Research Article

Investigation of the Effects of Unbalance and Bearing Wear on Shaft Vibration in a Natural Gas Turbine Plant

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Abstract
This work investigates the effect of imbalance and bearing wear on the vibration of rotating shafts at a southern Iraqi natural gas liquefaction plant. This experimental study examines the impact of wear on couplings and uneven weight on the vibration of a two-stage gas turbine's shaft, taking measurements during operation. The experimental procedure involves the use of proximity probes and the ADRE-408 Bentley Nevada system to measure vibrations along the X and Y axes. The study focuses on a two-stage gas turbine supported by four journal bearings and analyses the effects of coupling imbalance and erosion. The results show that adding 10 g and 20 g weights at 0° and 30° anticlockwise considerably increases the vibration amplitude, from 22.46 µm at 113.75 Hz to 24.35 µm at 117.5 Hz. Replacing worn couplings and bearings led to system stabilisation, vibration reductions, and a shift in critical frequencies. The data confirm that mass loss and bearing wear greatly affect the dynamics of rotating machinery. These findings emphasise the importance of predictive maintenance and diagnostic monitoring to prevent mechanical failures and maintain system stability.

Keywords: Bearing degradation, Dynamic response, Imbalance, Rotating shaft, Vibration analysis

1 Introduction

Vibration-based diagnostics enable the early detection of imbalance, wear, and lubrication issues in rotating machinery. Such detection supports predictive maintenance and enhances the reliability of systems such as gas turbines [1]. Turbine shafts are crucial to various industrial operations, including manufacturing, power generation, aviation, maritime, and other sectors [2], [3]. These shafts transfer mechanical energy from the turbine to system components, ensuring the continuous operation of machinery and infrastructure. The durability, reliability, and resilience of turbine system shafts depend on their structural integrity and dynamic performance [4]. Optimising operational efficiency and preventing catastrophic failures requires a thorough understanding and effective management of turbine shaft vibrations under dynamic loads [5]. A GE MS 3002 system for monitoring gas turbine faults was developed using spectral and vibrational analysis. The results show gas turbine monitoring equipment optimises performance and eliminates vibration modes [6]. Rotor dynamics relies on vibration analysis because uneven forces can cause and propagate rotor shaft cracks [7]. Recent studies have advanced vibration-based fault diagnosis by focusing on the instantaneous features of vibration signals. Using joint amplitude-frequency analysis and adaptive signal decomposition techniques, researchers have demonstrated improved accuracy in detecting bearing wear, imbalance, and degradation under dynamic conditions [8], [9]. These approaches contribute to real-time condition monitoring and improved system reliability under varying operational conditions.

In this study, the intrinsic frequencies, mode shapes, and system damping of a turbine shaft were calculated using both theoretical and experimental approaches. Engineers often use mathematical models and computer simulations to predict stress-induced vibrations. When operational frequencies coincide with a system's natural frequencies, resonance occurs, which amplifies vibrations and adversely affects the longevity of turbine components and overall system performance [10]. To mitigate shaft vibrations, various control methods are employed, including passive, active, and semi-active techniques. Passive control mechanisms—such as dynamic balancing and adjustable mass dampers—are simple and robust, but they rarely adapt to varying loads and operational conditions [11]. In contrast, active control systems utilise modern sensors and actuators to modify damping parameters in real time, enhancing vibration suppression while also increasing system complexity and maintenance requirements [12]. Nonlinearities and uncertainties in dynamic systems can be effectively addressed using fuzzy control for managing turbine shaft vibrations. Unlike traditional mathematical modelling, fuzzy control relies on heuristics and linguistic rules to make decisions with partial or uncertain inputs [13]. Fuzzy controllers are well-suited for responding to unexpected, time-varying dynamic demands. They adapt control inputs based on real-time vibration data to maintain shaft stability and performance [14]. Orbital journal bearings also play a crucial role in stabilising load-induced turbine shaft dynamics. These bearings help reduce friction and wear during complex shaft motions, such as axial and radial displacements. However, journal orbital motion itself may contribute to shaft vibration.

