Comparative Experimental Study of Different Bearings Vibration Using FFT and Statistical Feature Extraction

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In this study, vibration patterns of different types of bearings, such as deep groove ball bearings, cylindrical roller bearings (single row), and radial spherical plain bearings were studied. They're the most commonly used in mechanical parts in rotary machines under the effect of load and different angular speeds: 1488 rpm, 1900 rpm, and 2450 rpm. As well as the two cases, the first without load and the second with a load of 29.43 N, there was a three-disc mounted on the shaft. The vibration monitoring and analysis technique, which is a dependable and accurate assessment method to select a suitable bearing element, the Fast Fourier Transformation analysis was used for processing the vibration signal that was gathered from three different types of bearings for three different running speeds with two load levels. Additionally, several indicative parameters including the root mean square, kurtosis, peak-to-peak, and skewness were extracted from the vibration data using time-domain signal analysis. From experimental results, it was noticed that the deep groove ball bearing gives the lowest amplitude of vibration among all two bearings when the system is without loading effect, whereas the cylindrical roller bearing gives the least amplitude under loading operating conditions; finally, the radial spherical plain bearing was operated at its best at high rotational speeds.

Keywords: *Vibration Analysis, Bearings, Fast Fourier Transformation (FFT), Statistical features, Rotational Speed.*

1. INTRODUCTION

The oscillating movement of a particle or body around an equilibrium position is referred to as mechanical vibration. Because they result in higher strains and energy losses, vibrations in machinery and buildings are frequently undesirable. Therefore, it is crucial to make an effort to reduce or completely eradicate these vibrations through efficient design. In recent years, the move toward lighter structures and faster machinery has made vibration analysis more and more crucial. It is anticipated that this trend will continue, making vibration analysis an even more urgent issue going forward [1].

In order to allow relative motion between the contacting surfaces while supporting the load, a bearing is a machine element. A layer of lubricant is frequently applied to minimize wear, friction, and, under some situations, heat dissipation. Although vegetable oils, silicone oils, greases, and other lubricants may also be utilized, mineral oils that have been refined from petroleum are typically used [2].

The rolling elements in the bearings are frequently used in most of the rotating machines. A rolling element, an outer ring, an inner ring, and a cage are the four primary components of a standard rolling bearing. The cage allows the rolling elements to rotate freely, keeps them apart at regular intervals, and secures them inside the inner and outer raceways [3].

Because shafts, gears, and bearings rotate, every rotating machine has a different vibration signature. A crucial component of rotating machinery, rolling bearings have a big impact on machine vibrations. A bearing's structural components influence the system's vibration response to external time-varying forces by acting as both a mass and a spring inside the system. Bearings also produce excitation forces, which are time-varying forces that cause vibrations in the system. Although rolling bearings are designed to experience this excitement, flaws or deficiencies in bearing components can greatly increase these forces [4,5]. Where the most frequent reason for excessive vibrations in metal processing machines that lead to unacceptable surface quality is damage to individual bearing components [6].

Three primary elements contribute to rolling bearing vibrations: manufacture, design, and operation. Vibrations connected to design originate from the bearing's intrinsic construction. Variable elastic deformations at the ball-raceway contact points result from periodic changes in the number of rolling elements supporting the load during operation. These recurring load changes may induce significant self-oscillations. Depending on how lubrication affects the vibrations of the bearing, the rolling bearing first attains a beneficial lubrication mode in ball-outer ring contact before moving on to ball-inner ring contact.

Additionally, angular speed influences the lubrication regime far more than the applied load [7]. Point flaws in rolling components and raceways, contamination of bearings or lubricants, and variations in shape

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and size that arise during production or assembly - such as internal clearance, waviness, and surface roughness - all contribute to vibrations associated with manufac–turing. Significant vibrations can result from even little flaws. The friction and fatigue wear processes that take place inside the bearings during operation are associated with operational sources of vibration [8]. In some cases, the combined flaws (cage and ball fault, inner and outer race fault) are more serious, more susceptible to applied load and rotation speed, and have a bigger effect on the rotating system's total vibration level. This is explained by the fact that the combined fault situation produces the largest vibration amplitudes since it involves two faults operating simultaneously [9].

Combining AI with vibration analysis significantly improves defect identification as the need for increased machinery dependability increases in the current AI-driven age. In particular, ANNs have been widely used for defect categorization and identification. The sug–gested ANN models categorize many sorts of prob–lematic bearings, such as flaws in the inner race, outer race, or ball, and differentiate them from healthy ones, in contrast to previous ANN models that normally classify bearings as either faulty or healthy [10-12].

Factors like surface waviness and roughness, which vary with surface deterioration, affect the degree of vib–ration [13]. Vibration levels are also impacted by radial clearance in bearings, particularly in the middle and high-frequency ranges [14,15]. Vibrations at mul–tiples of the cage speed are caused by variations in ball size, and peak amplitudes decrease with increasing ball count [16]. A stiffer system with more rollers lessens the effect of fluctuating compliance frequencies as the oscillation center gets closer to zero [17]. Vibration amplitudes in tapered roller bearings are greatly inc–reased when the roller diameter error magnitude increases [18].

Plain bearings, on the other hand, are cheap to make, silent to run, and have vibration-damping qualities. They can handle foreign matter, take up a little radial area, and are simply machined. They are only appro–priate for relatively moderate temperatures and speeds, though, and they need a lot of lubricating oil. Because the lubricant coating over the bearing surface builds up gradually, the beginning resistance is significantly larger than the running resistance [19].

Random pressure changes between the lubricant film and surface asperities are important excitation sources for forced vibrations in the hydrodynamic lubrication of journal bearings. Operating parameters, including load, speed, and lubricant viscosity, cause these pressure variations to grow [20]. The stability of journal bearings is significantly impacted by the oil's viscosity, which has an impact on both the time and frequency domains. Vibrations can be lessened by techniques including using active lubrication, adding additives to maintain viscosity, or changing the oil's temperature [21].

