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Bearing Misalignment and Eccentric Wear: A Study on Condition Monitoring

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ABSTRACT

This study investigates ball bearing failures and evaluates the performance of various methods for detecting such failures. Experimental tests were conducted on a motor-generator system supported by ball bearings, where several types of real-time faults were intentionally introduced. Vibration signals were collected to capture the early stages of bearing faults. The monitored vibrations successfully predicted impending failures in the outer race, inner race, and ball of the bearings. To assess the effectiveness of the prediction methods, different stages of fault development were considered. Additionally, the feasibility of implementing these techniques in real-world systems and real-time applications was examined. The vibration monitoring technique proved to be suitable for analysing various defects in bearings and providing early indications of malfunction. Vibration measurements were recorded at each housing of the rotor bearings. The results indicated that a deteriorating or faulty ball bearing had a significant impact on the vibration spectra. To identify different defects in ball bearings, time domain, frequency domain, and Fast Fourier Transform (FFT) analysis were employed. The findings demonstrated the usefulness of each of these techniques in detecting problems associated with ball bearings.

Keywords: *Condition Monitoring, Bearings, Bearing faulty, Vibration analysis*

1. Introduction

Condition monitoring (CM) has gained significant traction in the industrial sector, particularly within engineering departments. Its application not only reduces maintenance costs and operational power but also enhances plant availability. Condition monitoring involves the continuous or periodic assessment of a working plant, equipment component, or structural element while in service. The primary goal of condition monitoring is to facilitate fault detection and prediction. The advantages of employing condition monitoring techniques are manifold. Firstly, it leads to a decrease in maintenance costs. Secondly, it enhances machinery availability, ensuring optimal operational performance. Lastly, it reduces the need for holding excessive spare parts in stock [1].

To identify areas for improvement in machinery availability and maintenance cost reduction, criticality and failure-made analysis techniques are commonly employed. These techniques involve selecting appropriate modes of condition monitoring based on machine criticality and failure modes. Online or offline vibration monitoring and online or offline performance monitoring are some of the condition monitoring modes utilized. The focus is on optimizing the condition monitoring system to achieve objectives effectively and with minimal overall cost. Today, criticality and failure mode analysis also consider total production output and plant efficiency, as these factors significantly impact operational costs and bottom-line profits of large-scale petrochemical and power generation facilities [1].

There are many rotary shafts in machines, and these machines have many problems, such as vibration. The vibration is the movement of the body from a certain point, moving on a particular path, and then returning to the same point at a certain time, and this is called full vibration. However, rotary vibration of the equipment occurs and it is a natural result of anybody that is unbalanced in terms of mass in the centre. The mechanical vibrations often occur on diffraction axes corresponding to objects rotating on the axis of rotation; for example, the erosion of metallic rings rotating about the shafts or damage in the bearing leads to the emergence of vibrations where the outcome of this causes a centrifugal force is trying to make the shaft from its chaos but the bearings installer for the shaft in this case and achieve to affect the force is transmitted to the bearings and causes the vibrations [2].

Many condition monitoring techniques are employed, one of them is vibration analysis. There are deferent methods deal with metallurgical failure, current spectral, and wear debris analysis among the array of condition monitoring techniques utilized. Performance monitoring offers additional perspectives which are not evaluable on through traditional condition monitoring as leakage, wear, and over-firing, [1].

Vibration signals captured on the external surfaces of machinery hold valuable insights into internal operations and the operational state of the machine. In optimal conditions, the vibration frequency spectra of machines showcase distinctive configurations. Nevertheless, with the emergence of faults, these spectra undergo alterations. Profound comprehension of vibration nuances and processes proves indispensable for proficiently configuring, scrutinizing, and calibrating vibrations within mechanical frameworks. The practical facets of vibration, encompassing utilization and design considerations for optimizing vibratory efficacy in mechanical constructions, are of particular focus [3].

Karacay and Akturk study [4] Examines the evolution of specific weaknesses within ball bearings by subjecting a freshly manufactured bearing to operational stress until complete failure occurs. In this experiment, vibrations were generated and various standard metrics were calculated such as the Root Mean Square (RMS) value, Peak-to-Peak value, Period of Frequency value, record high values, and Kurtosis. These scalar parameters provide general information about the vibration characteristics of the ball bearing, but specific details about the location or nature of the defect are not provided by such parameters. To investigate the defect location more precisely, spectrum analyses was conducted at designated points during the experiment. During the test, as the scale of the problem or failure increased, both the severity of the issue and the magnitude of vibrations also increased. However, a consistent relationship between vibration amplitude and failure size was not established. This lack of correlation can be attributed, at least in part, to the inherent variations in vibration characteristics across different systems. It's important to note that the test system in the experiment exhibited relatively severe imbalances in the rotating shaft, as well as misalignment in the ball bearing support and drive belt. Despite these challenges, the analysis successfully identified localized failure development in the ball bearing.

A study conducted by Maru et al. [5] investigated the impact of different viscosity oils on the vibration response of roller bearings. The researchers focused on two frequency bands and utilized a tribological parameter called the "l factor" to assess vibration behaviour. The study revealed several key findings. Firstly, the authors were able to detect changes in the lubrication regime of roller bearings by monitoring vibrations when using oils of different viscosity grades. Specifically, tests with ISO 32 and ISO 68 viscosity grades indicated a full film lubrication regime, while the ISO 10 grade suggested a lubrication regime close to a mixed type.

Moreover, the study found that variations in oil viscosity within the roller bearings, whether caused by using different oils or temperature fluctuations, only influenced bearing vibration in the high-frequency (HF) band ranging from 600 to 10,000 Hz. Overall, the study demonstrated that viscosity variations have a notable impact on vibration test results. In contrast, comparable trends are observed in the total RMS values of high-frequency bands. This dataset provides potential for further insights. Lower oil viscosity (V1) demonstrates a notable impact on vibration levels. Subsequent performance assessment indicated that alterations in oil viscosity, variations in ISO viscosity grades, or temperature fluctuations induced by high-frequency bands can influence the vibration characteristics of rolling bearings. [5].