Advanced designs for orbit journal bearings aim to enhance system damping and stiffness, thereby reducing vibration levels [15]. Because contact between the shaft and orbit journal bearings significantly influences turbine system dynamics, effective vibration control and analysis must account for bearing behaviour [16]. Both fuzzy control and orbit journal bearings contribute to damping turbine shaft vibrations. Fuzzy controllers adjust grease viscosity and bearing clearance in response to vibration patterns, helping to suppress vibrations, reduce resonance, and mitigate wear, thereby extending the lifespan of shafts and bearings [17]. Any effective vibration control strategy depends on preventive measures and frequent monitoring. For predictive maintenance, modern sensors and diagnostic tools are used to measure vibrations in real time. This proactive approach enables early fault detection, enhancing the safety and efficiency of turbine operations [18]. Beyond operational efficiency, vibration analysis of turbine shafts provides broader benefits. Failures in systems such as power plants and aircraft can have serious environmental, safety, and economic consequences. So, it's important to combine vibration analysis with fuzzy control and orbit journal bearings for these risky situations.

The moderating effect of fuzzy control and orbital journal bearings on dynamic turbine shaft vibrations has been studied through mathematical modelling, computer simulations, and experimental data. These results contribute to the improved stability and performance of turbine systems in industrial settings. Technological advancements derived from such studies have the potential to enhance both the safety and efficiency of turbine systems [19]. Maintaining the stability and efficiency of gas turbines and other rotating machines requires effective vibration control. However, vibrations resulting from unequal loading and bearing degradation may persist even after multiple corrective actions [20]. A previous study showed that a semi-active magnetorheological (MR) damper reduced vibrations by dynamically adapting its damping properties [21]. Another study highlighted the role of damping pads in minimizing vibration transmission and enhancing system stability [22]. Additional research emphasised the importance of condition monitoring for journal bearings, noting that wear and inadequate lubrication can significantly increase vibration levels. These studies align with current research by underscoring the impact of imbalance and bearing wear on vibration behaviour. Integrating advanced damping systems and real-time monitoring techniques has been shown to reduce mechanical failures and extend component lifespan. In the present study, a spinning shaft at a natural gas liquefaction plant in southern Iraq was analysed for imbalance-induced vibrations. The ADRE-408 system was used to collect and analyse data via dynamic response sensors. It was found that losing weight (10 g and 20 g at different angles) from worn couplings caused higher vibration levels, especially in tests with the fourth bearing. The replacement of worn couplings and bearings significantly improved system stability and reduced vibration levels.

This study highlights the importance of periodic component replacement and enhanced diagnostic monitoring to maintain the performance and stability of rotating equipment while also preventing unexpected failures. A research gap exists regarding the specific effects of erosion-induced mass loss on the dynamic behaviour of rotating machinery. This work addresses that gap through an experimental approach, with particular focus on the influence of asymmetric mass distribution caused by coupling wear on vibrational behaviour and critical speeds. The primary objective is to provide practical insights that can help improve maintenance scheduling and diagnostic practices in gas turbine operations.

2 Materials and Methods

This investigation identified the imbalance causing vibration in a spinning shaft at the North Al-Rumaila Natural Gas Liquefaction Plant in Basra, southern Iraq. The Basra Gas Company gas station (NR-NGL-BGC) uses the rotating shaft, and Figure 1. A schematic sketch depicts its dimensions. The four journal bearings support this two-stage gas turbine. The focus is on the unbalance occurring in the HP-side coupling hub and the effect of erosion in that area on the system's dynamic response.

