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Diagnosis of the Unbalanced Rotor Using Vibration and Noise Analysis in one and Two Planes

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ABSTRACT: *This article summarizes a comprehensive mechanical system defect diagnostics investigation focusing on system imbalance. This research used several mechanical defect diagnostic equipment to study mechanical systems. To identify and fix system imbalances. These technologies helped identify and explain mechanical defects, improving system performance and dependability. This research used mechanical fault diagnostic instruments to analyze the mechanical system. Measurement and analysis of system imbalance revealed opportunities for improvement. Understanding these discrepancies allowed the development of effective techniques to reduce them, improving mechanical system performance and lifetime. Upon increasing the rotor speed, the amplitude of vibration increased, followed by a subsequent decrease upon reaching the resonance of the system structure. The bump test results indicate that the structure's resonance occurs at 1200 rpm, resulting in the highest vibration amplitude for all measured values. The results suggest that there exists a favorable association between the speed of the rotor and the level of noise in the cases that were examined. In comparison to the balanced rotor, the unbalanced rotor displays a rise in noise level of 6.5%.*

KEYWORDS: *Vibration; Noise level; Unbalance ; Rotor ; Diagnosis.*

I. INTRODUCTION

Rotor unbalance widely produced in manufacturing and servicing. Generally, the unbalance is randomly distributed along the axis of rotation and is difficult to obtain completely. These unbalance cause an inertial force to excite the rotor vibration when running. Actually, it is not necessary to know the unbalance distribution in the rotor completely. As long as the inertial force caused by using a set of correction weights counteracts the vibration caused by the original unbalance of the rotor, the rotor will be balanced.

The influence coefficient and the correction weights can be identified by measuring the initial vibration of the different balancing planes and the vibration after installing the trial weights. This treatment was initially used to balance the plane's rotor on the shaft for the rotor; this purpose can be achieved by choosing two balancing planes at a balancing speed. American scholar Thearle [1] first proposed this method based on an experiment. Since this method can only be applied to a rotor, Baker [2] improved upon the method, but the experimental result was unsatisfactory.

A. Foppl (1895) formulated and solved the equations governing the response of a single mass rotor system [3]. His analysis showed that at speeds significantly higher than the critical speed the rotor would turn about its mass center; his undamped analysis predicted infinite response at the critical speed and a transient response at the critical speed frequency. H.H. Jeffcott [4] analyzed the fundamental nature of the response of a single mass flexible rotor to the imbalance in 1919. Proper instrumentation to experimentally verify these results would not exist for years. Rieger [5] references electronic and stroboscopic measurements first developed in the 1930’s.

Even before Jeffcott explained the fundamental response of rotor systems, balancing machines were in use, Rieger [5]; Martinson developed a balancing machine as early as 1870. The 'heavy' spot was marked by hand on this balancing machine. At low speeds (sub-critical), the 'heavy' spot would coincide with the 'high' spot of the whirl, according to Jeffcott's analysis. Early balancers such as S.H. Weaver [6] in 1928 realized that the balance weights and imbalances act as forces to the system. Weaver was aware that the forces at the bearing locations for a rigid rotor changed with the magnitude and phase of the imbalance weights. Later balancers would develop the notion of influence coefficients, though at first, they would not use this name.

Over the years, many rotors balancing techniques have been proposed in the literature [7, 8, 9]. Amongst all vibration-based rotor balancing techniques, the most predominant approach is the influence coefficient (IC) balancing method [10, 11]. IC balancing method is also known as field balancing because the balancing is performed at the site without disassembling the rotor from the machine. In this balancing approach, the rotor system is assumed to be linear, and the influence of the individual can be superposed to give the influence of a set of unbalances [7]. In addition, the IC method does not require any prior knowledge of the dynamics of the rotor [12]; it requires only the vibration response of the rotor at different trial masses. Therefore, the effect of the rotor unbalance in one plane, and two planes with their noise effect has not been reported in the previous literature. In the current investigation, the rotor balancing using one and two planes is studied and diagnosed using a vibration data collector; hence the noise analysis is investigated simultaneously in each case.