In their research on rolling element bearing diagnostics, Smith and Randall [6] emphasize the critical role of rolling element bearings in nearly all rotating machines. They highlight that failures in these bearings can lead to adverse consequences, including personal injuries and financial losses, particularly when faults go undetected and undiagnosed. Consequently, the field of vibration analysis places significant emphasis on investigating bearing failures, as they present an opportunity for significant benefits through early fault detection. In 1970, the introduction of Resonance Demodulation presented a fault diagnostic method for detecting rolling bearing damage. Following this development, SPM company pioneered tools that relied on nervous system responses to identify errors in measuring the resonance frequency using accelerometers, particularly for detecting rolling element bearing damage. Challenges arise in the initial scenario concerning the efficient demodulation of the echo signal to differentiate the fault signal from the surrounding background noise. Achieving optimal separation poses a significant issue. In a different case, utilizing the resonant frequency (approximately 33 kHz) as a carrier for the acceleration sensor is not always the most advantageous approach. This is particularly true when dealing with substantial machines and significant defects that induce resonances [6].

This study aims to investigate bearing faults and their impacts on vibration measurements for the monitoring of bearing faults. The research involves creating faulty bearings for testing on a rig, conducting vibration measurements to analyse bearing faults, and comparing these results with those obtained from standard bearings.

2. Materials and Methods

The test rig is conceived to explore the vibration-induced failures and unbalance behaviors in several types of bearings and rotor shafts. It comprises a motor generating approximately 2/8 HP and running at 1500 RPM using an AC power supply mounted on an iron structure, propelling a pair of flywheels in parallel alignment with their axes. These flywheels feature multiple perforations for adjusting mass distribution to induce imbalance, and they are affixed to a shaft via ball bearings (Figure 1). The electric motor is linked to a drive belt to harmonize motion and ensure the transmission of power across the entire assembly. The rig has a cover for operator safety (Table 1).

To monitor vibrations within the bearings, two accelerometer sensors are positioned vertically within the housing of the ball bearings. These sensors are wired to a data acquisition system, which interfaces with a computer running specialized software (YE7600).

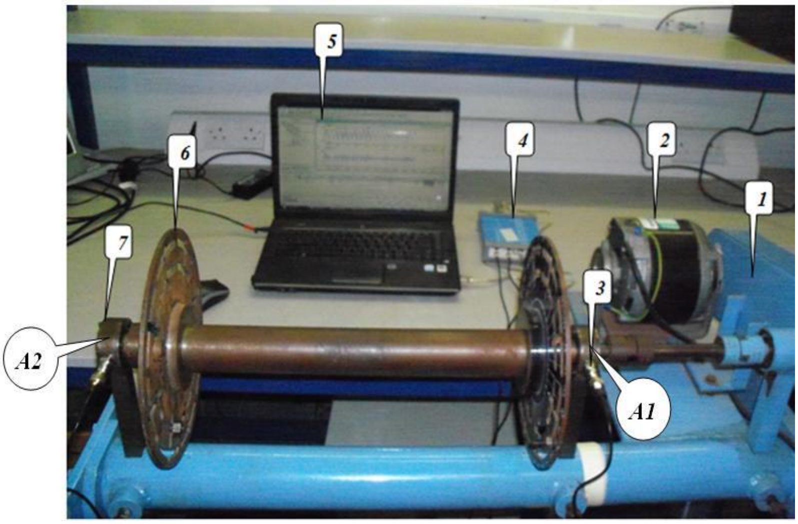

Figure 1: Experiment test rig

Table 1. Description of experiment test rig individual components

No.	Description
1	Belt and coupling to transfer speed between motor and drive shaft
2	Electric motor: RANCO, 220-240v,1/8 hp, 1,500rpm
3	Piezoelectric accelerometer: CA-YD-185TNC 50mV/Gtnc
4	Data acquisition: Sinocera Dynamic, YE6231
5	Laptop computer: HP
6	Flywheel: Angle scale with the shaft 0-360°
7	Bearing house
A1	Bearing with defects at bearing house one (A1) at sensor one
A2	Good bearing at bearing house two (A1) at sensor two

The experiment selected the single-row ball bearing, which comprises a solid outer ring, balls, an inner ring, and a cage or separator. (Figure 2). The primary emphasis of this specific research investigation centered on the behavior of the ball bearing assembly when subjected to a distributed radial load exerted on the inner surface or ring of the outer ring. Figure 3 and Table 2 provide the specific dimensions of the ball bearing used in this investigation. The analysis presented here assumes an ideal scenario where there is no sliding and the contact angle is equivalent to 0°.

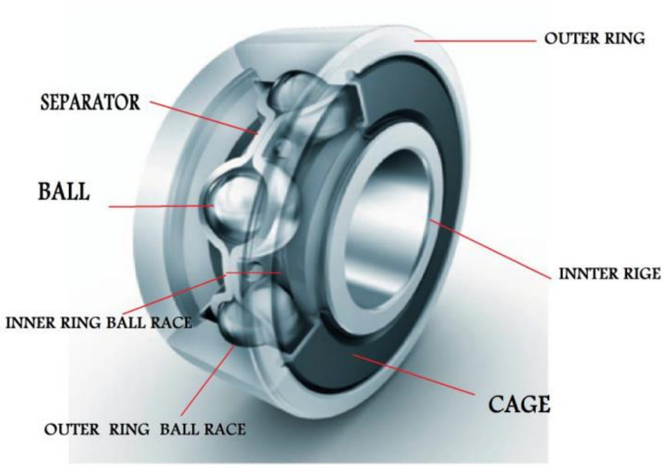

Figure 2: Ball bearing parts [7]

