Vibration Analysis of Rotating Shaft with Slant Crack on Surface.

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ABSTRACT

In this paper, study has done for surface slant cracks on solid shaft. In the current analysis, various methods have developed for crack detection of on rotating shaft using experimental analysis. FFT analyzer is used to record the results of vibration, which are used to analyze the vibration characteristics of rotating shaft. Finite Element Analysis of slant cracked shaft at various rpm is done in this paper. Healthy and slant cracked shaft is modeled and analyzed with numerical method. Vibration analysis is a Non- destructive method for the detection of crack, which is good than destructive methods.

Keywords : *Vibration analysis, Cracked shaft, FFT analyzer, Healthy shaft, Slant crack.*

I. INTRODUCTION

Shaft is a rotating element, due to rotations and loading conditions defect may produce in the shaft. As defect produces it directly affects on performance parameters of the shaft. Also, Efficiency of shaft may reduce. So, defects produced like cracks like slant, transverse, longitudinal and bends can be studied. In this thesis, slant crack is studied with cracks at different locations and with different speed conditions. Also, loading condition is applied for the crack study.

Dynamic analysis is a process of looking for anomalies and monitoring change from the established vibration signature of a system. The vibration of any object in motion is characterized by variations of amplitude, intensity, and frequency. These can correlate to physical phenomena, making it possible to use vibration data to gain insights into the health of equipment. Vibration analysis can be used to:

- Find a developing problem that can be repaired to increase machine lifetime
- Detect and monitor a chronic problem that cannot be repaired and will only get worse
- Establish acceptance testing criteria to ensure that installation/repairs are properly conducted.

II. LITERATURE SURVEY

Zhaohui Ren et. al.[1] have personated the study Crack fault diagnosis of rotor systems using wavelet transforms. As the traditional Fourier transformation method cannot process the non-linear and non-stationary vibration signal of the cracking rotor system, the 3-D waterfall spectrum in combination with reassigned wavelet scalogram method is presented to analyze the temporal frequency characteristics of the crack fault.

A.Bovsunovsky and C.Surace [2] have studded the non-linearities in the vibrations of elastic structures with a closing crack: The main purpose of the study is to illustrate the principal achievements of numerous researchers who have studied the non-linear effects caused by a closing crack in the most common types of structural elements such as beams, shafts and plates, the aim being to assess the potential and future prospects of using non-linear behavior to detect damage.

Qinkai Hann et. al.[3] has discussed on ear transmission systems have been widely used in various applications such as automobiles, and aircrafts, and are considered to be one of the most important mechanical components. As design of geared-based power train systems becomes more compact, the dynamics of geared rotor systems becomes a more critical factor to generated acoustic noise and structural durability. Thus, the vibration problems associated with geared rotor systems have been the focus of research for nearly three decades.

Qinkai Hann et. al. [4] has analyzed a geared rotor bearing system with slant breathing crack. In his study vibration problems associated with geared systems has been focused. With his research, torque is mainly transmitted by geared system and slant crack is more likely to appear on the gear shaft. Due to this slant crack the dynamic behavior of cracked geared system had different behavior than uncracked system.

R. Ramezanpour et. al [5] has investigated dynamic behavior of a Jeffcott rotor system with a slant crack under arbitrary crack orientations. Flexibility matrix and stiffness matrix of the system were calculated in this paper using fracture mechanics. In four directions system equations obtained, two transversal, one torsional and one longitudinal, and solved using numerical method. In this paper a symmetric relation for global stiffness matrix was presented.

III. SIMULATION

Modeling has been carried out in ANSYS software and it is meshed by using solid 186 element. 4084 number of elements and 4078 number of nodes are generated for slant cracked shaft. Similarly, 6705 number of elements and 6703 number of nodes are generated for healthy shaft.

Fig. 1 Meshed model of Cracked shaft

Explicit dynamic analysis has been carried out. Response of the assembly was obtained in 0.01 sec. Analysis was carried out in 500 number of sub steps. Response at each node was taken into consideration. Results validation is done at different values of RPM. After validation of results of first case, experimentation was carried out for remaining cases.

