Application of Matrix Method of Dimensional Analysis (MMDA) and Experimental Approach for investigation of Unbalance in Rotor Bearing

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Abstract

Rotor deformation, material in homogeneity, and manufacturing tolerance lead to unbalance inside the rotor-bearing system. Industrial maintenance follows condition monitoring to diagnose the faults present. This work employs the matrix method of dimensional analysis to identify the unbalance in the rotor-bearing system. Experimentation performed under the various operating conditions on the test rig reveals that the vibration amplitude increases as unbalanced mass increases; defect frequency corresponds to the shaft frequency of the rotating shaft, which conforms to the theoretical results. The method reduced the excessive number of variables considered due to simplicity, ultimately reducing the time and effort required. Also, the detection potential of the method for unbalance endorses application to industrial machines.

Keywords: rotor-bearing system; dimensional analysis; unbalance fault

1. INTRODUCTION

The rotor-bearing system is vital in modern industry and is predominantly used in machines like gensets, aero engines, fabrizer, and gas turbines. A series of failures will happen in bearing with localized or distributed faults such as unbalance, misalignment, crack, inner race, outer race fault, and ball pits. This ultimately results in increasing the ideal time of machinery and loss of production. In rotating machinery, unbalance creates excessive vibration and causes a catastrophic or sudden system failure. Unbalance will produce inside the system due to deformation of the shaft, in homogeneity in material, and manufacturing tolerance. Various techniques have been implemented for the diagnosis of the fault, such as condition monitoring using vibration analysis, acoustic analysis, and lubrication analysis.

Many products and processes rely on rotating systems, from machine tools to autos to rockets to ships to submarines to power plants to gadgets in the home to medical equipment. The dynamic systems need accurate and reliable predictions of the dynamic properties of their important parts as well as the detection of the corresponding fault parameters for proper working. Identifying these systems' multiple fault parameters (MFPs) is a crucial goal for researchers in this field.

Liu et al.[1] studied the rotating system with an outer ring defect is modeled using the energy technique. A unique strategy for rotor mass distribution was used to increase the model's accuracy with the experimental data. The dynamic equations could be solved using the Runge-Kutta method and the simulated vibration signal generated. Mufazzal et al.[2] examined vibration response of ball bearings using a modified two-DOF lumped parameter model. An extra deflection and multi-impact theory are used to simulate how healthy and faulty bearings behave under varying loads and speeds. The bearing response characteristics were studied using numerical simulations run at various speeds, loads, defect sizes, and locations. Suryawanshi et al. [3] analyzed rolling contact bearing under the influence of inclined surface fault. The Buckingham Pi theorem of DA is used to cultivate a mathematical model by considering the rotor speed, angle of the incline, surface fault, load, and other bearing parameters. Experiments are conducted to determine the impact of the slope and size inclined surface fault on the vibration characteristics of spherical roller bearings. Han et al. [4] use the rotor-bearing system unbalance parameters identified using an evolutionary algorithm and a kriging surrogate model. Yaxi Shen [5] implemented a linear elastic plate, actuators, and piezoelectric sensors to model a dynamic structural system. Simulation confirms that steady-state resonance vibrations are suppressed. Genetic Algorithm-designed PID and Hybrid Fuzzy-PID control aim to minimize performance output error. Wan [6] introduced Soft Competitive Learning as a novel approach to classification. The proposed diagnosis is implemented for faulty bearings using SFART, or Fuzzy Adaptive Resonance Theory. The closeness of neurons was measured using Yu's lateral inhibition theory. Sanches et al. [7] uses a finite element approach for modeling the dynamic system, and the faults were found utilizing time-domain rotor responses and correlation analysis. A simplified system model's Lyapunov matrix equation uses least-squares fitting to identify fault parameters. The damping of rotor and coupling is determined using a differential evaluation optimization. Sugumaran et al. [8], Using the retrieved features, created a rule set for a fuzzy classifier. To identify bearing fault scenarios in train data, a decision tree is used to select the best few histogram characteristics. A fuzzy classifier is created and evaluated on real-world data. The findings are positive.Tiwari and Chakravarthy [9] provided an approach based on force response measurements for identifying unbalance parameters. De Queiroz [10] presented an unbalance force identification approach based on harmonic response to determining unbalanced parameters. Li et al. [11] developed an expert system for identifying unbalance utilizing a acoustic signal and Artificial Neural Network (ANN). Harsha [12] studied the unbalanced rotor on roller bearings using nonlinear dynamic analysis. They demonstrated the dynamic response's appearance of chaos and instability as the speed of the system was altered. Shinde et al. [13] studied a matrix method of dimensional analysis to find the effect of unbalance and misalignment. The author uses a support vector machine for multi-fault classification in a rotor-bearing system. A support vector shows a promising performance for a given input parameter.