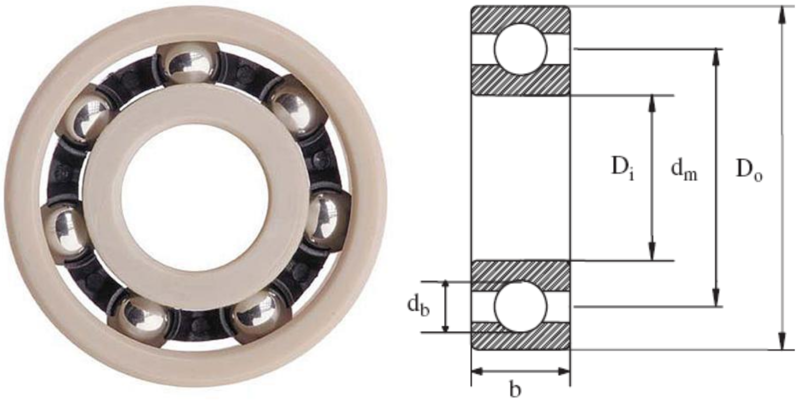

Figure 3: Ball bearing dimension [7]

Table 2: Dimension of ball bearing

Bearing parts	Dimension (mm)
Inner bore diameter D_i	17
Outer ring outside diameter D_o	40
Pitch diameter dm	29.31
Ball diameter d_b	6.7
Width b	12
Mass	60 g
Number of balls N_b	8 balls
Contact angle α	0°

A localized defect may manifest on various components within a rolling element bearing, depending on the prevailing working conditions. The equations and calculations for each unique failure scenario, each with its respective occurrence frequency and corresponding computed values, are presented below:
Standard frequency (healthy) is calculated for each part of the ball bearing: [8]
Shaft frequency (f_A)

$$f_A = \frac{shaft\ speed\ rpm}{60}$$

$$= \frac{1{,}500}{60}$$

$$= 25\ Hz$$

Then cage frequency (f_c) is calculated:

$$f_c = \frac{1}{2}\ f_A \left[1 - \frac{d_b}{d_m}\ \cos\alpha\right]$$

Where d_b is the ball diameter, d_m is the pitch diameter and α is the contact angle.

$$= \frac{25}{2}\left[1 - \frac{6.7}{20.31} \times \cos 0\right]$$

$$= 9.65\ Hz$$

The outer race frequency (f_o) is calculated:

$$f_o = \frac{N_b}{2}\ f_A \left[1 - \frac{d_b}{d_m}\ \cos\alpha\right]$$

Where N_b is the number of bearing balls.

$$= \frac{8}{2} \times 25 \left[1 - \frac{6.7}{20.31} \times \cos 0\right]$$

$$= 77.2\ Hz$$

The inner race frequency (f_i) is calculated:

$$f_i = N_b[f_A - f_c]$$

$$= 8[25 - 9.65]$$

$$= 110\ Hz$$

The roller elements frequency (f_r) is calculated:

$$f_r = \frac{d_m}{2d_b}\ f_A \left[1 - \left(\frac{d_m}{2d_b}\right)^2 cos^2\ 0\right]$$

$$= \frac{29.315}{2 \times 6.7} \times 25 \left[1 - \left(\frac{6.7}{29.31}\right)^2 cos^2\ 0\right]$$

= 51.83 Hz

The experiment was segmented into four stages. The initial phase focused on the data acquisition setup, comprising components like data acquisition hardware, a piezoelectric accelerometer, signal filtering mechanisms, and a laptop computer. The investigation targeted frequencies approximately at 20,000 Hz, facilitating the capture of vibration signals ranging up to 10,000 Hz. This frequency range was deemed sufficient for detecting faults in ball bearings, given that the failure frequency did not surpass 3,000 Hz under the test conditions.

In the second phase, the focus shifted to simulating various ball bearing faults. This involved placing a problematic bearing in bearing house A1 at sensor one and maintaining the second bearing house (sensor A2) with a fault-free bearing. This procedure was systematically repeated using different types of bearing failures such as cracks, misalignment, corrosion, cage damage, and normal fatigue. The motor and shaft undergo rotation along the axis of the shaft, which is upheld by bearings located at both ends. Vibration can manifest as a force exerted either radially or axially, and subsequent signal analysis is conducted.

The third phase leaved the bearing house one (A), initially identified as a bearing in good condition, and repeating the same procedure with the second bearing house (A2), where a bearing with a distinct failure was installed and assessed using the machine system test. Subsequently, faulty ball bearings were installed in both bearing houses, and the signals were collected for analysis. To ensure the proper functioning of the experimental setup, data from the healthy bearings were gathered after running the system approximately three times. The healthy data in the time domain (Figure 4) reveals RMS and peak-to-peak values ranging