The stiffness and vibration response of linear plain bearings will be impacted by the usage of hybrid pairs of materials with notably differing strengths, elastic properties, deformability, and wear resistance [22].

One method that has shown promise for evaluating journal-bearing performance is the use of features that are taken from vibration and sound inputs [23].

The aforementioned reviewed literature makes it ap–parent that design errors, manufacturing problems, and operational wear are the main causes of bearing vibrations. Elements such as surface roughness, radial clearance, and roller variations have a major effect on vibration levels. Rolling bearings are more suitable to work under the vibration effect, whereas plain bearings provide superior dampening and quieter operation but need more frequent maintenance. Viscosity management, optimal lubrication, and precision manufacture may all help to effectively limit vibration, which will increase the lifetime and performance of rotating machinery.

According to a review of literature studies, the following research gaps have been identified:

1. Most researchs focuses on a single type of bearing (such as roller bearings only), with no systematic comparisons with other types of bearings (rolling versus sliding) under the same operating conditions.
2. Most practical tests of bearings are conducted under constant loads and speeds, which does not reflect the realistic dynamic conditions under which these parameters change.
3. Previous research often addresses artificial defects (such as initial cracks) rather than defects resulting from operation, which may affect the vibration signals generated by the defect.
4. Despite the wide use of statistical features and the Fast Fourier Transform (FFT), few studies combine them to correlate behavior in the time and frequency domains for accurate fault identification.

To address these gaps, our study examines and ana–lyzes the vibrations generated by three different types of bearings: deep groove ball bearings, cylindrical roller bearings (single row), and radial spherical sliding bearings - under two load levels and three different ope–rating speeds, using fast Fourier transform and vibration characteristic extraction techniques. The novelty of this research is as follows:

1. Making a comparison between three different types of bearings - deep groove ball bearings, cylindrical roller bearings, and radial spherical sliding bearings - under different load and speed operating conditions.
2. Analyzing practical generated vibration signals utilizing both frequency-domain (FFT) and time-domain (statistical features) methods.
3. Develop design selection techniques to select the best bearings based on dynamic operating conditions.

The structure of this paper is as follows: The method for extracting features from the recorded vibration signals is discussed in Section 2; the test rig utilized for the experimental work is discussed in Section 3; the analysis of the vibration features is covered in Section 4; and the results of the Time domain vibration signal and Fast Fourier Transformation are discussed in Section 5. Section 6 concludes with a summary of the work and some closing thoughts.

2. VIBRATION FEATURE EXTRACTION

Vibration signals are often used to evaluate the likelihood of defects and track the condition of mecha-nicccal components. However, because raw vibration signals are often congested with noise, which increases the likelihood of an incorrect health evaluation, they are unable to offer useful and significant information about the machine's health state. Data mining, or feature extraction, can be done in a number of methods. Therefore, a popular and significant feature extraction technique is the application of statistical feature functions to vibration data. Therefore, in order to guarantee the accuracy of bearing vibration monitoring, this work uses statistical feature extraction as a data mining technique. Four feature functions - root mean square, kurtosis, peak-to-peak, and skewness - are selected in this case. A particular facet of the signal behavior was reflected in the selection of each feature. However, mechanical damage primarily impacts vibration signals, perhaps increasing the total strength of vibrations or causing an impulsive disruption. The characteristics listed above are defined and expressed mathematically as follows [24]:

The RMS, or variation in the signal's strength, is an amplitude-modulated Gaussian random process; it can be computed as:

$$RMS = \sqrt{\frac{1}{L}\sum_{i=1}^{L}\left|y_i\right|^2} \tag{1}$$

Kurtosis is a dimensionless feature that measures a deviation of the signals from the probability density function (PDF); it can be calculated through the following equation.

$$Kurt = \frac{\frac{1}{L}\sum_{i=1}^{L}\left|y_i-\mu\right|^4}{\left(\frac{1}{L}\sum_{i=1}^{L}\left|y_i-\mu\right|^2\right)^2} \tag{2}$$

Peak to Peak values show the vibration signal's highest amplitude. Vibration amplitudes are larger when the Peak to Peak values are higher; this can be calculated as:

$$PP = \left|\max\left(y_i\right)-\min\left(y_i\right)\right| \tag{3}$$

Skewness, which stands for the third moment of distribution, quantifies how asymmetrical the probability distribution is around its mean. It could be zero, positive, or negative, signifying symmetrical, left, or right distributions. It can be found as follows.

$$Skw = \frac{\frac{1}{L}\sum_{i=1}^{L}\left|y_i-\mu\right|^3}{\left(\frac{1}{L}\sum_{i=1}^{L}\left|y_i-\mu\right|^2\right)^3} \tag{4}$$

3. EXPERIMENTAL WORK

3.1 Experimental Setup

Figure 1 indicates that the experimental setup consists of a multi-disc rotor, pulleys, two types of bearings, and a single-phase AC motor. The motor has four grooves to make an adjustable center distance with the rotor shaft; a set rotational speed is up to 1350 rpm, and a power rating is 0.75kW. The rotor shaft, which is 300mm long and has crosssectional dimensions of Din=19.5 mm and Dout=25 mm, is supported by two different types of bearings with 260 mm separating them. Each disk weighs 1000 g and has an outer diameter of 76 mm. The multi-disc rotor is driven by the V-belt drive. Steel base plates are used to mount the bearing pedestals in addition to the AC motor.

Three different driver pulley radii (33.5 mm, 42.5 mm, and 56 mm) and a standard-length v-belt (650 mm) are used in the experimental setup to generate three distinct rotational speeds: 1488 rpm, 1900 rpm, and 2450 rpm, respectively.