The dimensional analysis is a powerful method to predict the vibration characteristics of a rolling element bearing (REB) under variety of conditions. The latest machine learning tools also increase fault diagnosis performance at different fault conditions [14-22].

2. DIMENSIONAL ANALYSIS (MATRIX METHOD)

The matrix method of dimensional analysis has been used to determine the unbalance rotor-bearing system characteristics. Fourteen dependent and independent variables have been used to generate a mathematical model under the influence of rotor unbalance. All variables affecting the dynamic behaviour of a taper roller contact bearing can be defined in terms of three fundamental dimensions: length (L), time (T), and either force (F) or mass (M). Variables used for the FLT (Force, Length, and Time) model are shown in table no.1 with their dimensionless unit. The functional relation between vibration amplitude in terms of velocity and 13 dependent variables is shown by the equation,

$$\dot{x} = f(d_m, D_b, B, K_d, \rho, E, \delta, c, N, W, F_u, K, m_r) \quad (1)$$

Details of each parameter are given in table no.1 Table 1 represents 14 dimensionless parameters

Table 1. Dimensions to study the system [14]

Symbol	Parameter (Unit)	Dimension
d_m	Diametric Pitch (mm)	L
D_b	Diameter of Ball (mm)	L
E	Modulus of Elasticity ($\frac{N}{mm^2}$)	FL^{-2}
K_d	Contact force for deformation ($\frac{N}{mm^{-1.5}}$)	$FL^{-1.5}$
ρ	Material Density ($\frac{kg}{m^3}$)	$FL^{-4}T^2$
δ	Bearing deflection (mm)	L
c	Coefficient of Damping ($\frac{Ns}{m}$)	$FL^{-1}T^1$
N	Speed of Shaft(rpm)	T^{-1}
W	Load (N)	F
U	Unbalance mass (kg)	$FL^{-1}T^2$
B	Bearing Width (mm)	L
Ks	Stiffness (N/m)	FT^{-2}
m_r	Rotor mass (Kg)	$FL^{-1}T^2$
$\dot{x}$	Velocity(Amplitude of Vibration- m/s)	LT^{-1}

Table 1 depicts a matrix with fourteen variables. The dependent variables are represented on the left side of the matrix, while the independent variables are represented on the right side of the matrix.

Figure 1 represents the following manner

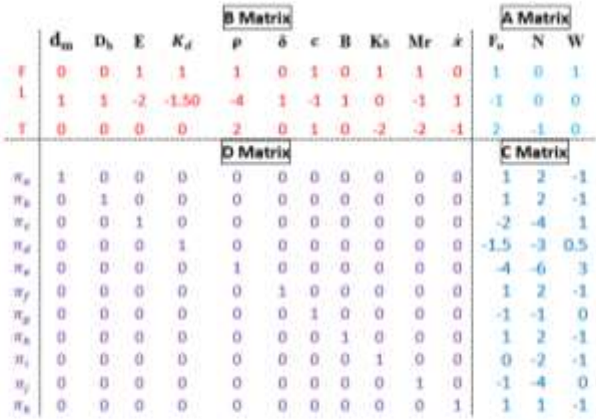
A matrix- Repeating variables

B Matrix- Dimensionless parameter

C Matrix- Derived from equation

$$C = -(A^{-1} B)^T \ [15]$$

D Matrix- No. of 'π' terms



	d_m	D_b	E	K_d	ρ	δ	c	B	Ks	Mr	$\dot{x}$	F_u	N	W
					B Matrix							A Matrix		
F	0	0	1	1	1	0	1	0	1	1	0	1	0	1
L	1	1	-2	-1.50	-4	1	-1	1	0	-1	1	-1	0	0
T	0	0	0	0	2	0	1	0	-2	-2	-1	2	-1	0
					D Matrix							C Matrix		
π_a	1	0	0	0	0	0	0	0	0	0	0	1	2	-1
π_b	0	1	0	0	0	0	0	0	0	0	0	1	2	-1
π_c	0	0	1	0	0	0	0	0	0	0	0	-2	-4	1
π_d	0	0	0	1	0	0	0	0	0	0	0	-1.5	-3	0.5
π_e	0	0	0	0	1	0	0	0	0	0	0	-4	-6	3
π_f	0	0	0	0	0	1	0	0	0	0	0	1	2	-1
π_g	0	0	0	0	0	0	1	0	0	0	0	-1	-1	0
π_h	0	0	0	0	0	0	0	1	0	0	0	1	2	-1
π_i	0	0	0	0	0	0	0	0	1	0	0	0	-2	-1
π_j	0	0	0	0	0	0	0	0	0	1	0	-1	-4	0
π_k	0	0	0	0	0	0	0	0	0	0	1	1	1	-1

Figure 1. Dimensionless Set [14]

From the above dimensionless matrix 11 π terms are obtained and listed in the Table No. 2.