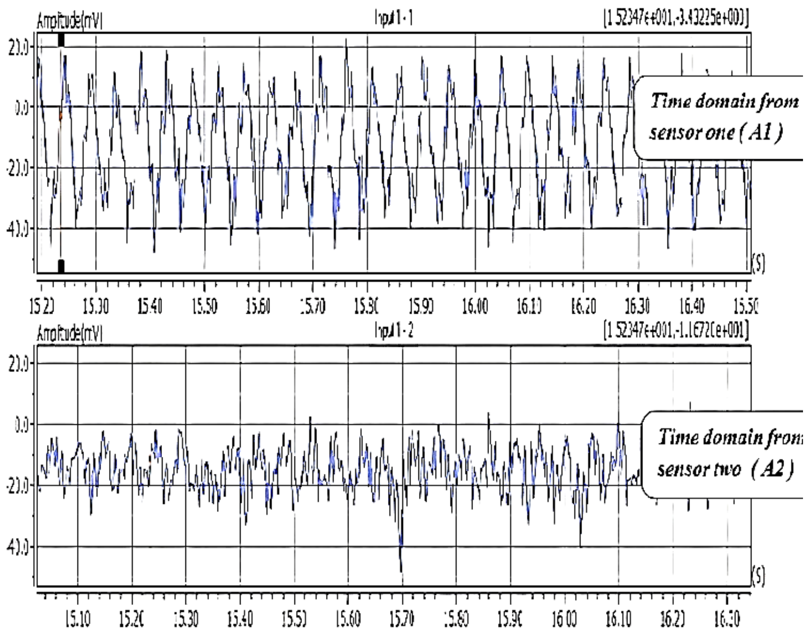

Figure 4: Time domain healthy data

approximately from 20 mV to -45 mV. Additionally, the time domain graph illustrates consistent vibration patterns over a 60-second duration, demonstrating signals from sensors one and two.

In the presented data, the frequency domain analysis (Figure 5) highlights the measured values associated with bearing failure. Specifically, the outer ring failure exhibits a frequency of 110 Hz with an amplitude of 4.22 mV, while the inner ring failure manifests at 77.5 Hz, indicating normal conditions devoid of any vibrations originating from the ball bearing supporting the shaft at locations A1 and A2.

Moreover, the frequency domain analysis reveals distinct regions of concern within the system. These include the shaft failure zone (S1) and areas pertaining to ball bearing faults, denoted as B1 for ball malfunction, B2 for inner race failure, and B3 for outer race failure (Figure 6).

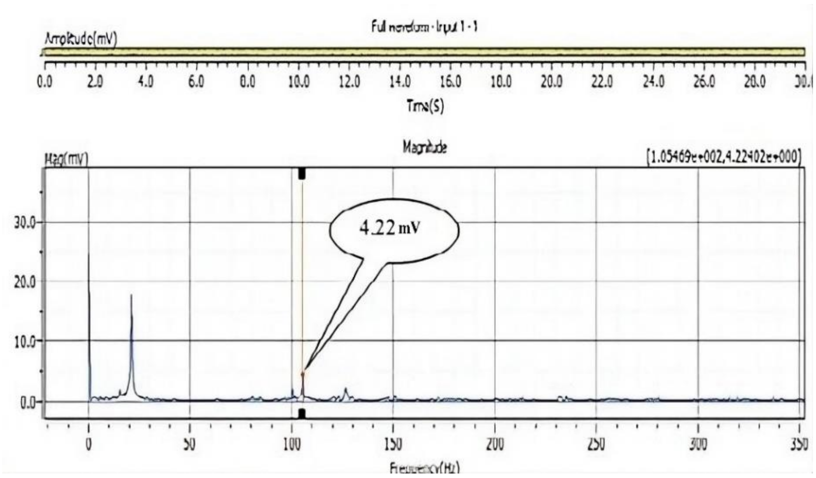

Figure 5: Signal of frequency domain (A1)

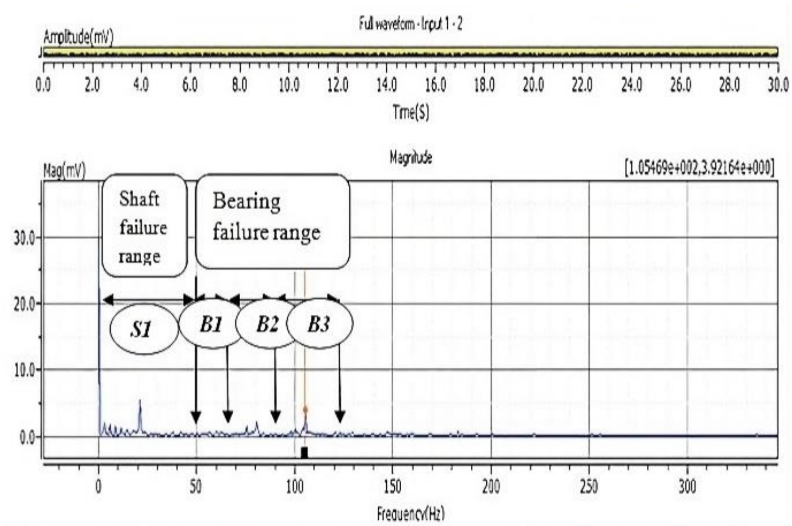

Figure 6: Signal of frequency domain signal (A2)

3. Results and Discussion

Case 1: Crack on outer ring

The initial issue pertains to a crack - it can be induced in the outer race through theapplication of a hammer. Analysis of the time domain signal from the first and second sensors (Figure 7) indicates fluctuating vibration spectrum or amplitude, with an observed increase in peak-to-peak values compared to the data from undamaged components, reaching approximately ±75 in amplitude.

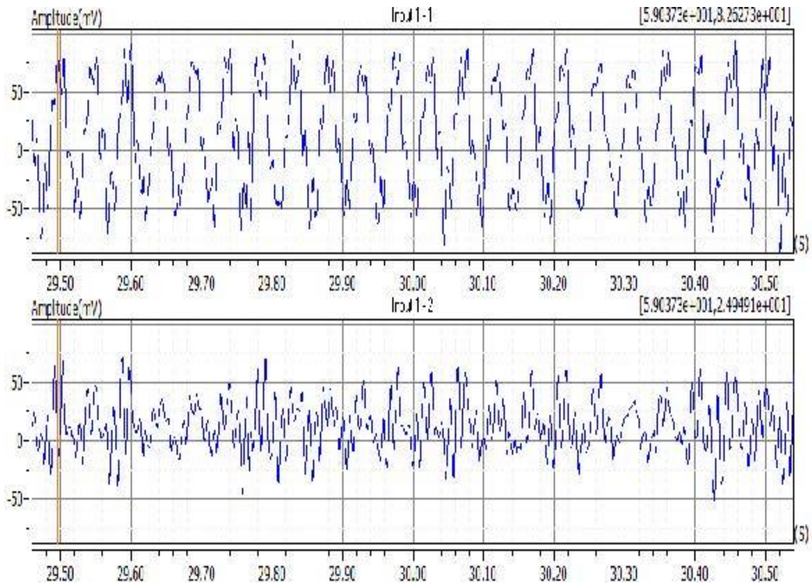

Figure 7: Time domain signal for case 1

The absence of time domain data in the vibration analysis signals necessitates the utilization of the frequency domain. In this context, an anomaly in the ball bearing is evident, specifically a shift in the outer race at approximately 110 Hz, indicating a fault within the bearing housed in location A1 (Figure 8). Furthermore, a slight deviation in the vibration signal from the bearing housed in location A2 suggests