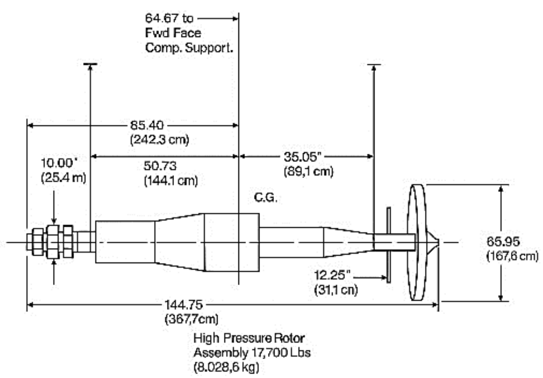

Figure 1: Schematic of a two-stage gas turbine with dimensions.

The Bentley Nevada device, labelled ADRE-408, was used to analyse the signal by operating the rotating shaft and evaluating the resulting vibrations, as well as monitoring and diagnosing the issue before it caused mechanical failure [23]. Figure 2 depicts the apparatus used in the vibration analysis, with its characteristics listed in Table 1.

Table 1: Specifications of the ADRE-408.

Parameter	Value
Single-ended Inputs	8
Differential Inputs	4
4-20 mA Inputs	8
Maximum Signal Input Range	-25 to +25 V
AC Coupled High Mode	1.6 Hz to 50 kHz ± 1.25% of Full-Scale Range
DC Coupled Low Mode	0.167 Hz to 20 kHz ± 1% of Full-Scale Range
A/D Converters	24-bit

Figure 2: ADRE Sxp data acquisition system.

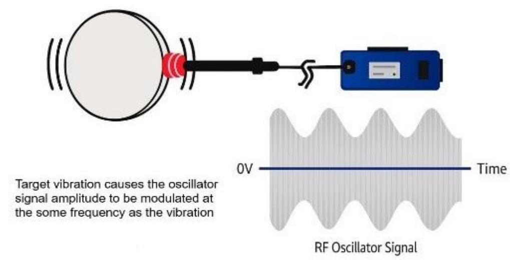

Figure 3: Sample vibration measurement data recorded from the gas turbine system using the ADRE-408 data acquisition unit.

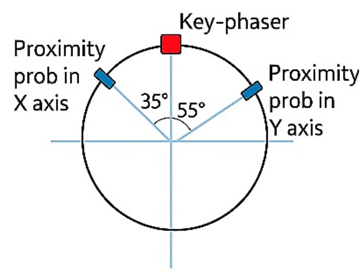

Figure 4: The sensor is oriented on the bearing.

A proximity probe sensor was employed to measure the rotor without physical contact. This sensor generates a gap voltage between the rotor and the probe tip via eddy currents. Figure 3 illustrates the process by which the data collector transforms this signal into a vibration waveform represented in both the time and frequency domains.

The measurement was carried out on one of the spinning shaft's bearings. The study focused on bearing number 4, which was subjected to vibration measurements using two sensors. One sensor was positioned in one direction and labelled as the X-axis, while the other was positioned perpendicularly and labelled as the Y-axis, as shown in Figure 4.

During the experiment, the rotating shaft was operated at a speed ranging from 5,714 to 16,630 RPM. A personal computer was used in conjunction with specialised software configured to process data from the Data Acquisition Card (DAQ). The DAQ was connected to the computer to capture the bearing signal, which was processed and analysed. Figure 5 illustrates a simplified diagram of the setup used to connect the laptop to the sensors and shows the location of the bearing.

Figure 6 shows an example of data readings that were acquired from the personal computer during the measurement process at the station.

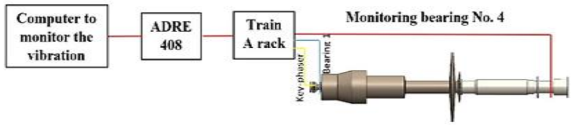

Figure 5: A schematic diagram of the process of connecting the laptop and card to the rotating shaft.

Figure 6: Sample of vibration measurement data recorded from the Gas turbine system using the ADRE-408 data acquisition unit.