Table 2. Dimensionless Variables

Variable	π-terms	Variable	π-terms
U, N, W and d_m	$\pi_a = \frac{d_m U\, N^2}{W}$	U, N, W and C	$\pi_g = \frac{C}{U\, N}$
U, N, W and D_b	$\pi_b = \frac{D_b\, U\, N^2}{W}$	U, N, W and B	$\pi_h = \frac{B\, U\, N^2}{W}$
U, N, W and E	$\pi_c = \frac{E\, W}{U^2 N^4}$	U, N, W and K_s	$\pi_i = \frac{K_s}{N^2\, W}$
U, N, W and K_d	$\pi_d = \frac{K_d\, W^{0.5}}{U^{1.5} N^3}$	U, N, W and Mr	$\pi_j = \frac{M_r}{U\, N^4}$
U, N, W and ρ	$\pi_e = \frac{\rho\, W^3}{U^4 N^6}$	U, N, W and $\dot{x}$	$\pi_k = \frac{\dot{x}\, U\, N}{W}$
U, N, W and δ	$\pi_f = \frac{\delta\, U\, N^2}{W}$		

Vibration amplitude is function of all the π terms and represented by equation no. 2,

$$\frac{\dot{x}\, U\, N}{W} = f\left(\frac{d_m U\, N^2}{W}, \frac{D_b\, U\, N^2}{W}, \frac{E\, W}{U^2 N^4}, \frac{K_d\, W^{0.5}}{U^{1.5} N^3}, \frac{\rho\, W^3}{U^4 N^6}, \frac{\delta\, U\, N^2}{W}, \frac{C}{U\, N}, \frac{B\, U\, N^2}{W}, \frac{K_s}{N^2\, W}, \frac{M_r}{U\, N^4}\right) \quad (2)$$

It is reasonably challenging to manage a number of π terms in such a way that dimensional reduction occurs in following way,

$$\pi_I = \frac{\pi_a}{\pi_b} = \frac{d_m}{D_b} \quad (3)$$

$$\pi_{II} = \frac{\pi_f}{\pi_h} = \frac{\delta}{B} \quad (4)$$

$$\pi_{III} = \frac{\pi_i \times \pi_j}{\pi_g} = \frac{W\ M_r}{K_s C\ N} \quad (5)$$

Eq. 3 indicates vibration amplitude after a sequence of reductions in the dimensionless parameter.

$$\pi_k = (\varphi\ \times \pi_I \times \pi_{II} \times \pi_{III}) \quad (6)$$

Where,φ is the dimensional groups which not assorted during experimental analysis. After putting the π terms we get

$$\frac{\dot{x}W^2}{U\ N} = \left(\varphi \times \frac{d_m}{D_b} \times \frac{\delta}{B} \times \frac{W\ M_r}{K_s C\ N}\right) \quad (7)$$

The above equation represents a mathematical equation for the rotor-bearing system under unbalance conditions.

3. EXPERIMENTATION

For the rotor-bearing setup, the dynamic response of the test setup is investigated by taking unbalanced mass and shaft speed into account. The schematic view of the test setup, as shown in figure 2, consists of a shaft with a disc supported between two bearings that are driven at operating speed by a DC motor via a dimmer stat. The shaft is coupled with a DC motor with a flexible coupling.

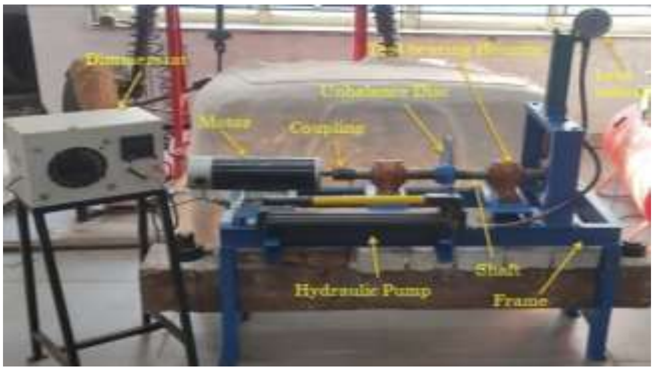


Figure 2. Experimental Setup

Table 3. Specifications of Bearing	
Bearing Type	SKF 6209-K
(d)- Bearing Inner Dia.	45 mm
(D)- Bearing Outer Dia.	85 mm
No. of Balls (N)	9
Diameter of balls	9 mm

Two deep groove ball bearings, specifications mentioned in table 3, are used for experimentation. To measure vibration signals, Adash VAPro 4400 Fast Fourier transforms (FFT) with an accelerometer of the piezoelectric type with a sensitivity of 100 mV/g was placed on the test bearing housing. The different combination of bearing speed and unbalanced mass was simulated between 500 rpm to 1300 rpm. Table 2 shows the bearing specifications that were used in the experiment.

