Study of vibration characteristics of unbalanced overhanging rotor

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Abstract. The rotor unbalance and misalignment are two major sources of vibration. Rotor unbalance is omnipresent in all rotating machinery, posing serious threat to machine life and operation. The present work is an attempt to investigate the vibration characteristics of rotating mechanical system, which has unbalanced rotor mounted on overhanging shaft. Vibration signals are acquired using accelerometer mounted on the bearing housing nearer to the rotor. The FFT analysis of the acquired data revealed the steady state response of balanced and unbalanced rotor under operating conditions. Numerical analysis of the system using ANSYS portrayed the modal frequencies, mode shapes; harmonic analysis illustrates the response of system for different mass unbalance. The results revealed that magnitude of vibration characteristics significantly increases with excitation frequency and overhang length. Campbell diagram illustrates the absence of critical speed within the selected operating range.

1. Introduction
Vibration analysis is exercised to monitor the status of machinery under operating conditions. A major advantage is that it can identify developing problems before they become critical and cause failure or unscheduled shutdown of system. Rotating machines are extensively found in various engineering applications such as propulsion systems, turbo machines, machine tools, aircraft engines and in routine applications such as pump, fan, blowers etc. The design criteria of such machines are towards light weight and higher speed of operation. The major concern in rotating machineries is vibration. The reason for these vibration stem from system faults which, in general, exist as rotor unbalance, misalignment, cracks and rubs. Therefore, explicit projection of the dynamic characteristics is vital in design of the rotating mechanical systems [1].

Unbalance is induced in any system when the axis of inertia of rotor does not coincide with the geometric axis. The rotor unbalance generates a visible centrifugal force which operates at machine rotating frequency. This causes deflection of shaft and induces stress, affecting the working efficiency of the system [2]. Rotating machines possessing rotational energy gets transferred to machine elements in form of vibration. The mechanism of transfer is explicitly represented by unbalance in system [3]. Vibration characteristics of unbalance response provide necessary guidance for vibration

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control. The response of the system depends on speed, geometric proportions and mass distribution of rotor, and on dynamic stiffness of shaft and bearings. The unbalance response is the vibration of the rotor system, and the unbalance is reflected by the dynamic vibration characteristics of the system [4].

The symptom of malfunctions is indicated by machine vibrations and carries useful information about the character and cause of vibration. In order to identify the faults, model and signal based approaches are used. Model based methods are used to identify the magnitude and location of faults [5]. Experimental investigation is one of the most reliable methods of determining the dynamic characteristics of system, as the real features are portrayed as it is. Reliable modelling of systems with rotating elements is a tedious work due to inherent uncertainties as to boundary conditions and operating parameters. The data acquired from modal testing, on further processing indicates the mode shapes that describe the dynamic behaviour of system [6]. Experimental modal analysis technique is most extensively used tool for identification of modal parameters of system which illustrates the natural frequencies and mode shapes in the frequency range of interest [7].

Modes are inherent properties of structure which are employed as efficient tool to characterize the resonant vibration. Resonant vibration is caused due to interaction of inertial and elastic properties of material within the system and is often cause of many vibration related problems occurring in machinery. In order to study vibration response it is essential to identify and quantify the resonances of the system which in most of cases is done by studying the modal parameters of the system [8]. Vibration signature obtained from vibration analysis helps in monitoring the health of the mechanical system and also helps to alert the equipment operators about the possible failure of the system in advance. It could be also useful to prevent the system breakdown [9].

Existence of minute amount of unbalance in high speed machinery may prove to be fatal causing catastrophic failure of machinery much before the anticipated life. Overhung rotors find existence in many engineering applications like pump, fans, propellers and turbo machinery. The vibration signature of overhung rotor is different from that of centre hung type. Thus the accurate prediction of vibration characteristics of overhung rotor is vital for efficient operation and greater life.

Vibration due to unbalance causes damage to critical machine elements like bearings, coupling, seals etc. Perfect balanced rotor are highly impossible attributing to the presence of porosity in casting, nonlinearities, non-uniform distribution of density, manufacturing tolerances and wear of parts during operation. The centrifugal force generated as an effect of mass unbalance must be reacted against by bearing and support structures [10].

Machines with a nonrotating shaft behave much like familiar structures. However, once the rotor starts spinning, the modes are no longer planar. With radially symmetric bearings, the rotor center traces out a circle. The rotor whirls either in the same direction as rotation, or against rotation, resulting in both forward and backward whirl modes. The frequencies are affected by both the mass and diametral mass moment of inertia. The mass has the greatest effect at points of large circular motion (anti-nodes), while the mass moment of inertia has the greatest effect at points of large rocking motion (nodes). Changes in mass precisely at a node do not change the corresponding natural frequency, and changes in mass moment of inertia at points of no conical motion do not change the corresponding natural frequency. The modes affected by the mass moment of inertia are strongly affected by changes in speed. Considering the bearing characteristics do not change, the backward whirl mode will decrease in frequency with increasing shaft speed, while the forward mode frequency will increase. The extent to which this occurs is related to both the mode shape and the ratio of the polar mass moment of inertia to the diametral mass moment of inertia. Thus, a machine with a big disk/fan blade will probably show strong speed dependent effects in at least some modes [11].

