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# AN EXPERIMENTAL INVESTIGATION ON THE EFFECT OF INNER RACE DEFECT ON ROLLING ELEMENT BEARINGS

**Author(s):**
Mohsin Albdery[1,2], István Szabó[1]

**Affiliation:**
[1] Institute of Technology, Hungarian University of Agriculture and Life Sciences, 2100 Gödöllő, Páter Károly u. 1., Hungary;
[2] Doctoral School of Mechanical Engineering, Institute of Technology, Hungarian University of Agriculture and Life Sciences, 2100 Gödöllő, Páter Károly u. 1., Hungary.

**Email address:**
mohsinalbdery@gmail.com; Szabo.Istvan.prof@uni-mate.hu

**Abstract:** This article investigates the detection and monitoring of bearing defects in rotating machinery using vibration analysis techniques. The study employs a specially designed test bench set up at the Institute of Technology, Hungarian University of Agriculture and Life Sciences. The test bench incorporates an asynchronous motor and uses the SKF 1209 EKTN9 bearing model for analysis. Vibration data are captured using the SKF Micrologdevice equipped with accelerometer sensors. The study emphasizes the importance of early detection to prevent costly breakdowns and enhance operational efficiency. Experimental investigations reveal that as Inner Race Defects (IRD) severity increases, the vibration amplitude also rises, indicating a direct relationship between IRD severity, bearing damage, and vibration characteristics. The results demonstrate that higher IRD severity levels and rotation speeds increase vibration amplitude, providing valuable insights for predictive maintenance models. Vibration measurement techniques, such as analysing the vibration amplitude using gE True peak values, are explored as reliable methods for assessing the health status of rolling element bearings. Experimental investigations examined the effect of inner race defects (IRD) on bearing vibration. The results demonstrated a direct relationship between IRD severity, bearing damage, and vibration characteristics, with higher severity and rotation speeds leading to increased vibration amplitudes.

**Keywords:** Inner race defect; Rolling element bearings; Bearing defects; Fault detection; Vibration measurement

## 1. Introduction

Due to its cost efficiency, performance, and robustness, rotating machinery has become a staple in industrial settings. However, these machines often operate under strenuous conditions, including high load, speed, and limited lubrication. The rolling element bearing is the most susceptible component to wear and tear in these machines. Operating under harsh and potentially dangerous conditions often leads to component failure, posing a risk to worker safety and leading to financial losses. In fact, bearing failures account for over 42% of mechanical failures [1].

As Heng et al. pointed out, bearing failures are the primary cause of mechanical breakdowns, leading to an upsurge in warranty and maintenance costs. In some cases, these failures can even precipitate a total machinery shutdown. [2] Given the implications, the diagnosis of rolling element bearings health has gained significant attention, especially with the advent of techniques such as machine learning. The vibration signature of these bearings often provides early indicators of a problem. The main causes of rolling element bearing failures are imbalance shaft faults, ball bearing defects, inner, outer, and cage faults. The field of defect diagnosis in rolling element bearings has recently undergone substantial advancement, with vibration techniques such as time domain, frequency domain, time-frequency domain, shock pulse method, and acoustic emission technique playing a significant role. Numerous studies on vibration signal analysis techniques have been conducted, and numerous reviews have been made to categorise defect and fault diagnosis techniques [3]. The ability to accurately predict a machine or component's lifespan and potential failures through signal levels is a crucial focus of this research. The goal is to extend the life of the machinery by identifying faults at early stages, enabling the implementation of an effective maintenance schedule for corrective measures [4].

Additionally, the frequency domain technique provides valuable data to pinpoint the exact location of a fault in rolling element bearings. This is achieved by analysing the vibration peaks at characteristic frequencies of the bearings, allowing easy detection of defects [5]. It has been concluded from a study of various literature that for accurate fault analysis and condition monitoring, it is more beneficial to use a combination of condition monitoring techniques along with vibration analysis.

