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Vibration Characteristics in Ball Bearing due to Unbalanced Mass

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Abstract

Bearings are the critical element in an equipment, where rotary motions are mandate to full fill its operation. Condition at which the bearing fails to contribute its purpose, leads to system breakdown and hence a major setback in successful operation of the equipment. Monitoring and maintaining the continuous and uninterrupted operational condition of the bearing in a rotary system becomes primary task. Diagnosing and preventing the unexpected failure of bearing is a prime goal of the industrial personnel, for which they must have the information of the present working condition of the bearing. There are several ways to gauge the present working condition of the bearing, like knowing the vibrations, temperature and noise at the bearing, during its operation. In context with this, an experimental study is designed to simulate the actual shop floor condition to obtain the bearing vibration characteristics, influenced by the predefined parameters such as speed, mass and angular orientation of the mass with respect to shaft axis. The experimental study could be able to yield the bearing characteristic, operated under the combination of these parameters. The study supports the decision-making process in bearing failure analysis through vibration characteristics.

Major Findings: *The investigation is engrossed on finding the vibrational behavior of the ball bearing, subjected to combinations of speed, counter mass and angular position of the counter mass. The study enlightens on the supremacy of axial and tangential accelerations due to counter mass and its angular location from the shaft axis.*

Keywords: Bearing, Decision Making, Diagnose, Vibrations

1.0 Introduction

Bearings are crucial elements in the rotating machinery. They drive, control and monitor the functions of the rotating system. Awareness of attributes relating the behavioural changes in the working principle of bearing, at a given time of measurement provides an ample opportunity to save the system from catastrophic failure. The best method of monitoring is to predict and attend the brewing problems of failure while it is being in use[1]. The elementary level of understanding on bearing behaviour used to support a shaft with disc or pulley increases the confidence in predicting the failure modes of bearing. Daniel *et al*., demonstrated the amplitude of the vibration in the bearings using test rig. The insight of the work estimated the maximum amplitude in tapered roller bearing compared to ball and thrust bearing[2]. Khadersab and Shivakumar[3] studied the vibration behaviours of