4. RESULTS AND DISCUSSION

Different combinations of unbalance mass of 50 gm to 125 gm and shaft speed of 700 rpm to 1300 rpm are considered for experimentation. Sixteen experiments were performed on the test setup, and frequency responses were obtained below.

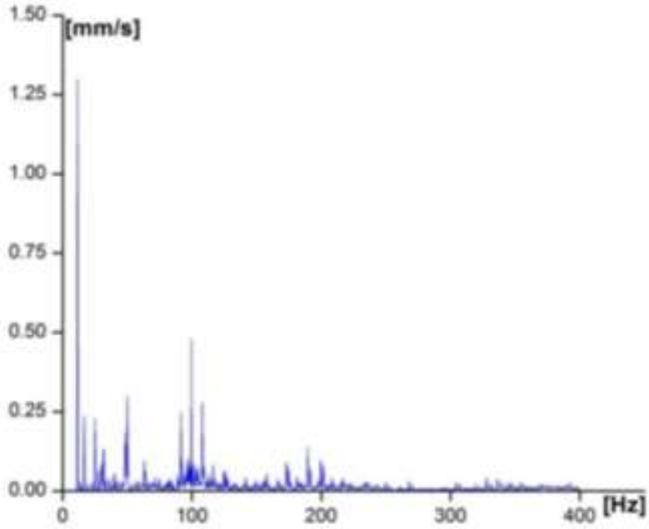


Figure 3. Vibration characteristics for trial-2

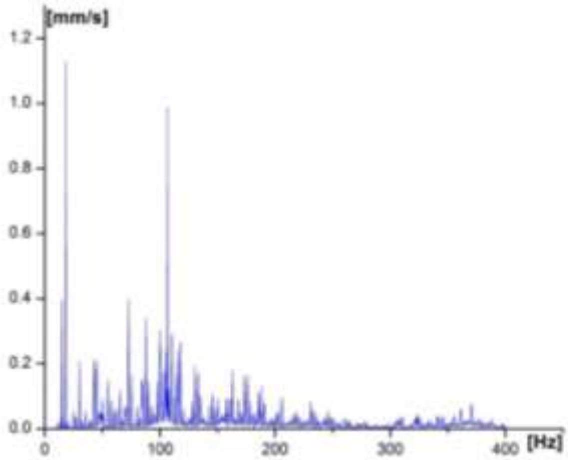


Figure 4. Vibration characteristics for trial-8

An unbalance mass of 125 gm and speed of 700 rpm is a combination for trial 2 is considered. The vibration response obtained using the test is shown in figure 3. The figure depicts that vibration spectrum is obtained at first harmonics of shaft frequency ($1 \times f_s$). The vibration amplitude for a given unbalance mass is 1.29 mm/s.

A combination of unbalance mass and speed for trial 2 is 75 gm and 900 rpm, respectively, shown in figure 4. The vibration response obtained using the test is shown in figure 4. The vibration amplitude for conducted trial is 1.18 mm/s. As the unbalance mass increase with speed, vibration amplitude also increases.