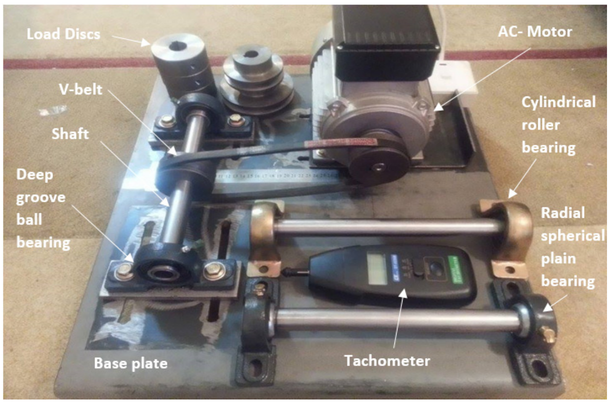

Figure 1. Experimental Rig.

3.2 Instrumentation and Sensory System

The three main categories of vibration measurement are usually prognosis, diagnosis, and detection [5]. Through constant vibration level monitoring and signal analysis, engineers can pinpoint the root causes of possible problems and take corrective action before significant damage or equipment failure happens. This proactive strategy improves safety, lowers maintenance costs, and increases equipment reliability [25].

A triple-axis acceleration board (LSM330DLC) is utilized in this study's instrumentation to monitor vibrations on bearing pedestals. Both static gravity acceleration and dynamic acceleration brought on by shock or vibration can be detected by this system-in-package, which combines a 3D digital accelerometer and a 3D digital gyroscope. A signal processing circuit converts the sensor's output into an analog voltage proportionate to the vibration level.

To guarantee control loop stability, the LSM330DLC has a specialized, programmable signal processing path that offers low latency, minimal noise, and customized filtering. An auxiliary SPI interface that is adjustable for the accelerometer and gyroscope can be used to provide data from this dedicated signal line. System designers choose the LSM330DLC for the development and production of dependable and adaptable products because of its excellent performance, high quality, small size, low power consumption, and strong resilience to mechanical shock [26].

The signals produced by the acceleration sensor were gathered and then sent to a PC. For every type of bearing and loading, these obtained signals are subsequently processed using Fast Fourier Transform (FFT) analysis. A tachometer is also used to measure the rotational speed.

3.3 Types of Bearings

In this work, different types of bearings were studied under the effect of vibration as follows [27]:

3.1.1 UCP 205 Pillow block ball bearing

The insert bearing in a cast iron housing that may be fastened to a support surface makes up pillow (Plummer) block ball bearing units as shown in Figure 2. Applications involving both continuous and alterna–ting rotational orientations can benefit from this robust and rigid variation. It is simple to attach since it has an inner bush that is expanded at both ends and is fixed on the shaft through screw tightening between the inner bush and the shaft.

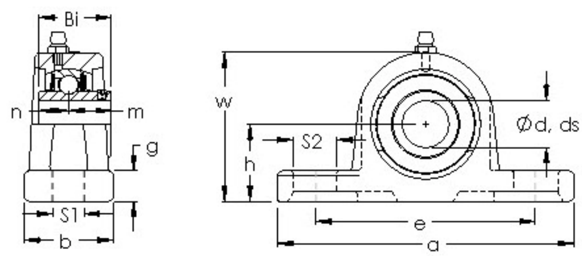

Figure 2. UCP 205 Pillow block ball bearing

3.1.2 NJ 305 Single-row cylindrical roller bearing

Single-row cylindrical roller bearings shown in Figure 3, are made to withstand high speeds and strong radial loads. NJ design bearings can support axial movement in a single direction because they have two integrated flanges on the outer ring and one on the inner ring. The detachable design, which makes mounting easier and allows the bearing components to be swapped out, is a crucial feature. This bearing was inserted in the INA Pillow Block Unit (2 Bolt) for fixing the base plate.

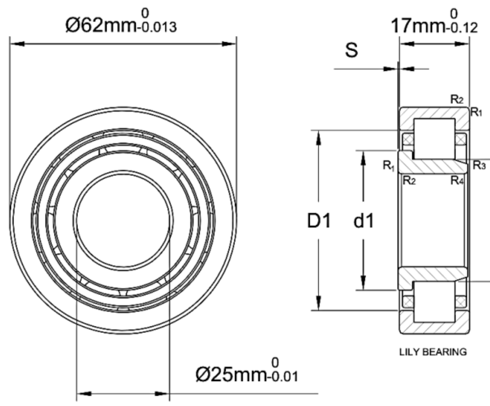

Figure 3. NJ 305 Single-row cylindrical roller bearing

3.1.3 Radial spherical plain bearing

The spherical plain bearings shown in Figure 4, require no maintenance and feature a bronze contact surface combination. The sliding surfaces must be shielded from impurities on the outside. Higher load ratings and greater tilt angles are made possible by the availability of these bearings with a wider inner ring and a bigger outer diameter. To fasten it to the base plate, this bearing was mounted in the Pillow Block Unit (2 Bolt).