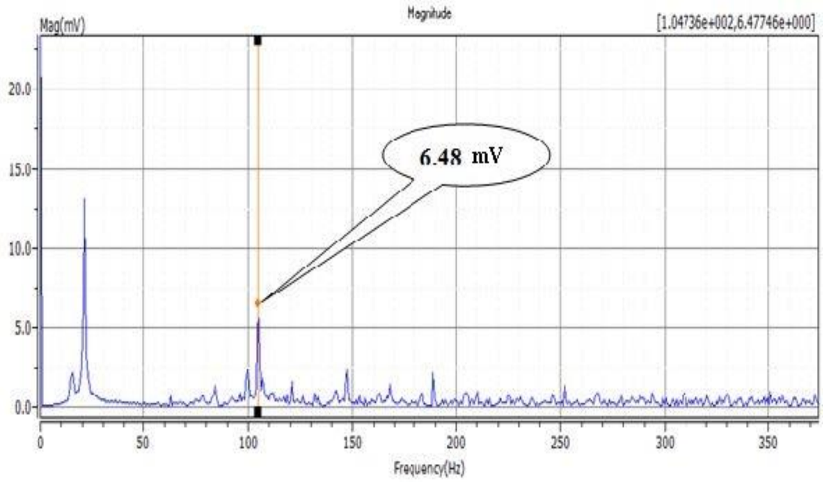

Figure 8: Signal of frequency domain (case 1)

Case 2: Corrosion on the outer ring

its susceptibility to the vibrations originating from the faulty bearing at location A1. A comparison between the vibration magnitude in the healthy dataset (4.22 mV) and the vibration amplitude at the outer race (6.48 mV) elucidates that the system containing the faulty ball bearing exhibits higher vibration levels.

In instances where machinery or equipment remains inactive for prolonged periods, corrosion may develop, leading to failures. In this investigation, a ball bearing experiencing corrosion-related failure was installed in bearing housing A1 of a test machine. Signals were then captured from sensors. Analysis of the time domain signal (Figure 9) revealed irregular vibrations, with peak-to-peak amplitude ranging approximately from +45 to -20. Interestingly, the vibrations observed in the second sensor (A2) were notably smaller in magnitude.

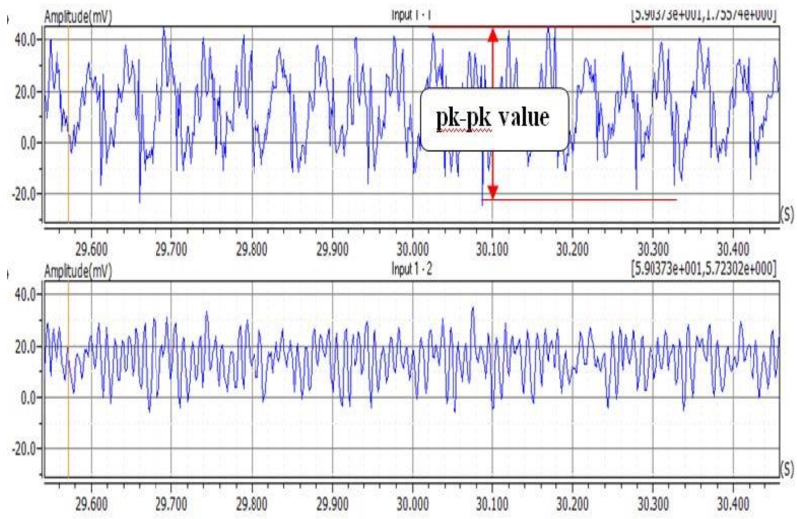

Figure 9: Signal of Time domain (case 2)

The data indicates that the frequency associated with ball bearing failure is approximately 104 Hz with an amplitude of 6.46 mV (Figure 10). A comparison

(fig 11)

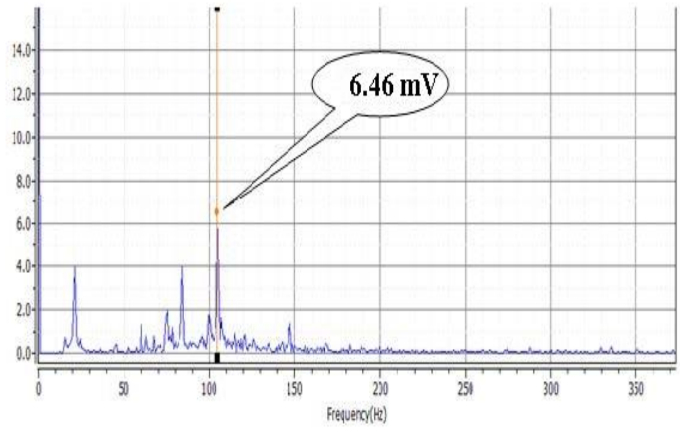

Figure 10: Signal of frequency domain at bearing house A1 (case 2)

With the frequency of the outer race in a healthy state, which stands at 110 Hz, confirms that the bearing within housing A1 is experiencing outer ring failure.