The health of rolling element bearings can be swiftly determined through vibration monitoring, as it uncovers crucial data about the early onset of faults. Given the serious implications of such faults, it's essential to improve our understanding of early bearing defects through these techniques [6]. The process begins with data acquisition from the rotating machinery system via sensors. Following this, signal pre-processing and feature extraction are performed to reduce raw data's dimensionality and glean valuable information from the signal. The efficacy of machine fault diagnosis heavily relies on the right feature extraction and selection techniques. The design of this device caters to the testing of a diverse range of mechanical drives, clutches, and rotating elements. The test bench's grooved table affords an array of possibilities for positioning both the drive and driven units. Throughout the measurement process, all drive parameters can be fixed using a data collector and accurately defined through a programmable logic controller (PLC), as depicted in figures 1 and 2. For an accurate interpretation of the bearing's condition, measuring and analysing signals from the bearing is necessary. This requires instrumenting the bearing with a suitable accelerometer sensor to measure the mechanical parameter that is manifested as a signal from the bearing [7].

## 2. Vibration Monitoring Techniques

### *2.1. Frequency Domain Data Analysis*

Frequency domain analysis refers to the study or visualisation of vibration data in relation to frequency [8]. To transform the time domain vibration signal to the frequency domain, a Fourier transform is typically used, frequently in the form of a fast Fourier transform (FFT) algorithm. Frequency domain analysis refers to the study or visualization of vibration data in relation to frequency. In the case of rolling element bearings, we have four types of frequency measurements as follows:

**A.** Ball Pass Frequency Outer Race (BPFO): BPFO signifies the basic vibration frequency generated when the ball passes over a defect, such as a crack, in the bearing's outer race [9]. The frequency of ball passes in the outer race can be computed as per the provided equation (1):

$$\text{BPFO} = \left(\frac{N}{60} \times \frac{n}{2}\right) \times \left(1 - \frac{d}{D}\cos\,\theta\right) \tag{1}$$

When D denotes the pitch diameter, d represents the ball diameter, N indicates the shaft rotation speed in revolutions per minute, n is the number of balls, and θ is the contact angle.

**B.** Ball Pass Frequency Inner Race (BPFI): BPFI is the vibration frequency induced by the ball passing over a defect, such as a crack, in the bearing's inner race. The frequency of ball passes in the inner race can be calculated using the provided equation (2) [10]. The frequency of ball passes in the inner race can be calculated as:

$$\text{BPFI} = \left(\frac{N}{60} \times \frac{n}{2}\right) \times \left(1 + \frac{d}{D}\cos\,\theta\right) \tag{2}$$

When D denotes the pitch diameter, d represents the ball diameter, N indicates the shaft rotation speed in revolutions per minute, n is the number of balls, and θ is the contact angle.

**C.** Ball Spin Frequency (BSF): BSF refers to the rate of repetition of pulses each time a defective roller or ball passes over other elements of the rolling element bearing. The Ball Spin Frequency (BSF) can be calculated using the provided equation (3) [11]. The Ball Spin Frequency (BSF) can be calculated as follows:

$$\mathrm{BSF} = \frac{N}{60} \times \frac{D}{d} \times \left(1 - \left(\frac{d}{D}\cos\,\theta\right)^2\right) \tag{3}$$

When D denotes the pitch diameter, d represents the ball diameter, N indicates the shaft rotation speed in revolutions per minute, n is the number of balls, and θ is the contact angle.

**D.** Fundamental Train Frequency (FTF): FTF denotes a defect that occurs in the cage of a rolling element bearing. The Fundamental Train Frequency (FTF) can be calculated using the provided equation (4) [12]:

$$\mathrm{FTF} = \left(\frac{N}{60} \times \frac{1}{2}\right) \times \left(1 - \frac{d}{D}\cos\,\theta\right) \tag{4}$$

When D denotes the pitch diameter, d represents the ball diameter, N indicates the shaft rotation speed in revolutions per minute, n is the number of balls, and θ is the contact angle.

The contact angle ($\theta$) in the context of rolling element bearings is a critical geometric and mechanical parameter. It represents the angle between the line of contact formed by the rolling element (usually a ball or a roller) and the raceways, and a plane perpendicular to the bearing axis.

*Parameters Definition:*

*D*: Pitch Diameter

The diameter of an imaginary circle that runs through the center of the balls when they are in contact with the races. It is a critical geometric parameter of the bearing. Usually measured in millimeters (mm) or inches (in).

*d*: Ball Diameter

The diameter of the individual rolling element, typically a ball. Also usually measured in millimeters (mm) or inches (in).

