous vibration takes place which means that the system experiences unbalance. The highest amplitude is (0.048g) at frequency (200Hz) because of the unbalance external force, fan defect.

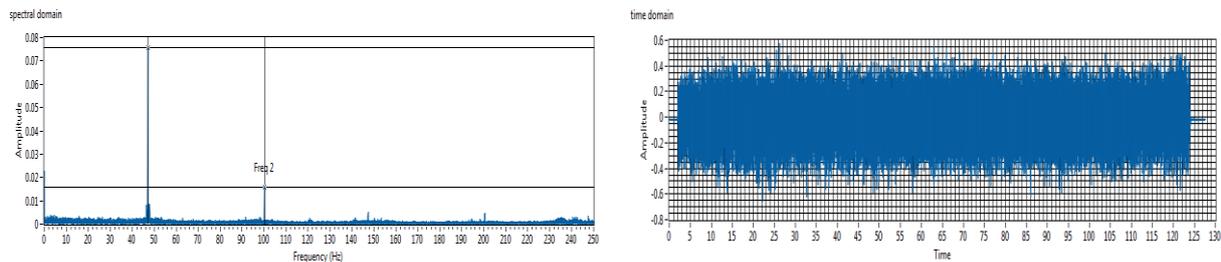
**Fig. 10. Defected cooling fan**



a. spectral domain
b. Time domain
Fig. 11. Unbalance mode with flow rate of 25 liter/min. in x-direction

3.7. Pedestal looseness mode without flow in y-direction

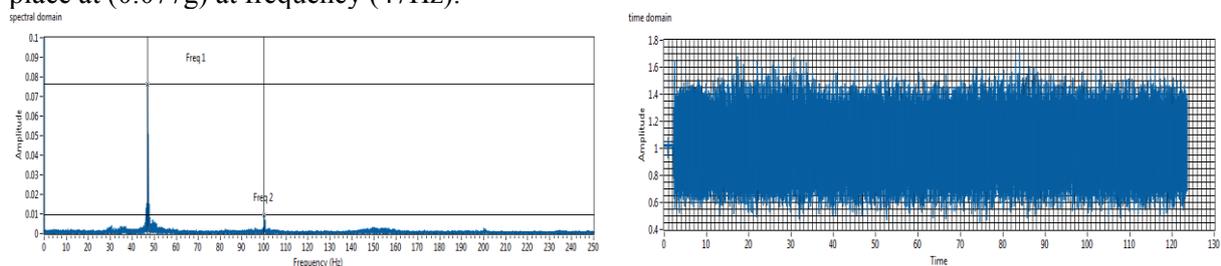
In a rotor bearing system, the loosened bolt of the pedestal will reduce pedestal stiffness and mechanical damping. This will lead to violent vibration of the whole system. When the looseness fault is serious, it may cause other faults such as rub impact fault of the rotor-stator. This may even lead to disastrous accidents. Therefore, the research on pedestal looseness is significant in engineering practice for the safe operation of rotating machinery. It affects service life and work efficiency. It is shown in (Fig.12) that some peaks appeared at (100, 150 and 200) Hz. The highest amplitude took place (0.076g) at frequency (47Hz) due to dry run which causes overheating and damages the bearings, shaft and impeller in axial and radial directions.



a. spectral domain
b. Time domain
Fig.12. Pedestal looseness mode without flow in y-direction

3.8. Pedestal looseness mode without flow in z-direction

It is shown in (Fig.13) that a small peak appeared at frequency (100Hz) and the highest amplitude took place at (0.077g) at frequency (47Hz).



a. spectral domain
b. Time domain
Fig. 13. Pedestal looseness mode without flow in z-direction

3.9. Pedestal looseness mode with flow rate of 25 liter/min. in z-direction

The system will run with full flow, this is the most dangerous case in the current study. There are some peaks appeared at (100Hz and 200Hz) and the highest amplitude is (0.104g) at frequency (47Hz) as shown in (Fig14). This is because the system runs with full load and all forces affect the pump without any fixation. This results in system instability. The highest amplitude will be at 1X.