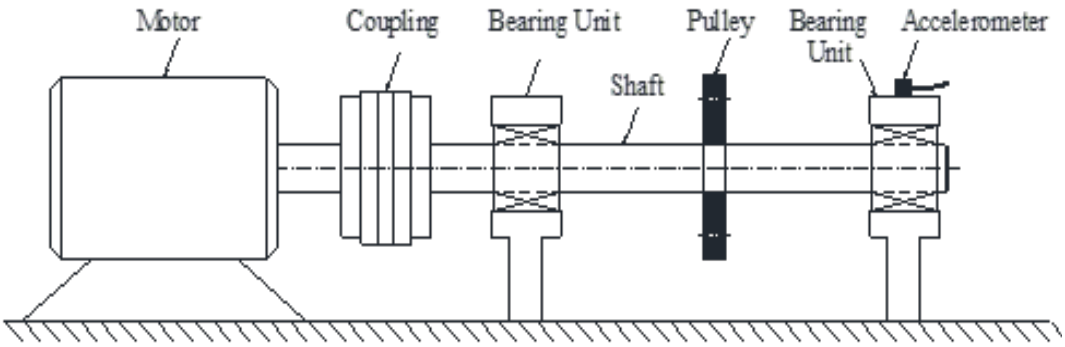
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faulty and healthy bearings. The findings generated a methodology for assessing the faulty bearing in rotating machinery[3]. Brusa *et al.* developed modular bearing test rig to test different bearings by changing the adapters. The system was tested, and the behaviour was found to be consistent with the real bearing test rig[4]. Saruhan *et al.*, investigated the vibrational behaviour of rolling element bearings subjected to static loads and running speeds. The bearings were tested using a test rig to understand the magnitude of the vibration amplitude in normal and different defect conditions such as defects in ball bearing, inner race and outer race[5]. Daniel *et al.*, generated a comparative study between three types of bearings on the vibrational characteristics. The experimental work revealed that the vibrations in ball bearing are lesser than that of tapered roller bearing[6]. Nistane and Harsha presented the lifetime bearing failure statistics for ball bearings. The exertion provides a platform of vibrational behaviour which represents the failure evaluation and location of the defect in the ball bearing[7]. Pondkule *et al.*, examined the vibrations patterns in the ball bearing of two-wheeler front wheel axel under constant speed and different loading conditions. The irregularities in the vibration spectrum revealed location of the defect in the bearing[8]. Ahobal and Ajit Prasad attempted to probe the vibration characteristics in the unbalanced rotating system. The investigation revealed the vibration increases with increase in unbalanced mass and overhang length. They also presented the absence of critical speed within the selected range of speeds using Campbell diagram[9]. Mya Thet Hnin and Thein Min Htike, estimated the unbalance in rotating rotor system using frequency response function of the rotor and deflection shape. The proposed estimated technique is able to achieve 0.084% variation from the prescribed unbalance value[10]. Prashant H Jain and Santosh P Bhosle, analysed the effect of disc eccentricity, rotor speed, radial clearance and radial load on the vibration responses for a ball bearing system. The simulation analysis exhibits the least influence of radial load on vibration response[11]. Minhas *et al.* indulged into reviewing techniques and methods to evaluate the rolling element bearing vibrational behaviour in good and fault detection. They have proposed finite element analysis is one of the reliable tools for bearing vibrational analysis[12]. Yang *et al.* conducted the experimental validation on the influence of resonance characteristics on the rigid bearing rotor system. They evaluated the favourable and unfavourable criteria of the resonance and rotor eccentric excitation on the quality of the bearing fault vibration signal[13]. Hariharan and Srinivasan investigate the vibration spectrum in the shaft misalignment. They derived that the misalignment effect can be amplified, and the speed is pronounced in the frequency spectrum[14] Xu *et al.*, investigated the vibrations of bearing under the dynamic gear meshing force. They have evaluated the bearing vibration characteristics through a modulation signal bi-spectrum sideband estimator indicator which is better than the ball pass frequency on outer race[15]. Bearing is meant to support the rotating element during power transmission and sustain the various loads exerting on it. As the bearing subjected to various loads during its usage, it yields vibratory motion causing wear and tear in its functional elements leading to its inability to perform at its best. The vibrations in the bearings are generally influenced by several factors such as speed, alignment of the bearing elements and loads. Consider one such situation in ball bearing, wherein we observe the variation in loading pattern, causing unbalance in the rotating system. An attempt was made to realise the unbalance through experimental setup and explore the vibrational behaviour of the bearing.

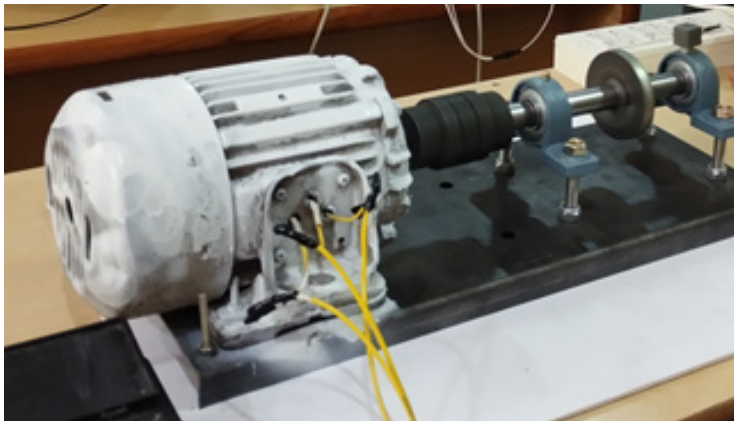
2.0 Materials and Methods

Power transmission in construction machinery, material handling systems and special purpose machineries is majorly supported by ball bearings, especially UCP 205 bearing units. UCP 205 are pillow block ball bearing units, mounted on support surfaces using cast iron housing. This bearing unit is designed for a constant and alternate rotation. The application of this kind of bearing needs an attention in monitoring the good functional behaviour, especially with unbalanced masses. To address the challenges in monitoring the good operating conditions, an experimental methodology was developed to investigate the vibrational behaviour of UCP 205 bearing, bound by unbalanced masses, acting at a distance from the axis of the shaft.