II. EXPERIMENTAL ANALYSIS

Figure 1 depicts a schematic representation of the rotor unbalance. The experimental arrangement comprises two discs, one coupling, two journal bearings, and a variable speed control mechanism affixed to the AC motor. To investigate the issue of unbalance, a bolt was affixed onto the disc and regulated by frequency inverters to display the test components. An experiment was conducted wherein a motor was coupled with an unbalanced disc to produce a unique source of vibrations. The experiment was repeated using two discs and a COMMTEST Vb7 data collector. The aforementioned apparatus was utilized in tandem with an optical sensor tachometer boasting a precision of under 0.1%, and a piezoelectric accelerometer sensor with a sensitivity of 100 mV/g, suitable for employment at frequencies ranging from 0.8 Hz to 10 kHz. The sound level meter utilized a noise sensor to quantify the amplitude of sound generated at various velocities. The investigation employs a test rig, as depicted in Figure 2, which is representative of the standard equipment utilized.

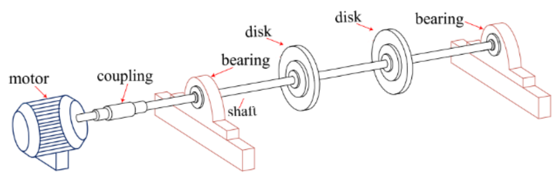

Figure 1:Schematic of experiment system.

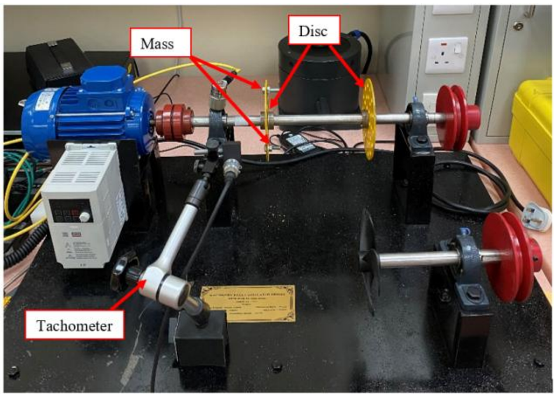

Figure 2: Typical experimental apparatus used to study the system

INITIAL MEASUREMENTS

At the outset, the vibration levels across the globe were evaluated in terms of millimeters per second root mean square (RMS). This unit of velocity possesses the attribute of being more closely linked with the vibrational energy that is expressed as a factor of centrifugal force and mobility. All measurements are contingent upon the frequency alteration, which is subsequently converted to revolutions per minute (RPM) by multiplying by 60. The rotational velocities utilized in this study are 600, 900, 1200, and 1500 revolutions per minute, as presented in Table 1. The installation of the piezoelectric accelerometer sensor is oriented radially with respect to the bearing and is positioned in proximity to the initial unbalanced disc.

STATIC BALANCE

In the beginning, we start this experiment without any mass to see the origin case 1 and compare this case with the other cases. The relationship between speed and amplitude is clear when speed increases; the amplitude increases from 600 until 1200 RPM, then the amplitude decreases when speed reaches the maximum 1500 RPM.

UNBALANCE ONE PLANE

Initiating with the introduction of mass at the initial plane, it creates an imbalance within the system. In the second scenario, the correlation between speed and amplitude is evident. Specifically, the amplitude proportionally increases as the speed escalates from 600 to 1200 RPM. However, the amplitude subsequently decreases upon reaching the maximum speed of 1500 RPM. It is evident that the amplitude exceeds 15 millimeters per second.

UNBALANCE TWO PLANES

In this instance, a secondary mass was introduced at the second plane, resulting in a discernible correlation between velocity and magnitude. Specifically, it is evident that as velocity escalates, the magnitude experiences an increase from 600 to 1200 RPM, followed by a subsequent decrease once the maximum velocity of 1500 RPM is attained. It is evident that the amplitude exceeds 17 mm/s, representing the maximum amplitude obtained.

BALANCE TWO PLANES

In Case 4, a mass is placed on the second plane in a direction opposite to the initial mass, with the aim of achieving equilibrium. The correlation between speed and amplitude is evident, whereby an increase in speed results in a corresponding increase in amplitude ranging from 600 to 1200 RPM. However, the amplitude subsequently decreases upon reaching the maximum speed of 1500 RPM.

BALANCE ONE PLANE

In Case 5, there is a presence of another mass at the first plane, positioned in an opposing direction, with the aim of attaining equilibrium. The correlation between speed and amplitude is evident, whereby an increase in speed results in an increase in amplitude from 600 to 1200 RPM. However, the amplitude subsequently decreases when the speed reaches its maximum of 1500 RPM.The equilibrium state of the system with a balance of 5 closely approximates the static equilibrium state observed in case 1, and exceeds that of case 4.