*N*: Shaft Rotation Speed

The speed at which the shaft of the machinery, to which the bearing is attached, is rotating. It is measured in revolutions per minute (RPM).

*n*: Number of Balls

The total number of rolling elements (balls) in the bearing assembly.

*θ*: Contact Angle

The angle at which the rolling elements (balls) make contact with the races. It's usually expressed in degrees (∘).

1/60 and 2*n*/2 Conversion Factors

1/60 is a conversion factor to change RPM to revolutions per second.

2*n*/2 signifies the number of times the ball makes contact with either race within a single revolution. It's half the number of balls because each ball contacts either the inner or outer race once per revolution.

### *2.2. Time-Frequency-Domain Technique*

The time domain approach, as described by Tse et al., is utilised to visualise or scrutinise vibration data with respect to time. [13] Several time-frequency domain methods have been developed which show potential for detecting and diagnosing bearing issues in some of the more complex rotating machines. This is especially when the ratio of noise level to vibration signal is low and numerous frequency components are present. The time-frequency technique can display the frequency segments of a vibration signal and differentiate their time variation characteristics. Techniques in the time-frequency domain can handle both non-stationary and stationary vibration signals. This is the main advantage of time-domain techniques over frequency-domain techniques.

## 3. Methodology

The primary objective of this research is to investigate the effect of inner race defects (IRD) on the vibration characteristics of rolling element bearings. The study aims to provide a comprehensive understanding of how different sizes of IRD influence the vibration amplitude at various rotational speeds. The motivation behind this research stems from the critical role that rolling element bearings play in industrial machinery. Early detection of bearing defects can prevent costly breakdowns, improve operational efficiency, and ensure worker safety. Understanding the relationship between IRD and vibration characteristics can contribute to the development of predictive maintenance models.

The methodology involves the following steps:

1. Experimental Setup: A test bench replicating industrial conditions is set up, featuring an asynchronous motor and SKF 1209 EKTN9 bearings. The setup is instrumented with vibration measurement tools like the SKF Microlog device.
2. Defect Creation: Controlled defects of sizes 0.5 mm, 1 mm, and 2 mm are created on the inner race of the bearings using an Electrical Discharge Machine (EDM).
3. Alignment: The Fixturlaser XA system is used to ensure proper alignment of the motor and bearing assembly.
4. Vibration Measurement: Vibration data is collected using accelerometers attached to the bearing housing. The data is then analyzed using MATLAB software.
5. Data Analysis: The vibration data is analyzed in both time and frequency domains to understand the effect of IRD on vibration characteristics.
6. Result Interpretation: The gE True peak values are used to quantify the vibration levels for bearings with different sizes of IRD at various speeds.

Creating defects on the inner race of a bearing is a precise task that requires strict adherence to procedures to achieve a 1 mm defect size. One common method for defect creation is using an Electrical Discharge Machine (EDM). The EDM process involves creating a high-frequency electrical spark between an electrode and the bearing's submerged inner race, which is immersed in dielectric fluid. This spark generates intense heat that melts the localised area, leading to the removal of material and the creation of a defect [14]. The controlled creation of defects using EDM enables researchers to study bearing failures under specific conditions, facilitating the development of predictive maintenance models and advancing our understanding of bearing behaviour in real-world scenarios.

### *3.1. Experimental setup*

A specially designed test bench was set up in the Department of Machine Construction at the Institute of Technology, Hungarian University of Agriculture and Life Sciences, with the primary focus of analyzing bearings, as shown in Figure 1. This setup is a replica of the one explained in [15], intended to detect the initial stages of rolling element bearing failure.

The test bench incorporates an asynchronous motor frequently used in industrial applications such as pumps and fans. The bearing model selected for analysis was SKF 1209 EKTN9, renowned for its significant radial load endurance and minimal axial load capacity in both directions. The complete test bench assembly comprised a motor unit, rolling bearing elements, an electrical control board, and a measuring system (refer to Fig. 1). The bench structure was versatile, consisting of two independent grooved tables that could be positioned as needed.