The frequency domain signal demonstrates a slight alteration in the vibration pattern within bearing location A2; however, it is increasingly impacted by the system's vibration, leading to an imminent failure due to radial forces in a short timeframe (Figure 11).

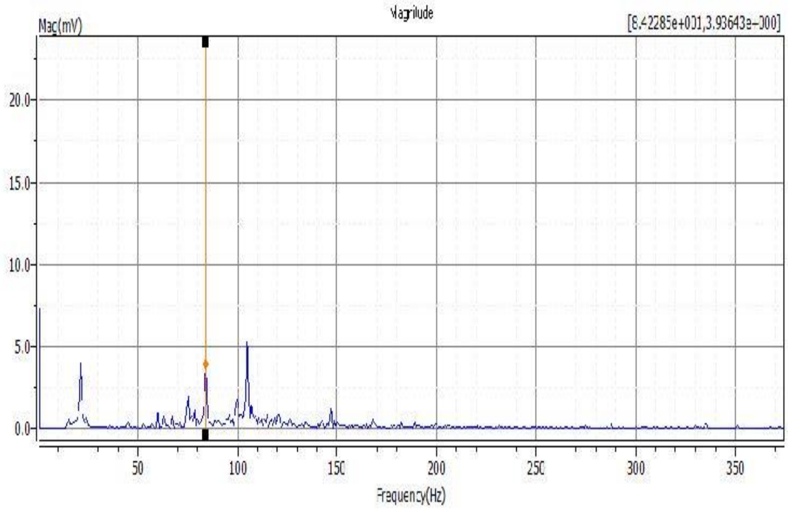

Figure 11: Frequency domain signal at bearing house (A2) for case 2

Case 3: Presence of misalignment of the outer ring and inner race

Misalignment failure between the outer race, inner race, and the ball in the outer ring leads to misalignment. This occurs when the shaft contacts the bearing incorrectly. A noticeable increase in vibration from healthy data (4.22 mV) to failure data (6.41 mV) demonstrates elevated levels in the presence of misalignment. The time domain signal shows the random vibration’s amplitude with a range of peak-to-peak of ±55 mV, however; the time domain analysis is limited in its ability to identify intricate failures; it primarily validates the presence of vibration within the system. (Figure 12).

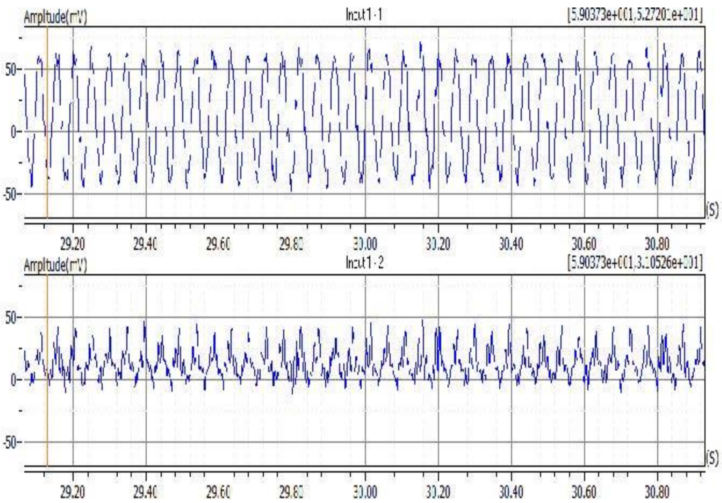

Figure 12: Signal of time domain (case 3)

The frequency domain analysis of the signal reveals a distinct fault in the bearing, with a clear indication of the failure location main. Specifically, the spectrum identifies the fault positions as follows: the outer race failure is located at 107 Hz with a magnitude of 6.41 mV, while the inner race fault is situated at 75 Hz with a magnitude of 3.8 mV (Figure 13).

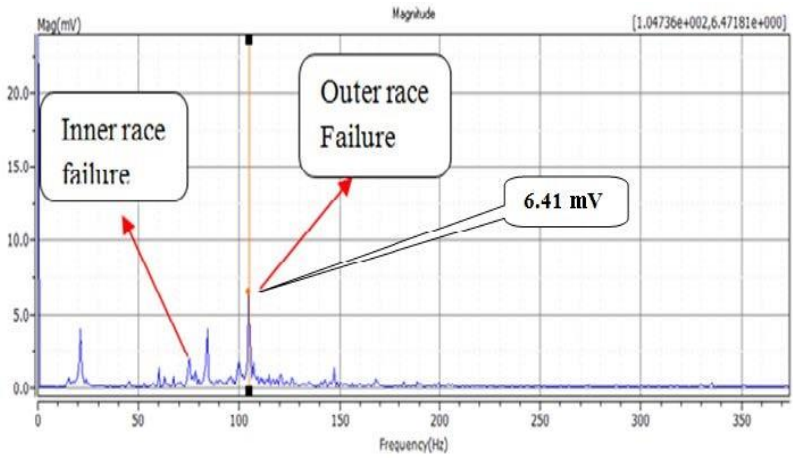

Figure 13: signal of frequency domain (case 3)

Case 4: Cage failure

The failure happens when a bearing is installed through hammering or when a weak material is used for the cage. In the time domain representation, the spectrum appears abnormal, indicating a high vibration signal within the system. However, the origin of this vibration signal is not clearly discernible in the time domain data (Figure 14).