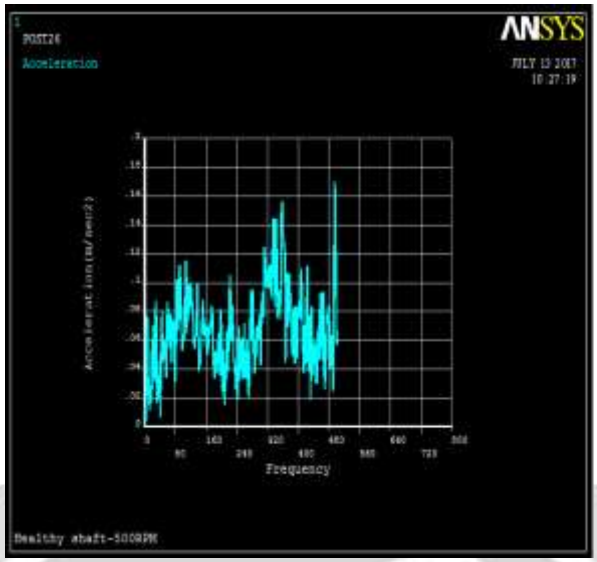

Fig. 2 Time response of healthy shaft at 500 rpm speed.

The top node at bearing surface was selected for getting accurate response. Fig. 2 shows the time signals of undamaged shaft which rotates at 500 rpm. During experimentation results were obtained by mounting probe of FFT analyzer at the same location.

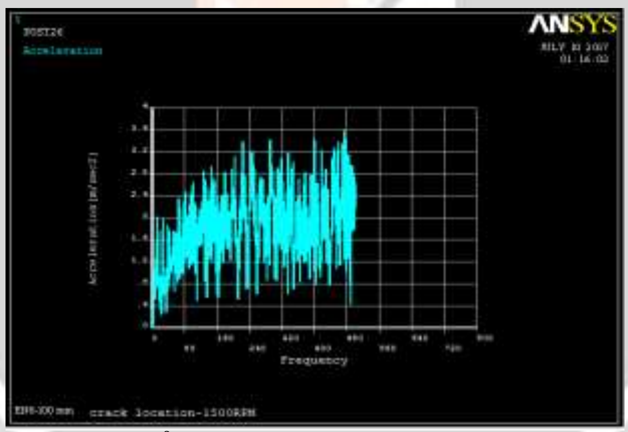

Fig. 3 Time response of 45^0 crack at 100 mm from bearing end at 1500 rpm speed.

Fig.3 is time domain result of explicit dynamic analysis of shaft having crack at 100 mm from bearing end and rotating at 1500 rpm.

Similarly the study is carried out for various speed and crack location. to validate the results experimentation is carried and finally the simulation and experimental results for various speed of shaft and crack angle with venation in crack location is out. These results have discussed in result and discussion part of this paper.

V. EXPERIMENTATION

Study of dynamic behavior of rotating shafts under the influence of defect as slant crack on surface of shafts has main focus of attention. The presence of a defect may lead to a dangerous effect on the rotating machinery. Therefore, the timely detection of defects would potentially avoid severe damage and expensive repairs due to the failure of rotating machinery. It also helps to avoid any human causality due to catastrophic failure. The experimentation is carried out on an intact shafts and defects developed shafts with experimental set up as shown in fig.1.

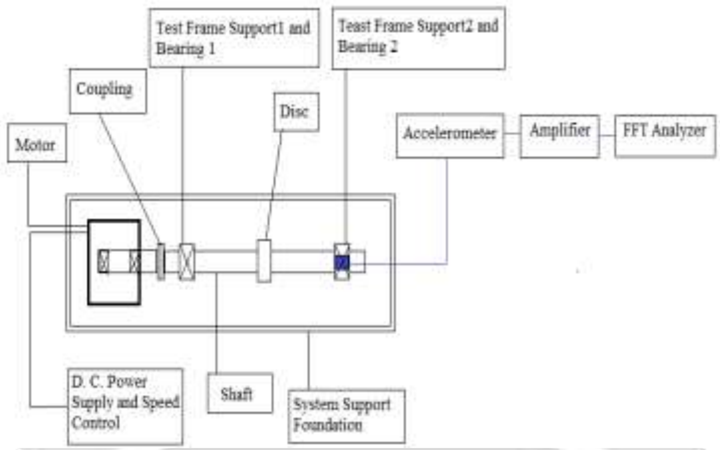

Fig. 4. Experimental arrangement

Pedestal bearing is used to support the shaft while its rotation. Number of balls: 8, Ball diameter: 0.00795 m, Radial clearance: $11.285*10^{-6}$ m, Bore diameter: 0.02 m, Outside diameter: 0.047 m, Pitch diameter: 0.03350 m.

Shaft: The Shaft is used to transmit the motion from motor to the test bearing. Materials: EN8, Diameter (mm): 21, Length (mm): 700 (Working Length) Crack orientation: 29^0, 45^0 55^0 with longitudinal axis of shaft Crack location : 100 mm and 300 mm.(from Bearing side)[7].