The specifications for the SKF 1209 EKTN9 bearing are as follows:

- Type: Self-aligning ball bearing
- Model: SKF 1209 EKTN9
- Boundary dimensions:
  - Bore diameter (d): 45 mm
  - Outer diameter (D): 85 mm
  - Width (B): 19 mm
- Dynamic Load Rating (C): Approximately 22.9 KN
- Static Load Rating (Co): Approximately 7.8 kN
- Limiting Speed: Approximately 11,000 RPM with grease lubrication
- Weight: Approximately 0.47 kg

The mounting of SKF 1209 EKTN9 demands meticulous attention to detail to ensure optimal performance and longevity [16]. Initially, the workspace, including the shaft and housing bore, needs to be thoroughly cleaned. If necessary, SKF-approved grease should be evenly applied to the bearing's interior and the shaft.

However, excessive grease should be avoided as it could cause overheating. The bearing should be placed on the shaft such that it slides on smoothly, and it should be tapped into its correct position on the shaft using an SKF fitting tool or a sleeve and a soft mallet. After properly aligning the bearing, it should be secured following the machine's design specifications, typically involving a locknut or an end plate. Finally, the bearing should be manually rotated to check for smooth operation without any unusual noise or friction. Always ensure that all safety protocols are followed, and appropriate personal protective equipment is used throughout the process.

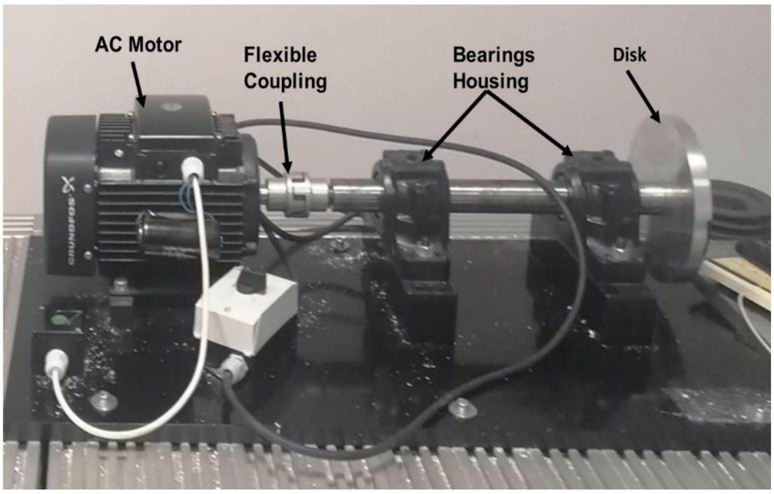


*Figure 1.* Experimental Setup of the Rolling Element Bearing Test Bench

### *3.2. Alignment Procedure*

Alignment of rotating machinery is essential to ensure the proper functioning of interconnected machines and prevent misalignment-related issues. In standard operational conditions, the rotational centers of the shafts align perfectly, demonstrating collinearity [17]. Alignment involves adjusting the relative positioning of two interconnected machines, such as a motor and a pump, to center the moving machine's shafts with those of the stationary machine. During regular machine operation, the axial centerlines converge, necessitating the determination and adjustment of the relative positioning of the machines [18].

The Fixturlaser XA system is commonly utilised for assessing and rectifying misalignment. This system allows for the adjustment of the front and back pairs of the motor's feet, both vertically and horizontally, to align the shafts within specified tolerances. A tolerance table provided in the device manual guides this alignment process. The Fixturlaser XA system incorporates two measuring units positioned on each shaft using fixtures provided with the system, as depicted in Fig. 2. The system calculates the relative distance between the two shafts in two planes by rotating the shafts into various measuring positions with the bearings. Measurements from the bearing to the coupling and the motor feet are inputted into the system. Subsequently, the Fixturlaser XA display provides information on the actual alignment condition and the motor's position, enabling explicit adjustments to be made based on the displayed values. The alignment results are stored in the memory manager for record-keeping, and the data can be easily transferred to a PC for further analysis [19].

In Fig. 2 shows the To initiate the defect-creation process, thorough cleaning of the bearing is essential to remove any grease or contaminants. Once cleaned, the bearing is positioned in the EDM machine, ensuring proper inner race alignment with the electrode. Parameters such as pulse duration, discharge current, and voltage are carefully set to create the desired 1 mm defect without causing unintended damage to the rest of the bearing. Close monitoring of the process is necessary to prevent over-cutting and ensure precise defect size [20]. After the defect is created, the bearing should undergo another round of cleaning to eliminate any residual particles. To confirm the size of the defect, precise measuring equipment like a micrometre or optical comparator is used for accurate measurements [21].