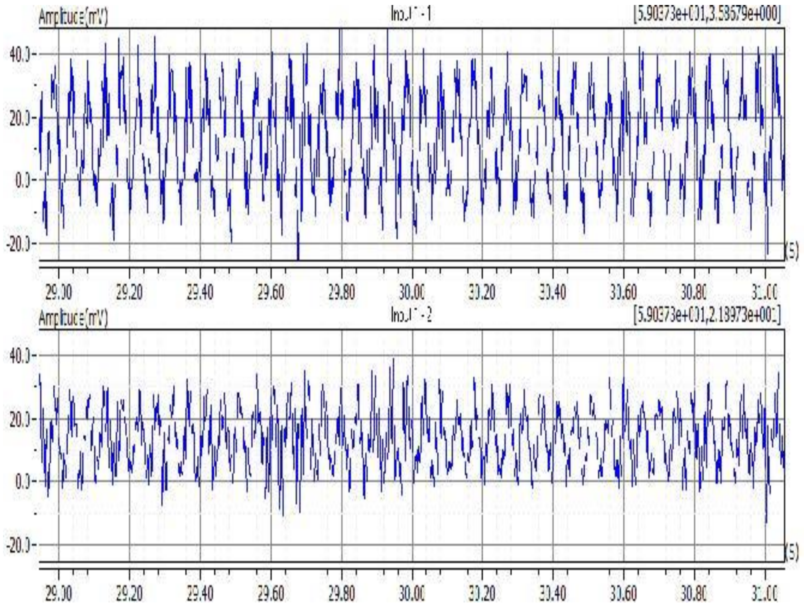

Figure 14: signal of time domain (case 4)

The frequency domain analysis provides a distinct finding that identifies the source of vibration as originating from the bearing, particularly from both the outer and inner race of the bearing (Figure 15). Additionally, in the alternate bearing housing, an issue is detected in the outer race of the bearing, leading to an increase in shaft vibration levels to 14.8 mV and 6.57 mV for the shaft and outer ring, respectively (Figure 16).

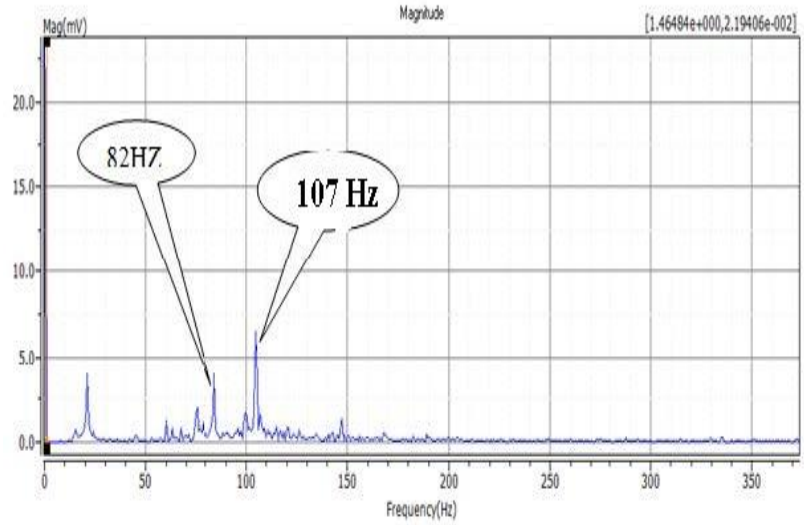

Figure 15: Signal of Frequency domain (case 4)

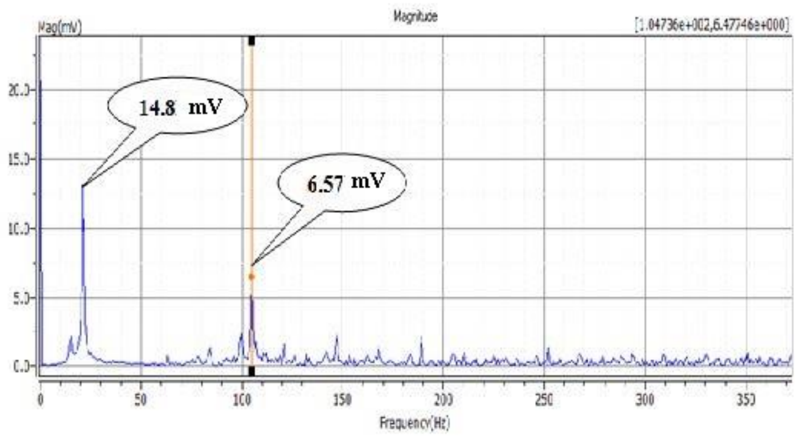

Figure 16: Signal of frequency domain at bearing house A2 (case 4)

Case 5: Inner ring typical fatigue

The issue of abnormal fatigue occurring at the inner ring is observed when the angular contact deviates from 0° and when the bearing exceeds the standard operational hours. Consequently, the bearing undergoes physical changes, resulting in the formation of small chips that accumulate between the ball and the inner race. This situation is indicated by a significant vibration signal in the equipment in the time domain, and those vibrations in both graphs are random vibration. The peak-to-peak amplitude is +59 to -57 (Figure 17).

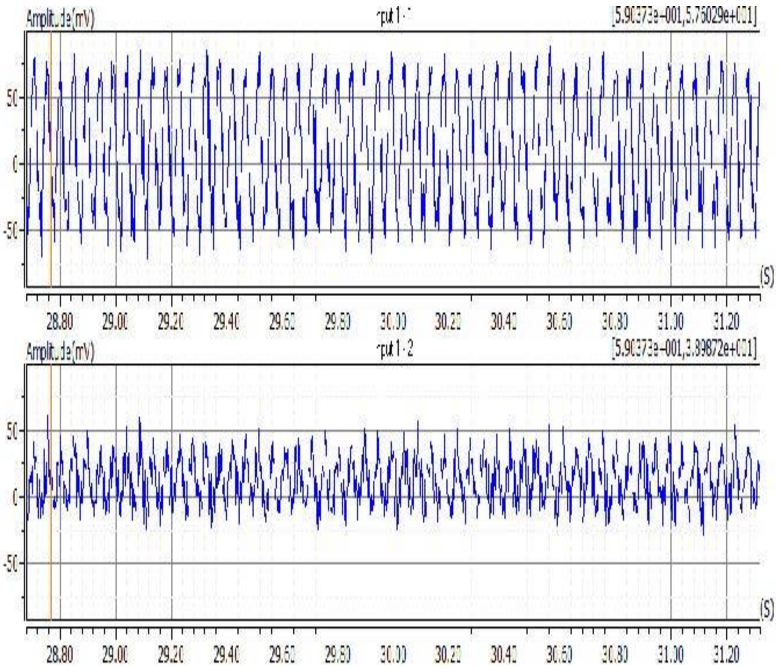

Figure 17: Signal of Time domain (case 5)

On the other hand, the frequency domain signal precisely indicates the specific point that caused this vibration in the inner race at a particular value 84 Hz, and magnitude 3.9 mV (Figure 18).