Experimental Procedure:For experimentation shafts are mounted on experimental setup for readings. For this paper work healthy shaft of EN8 material is taken. Also, of same material and slant crack location at 100 mm from bearing 1 is taken. With the help of 1 H. P. motor these healthy and slant cracked shafts are controlled over speed and readings are taken at 500 rpm, 1000 rpm, 1500 rpm and 2000 rpm with the help of FFT.

VI. RESULTS AND DISCUSSION

The experimentation has carried out with above experimental setup. The vibration spectrums of FFT analyzer for various cases are maintained as below.

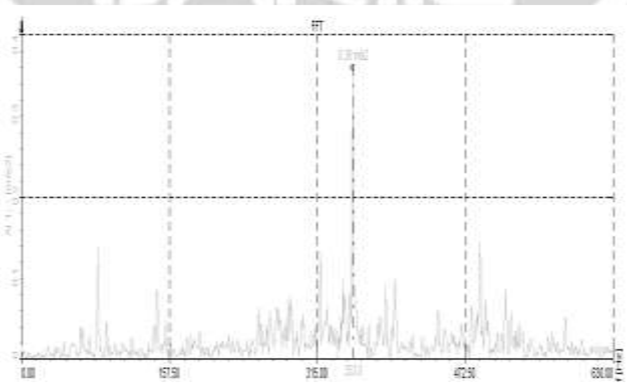

Fig. 5 Experimental frequency spectrum from FFT analyzer for healthy shaft at 500 rpm speed with 0.35 kg disc weight

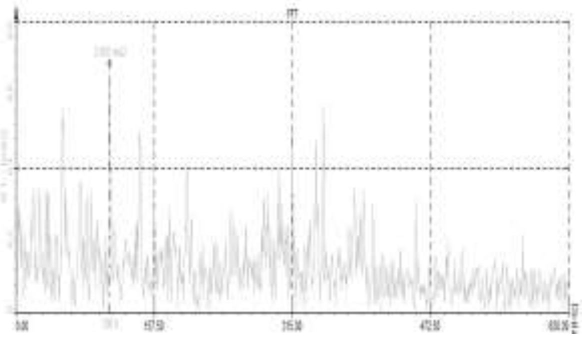

Fig. 6 Experimental frequency spectrum from FFT analyzer for healthy shaft at 1000 rpm speed with 0.35 kg disc weight

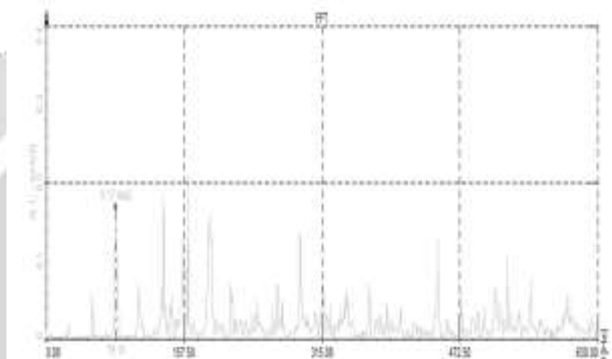

Fig. 7 Experimental frequency spectrum from FFT analyzer for 29^0 crack orientation at 100 mm crack location for 1500 rpm speed with 0.5 kg disc weight

The experimental results are obtained by using FFT analyzer. The experimental readings are taken for intact shafts and cracked shafts with speed variation. In addition to this, the frequency response is obtained for cracked shaft.
The various parameters selected for experimental work are:
Location of crack - 100 mm and 300 mm. from Bearing side for EN8 material.
Orientations of crack - 29^0, 45^0, 55^0 with longitudinal axis of shaft for EN8 material.
Shaft speed variations - 500 rpm, 1000 rpm, 1500 rpm and 2000 rpm.
Central disc weight - 0.25 kg, 0.35 kg and 0.5 kg.

Table 1 Amplitudes of vibration at different speed of undamaged shaft with 0.5 kg disc.

Speed (rpm)	**Experimentation Amplitude X1 (m/s^2)**	**Simulation Amplitude X2 (m/s^2)**	**Percentage Error $\frac{(X1-X2)}{X1}\times$ 100**
500	0.1431	0.1702	-18.93
1000	1.1505	1.3903	-20.84
1500	5.3005	4.9320	6.95
2000	1.1269	1.1901	-5.608

It is observed from table 1 and fig. 8 that there is 20.84 maximum percentage deviations in results of experimentation and results of simulation for healthy as shown in below table.