The experimental work was carried out by using a test rig as shown in the Figure 1 (a) and (b). The test rig consists of a shaft of diameter 25 mm and length of 800 mm. The shaft is connected to the motor through a flexible



(a) Schematic diagram of test rig.



(**b**) Actual Testing Setup.

Figure 1. Experimental set of test rig.

coupling to minimise the effect of the high frequency vibration generated by the motor. The motor is connected to variable speed control unit for selecting the variable speeds. The maximum speed of the shaft is 3200 rpm.

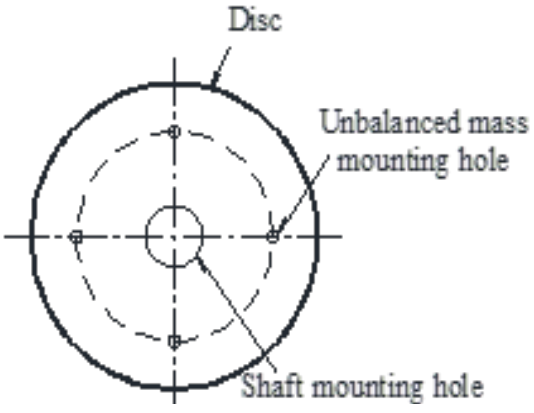


Figure 2. Details of the disc.

Two UCP 205 normal condition bearings are mounted on either side of the shaft. A static mass is applied to the shaft through a hot rolled steel disc mounted at the mid span of the shaft. The self-weight of the disc is 540 grams. The detail of the disc is as shown the Figure 2. Provisions were made in the disc to attach mass and develop the unbalanced mass conditions on shaft.

The position and amount of mass in grams, added in particular orientation is as illustrated in the Figure 3. Four mass configurations and three shaft speeds conditions were considered to evaluate the unbalanced masses and vibrational behaviour in the bearing. The vibrations in the right end bearing were acquired by using tri-axial accelerometer. Nine experimental trials were conducted with different combinations of speed and unbalanced masses. The acceleration for each trail was recorded and tabulated as shown in the Table 2.

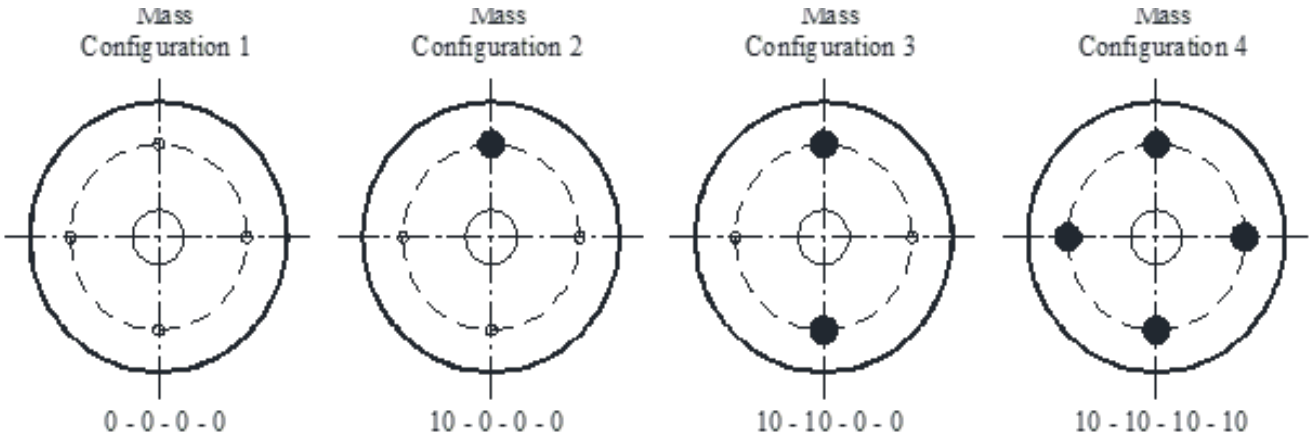


Figure 3. Unbalanced mass configuration.