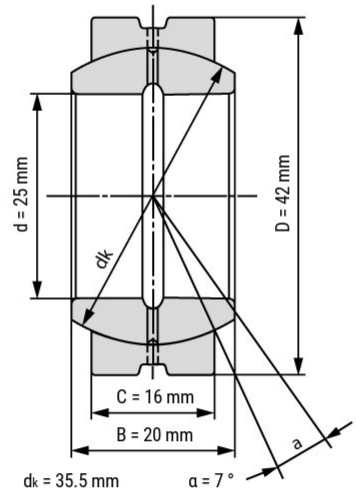

Figure 4. Radial spherical plain bearing

4. VIBRATION FEATURES ANALYSIS

In this study, the system monitoring with different bearing types and working conditions was performed by analyzing the vibration signals and extracted features. Namely, the shaft vibration has been monitored under the effect of load and different angular speeds: 1488 rpm, 1900 rpm, and 2450 rpm. As well as the two cases, the first without load and the second with a load of 29.43 N, there was a three-disc mounted on the shaft. Figures (5 up to 16) focus on the vibration features (RMS, Kurtosis, Peak to Peak, and Skewness) for each type of bearing (Deep Groove Ball Bearing, Cylindrical Roller Bearings, and Radial spherical plain Bearing) under different conditions (with and without load) at three speeds (1488 rpm, 1900 rpm, and 2450 rpm).

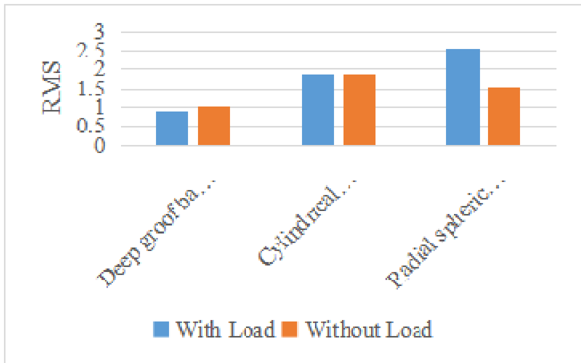

Figure 5. The RMS feature for different bearing types at the rotational speed of 1488 rpm.

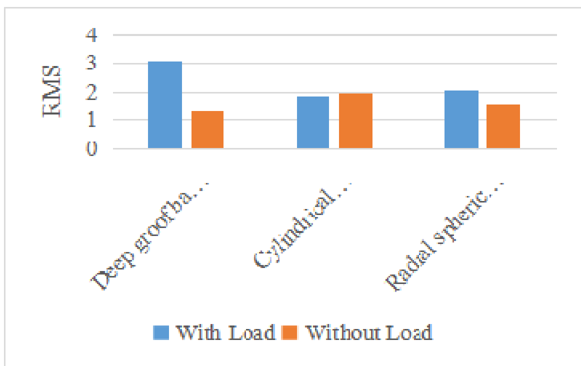

Figure 6. The RMS feature for different bearing types at the rotational speed of 1900 rpm.

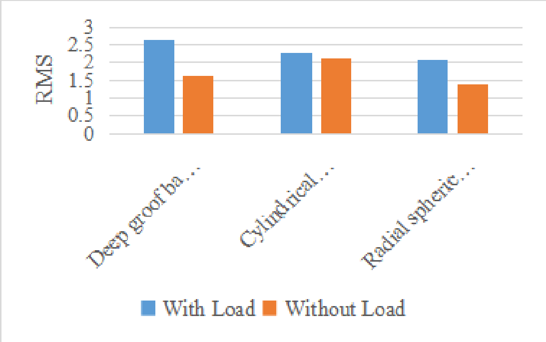

Figure 7. The RMS feature for different bearing types at the rotational speed of 2450 rpm.

From the figures (5 to 7), the following can be observed:

- Particularly at lower speeds (1488 rpm), the Radial spherical plain Bearing exhibits the highest sensitivity to load.
- Higher speeds (1900 rpm and 2450 rpm) increase the sensitivity of the Deep Groove Ball Bearing to load.
- Because they are the most stable under all conditions, the Cylindrical Roller Bearings are a suitable option for applications involving a range of loads and speeds.

From Figures (8 to 10), one can achieve the following:

- Particularly when there is no load, the Deep Groove Ball Bearing has the greatest kurtosis values, indicating that it could be more vulnerable to impacts or flaws.
- Depending on the load and speed, the kurtosis values of the Cylindrical Roller Bearings and Radial spherical plain Bearings vary substantially.

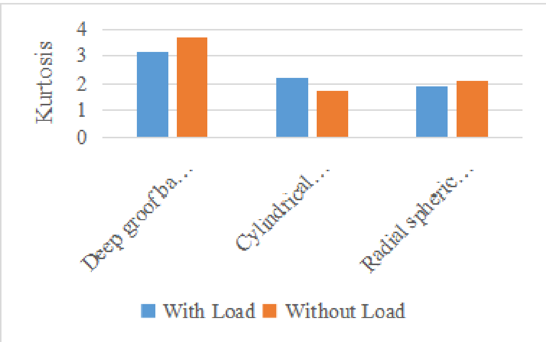

Figure 8. The Kurtosis feature for different bearing types at the rotational speed of 1488 rpm.

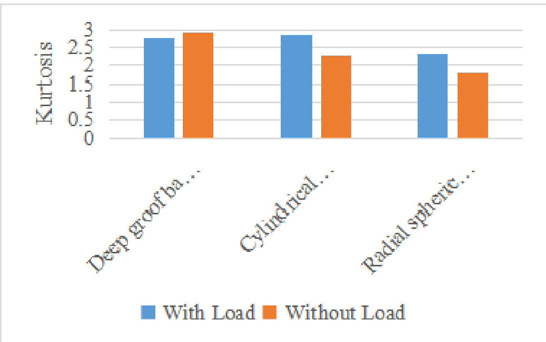

Figure 9. The Kurtosis feature for different bearing types at the rotational speed of 1900 rpm.

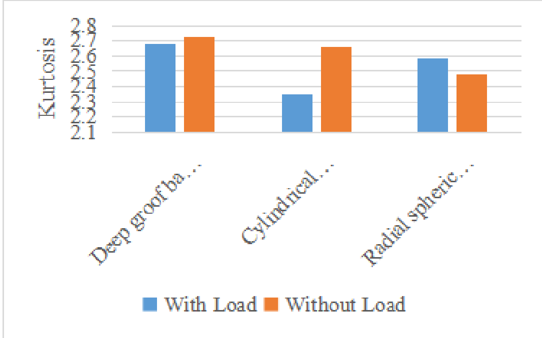

Figure 10. The Kurtosis feature for different bearing types at the rotational speed of 2450 rpm.