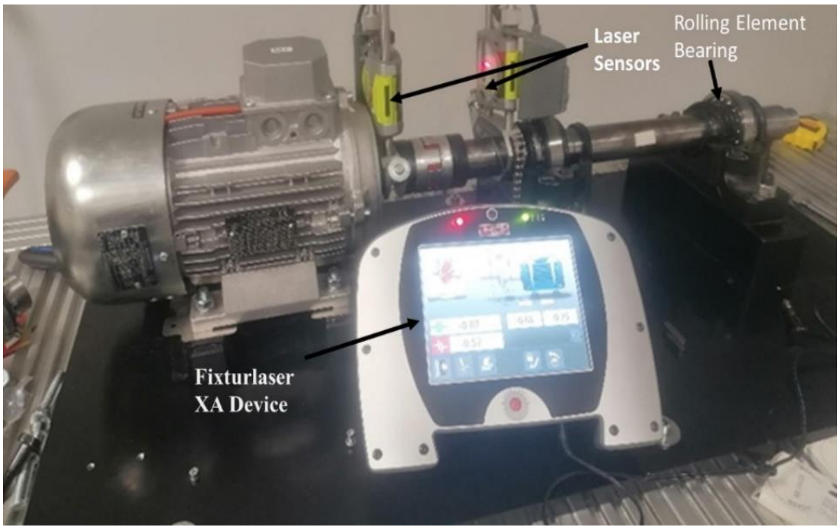


*Figure 2.* Using Fixturlaser XA system to check the alignment of shaft bearings system

### *3.3. Vibration Measurement*

Vibration measurement plays a crucial role in assessing the health of bearings across various types, and it can be effectively conducted using a Microlog analyser (Fig. 3). To begin the measurement process, one end of an accelerometer is connected to the analyser's Fast Fourier Transform (FFT) port. Then, the other end of the accelerometer is attached to the bearing housing in both axial and radial directions, allowing for the capture of vibration signals in the form of time-domain and frequency-domain curves. The FFT analyser and vibration analysis software interface makes diagnosing bearing defects possible [22]. Alignment adjustments are typically performed in the horizontal plane while the motor operates at full speed and under full load conditions. To obtain optimal readings, it is advisable to position the accelerometer sensor near the vibration generation point, which is usually near the bearing location. This is crucial since issues like shaft misalignment can generate forces that impact the bearing [23].

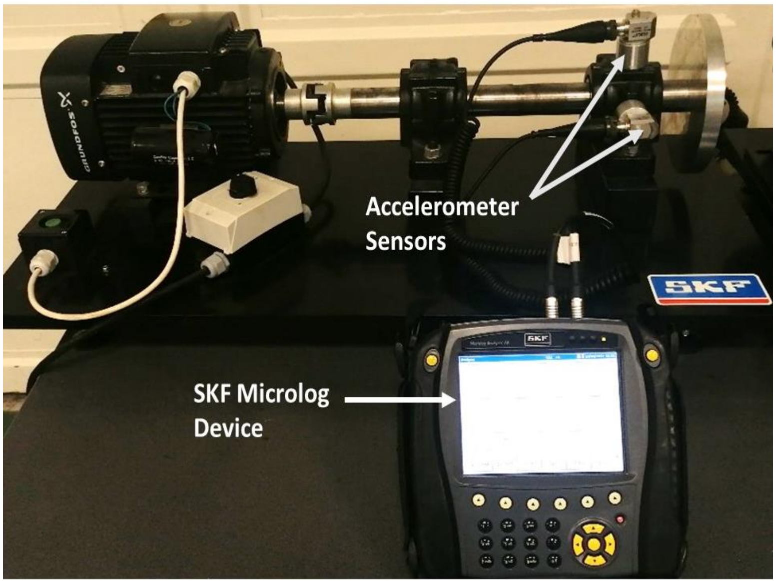


*Figure 3.* Vibration Measurement Setup using Microlog Analyzer

The experimental procedure for vibration measurement is outlined as follows:

i. Affix an accelerometer device to the bearing housing in both axial and radial directions.

ii. Connect the accelerometer to the Microlog analyser device using cables.
iii. Gather vibration data by running the bearings at various loads and speeds.
iv. Analyse the data from the Microlog analyser device using MATLAB software, particularly the Signal Processing Onramp tool.
v. Analyse the vibration data using both time-domain and frequency-domain techniques.
vi. Measure the torque through the bearing under different misalignment cases.