III. RESULTS

The system's behavior under varying rotational speed values is elucidated in Figures 3 and 4. The experimental procedure involved a gradual increase in rotational velocity, beginning at 600 rotations per minute (rpm) and incrementally increasing by 300 rpm until reaching a final speed of 1500 rpm. In the first scenario, the system was activated without any accompanying specifications. It can be inferred from the observation that an increase in rotational speed leads to a corresponding increase in vibrating amplitude. At a rotational speed of 1200 revolutions per minute, a significant increase in the overall value can be observed, indicating the occurrence of resonance at this particular speed. Additional tests will be conducted to monitor the system's behavior at 1200 rpm (20 Hz). Figures 5 and 6 depict the operational characteristics of the system upon activation, subject to a particular load applied to one of the rotating disks at a specific radial distance. In comparison to the first case, it is evident that a significant value increased the vibration readings. The persistence of the elevated global value at 1200 rpm provides confirmation of the resonance phenomenon transpiring at a frequency of 20 Hz.

Table 1: Vibration amplitude measurements

Speed RPM	Vibration amplitude			
	mm/s	mm/s	mm/s	mm/s
	Balance	Unbalance one plane	Unbalance two planes	Balance two planes
600	1.4	2	3.7	1.6
900	2.4	4	5.8	2.6
1200	6.1	15.16	17.9	9.6
1500	4.35	7.9	17.6	9.4