3 Results and Discussion

This experiment examines how unbalance affects rotating shaft vibration. The stability and performance of rotating equipment are directly influenced by unbalance. This study identifies mass loss due to coupling erosion as a cause of rotational unbalance. Erosion leads to uneven mass distribution around the shaft's rotational axis, resulting in increased vibration amplitudes and reduced system stability due to centrifugal forces. The experiment revealed a coupling issue around bearing 4, which could compromise the precision and reliability of the results. To ensure accurate measurements, the coupling was replaced. To determine how mass distribution affects vibrational behaviour, the study evaluated the system under different mass configurations. Trial runs were meticulously conducted to capture the system's dynamic behaviours under various conditions. The study's findings offer new perspectives on unbalance and vibration, suggesting improvements for the design and maintenance of rotating equipment systems. Regular high-speed bearing diagnostic testing helps maintain system performance and prevent malfunctions. Examining the bearings closely, checking them in real-time, and utilising advanced tools helps identify wear early and take prompt action, which can prolong the lifespan of bearings and related components. Figure 7 shows high-speed coupling wear, as analysed through mass loss.

Figure 7: Surface degradation and mass loss are observed in the worn coupling near Bearing 4, contributing to system imbalance.

Orbit Plot Analysis: This plot depicts the shaft's motion within the bearing using data from proximity sensors that measure the distance between moving parts. Changes in the shape of the orbit show that there might be an imbalance or uneven friction, which could mean the bearing surface is worn out or not getting enough lubrication [24].

Waveform Analyses: Irregular waveforms and increased peak amplitudes indicate the presence of mechanical faults, such as damage to rolling elements or abrasive contact surfaces caused by wear. Possible causes of wear include:

1) High-speed operation: Continuous stress on bearing components due to prolonged high-speed rotation.

2) Inadequate lubrication: Using incorrect or insufficient lubricants can increase friction and accelerate wear.

3) Material limitations: The materials used in bearing manufacturing may be incapable of withstanding the loads associated with high-speed operation [25].

After replacing the old coupling with a new one, vibration data was collected and analysed. As illustrated in Figure 8, the comparison between the worn and new coupling conditions shows a significant difference in the system's behaviour. Figure 8(a) shows the worn coupling condition, where higher vibration levels, instability, and uneven movement are clear because of surface wear. On the other hand,

Figure 8(b) shows how the system behaves after the coupling is replaced, with smoother waveforms and orbits, less vibration, and better overall performance and stability. Rotational unbalance, caused by unequal mass distribution, is a key factor contributing to excessive vibration in rotating machinery. This imbalance results in centrifugal forces during rotation, which can amplify vibration amplitudes and lead to potential mechanical failure if not corrected.

(a) (b)

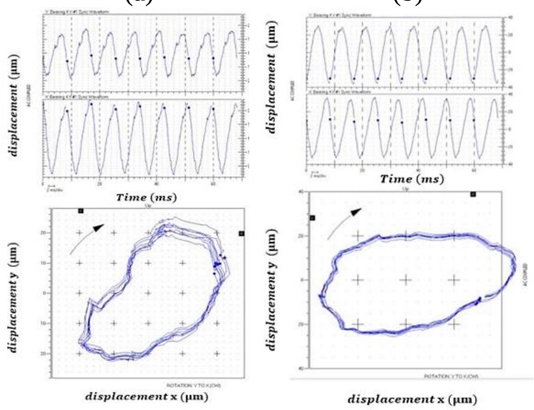

Figure 8: Vibration analysis of 4-axis bearing for (a) worn coupling and (b) new coupling.