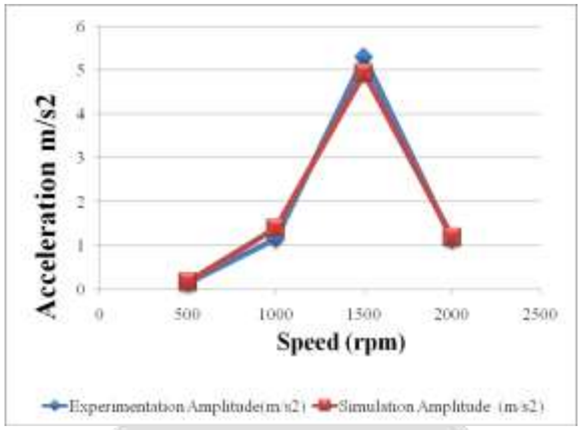

Fig. 8 Numerical and experimental acceleration values compression at different speed with 0.5 kg disc for healthy shaft

Table 2 Amplitudes of vibration at different speed of 45^0 crack orientation at 100 mm with 0.5 kg disc.

Speed (rpm)	**Experimentation Amplitude X1 (m/s^2)**	**Simulation Amplitude X2 (m/s^2)**	**Percentage Error $\frac{(X1-X2)}{X1} \times 100$**
500	0.2332	0.2205	5.44
1000	0.7610	1.0421	-36.93
1500	3.8907	3.4991	10.065
2000	1.2802	1.3223	-3.28

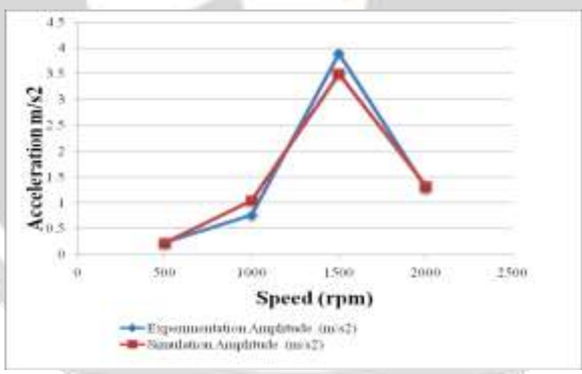

Fig. 9 Numerical and experimental acceleration values compression at different speed with 0.5 kg disc for 45^0 crack at 100mm.

Above figure shows results for varying speed as 500, 1000, 1500 rpm and 2000 rpm for 45^0 crack at 100 mm from bearing side. The result shows that the amplitude of vibration increases up to 1500 rpm and suddenly decreases for 2000 rpm. it may be due to resin, that as frequency increase above some value amplitude get reduces.

There is 33.39 maximum percentage deviations in results of experimentation and results of simulation for 60^0 Cracked shaft at location 100mm.

Table 3 Amplitudes of vibration at different speed for 600 crack at 100 mm with 0.5 kg disc.

Speed (rpm)	**Experimentation Amplitude X1 (m/s^2)**	**Simulation Amplitude X2 (m/s^2)**	**Percentage Error $\frac{(X1-X2)}{X1} \times 100$**
500	0.0551	0.03670	33.39
1000	0.1222	0.1040	14.89
1500	0.6451	0.5171	19.84
2000	0.9342	0.9432	-0.96

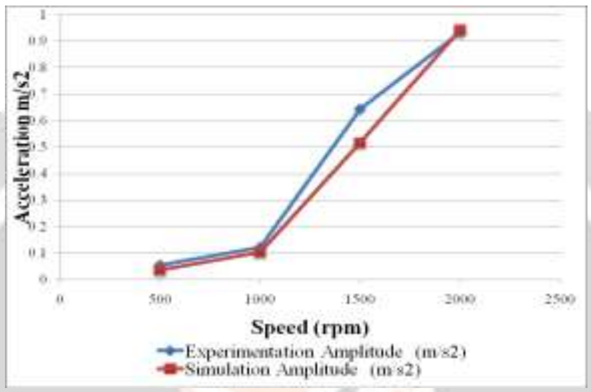

Fig. 10 Numerical & experimental acceleration values compression at different speed with 0.5 kg disc for 60^0 crack at 100mm.

The fig 10 shows the experimental and numerical results for 60^0 crack orientation at 100 mm location with the disc weight as 0.5 kg for speeds of 500, 1000, 1500 and 2000 rpm. From above table 3, result shows that amplitude for higher speed of 2000 rpm. As speed increases amplitude also increases. It is observed that there is 33.39 maximum percentage deviations in results of experimentation and results of simulation for 60^0 Cracked shaft at location 100mm.

1. Effect of crack orientation on vibration amplitude:

To study the effect of crack orientation on shaft vibration various angle on shaft have manufactured and for each angle the experimentation is carried out. This experimentation is done by crack location at 100 mm from bearing side. The result of this is summarized in below table and graphical representation of the same.