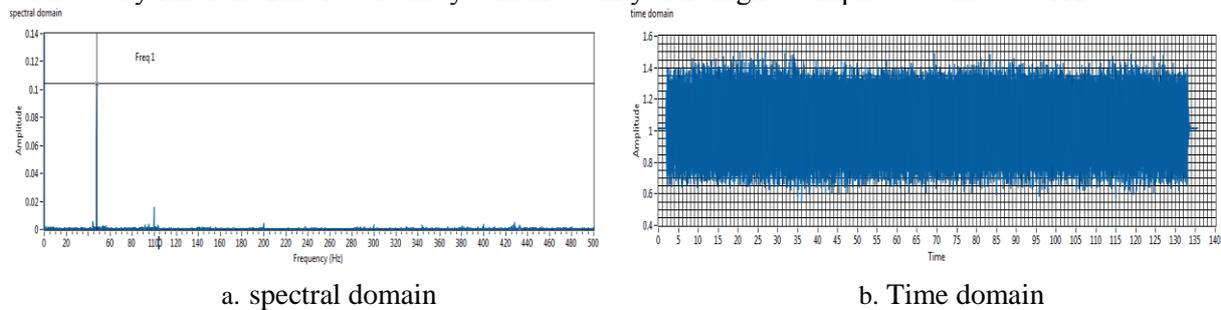


Fig. 14. Pedestal looseness mode with flow 25 liter/min. in z-direction

3.10. Pedestal looseness and unbalance mode with flow rate 25 liter/min. in x-direction

This case combines between unbalance and pedestal looseness as shown in (Fig.15). There are a lot of peaks appear at spectral domain graph. The highest amplitude is (0.06g) at frequency (50Hz) because all the system move due to the pedestal looseness. The pump vibrates due to fan defect so the frequency for highest amplitude is 1X.

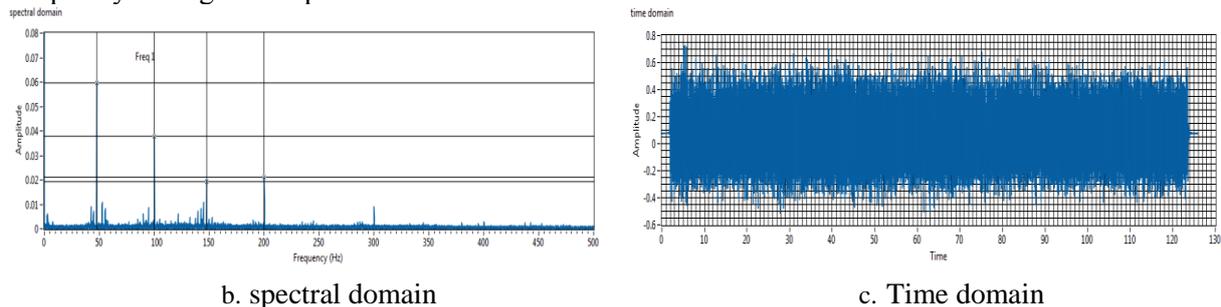


Fig. 15. Pedestal looseness & unbalance mode with flow 25 liter/min. in x-direction

3.11. Fault impeller mode with flow 25 liter/min. in x-direction

Finally the impeller is malfunctioned by breaking 7 blades as shown in (Fig.16) and the system runs with full load.