(a) (b)

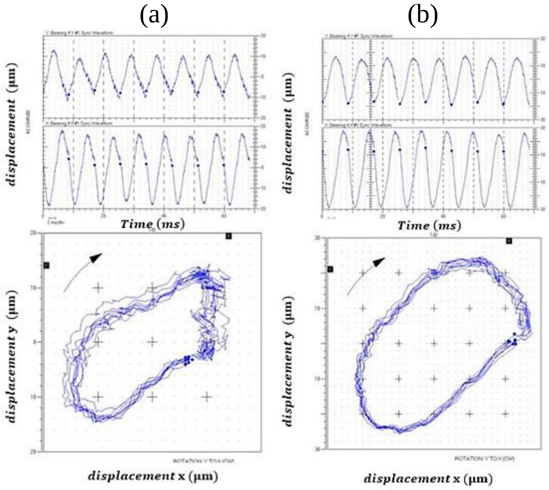

Figure 9: The vibration analysis of a 4-axis bearing was conducted under different mass loss conditions : (a) 10 g was removed at 0° CCW, and (b) 20 g was removed at 30° CCW.

The impact of losing weight during coupling was also looked at by taking away two balancing blocks: 10 g at 0° anticlockwise (CCW) and 20 g at 30° CCW from the Key-phaser. The dynamic responses under these two imbalance conditions are illustrated in Figure 9, where Figure 9(a) shows the vibration response resulting from the 10 g mass removed at 0° CCW. Figure 9(b) presents the response due to a 20 g mass removed at 30° CCW.

As observed, doubling the removed mass from 10 g to 20 g leads to a proportional increase in centrifugal force. This force is directly influenced by the mass magnitude, eccentricity, and the square of the rotational speed [26]. Also, changing the angle of the removed mass from 0° to 30° anticlockwise disrupts the system's balance even more, creating extra unbalanced forces. This asymmetrical mass distribution increases the dynamic load on the bearings, resulting in significantly higher vibration amplitudes.

Frequency domain analysis is critical for vibration monitoring and mechanical diagnostics because it enables the identification of characteristic frequencies associated with specific system behaviours. Unlike time-domain analysis, it decomposes signals into their frequency components using techniques such as the Fast Fourier Transform (FFT), allowing the detection of resonances, imbalance, and structural anomalies. To evaluate the system's dynamic response under different mechanical conditions, measurements were taken in both the X and Y directions using proximity sensors. These measurements compared the system's performance under two conditions: 1) with bearing wear and 2) after replacing the worn coupling. The results are shown in Figure 10, where Figure 10A illustrates the frequency domain response in the presence of bearing wear, showing significant amplitude increases, harmonics, and sidebands. Figure 10(b) presents the response after coupling replacement, indicating reduced vibration and improved spectral stability.

The wear condition (Figure 10(a)) leads to irregular force distribution and clearance variations, causing high-amplitude peaks at fundamental and harmonic frequencies. Side bands and spectral modulations suggest localised mechanical faults or lubrication issues. Comparing the spectra in both orientations reveals that the Y-direction exhibits additional frequency components, which may result from asymmetric loading and stiffness distribution — an indication of anisotropic system behaviour. The X-direction spectrum tends to show higher amplitudes at the first critical speed, suggesting more pronounced effects of imbalance or misalignment. At the same time, the Y-direction has a wider range of frequencies, which means it is more affected by wear issues, especially at higher critical speeds. These results reinforce the importance of regular frequency-domain monitoring for early fault detection and maintenance planning.

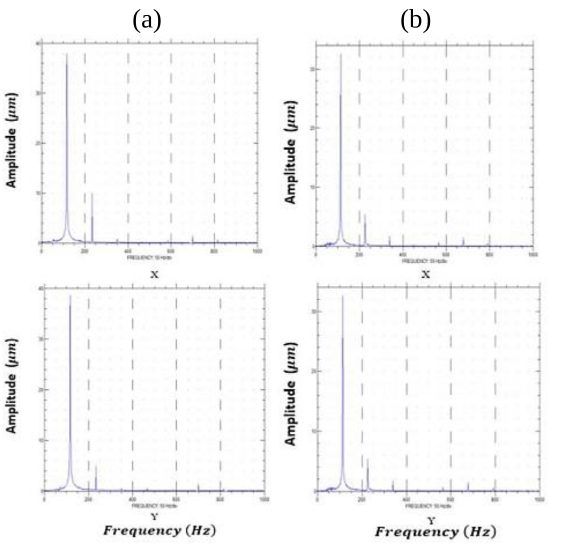

Figure 10: The dynamic response of the journal bearing in the X and Y directions is illustrated as follows: (a) worn part, (b) new coupling.