Table 1. Testing trials data

Experimental Trials	Conditions	Speed (rpm)	Mass configuration on the shaft (grams)	Accelerations (g)		
				X	Y	Z
1	Increase in speed	1160	540 - 0 - 0 - 0 - 0	1.558	0.835	0.522
2		1300	540 - 0 - 0 - 0 - 0	1.664	1.039	0.605
3		1420	540 - 0 - 0 - 0 - 0	4.227	1.392	0.969
4	Increasing mass positions	1300	540 - 0 - 0 - 0 - 0	1.664	1.039	0.605
5		1300	540 - 10 - 0 - 0 - 0	1.602	0.772	0.608
6		1300	540 - 10 - 10 - 0 - 0	2.393	1.113	0.669
7	Increasing mass	1300	540 - 0 - 0 - 0 - 0	1.664	1.039	0.605
8		1300	540 - 10 - 10 - 0 - 0	2.393	1.113	0.669
9		1300	540 - 10 -1 0 - 10 -10	1.911	0.834	0.633

3.0 Results and Discussions

The vibration characteristics of UCP 205 bearing unit was interpreted under the circumstances of different speed and mass configurations. The vibration of right end bearing unit was acquired using a tri-axial accelerometer. The mounting and representation of measurement of vibration in all three directions is as illustrated in the Figure 4. The vibrational behaviour of the bearing was discussed with respect to three different shaft speeds and four unbalanced mass configurations.

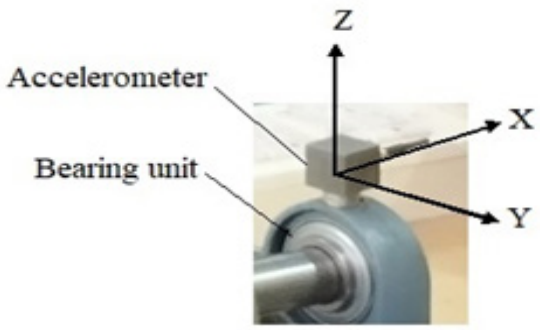


Figure 4. Accelerometer location and direction of each axis of measurements.

3.1 Vibration and Speed Interaction

In Figure 5, the acceleration of the bearing unit depicts increasing trend with respect to increase in speed. The accelerations in Y and Z directions are less compared to X direction. The acceleration in X axis exhibits two modes, less and more variation between the speeds 1160rpm to 1300rpm and 1300rpm to 1420rpm respectively. The increased unbalanced force at higher speeds could be the reason for this variation.

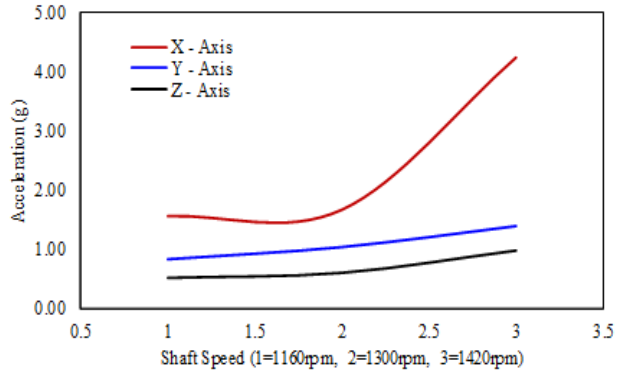


Figure 5. Accelerations with increase in the speed of the shaft.

3.2 Vibration and Mass Interaction – Case 1 and Case 2

The unbalanced mass on the shaft has been introduced through the disc. The mass of 10 grams as load are added on the shaft with different configurations as shown in the Figure 3. The addition of these mass configurations has been illustrated in two cases, case 1 and case 2. The unbalanced masses were mounted at 90° and 180° apart in case 1 and at 90° apart in case 2.