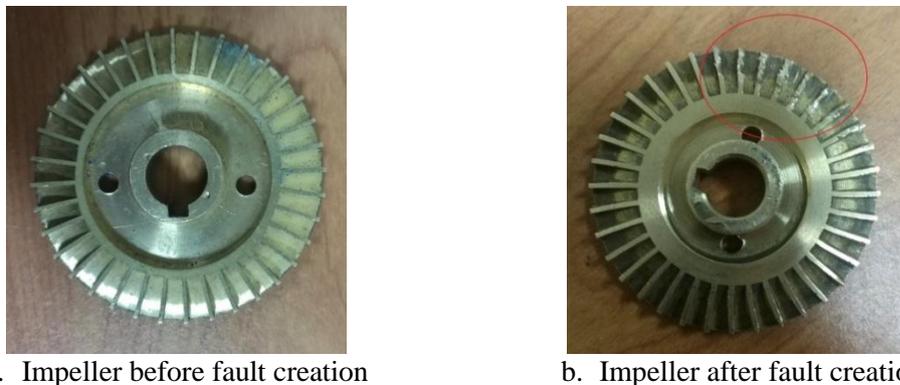


Fig. 16. Impeller before and after fault creation

From (Fig.17) and (Fig.18), it is shown that the system has subharmonic response and the amplitude massively increased to high values. Certainly the amplitude increase at 50 HZ, 100HZ... etc., but at

this case the major aim is to define the highest amplitude to set the alarm and the trip of the pump to preserve remnant parts of the system. The frequency that has major amplitude could be determined by blade pass frequency equation. The result compared with the frequency marked at (Fig.17).

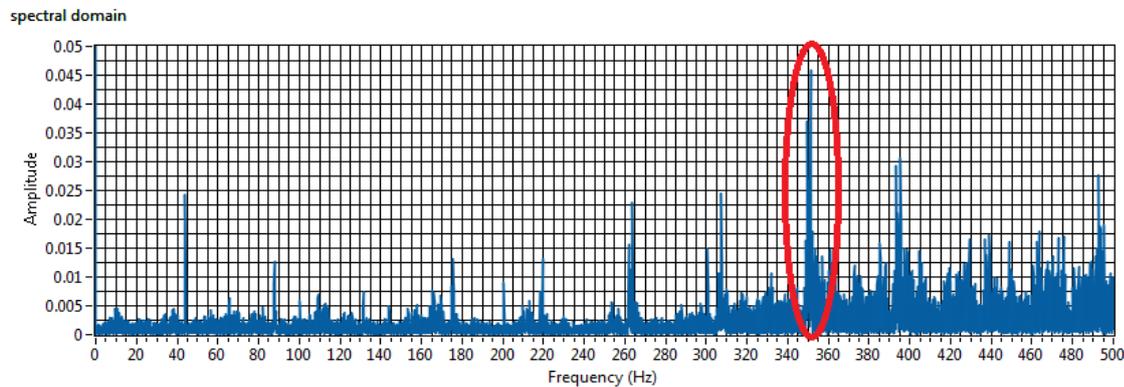


Fig. 17. Spectral domain for fault impeller mode with flow 25 liter/min. in x-direction