We also investigated how different levels of mass unbalance influence the system's frequency response. We focused our analysis on comparing the dynamic behaviour when we removed 10 g and 20 g masses at specific angular positions on the coupling. The frequency spectra in Figure 11 illustrate the system's response: Figure 11(a) corresponds to a 20 g mass removed at 30° CCW from the Key-phaser. Figure 11(b) corresponds to a 10 g mass removed at 0° CCW. Both plots display the emergence of critical speeds, considered pronounced peaks within the 100–200 Hz range, which represent the system's fundamental resonance frequencies. Additionally, smaller peaks appear at higher frequencies, likely indicating harmonics or structural resonances. As expected, taking away a larger weight (20 g) greatly increases the main peak, showing that there is a direct link between how much unbalance there is and how strong the vibrations are. These results emphasise the necessity of maintaining precise balance in rotating components to avoid resonance and potential mechanical failure.

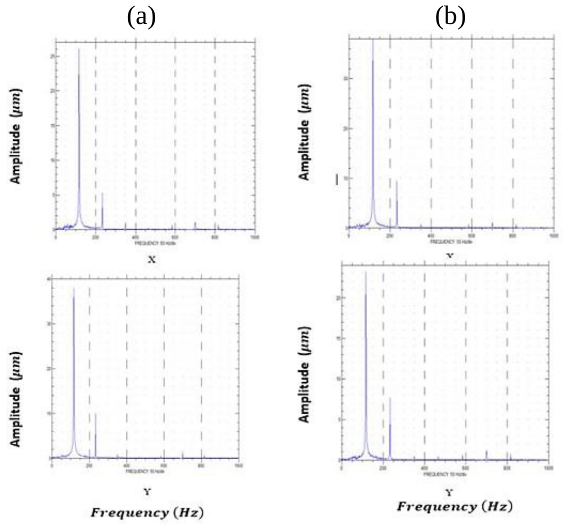

Figure 11: The frequency domain response varies under different unbalance conditions: (a) 20g is removed at a 30° anticlockwise angle, and (b) 10g is removed at a 0° anticlockwise angle.

Table 2 shows the important speeds and the highest vibration levels in different situations, such as before and after the coupling was replaced, and when extra weights of 11 g and 20 g were added. The results demonstrate how structural modifications and increased unbalance influence the system's dynamic behaviour.

Table 2: Critical speed and amplitude for all cases.

Cases	1st Critical	2nd Critical	3rd Critical
Wear in the coupling	113.75 Hz 22.46 um pp	226.25 Hz 5.62 um pp	340 Hz 1.33 um pp
New coupling	117.5 Hz 55.1 um pp	233.75 Hz 11.1 um pp	351.25 Hz 1 um pp
With 10 g	117.5 Hz 39.1 um pp	233.75 Hz 10 um pp	351.25 Hz 0.7 um pp
With 20 g	117.5 Hz 24.35 um pp	233.75 Hz 5.1 um pp	351.25 Hz 0.9 um pp

The second critical speed before modification is 226.25 Hz, with a vibration amplitude of 5.62 µm peak-to-peak (pp). After the improvement, the second critical speed increases to 233.75 Hz, and the vibration amplitude rises to 11.1 µm pp, indicating changes in the system's stiffness and damping characteristics. Under an 11 g imbalance, the second critical speed remains at 233.75 Hz, while the vibration amplitude decreases significantly to 10 µm pp. With a 20 g imbalance, the second critical speed remains unchanged at 233.75 Hz, and the vibration amplitude further decreases to 5.1 µm pp, matching the amplitude observed at the first critical speed.