Graph can be plotted for presentation of crack orientation with amplitudes with shaft speed variations by taking loading disc weight 0.25 kg for each case as below:

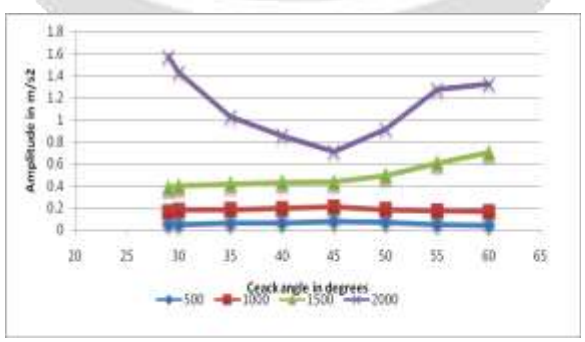

Fig. 11 Graphical representation of shaft speed variations angle orientations and amplitude values for 0.5 kg disc weight

From above fig.11, as speed of shaft increases, for every angle of the amplitude of vibration increases. Also, for speed (500 rpm, 1000 rpm) for all angle of orientations the amplitude remains constant. For 1500 rpm as angle of orientation increases the amplitude of vibration increases.

2. Effect of Disk weight on vibration amplitude

The effect of disc weight on dynamic condition of cracked shaft has checked and represented as follows

Graph has plotted for presentation of disc weight variations with amplitudes by taking 45^0 angle orientation for each case as below:

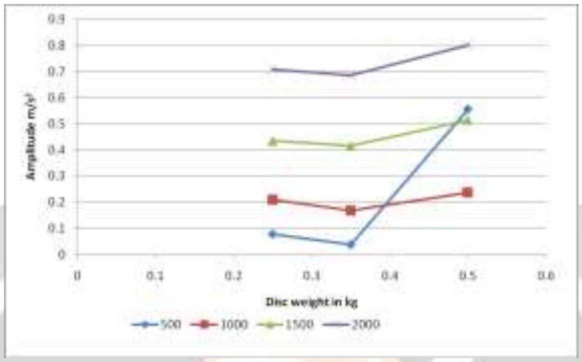

Fig. 12 Graphical representation of shaft speed variations for shaft with disc weight variation and amplitude values for 45^0 crack orientation

From above fig. 12, as the weight of disc increases from 0.25 kg to 0.35 kg the amplitude of vibration decreases for every speed (500 rpm to 2000 rpm) but as weight of disc increases from 0.35 kg to 0.5 kg the amplitude of vibration suddenly increases for every speed (500 rpm to 2000 rpm). In addition to this, as speed of rotating shaft increases the amplitude of vibration increases for every disc weight.

Table No.4 Results for varying disc weights from FFT analyzer with 29^0, 45^0 and 55^0 crack orientation on shaft at 100 mm crack location

Shaft Speed (rpm)	Crack orientation in degrees							
	29	30	35	40	45	50	55	60
500	0.0496	0.0504	0.0592	0.06212	0.07757	0.07125	0.04867	0.4131
1000	0.1634	0.1789	0.1851	0.1954	0.2082	0.1825	0.1722	0.1674
1500	0.3858	0.3974	0.4158	0.429	0.4348	0.4935	0.6071	0.7035
2000	1.5695	1.4287	1.0252	0.8542	0.7092	0.9124	1.2734	1.3241

V. CONCLUSION

Experimental work and Simulation is carried out for present work. Following conclusions are drawn:

It is observed that results obtained by finite element analysis are in good agreement with results of experimentation.

1 As speed of rotating shaft increases from 500 rpm to 1500 rpm, the amplitude of vibration increases for crack locations at 100 mm from bearing side. But after 1500 rpm, the sudden drop in amplitudes of vibrations is observed. This is due to non linear vibrations are induced at high speed because of whirling of shaft.

2 Crack produces more oscillation of the shaft. Hence, the amplitude of vibration increases. The above information can be used to predict failure of shaft in rotor system and preventive action can be taken.

3 For every angle of orientation of slant crack the amplitude of vibration increases as shaft speed increases.
4 As the weight of disc increases from 0.5 kg to 1.5 kg the amplitude vibration decreases for every speed (500 rpm to 2000 rpm) but as the weight of disc increases from 0.35 kg to 0.5 kg the amplitude of vibration suddenly increases for every shaft speed (500 rpm to 2000 rpm).
5 Due to disc weight attached at centre of shaft for loading condition, torque develops and suddenly amplitude reduces at higher speed. (i. e. at more than 1500 rpm). This is due to whirling of shaft.

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