The observation of figures (11 to 13) leads to the following:

- In terms of peak-to-peak values, the Deep Groove Ball Bearing and Radial spherical plain Bearing exhibit the highest load sensitivity, particularly at higher speeds.
- With the fewest variations in peak-to-peak values under various conditions, the Cylindrical Roller Bearings are the most stable.

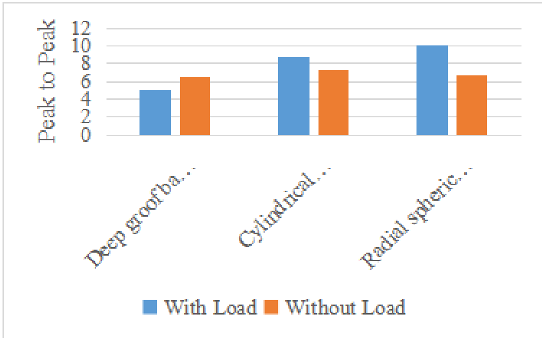

Figure 11. The Peak to Peak feature for different bearing types at the rotational speed of 1488 rpm.

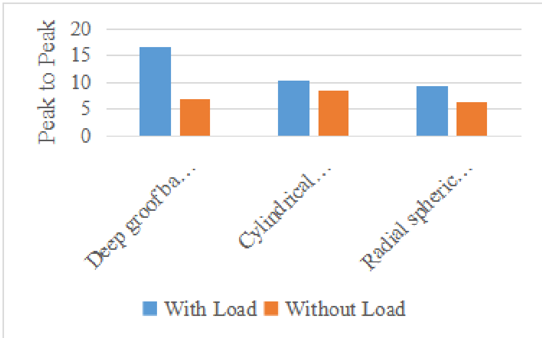

Figure 12. The Peak to Peak feature for different bearing types at the rotational speed of 1900 rpm.

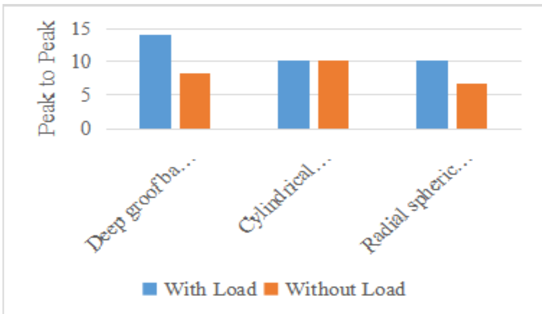

Figure 13. The Peak to Peak feature for different bearing types at the rotational speed of 2450 rpm.

Finally, from figures (14 to 16) can be concluded the following:

- Particularly while under load, the Deep Groove Ball Bearing exhibits the greatest variety in skewness values, signifying shifts in the distribution of vibration signals.
- Stable vibration signal distributions are suggested by the more constant skewness values displayed by the Cylindrical Roller Bearings and Radial spherical plain Bearings.

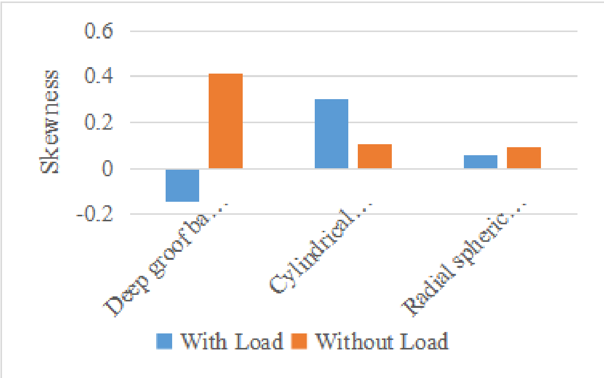

Figure 14. The Skewness feature for different bearing types at the rotational speed of 1488 rpm.

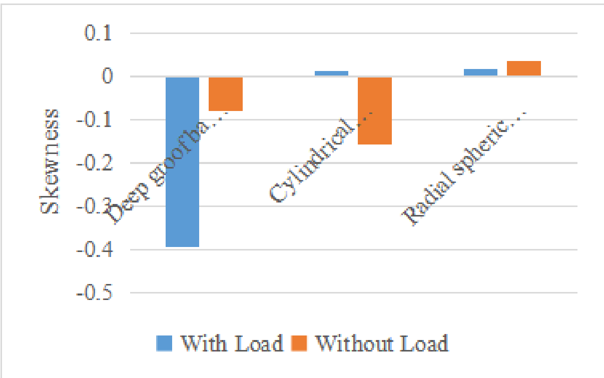

Figure 15. The Skewness feature for different bearing types at the rotational speed of 1900 rpm.

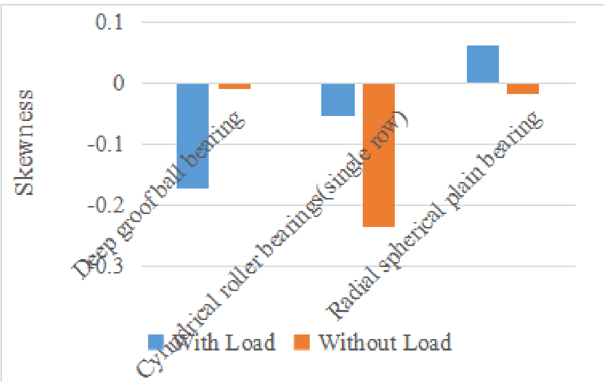

Figure 16. The Skewness feature for different bearing types at the rotational speed of 2450 rpm.