Vibration characteristics measured at the bearings indicate the influence of unbalance on the system. A shaft can fail, if it is operating continuously at vibration level, which can induce stress levels beyond endurance stress. Hence it is essential to predict amplitude of vibration of shaft from the vibration signals sensed at bearings. In any rotating machinery the bearings are the structures which support the system and the forces from rotor are transmitted to the bearings where the data is acquired.
In view of the significance of influence of vibration characteristics on the performance of a mechanical system, the present work is a small attempt to analyse the effects of operating parameters of a rotating system on its vibration characteristics, experimentally. The study also includes numerical analysis using standard FEA tool [ANSYS] in order to obtain the modal frequencies and corresponding mode shapes including harmonic response.

2. Experimental details

Figure 1 shows the experimental setup used in present analysis. The shaft is connected to 0.37kW 3 phase induction motor through flexible coupling. The motor can be operated to a maximum speed of 1400rpm and the operating speed is controlled with help of Variable Frequency Drive (VFD). The free end of shaft is mounted with mild steel disc of diameter 145mm and 6mm thick, having provision to add the unbalance masses at radius of 60mm. Accelerometer used to measure the vibration characteristics is mounted on bearing housing near the free end of shaft. The accelerometer is connected to data acquisition system for acquiring and processing of vibration data. Table 1 shows the details of operating parameters in present study.

The vibration data is acquired for both balanced and unbalanced conditions of shafts at different speeds. The signals acquired in time domain is processed and converted into frequency domain signals using Fast Fourier Transform (FFT) technique. Acceleration and displacement of vibration, in frequency domain are used for analysis of the system response.

2.1 Numerical analysis

Numerical analysis is carried out by using FEA tool-ANSYS 16.2 to determine the modal frequencies, mode shapes and steady state response of system under different unbalanced conditions. Further to determine the critical speeds of system Campbell diagram is constructed. Figure 2 shows the 3-D model created, with bearings specified in FEA tool, for the numerical analysis of the system. Automatic mesh method was used to mesh the structural model of system. The material specified is structural steel with properties of elastic modulus E=210GPa, Poisson ratio=0.3, Density=7850kg/m³. The bearing stiffness was set to constant value of 1000N/mm, because the bearing load and influence of rotation speed on bearing stiffness were not considered. Figure 3 shows the rotating force applied to the system by specifying the unbalance mass with eccentricity.
3. Results and discussion

3.1 Vibration characteristics of system

Some of the typical results obtained from experimental investigation are shown in figure 4 and figure 5. It can be observed from the FFT plots that, distinct peak is not visible at excitation frequency in case of balanced system. But in case of both unbalanced conditions dominant peaks are visible at machine excitation frequency. The magnitude of vibration characteristics increase with increase in unbalance mass, the reason for which can be attributed to increase in centrifugal force developed during operation.

It can be observed that for identical unbalanced masses the magnitudes of vibration characteristics (acceleration and displacement) are higher in case of 250mm overhang when compared with 150mm overhang. This is due to increase in moment induced because of unbalance force for higher overhang, thus inducing higher reaction force at bearing nearer to the disc. Therefore it is clear from experimentation, to curb the unduly vibration induced in mechanical system having overhung shaft, the measure of overhang length is vital criteria and is to be as minimal as possible.
Figure 4. Vibration Characteristics [150mm overhang length, 550 rpm-9.17 Hz], (a) acceleration balanced; (b) displacement balanced; (c) acceleration-25g unbalanced mass; (d) displacement-25g unbalanced mass; (e) acceleration-50g unbalanced mass; (f) displacement-50g unbalanced mass.
Some minor peaks are visible in FFT plots at higher frequencies; this is attributed to components of higher harmonics of excitation force. Since the magnitude of these peaks at higher frequencies is small when compared to those at actual excitation frequency, these are considered to be insignificant.
3.2 Variation of acceleration with excitation frequency

Variation of acceleration with excitation frequency is shown in figure 6 and figure 7. It can be seen from plots that the magnitude of response increases with increase in excitation frequency. Also the magnitude of acceleration increases with increase in unbalance masses.