In the study, the SKF Microlog device as shown in figure, equipped with two accelerometer sensors, will be employed to capture vibration data from bearings subjected to axial and radial loads. The investigation will also explore the effects of machine and motor imbalances on both roller and sliding bearings, as imbalances can generate excessive forces that affect bearing performance. Additionally, the impact of misalignment on sliding bearings will be studied. Data collected from the HBM Spider8 acquisition and SKF Microlog devices will be analysed using MATLAB software. The SKF Microlog analyser and SKF sensors will be utilised to examine the effects of inner race defect in three size 0.5 mm,1 mm and 2 mm on the bearing.

## 4. Experimental Result

### *4.1. Vibration measurement for bearing defect and healthy bearing*

Table 1 presents the True Peak Values (TPV) obtained from vibration measurements for different bearing conditions, including a healthy bearing (HB) without any defect and bearings with inner race defects (IRD) of varying sizes (IRD0.5 with a 0.5mm defect, IRD1 with a 1mm defect, and IRD2 with a 2mm defect). The gE True peak values are reported for different speeds ranging from 500 RPM to 2500 RPM. At a speed of 500 RPM, the gE True peak value for the healthy bearing (HB) is 0.354. As the size of the inner race defect increases, the gE True peak values also increase. The gE True peak value for IRD0.5 is 2.07, IRD1 is 1.64, and IRD2 is 1.1. These results indicate that even a small defect size of 0.5mm can significantly impact the vibration levels in the bearing. Moving to a higher speed of 1000 RPM, the gE True peak value for the healthy bearing is 0.465. Similar to the previous speed, larger defect sizes increase the gE True peak value. The gE True peak value for IRD0.5 is 1.28, IRD1 is 2.31, and IRD2 is 2.63. These findings further emphasise the correlation between defect size and vibration levels, highlighting the increased impact of larger defects on the bearing's vibration signature. At 1500 RPM, the gE True peak value for the healthy bearing is 0.643. The gE True peak values for IRD0.5 is 1.4, IRD1 is 1.87, and IRD2 is 1.22. It is notable that the gE True peak values for IRD1 and IRD2 are closer to the gE True peak values of the healthy bearing at this speed. This suggests that the influence of defect size on vibration levels may vary depending on the operating speed.

As the speed increases to 2000 RPM, the gE True peak values for the healthy bearing significantly rises to 1.43. Similarly, the gE True peak values for the bearings with defects also increase. The gE True peak values for IRD0.5 is 3.69, for IRD1 is 1.9, and for IRD2 is 3.64. These results indicate that higher speeds amplify the effect of defects on vibration levels, resulting in more pronounced differences between healthy and defective bearings. Finally, at 2500 RPM, the TPV for the healthy bearing is 1.59. The gE True peak values for IRD0.5 is 2.54, IRD1 is 3.54, and IRD2 is 4.58. These findings demonstrate a clear trend of increasing gE True peak values with larger defect sizes at higher speeds.

*Table 1.* True peak values (TPV) of Bearings with heatley bearing and inner Race Defects

| Speed RPM | HB | IRD0.5 | IRD1 | IRD2 |
|---|---|---|---|---|
| 500 | 0.354 | 2.07 | 1.64 | 1.1 |
| 1000 | 0.465 | 1.28 | 2.31 | 2.63 |
| 1500 | 0.643 | 1.4 | 1.87 | 1.22 |
| 2000 | 1.43 | 3.69 | 1.9 | 3.64 |
| 2500 | 1.59 | 2.54 | 3.54 | 4.58 |

### *4.2. Vibration Measurements for inner race defect in Vertical Direction*

Table 2 presents the effect of inner race defect (IRD) on rolling element bearings at various speeds. The table consists of five columns and six rows. The first column represents the rotation speed of the rolling element bearing in RPM (revolutions per minute). The position of the Inner Race Defect (IRD) after mounting the bearing into the housing can be a critical factor in the experimental setup and subsequent data analysis. The second, third, and fourth columns represent the IRD severity levels of 0.5, 1, and 2, respectively. The fourth column displays the gE True peak values, which are measures of the vibration of the rolling element bearing. The results in Table 2 demonstrate the impact of different IRD severity levels and rotation speeds on the vibration of the rolling element bearing. As the severity of the IRD increases, the gE True peak values also increase, indicating a higher level of vibration. This observation implies that the inner race defect's severity directly influences the bearing's vibration characteristics. Furthermore, it is evident that the gE True peak values vary with the rotation speed of the bearing. Some speeds exhibit higher vibration values compared to others for the same IRD severity level.