Together with the graphical representations, Table (1) has been included to allow for a more transparent comparison of the extracted vibration features. The RMS, kurtosis, peak-to-peak amplitude, and skewness numerical values for each bearing type under various loading scenarios and speeds are shown in this table.

Table1. The Vibration Extracted Features for different bearing types and operating conditions.

Rotational Speed	Vibration Features	Bearing Type: Deep Groove Ball Bearing	Cylindrical Roller Bearing	Radial Spherical Plain Bearing	Operating Condition
1488 rpm	RMS	1.022	1.859	1.543	Without loading
	Kurtosis	3.720	1.748	2.137	
	Peak to Peak	6.474	7.409	6.759	
	Skewness	0.411	0.107	0.087	
1900 rpm	RMS	1.310	1.964	1.574	
	Kurtosis	2.928	2.266	1.782	
	Peak to Peak	6.7	8.343	6.316	
	Skewness	-0.079	-0.1571	0.033	
2450 rpm	RMS	1.623	2.110	1.380	
	Kurtosis	2.730	2.664	2.478	
	Peak to Peak	8.364	10.206	6.669	
	Skewness	-0.0097	-0.236	-0.0175	
1488 rpm	RMS	0.90791	1.863	2.577	With loading
	Kurtosis	3.141	2.245	1.880	
	Peak to Peak	5.006	8.882	10.123	
	Skewness	-0.1444	0.3003	0.0529	
1900 rpm	RMS	3.1108	1.8388	2.0516	
	Kurtosis	2.757	2.833	2.329	
	Peak to Peak	16.505	10.338	9.133	
	Skewness	-0.396	0.011	0.0164	
2450 rpm	RMS	2.651	2.277	2.100	
	Kurtosis	2.677	2.351	2.581	
	Peak to Peak	14.028	10.238	10.266	
	Skewness	-0.172	-0.054	0.0626	

For General Conclusions:

First, the Deep Groove Ball Bearing:

- Extremely load-sensitive, particularly at higher speeds (1900 rpm and 2450 rpm).
- RMS, kurtosis, and peak-to-peak values under stress all exhibit notable variations.
- Extremely high kurtosis readings suggest that it may be vulnerable to influences or faults.

Second, Cylindrical Roller Bearing:

-The most stable under all conditions, having the least amount of variation in RMS, Kurtosis, and Peak to Peak values.

-Appropriate for applications with different speeds and loads.

Third, Radial Spherical Plain Bearing,

-It is particularly sensitive to load at lower speeds (1488 rpm), which is the third component.

- The RMS and Peak to Peak readings under load exhibit notable variations.

- Possibly less appropriate for applications requiring a lot of load.

This analysis can assist in choosing the best bearing type depending on the intended vibration features and the conditions of operation (load and speed).

5. FAST FOURIER TRANSFORM ANALYSIS

The acceleration amplitude values of three different bearings - a deep groove ball bearing, a single-row cylindrical roller bearing, and a radial spherical plain bearing - have been gathered for this study. For each bearing, three levels of rotational speed are taken into consideration, both with and without loading.

In this work, the Fast Fourier Transform (FFT) is utilized as a computational method for signal processing that converts time-domain data to the frequency domain. This calculation makes it possible to see the remarkable frequency peaks of the signal instead of viewing the total signal components [28,29].

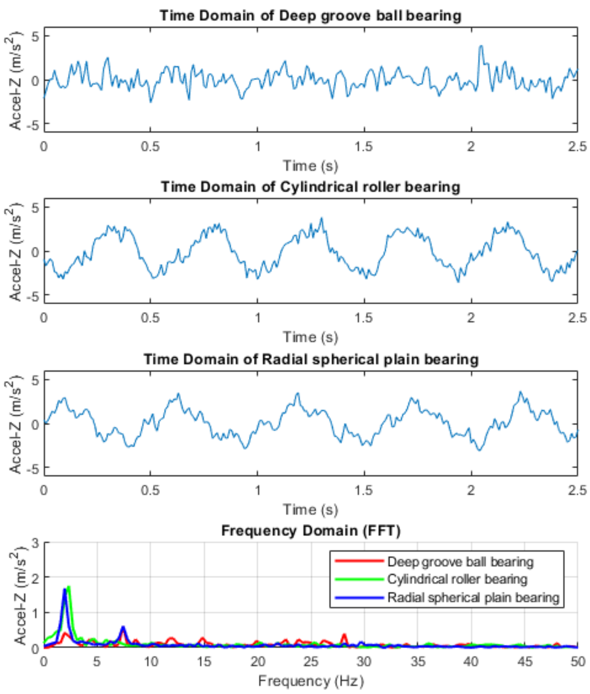

Figure 17. FFT analysis and Comparison between Deep groove ball bearing, cylindrical roller bearing (single row), and Radial spherical plain bearing at speed (1488 rpm) without loading.

Figures (17 up to 19) illustrate the Fast Fourier Tra–nsform (FFT) of measured vibration in the z-direction when using deep groove ball bearings, cylindrical roller bearings (single row), and radial spherical plain bea–rings at speeds of 1488, 1900, and 2450 rpm without applying load on the rotating shaft. Figures (20 up to 22) illustrate the Fast Fourier Transform (FFT) of measured vibration in the z-direction when using a deep groove ball bearing, a cylindrical roller bearing (single row), and a radial spherical plain bearing at speeds of 1488, 1900, and 2450 rpm with an applied load on the rotating shaft.

Figures (17 to 19) illustrate the vibration analysis in both the time domain and frequency domain (FFT) for three types of bearings. The first three graphs in each fi–gure display the acceleration in the z-direction over time for each type of bearing without loading conditions.