![Figure 6. Acceleration vs Excitation frequency for 150mm overhang.](image1)

![Figure 7. Acceleration vs Excitation frequency for 250mm overhang.](image2)

Table 2 shows the modal frequencies extracted from the numerical modal analysis of the two shafts. The first modal frequencies of the two systems are at around 52 Hz and 30 Hz. Since the stiffness of the shaft is reduced due to increased overhang length there is reduction in natural frequency of shaft at 250mm overhang compared to 150mm.

| Mode | Modal Frequency [Hz] |
|------|----------------------|
| 150 mm overhang | 250 mm overhang |
| 1. | 52.696 | 30.155 |
| 2. | 75.698 | 35.277 |
| 3. | 144.98 | 120.1 |
| 4. | 317.4 | 256.49 |
| 5. | 430.54 | 285.97 |
| 6. | 431.17 | 434.67 |

It can be observed from the experimental results [figures 4&5], the presence of distinct peaks at around 50 Hz and 30 Hz in the systems, indicating the excitation of first natural modes by the harmonics of excitation force. The trends observed in figure 6 and figure 7 illustrate that the experimental vibration characteristics are tending towards their peak value as the excitation frequency gets closer to first modal frequency. The trends in figure 6 and figure 7 would have reached their peaks and identified first modal frequency experimentally, but for the limitations of the operating speed of the drive motor (24 Hz).

3.3 Mode shapes and harmonic response

Some typical mode shapes obtained from modal analysis are shown in figure 8 and figure 9. The mode shapes of shafts display some dominant shaft bend modes and disk modes.
The typical harmonic responses of the system [150mm overhang] under two unbalance conditions are shown in figure 10 and figure 11. The response plots for two unbalanced conditions illustrate the amplitude of displacement tending towards peak value around their natural frequencies. It can be observed that the response tending towards peak value at 80 Hz and 320 Hz which are the first and second bending modes in vertical direction. Deformed shape of shaft at 80 Hz and 320 Hz are shown in figure 12 and figure 13.
3.4 Identification of critical speed using Campbell diagram

Campbell diagrams are constructed for the two systems investigated as shown in figure 14 and figure 15. It can be seen that critical speeds (i.e. intersection of excitation frequency line with modal frequency line) are not observed in operating range of interest. However the higher harmonics of excitation frequency (4X, 5X…etc.) intersect the first and second modal frequency lines. The magnitude of vibration characteristics at higher harmonics are low and hence can be neglected.

4. Conclusions

Following conclusions can be drawn from the experimental and numerical analysis carried out to investigate the effect of operating parameters on vibration characteristics of a rotating system.

- The amplitude of vibration is proportional to the excitation force and the magnitude of vibration characteristics increase with increase in excitation frequency, unbalance masses and overhang lengths.
- The FFT plots indicate the presence of first modal frequency, at which the harmonics of vibration characteristics tend towards their peak value.
The results obtained from numerical analysis illustrate the modal frequencies, corresponding mode shapes and frequency response of system indicates the frequencies corresponding to high deformation, illustrating the regions subjected to large deformation.

Campbell diagram indicated the absence of critical speeds within the operating range considered.

References

[1] Ritto T G and Rubens S 2005 The effects of unbalance and clearance on the bearings of an overhung rotor Proc. of COBEM 18th Int. Congress of Mechanical Engineering (Ouro Preto, Brazil).
[2] Larsen J S and Santos I F 2015 On the nonlinear steady-state response of rigid rotors supported by air foil bearings-theory and experiments J. Sound Vib. 346 284-297.
[3] Muszynska A 1995 Vibrational diagnostics of rotating machinery malfunctions Int. J. Rotating Mach. 1 237-266.
[4] Didier J, Sinou J J and Faverjon B 2012 Study of the non-linear dynamic response of a rotor system with faults and uncertainties J. Sound Vib. 331 671-703.
[5] Markert R, Platz R and Seidler M 2001 Model based fault identification in rotor systems by least squares fitting Int. J. Rot. Mach. 7 311–321.
[6] Bucher I and Ewins D J 2001 Modal analysis and testing of rotating structures Philosophical Transactions of Royal Society of London A 359 61-96.
[7] Irretier H D 2002 History and development of frequency domain methods in experimental modal analysis J. Phys 12 91-100.
[8] Hisham A H, Al-Khazali and Md Askari R 2012 The experimental analysis of vibration monitoring in rotor dynamic system with validation of results using simulation data International Scholarly Research Notices (Article ID 981010) 1-18.
[9] Algule S R and Hujare D P 2015 Experimental study of unbalance in shaft rotor system using vibration signature analysis Int. J. Emerging Engineering Research and Tech. 3 124-130.
[10] Md. Abdul Saleem, Diwakar G and Satyanarayana M R S 2012 Detection of unbalance in rotating machines using shaft deflection measurement during its operation IOSR J. of Mech. and Civil Engineering (IOSR-JMCE) 3 8-20.
[11] Swanson E, Powell C and Weissman S 2005 A practical review of rotating machinery critical speeds and modes J. Sound Vib 39 10–17.