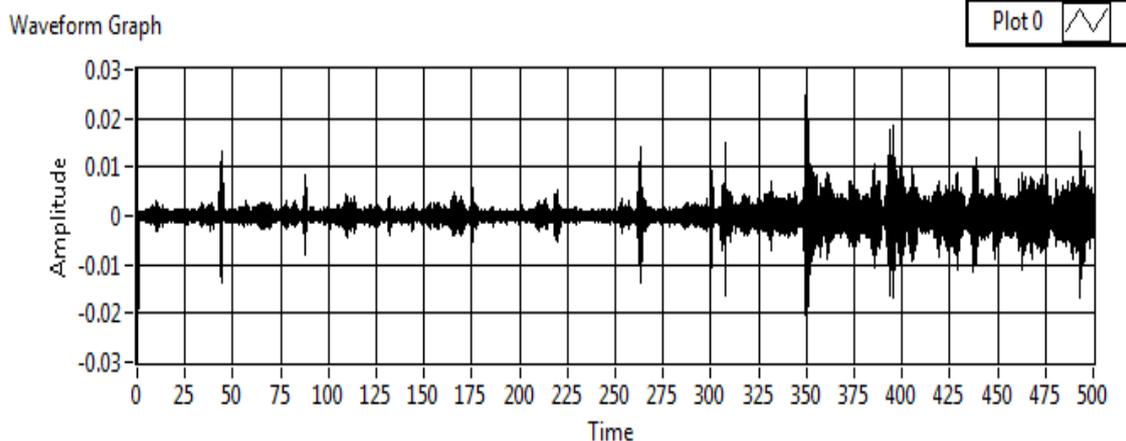


Fig. 18. Waveform graph for defected impeller mode with flow rate of 25 liter/min. in x-direction

Blade pass frequency^[14]:

$$\text{major frequency} = \frac{\text{no.of defective blades} \times \text{speed}}{60} \quad (1)$$

$$\frac{7 * 2850}{60} = 332.5 \text{ Hz}$$

By comparison between analytical and experimental results (332.5Hz and 350Hz) sequentially, it's possible to predict from the number of defected blades at which frequency the highest amplitude will take place.

4. Concluding result of measurements

All data can be collected from spectral domain by taking the amplitude at the referred frequencies in each direction separately.

It's possible to extract three circumstances from (Fig.19), (Fig.20) and (Fig.21). Firstly, studying each case separately to illustrate where the maximum amplitude takes place and at which frequency. Secondly, analyzing each frequency independently to illustrate where the maximum amplitude takes place and at which case. Finally, investigate the most dangerous case in absolute at each direction.

At x-direction:

Table 3. Amplitude values for each frequency at x-direction

		Amplitude values								
Frequency	50Hz	0.032	0.041	0.033	0.049	0.029	0.016	0.021	0.014	0.06
	100Hz	0.02	0.019	0.016	0.03	0.034	0.036	0.036	0.038	0.038
	150Hz	0.005	0.0025	0.003	0.008	0.001	0.002	0.004	0.0045	0.019
	200Hz	0.054	0.018	0.014	0.025	0.023	0.048	0.024	0.014	0.021

As shown in (Table.3) the highest amplitude colored in yellow and the most dangerous case in vertical direction which colored in red is unbalance and pedestal looseness with full flow case at frequency (50Hz) with amplitude (0.06g) because all the system move due to the pedestal looseness. The pump vibrates due to fan defect so the frequency for highest amplitude is 1X.

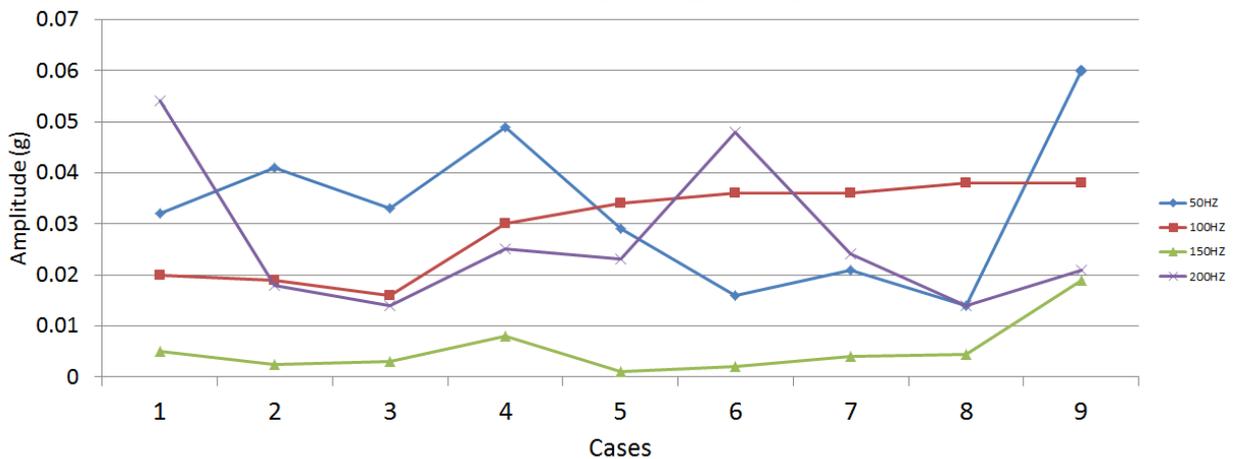


Fig. 19. Reading at each step with 50, 100, 150, 200 Hz at x-direction

At y-direction:

Table 4. Amplitude values for each frequency at y-direction

		Amplitude values								
Frequency	50Hz	0.011	0.021	0.022	0.026	0.015	0.026	0.076	0.039	0.018
	100Hz	0.0025	0.005	0.004	0.004	0.005	0.0025	0.016	0.006	0.013
	150Hz	0.009	0.0013	0.001	0.0025	0.0015	0.002	0.006	0.002	0.018
	200Hz	0.005	0.003	0.0029	0.003	0.003	0.004	0.005	0.002	0.012

All cases have highest amplitude at frequency (50Hz) compared with remnant frequencies. It's obvious from (Table.4) that the most dangerous case in axial direction which colored in red is pedestal looseness without flow case at frequency (50Hz) with amplitude (0.076g). When the system run in pedestal looseness case the pedestal stiffness and mechanical damping reduced and when run without flow the bearings, shaft and impeller damages due to overheating and. However, when the system runs with flow, the flow creates a balance force at axial direction that makes the system more stable.

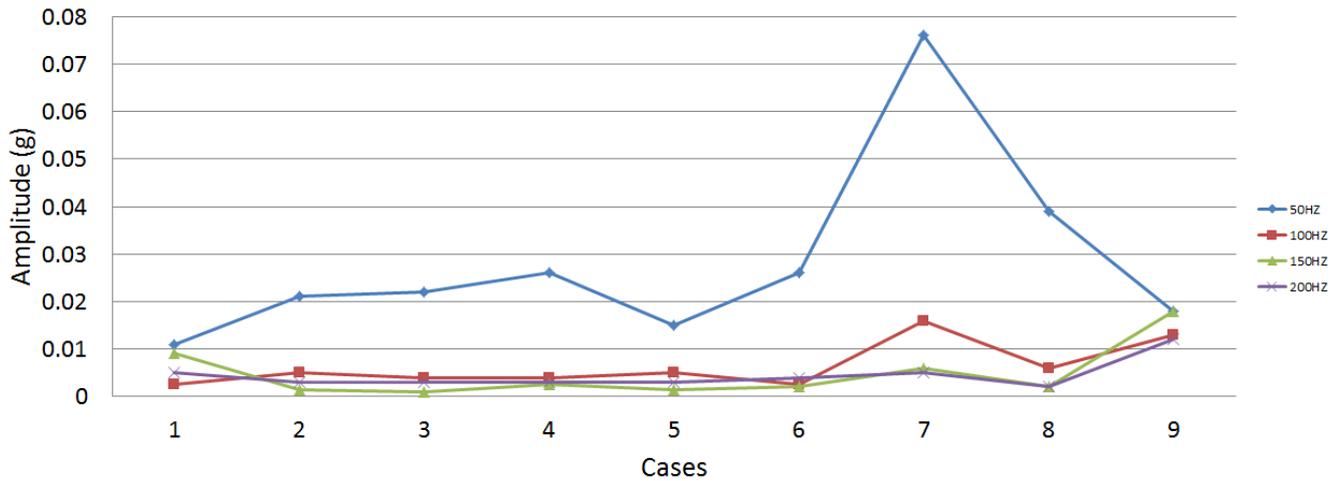


Fig. 20. Reading at each step with 50, 100, 150, 200 Hz at y-direction

At z-direction:

Table 5. Amplitude values for each frequency at z-direction

		Amplitude values								
Frequency	50Hz	0.028	0.049	0.056	0.056	0.067	0.034	0.077	0.104	0.033
	100Hz	0.011	0.007	0.007	0.004	0.016	0.006	0.01	0.018	0.017
	150Hz	0.009	0.001	0.001	0.001	0	0.001	0.004	0.002	0.0025
	200Hz	0.015	0.006	0.005	0.0035	0	0.002	0.003	0.004	0.005

All cases have highest amplitude at frequency (50Hz) compared with 100Hz, 150Hz and 200Hz frequencies. As shown in (Table.5) the maximum amplitude at each frequency colored in yellow and the most dangerous case in radial direction which colored in red is pedestal looseness with full flow case at frequency (50Hz) with amplitude (0.104g) because the system runs with full load and all forces affect the pump without any fixation. This results in system instability. The highest amplitude will be at 1X.

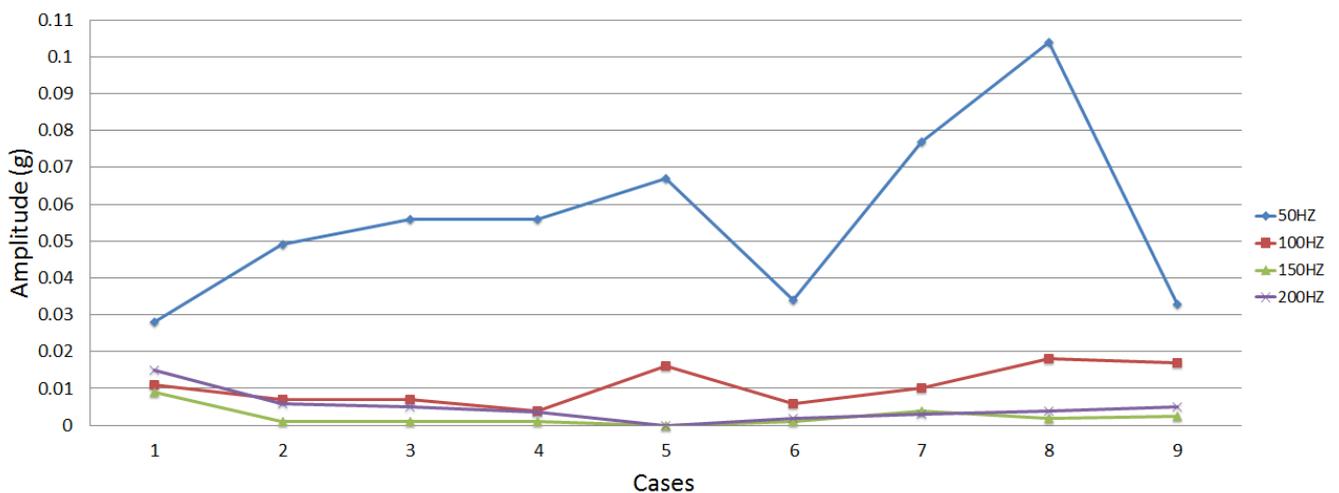


Fig. 21. Reading at each step with 50, 100, 150, 200 Hz at z-direction

5. Conclusion

The present work is ten cases of pump operation modes. Some defects were made to visualize the change of vibration response. The vibration analysis reveals that the unbalance and pedestal looseness is the most dangerous case in x-direction at frequency (50Hz) with amplitude (0.06g). The most dangerous case in y-direction is pedestal looseness without flow case at frequency (50Hz) with amplitude (0.076g). The most dangerous case in z-direction is pedestal looseness with full flow at frequency (50Hz) with amplitude (0.104g). When the impeller was broken, the frequency at which maximum amplitude occurs can be expected from the number of defected blades. The analytical result was validated experimentally.

Vibration monitoring can be used to early detect faults during pumps operation and, hence, save time, cost and extend the service life of the pumps.

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