Figures 12 and 13 illustrate the X and Y dynamic responses of a rotating system under both healthy and worn conditions. In Figure 12, the healthy response (blue) exhibits a peak at a frequency corresponding to an amplitude of approximately 100 µm, indicating system stability. In contrast, the worn response (orange) shows shifts in the frequency spectrum and irregular amplitude increases, signalling potential component wear or damage.

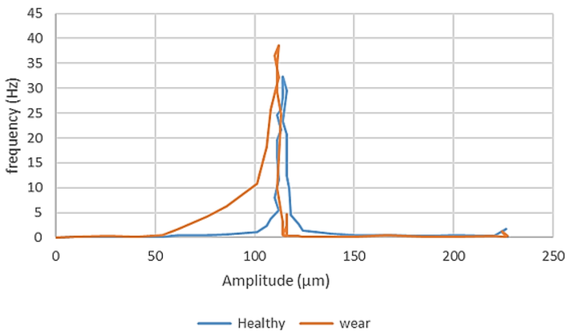

Figure 12: Comparison between the healthy and wear dynamic responses in the directions (X).

In Figure 13, the healthy Y-direction response exhibits a steady relationship between frequency and amplitude, whereas the worn response displays unpredictable movements and a greater number of frequency components, likely due to misalignment, wear, or uneven load distribution. These variations in the frequency spectra indicate the need for regular performance monitoring to detect system faults that could compromise stability and functionality.

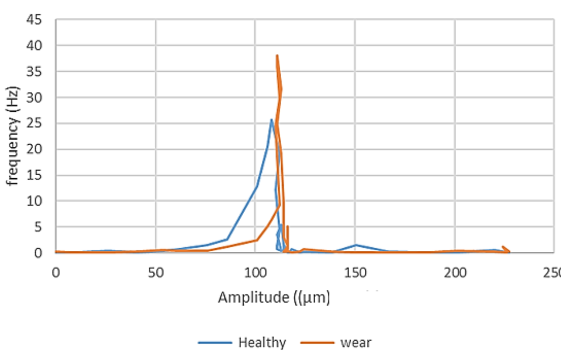

Figure 13: The dynamic response in the directions (Y) is compared between healthy and worn conditions.

4 Conclusions

This study aimed to investigate the effects of coupled erosion-induced mass loss and asymmetric mass distribution on the vibrational behaviour of a rotating shaft system. Experimental results indicated that an increase in missing mass from the coupling significantly increased vibration amplitude. In one experiment, adding 10 g and 20 g masses at 0° and 30° anticlockwise led to a noticeable rise in vibration amplitude. Before modification, the vibration amplitude at 113.75 Hz was 22.46 µm. After adding 20 g of mass, it increased to 24.35 µm at 117.5 Hz. Replacing worn couplings and bearings significantly reduced vibration levels. For example, the first frequency vibration amplitude was 22.46 µm before replacement and increased to 55.1 µm after modification. Following the adjustment, the critical frequency shifted from 113.75 Hz with an amplitude of 22.46 µm to 117.5 Hz with an amplitude of 55.1 µm, indicating improved system stability. These findings demonstrate that mass loss or component wear can substantially affect system dynamics. Therefore, maintaining performance and stability requires replacing worn components. Continuous monitoring and enhanced diagnostics are essential for ensuring system stability and preventing mechanical failure.

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Author Contributions

A.A.N. conceptualised the study, conducted the investigation, and edited the manuscript; E.S.A. handled methodology and drafting; N.A.J. designed the research and analysed the data; A.A.F.O. managed data curation, funding, and project administration; Z.K.H. contributed to the investigation and writing; E.K.N. supervised and reviewed the manuscript. All authors approved the final version.

Conflicts of Interest

The authors declare no conflict of interest.

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