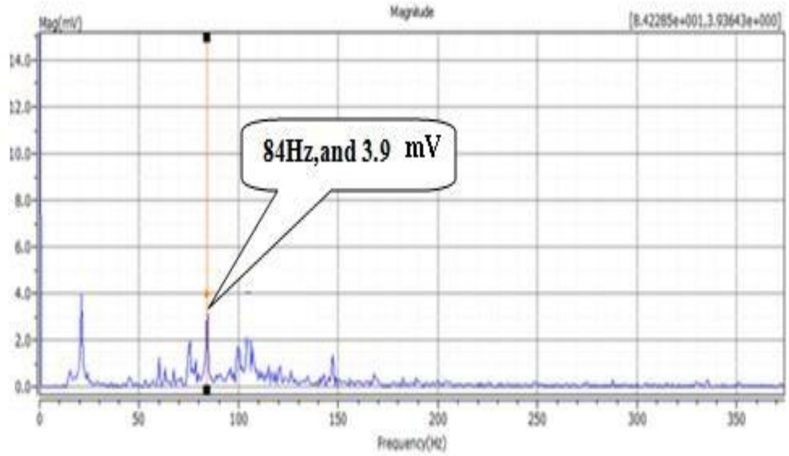

Figure 18: Signal of frequency domain signal (case 5)

During the stages of the time domain analysis, it becomes challenging to identify defects or obtain a clear signal in a ball element bearing. The presence of defects in the bearing may increase as the operation continues. To explore this further, this study investigates how the number of defects affects the time-domain parameters by introducing up to four defects on the outer ring. By incrementally increasing the number of defects on the outer ring, we aim to observe the influence of defect quantity on the time domain characteristics.

Another commonly employed approach involves processing vibration signals in the frequency domain. A key aspect of this analysis is identifying the typical defect frequencies exhibited in the frequency domain. These frequencies depend on the rotational speed and the specific location of the defect within the bearing. The presence of one or more of these defect frequencies in the direct or processed frequency spectrum serves as a primary indicator of a fault.

Vibration is a prevalent characteristic observed in various industrial equipment. When vibration levels exceed normal thresholds, it can indicate normal wear or necessitate further investigation into its underlying causes, leading to potential maintenance actions. This study focuses on the condition monitoring of rolling element bearings, aiming to contribute insights into assessing vibration levels and addressing maintenance needs.

Real failures have occurred in five models of single-row ball bearings, including issues such as cracks, corrosion, misalignment, cage damage failure, and normal fatigue. A model for predicting ball bearing failures was formulated, focusing on utilizing experimental data from accelerometers. The key contribution of this project is the ability to precisely foresee the remaining useful life of ball bearings based solely on measured accelerometer data. The analysis utilized bearing data obtained from the YE 7600 software, which conducted five run-to-failure tests under normal load conditions. Accelerometers were strategically placed in a horizontal direction on the bearing house to capture data. The tests revealed failures in the outer race, inner race, and ball and cage components of the ball bearing. Consequently, the tool created in this investigation underwent testing with experimental data derived from these trials to confirm its efficiency in detecting bearing failures and predicting remaining useful life.

The developed tool incorporates a signal of frequencies, enabling it to compute the frequencies of bearing faults and various trend condition metrics encompassing both time and frequency domains of the malfunctioning frequency. In the case of breakdown of the outer race, inner race, or ball, the frequency associated with the respective failure mode was selected for prediction of lifespan. The prediction of remaining useful life relied on tracking the progression of a particular degradation indicator, which was recognized as the spectral vibration level corresponding to the faulty frequency. Subsequently, the mean value derived from these frequency values corresponding to ball bearing failures was calculated, forming the degradation signal trend over time.

4. Conclusions

Ball bearings can be effectively analysed for defects by examining their frequency and vibration spectrum. By observing trends in the spectrum, one can determine the intensity of vibrations in bearings that are defective. Additionally, the vibration domain spectrum allows for the identification of amplitudes associated with specific defect frequencies, facilitating the prediction of defects in both the inner race and outer race of roller bearings. The time domain can illustrate the system's vibration using the RMS value and the peak-to-peak value. Conversely, the frequency domain is particularly effective for visualizing and detecting vibrations within complex signals, enabling their identification. The identification of issues in ball bearings is facilitated by the unique and distinct behaviors exhibited by vibration signals in cases of failure in the inner race, outer race, and ball components.

To further investigate ball bearings failure, several important considerations should be taken into account. Firstly, it is essential to create controlled failures at various stages in order for the purpose of contrasting various techniques and assessing their efficacy in spectral analysis. Additionally, expanding the comparison to include other vibration analysis techniques would be valuable for comprehensive understanding and assessment.

Secondly, investigate a variety of bearings, including Journal bearings, and conduct comparative analyses with alternative methods. Utilize a multi-speed motor to assess bearing vibrations across varying speeds. Lastly, apply these methodologies within operational systems like multi-stage pumps and small turbines instead of relying solely on test benches.

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