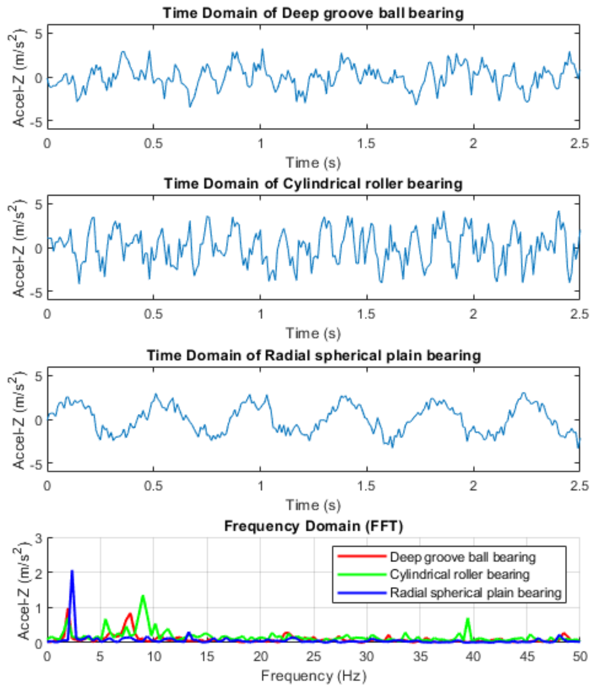

Figure 18. FFT analysis and Comparison between Deep groove ball bearing, Cylindrical roller bearing (single row), and Radial spherical plain bearing at 1900 rpm without loading.

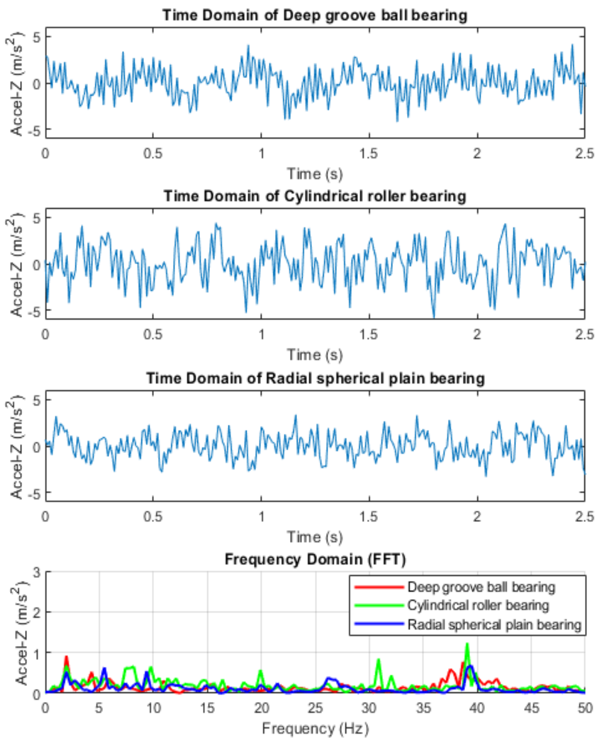

Figure 19. FFT analysis and Comparison between Deep groove ball bearing, Cylindrical roller bearing (single row), and Radial spherical plain bearing at 2450 rpm and without loading.

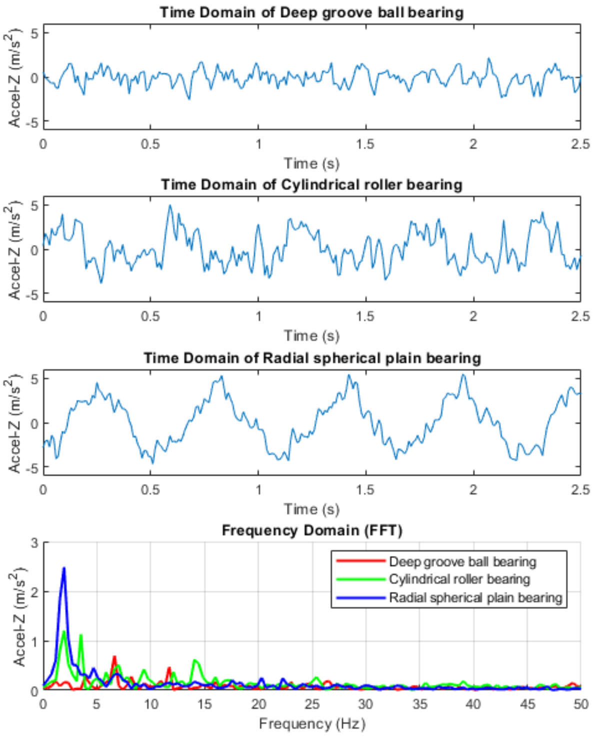

Figure 20. FFT analysis and Comparison between Deep groove ball bearing, cylindrical roller bearing (single row), and Radial spherical plain bearing at speed (1488 rpm) with loading.

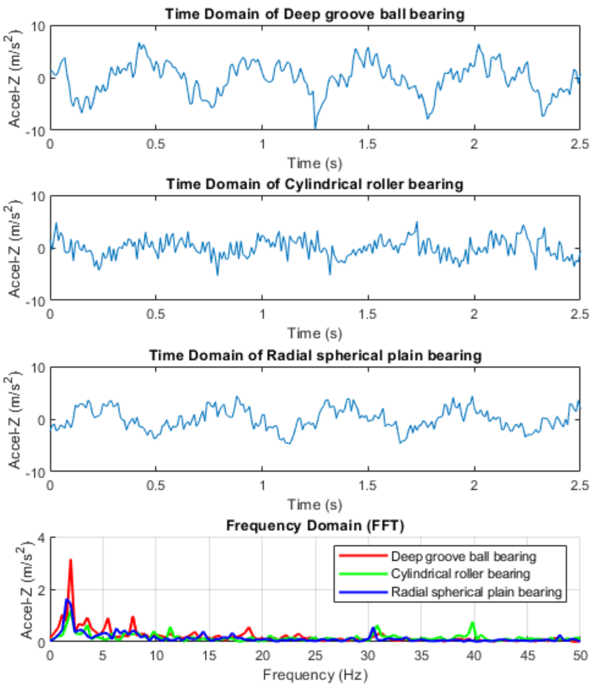

Figure 21. FFT analysis and Comparison between Deep groove ball bearing, Cylindrical roller bearing (single row), and Radial spherical plain bearing at 1900 rpm with loading.