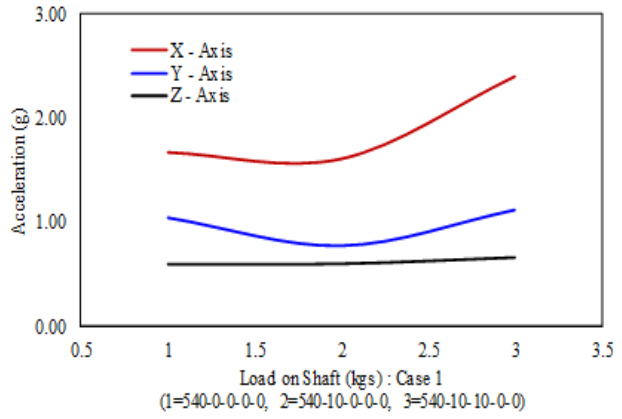


Figure 6. Accelerations with different mass positions on the shaft – Case 1.

The effect of these mass configurations on the vibration behaviour of the bearing is presented in the Figure 6 and Figure 7 respectively. In case 1, the disturbing mass acting at 90° and 180° on the shaft through the disc along with the self-weight of the disc, as depicted in the Figure 6. This position of the mass creates unbalance force acting on the shaft which is rotating at a speed of 1300rpm.

The vibration found to be minimum with disc only condition and increases as the mass position moves from 90° to 90° and 180°. A considerable variation in the accelerations were observed in the X and Y directions compared to Z direction. The Z axis defines the radial direction, where the impact applied mass will be negligible.

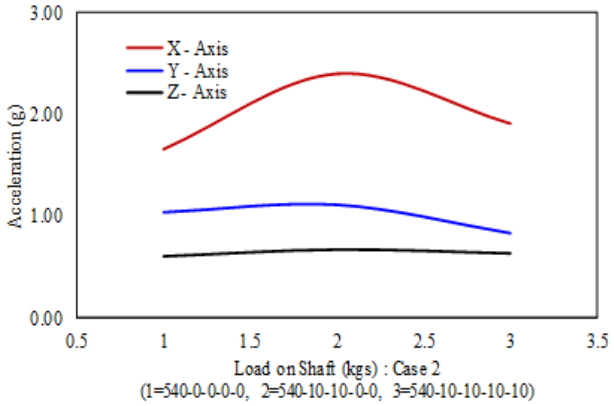


Figure 7. Accelerations with different mass positions on the shaft – Case 2.

Increase in vibration due to unbalanced mass is evident in the case 2. The mass configuration illustrated in the Figure 3 shows mass increasing from zero to 20 grams where two 10 grams masses are placed at 180° apart. This increase in mass induces unbalanced forces on the shaft and produce more vibration as detailed in the Figure 7. The variation in the vibration as indicated in the Figure 7 also conveys that the vibration decreases as it attains balanced mass distribution, where four 10 grams masses are placed at equal orientation of 90° apart.

The discussion encapsulates the ascendancy of unbalanced mass on vibration in the shaft. The

experimental evidence reveals that the unbalanced mass contributes to a considerable rise in vibration in axial (X) and tangential (Y) directions in the bearing unit, whereas it has minimum effect in the radial (Z) direction.

4.0 Conclusions

The experimental investigation was performed to bring out the vibrational behaviour of ball bearing subjected to unbalanced forces. The unbalanced force was induced in the bearing by the masses attached in the disc at predefined orientation from the axis of the shaft. The shaft was allowed to rotate at three different speeds with four mass configurations. The acceleration of the bearing was acquired using tri axial accelerometer. The axial (X), tangential (Y) and radial (Z) accelerations of the bearing justify the impact of unbalanced masses and provide insight of the vibration pattern. The vibration in the radial (Z) direction was found to be minimum, signifying the least effect of unbalanced mass. The supremacy of axial (X) vibration over the tangential (Y), divulge the impact of unbalanced masses in line with the axis of the shaft. The variation in the magnitude of axial and tangential vibrations indicates the authority of the unbalanced masses in these two respective directions. The investigation can be extended further for different mass configurations and multiple planes.

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