To visually represent the vibration measurements, Fig. 4 depicts the relationship between the rotation speed and the gE True peak values for different IRD severity levels in the vertical direction. The graph highlights the increasing trend of vibration with higher IRD severity levels and demonstrates the influence of rotation speed on the vibration levels.

*Table 2.* True peak values (TPV) for inner race defect in vertical Direction

| Rotation Speed RPM | IRD0.5 | IRD1 | IRD2 |
|---|---|---|---|
| 500 | 2.07 | 1.64 | 1.1 |
| 1000 | 1.28 | 2.31 | 2.63 |
| 1500 | 1.4 | 1.87 | 1.22 |
| 2000 | 3.69 | 1.9 | 3.64 |
| 2500 | 2.54 | 3.54 | 4.58 |

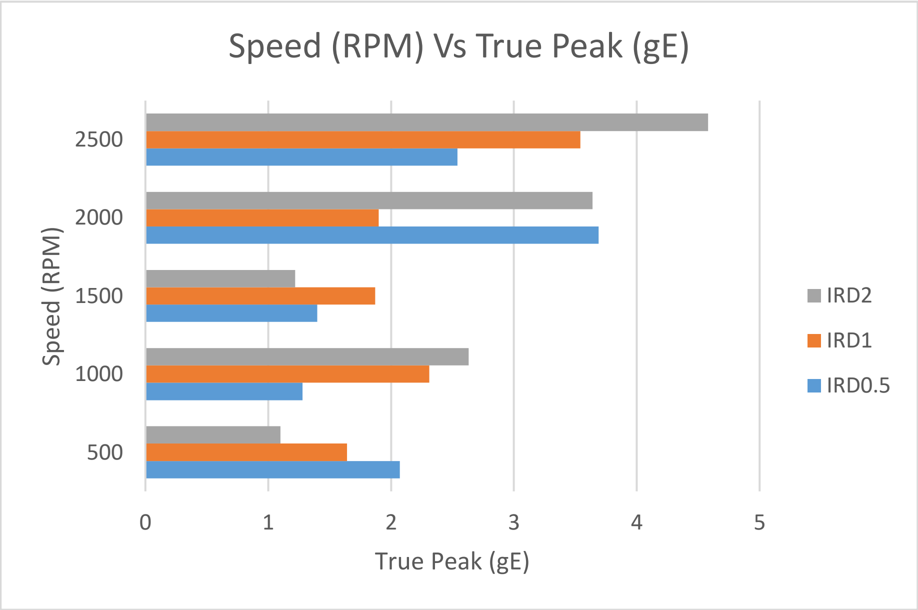


*Figure 4.* True peak values (TPV) in vertical direction for Inner Race Defects

### *4.3. Vibration Measurements in Horizontal Direction*

Table 3 presents the effect of inner race defects (IRD) on rolling element bearings at various speeds. The table consists of four columns, with the first column representing the rotation speed of the bearing and the remaining three columns representing the IRD severity levels (IRD0.5, IRD1, and IRD2). The values in the table represent the gE True peak values, which serve as a measure of the vibration amplitude of the bearing. The results obtained from experimental investigations indicate that as the speed of the bearing increases, the vibration amplitude also increases, suggesting a higher level of IRD severity. Additionally, it can be observed that as the severity of IRD increases (from IRD0.5 to IRD2), the vibration amplitude also increases, indicating a greater level of damage to the bearing. To visualise the relationship between the rotation speed, IRD severity, and vibration amplitude, Fig. 5 illustrates the trend of increasing vibration amplitude with higher IRD severity and rotation speed. This graph clearly represents the impact of IRD severity and speed on the vibration characteristics of rolling element bearings. The presented table and graph in Figure 7 serve as valuable references for diagnosing the severity of IRD in rolling element bearings based on the measured vibration amplitude using gE True peak values. By analysing the vibration data at different speeds and IRD levels, engineers and technicians can effectively assess the condition of bearings and identify potential issues early on, enabling proactive maintenance and reducing the risk of bearing failure.