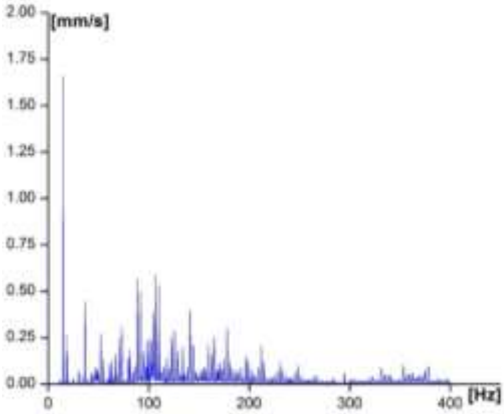


Figure 5. Vibration characteristics for trial-13

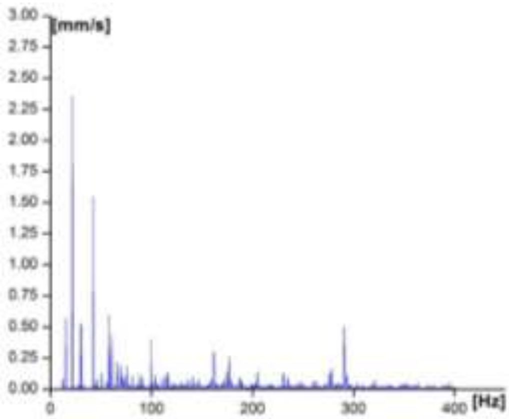


Figure 6. Vibration characteristics for trial-15

The above trial is conducted for rotor speed of 1100 rpm and 75 gm of unbalance mass. The peak amplitude is obtained for a given speed, and the mass is 1.65 mm/s. The peak amplitude is obtained at first harmonics of shaft frequency. The vibration response for the above trial is shown in figure 5.

Similarly, a combination of unbalance mass and speed of 1300 rpm and 50 gm unbalance, the vibration responses are shown in figure 6. The vibration amplitude for conducted trial is 2.354 mm/s.

Table 4. Summary of the vibration responses					
Trial No.	**Speed**	**Unbalance Mass**	**Experimental vibration amplitude (mm/s)**	**Model vibration amplitude (mm/s)**	**% Error in amplitude**
1	700	50	0.927	0.91	1.83
2	900	75	1.18	1.12	5.08
3	1100	100	2.07	2.03	1.93
4	1300	125	3.4	3.31	2.65
5	700	75	0.997	1.04	4.31
6	900	100	1.49	1.43	4.03
7	1100	125	2.35	2.3	2.13
8	1300	50	2.354	2.28	3.14
9	700	100	1.09	1.05	3.67
10	900	125	1.58	1.5	5.06
11	1100	50	1.59	1.52	4.40
12	1300	75	2.89	2.84	1.73
13	700	125	1.29	1.26	2.33
14	900	50	1.02	0.97	4.90
15	1100	75	1.65	2.06	0.48
16	1300	100	3.17	3.15	0.63

Similarly, all trials are conducted for different speeds and unbalanced mass. The combination of unbalanced mass and speed and respective vibration response is reported in table 4. It is observed from trials that as unbalanced mass is introduced with combinations of speed increases the vibration levels of the system and shows a vibration peak at first harmonics of shaft frequency, i.e. (1 × fs). Speed, geometric proportions, and mass distribution of the rotor and the shaft, bearings, and foundation's dynamic rigidity primarily determine the unbalance response. The disc unbalance is created by attaching the unbalance mass to the disc.

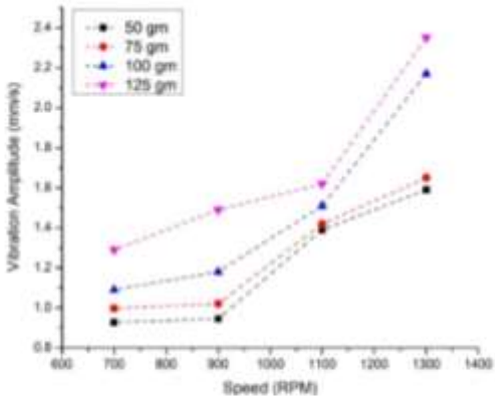


Figure 7. Influence of unbalance mass to speed of the shaft

Figure 7 shows the vibration amplitude vs. speed of the rotor in rpm. The maximum vibration amplitude is obtained at 125 gm of unbalance and at 1300 rpm of speed, while at 700 rpm of speed and at 50 gm of unbalance amplitude of vibration is minimum. It is observed from the pilot experiments that the amplitude of vibration increases as speed increases. Also, defect frequency corresponds to the shaft's rotating speed fs, which conforms to the theoretical results.

5. CONCLUSION

A model-based matrix method (MMDA) is proposed for the rotor-bearing system under the influence of unbalance. An unbalance mass can cause complete failure of the rotating machine. The proposed method is found to be accurate and effective in obtaining the vibration amplitude under unbalance conditions.

The following findings are noted.

- The difference between the vibration amplitude measured experimentally and the amplitude calculated using a mathematical model demonstrates a close match with a marginal error.
- Experimentation shows that the dominating peak is obtained at first and several harmonics of the shaft frequency.
- The mathematical model proposed is consistent with the results found experimentally.
- The findings indicate that the vibration amplitude is enhanced as the unbalance mass increases.

ACKNOWLEDGEMENT

This work is financially supported by the Chhatrapati Shahu Maharaja National Research fellowship 2020, SARTHI, Pune.

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