Figure 3: Case 1: Schematic of Static balance

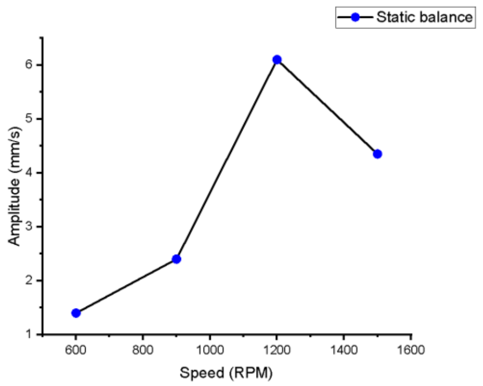

Figure 4: Case 1: Chart of Static balance

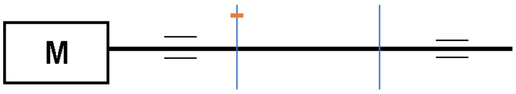

Figure 5: Case 2: Unbalance one plane schematic

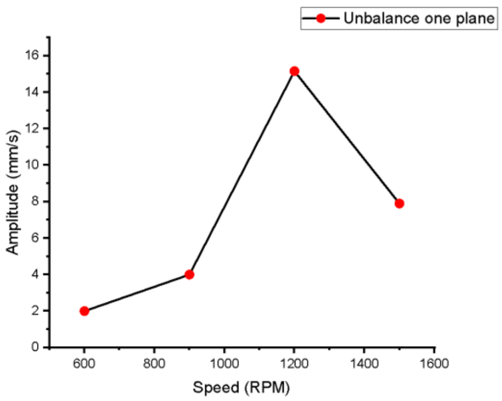

Figure 6: Case 2: Chart for Unbalance one plane

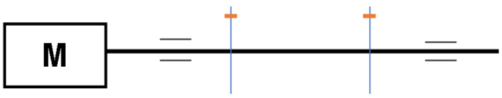

Figure 7: Case 3: Unbalance two planes schematic

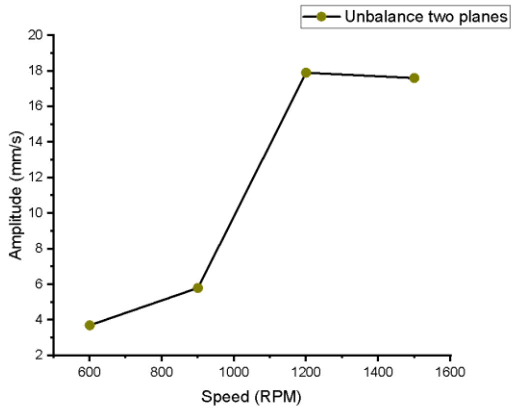

Figure 8: Case 3: Chart for Unbalance two planes

In the third scenario, an additional load has been introduced on the same side of the second rotating disk, resulting in a dual-plane imbalance. Figures 7 and 8 depict the response of the system to imbalances in two planes. It is evident that the magnitude of the vibration has escalated to exceedingly high levels, potentially surpassing acceptable thresholds at elevated rotational velocities. The persistence of the elevated global value at 1200 rpm provides confirmation of the resonance taking place at a frequency of 20 Hz. An experimental approach has been implemented in case 4 as a prospective resolution for case 2. Specifically, the load on the second disk has been repositioned to the opposing side of the first disk's load, as illustrated in Figures 9 and 10, which depict the vibration response outcomes. It is evident that the amplitude of vibration in case 4, which pertains to balance in two planes, exhibits a slight reduction in comparison to the values observed in case 2, which pertains to unbalance in one plane. However, there remains a discernible distinction between case 4 and case 1, which pertains to static balance. Figure 11 illustrates a comparison between the unbalance in a single plane and in two planes. The results indicate that the vibration amplitude in the two-plane unbalanced system is greater than that of the one-plane system, which is a logical outcome.

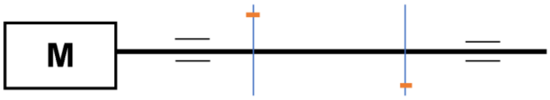

Figure 9: Case 4: Balance two planes' schematic

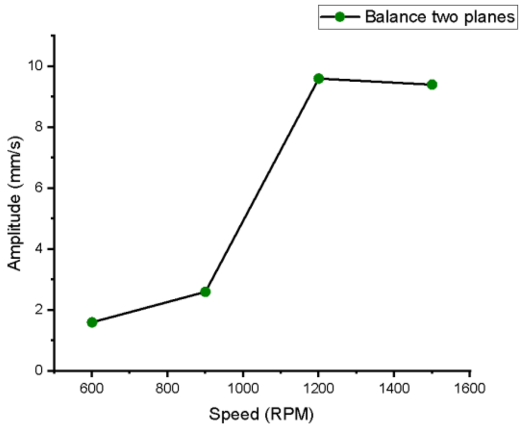

Figure 10: Case 4: Chart forthe balance of two planes

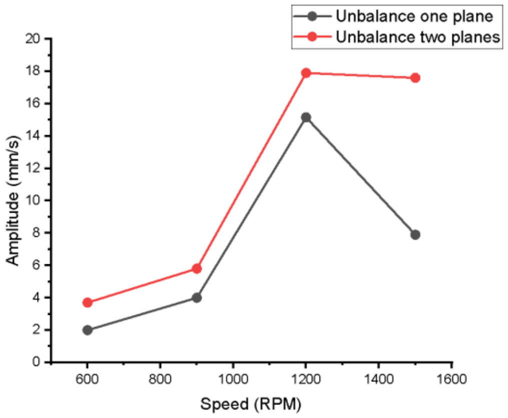

Figure 11: Comparison between cases 2 and 3

Noise analysis

Table 2 illustrates the performance of noise measurement for both balanced and unbalanced one-plane scenarios. It was observed that the noise level of the cases under investigation increased proportionally with the rotor speed. According to the data presented in Figure 12, the noise level in the unbalanced rotor is observed to be 6.5% higher than that in the balanced rotor.

Table 2: Noise measurement.

Speed	Noise	
RPM	dB	dB
	Unbalance one plane	Balance one plane
600	67	67
900	73	71
1200	78	74
1500	81	76
1800	83	78

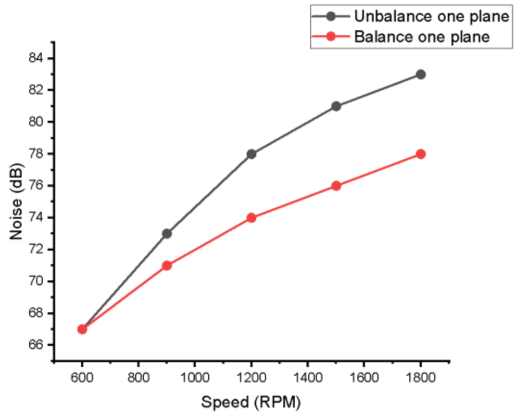

Figure 11: Comparison between the noise in the balanced rotor and unbalanced rotor for the same plane

IV. CONCLUSION

The current work performs the balancing procedures for rotating discs in one and two planes. The following conclusions are derived:

- As the rotor speed increased, the vibration amplitude increased and decreased after reaching to system structure resonance.
- The bump test results revealed that the structure resonance is obtained at 1200 rpm; thus, the maximum vibration amplitude is located at this value for all measured values.
- The findings indicate that there is a positive correlation between rotor speed and noise level in the investigated cases. The unbalanced rotor exhibits a 6.5% increase in noise level compared to the balanced rotor.

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