*Table 3.* True peak values (TPV) for inner race defect in horizontal direction

| Rotation Speed RPM | IRD0.5 | IRD1 | IRD2 |
|---|---|---|---|
| 500 | 1.22 | 1.44 | 1.12 |
| 1000 | 1.26 | 1.68 | 1.24 |
| 1500 | 1.92 | 2.26 | 1.79 |
| 2000 | 3.95 | 3.77 | 5.63 |
| 2500 | 6.7 | 2.71 | 6.51 |

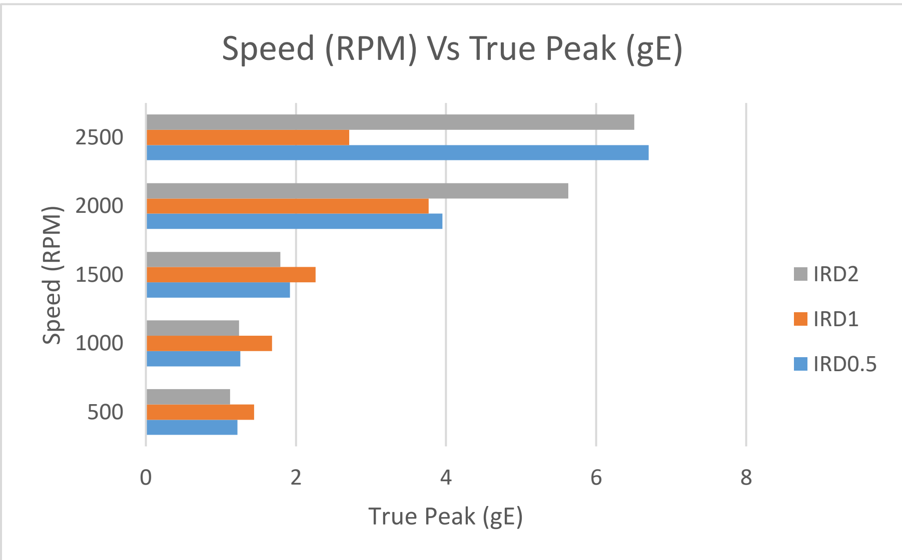


*Figure 5.* True peak values (TPV) for inner race defct in horizontal for Inner Race Defects

The results show that the severity of IRD significantly affects the vibration of the rolling element bearing, with higher IRD severity levels resulting in higher gE True peak values. The gE True peak values also vary with the speed of the rolling element bearing, with some speeds having higher values than others for the same IRD severity level.

## 5. Conclusion

In conclusion, this article has highlighted the significance of early detection and monitoring of bearing defects in rotating machinery. Vibration measurement techniques, such as analysing the vibration amplitude using gE True peak values, have proven to be reliable and non-invasive methods for assessing the health status of rolling element bearings. Through experimental investigations, the effect of inner race defects (IRD) on bearing vibration has been examined. The results obtained from the experimental works and the presented table have demonstrated that the vibration amplitude of rolling element bearings increases with higher IRD severity and rotation speed. This indicates a direct relationship between the severity of IRD, bearing damage, and vibration characteristics. By monitoring the vibration amplitude using gE True peak values, engineers and technicians can assess the severity of IRD and detect potential bearing failures at an early stage.

Overall, this article emphasises the importance of vibration measurement techniques and their role in detecting and monitoring bearing defects. By leveraging these techniques, engineers and technicians can implement effective maintenance strategies, improve operational efficiency, and enhance the reliability and lifespan of rotating machinery. The graphs highlights the increasing trend of vibration with higher IRD severity levels and demonstrates the influence of rotation speed on the vibration levels Further research is recommended to explore additional techniques and technologies for bearing fault detection and diagnosis to enhance the reliability of rotating machinery in industrial settings. Finally, the paper provides valuable insights into the effect of inner race defects on rolling element bearings and highlights the importance of vibration

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