The vibrations in a deep groove ball bearing appear to be relatively low in amplitude with minor fluctu–ations, which means stable behavior, while a cylindrical roller bearing shows more pronounced and irregular fluctuations compared to the deep groove ball bearing, indicating possible sensitivity to dynamic forces or surface irregularities. In a radial spherical plain bearing, it exhibits slower and smoother variations over time, suggesting it dampens vibrations more effectively and may be better for applications requiring lower vibration levels in many systems.

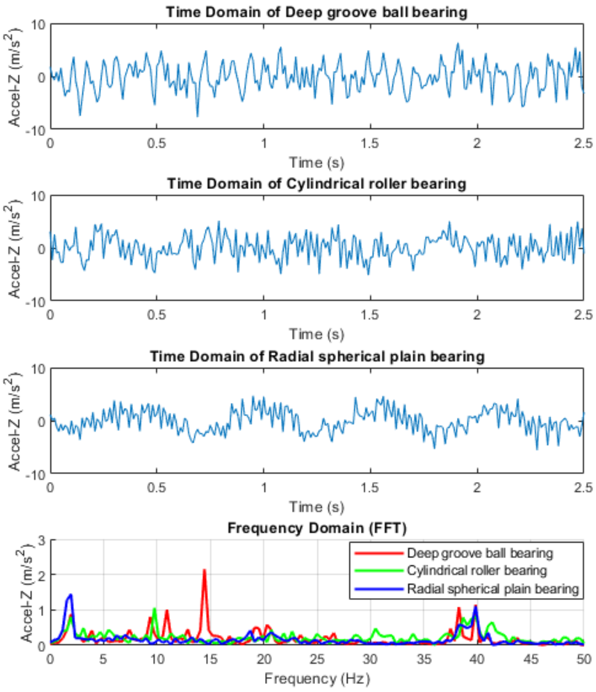

Figure 22. FFT analysis and Comparison between Deep groove ball bearing, Cylindrical roller bearing (single row), and Radial spherical plain bearing at 2450 rpm and with loading.

The graphs of frequency domain (FFT) analysis in figures (17 to 19) show that the deep groove ball bea–ring peaks are concentrated at lower frequencies, indi–cating higher sensitivity to low-frequency components, possibly due to small defects or imbalances in the system. The cylindrical roller bearing displays notice–able peaks at specific mid-range frequencies, reflecting resonance or interaction effects within the bearing com–ponent, while the radial spherical plain bearing exhibits significantly lower peaks overall, suggesting effective vibration damping across a broad frequency range.

In other words, deep groove ball bearings operate best when the shaft rotates without loads applied; at rotational speeds of 1488 and 1900 rpm, they have the lowest response levels when compared to other bearing types (see figures (17) and (18)). All bearing types exhibit comparable response levels at higher speeds, particularly 2450 rpm, which are noticeably lower than those noted at lower rotational speeds (see figures (19) and (22)).

Figures (20 to 22) illustrate the vibration behavior of three types of bearings at a range of speeds with load effects. When the system works under load effect, the amplitude of vibrations of the deep groove ball bearing increases compared to the unloaded case. This indicates that the bearing is more sensitive to external forces and load-induced stresses.

In a cylindrical roller bearing under load, there is a noticeable increase in irregular fluctuations in the vibration pattern, which could be attributed to the rollers handling the additional load.

The vibration profile of a radial spherical plain bearing under load and high speed remains smooth compared to the other bearings, but there is a slight increase in amplitude, which shows some sensitivity to the load.

Also, in figures (20) and (21), upon applying a load to the rotating shaft, single-row cylindrical roller bea–rings outperform other types of displaying the lowest response levels. Conversely, deep groove ball bearings exhibit the highest response under loading conditions (see Figure (21)). This disparity can be attributed to the stif–fness of roller bearings, which is approximately five ti–mes greater than that of ball bearings; consequently, the displacement resulting from applied loads in roller bea–rings is significantly less than that in ball bearings [30].

Furthermore, the hydrodynamic effects on the oil film layer between the shaft and the bearing cause radial spherical plain bearings to operate at their best at high rotational speeds (see figures (19) and (22)). Shaft speed has a direct bearing on this occurrence since higher speed results in higher oil pressure. Because of their increased damping coefficients and reduced stiffness, which improve system stability with less reaction disp–lacement, fluid film journal bearings are typically used in high-speed and heavy-duty applications [30].

6. CONCLUSION

The bearing element represents the most important mechanical part to take into account, due to its essential function in the dynamic response of most rotary mac–hines. Rotating machines are composed of many parts that could be failed. Vibration analysis is one of the most effective techniques for continuously monitoring the state of the rotating system. In this study, in normal state conditions, three types of bearings were tested under different shaft running speeds (1488, 1900, and 2450 rpm) with and without load. The data gathered du–ring the running tests were observed, analyzed by vibra–tion feature extraction and FFT analysis, and presented to find the following:

a- Without load condition

1. The deep groove ball bearing shows stable performance but is more affected by low-frequency vibrations.
2. The cylindrical roller bearing demonstrates higher sensitivity to a wider range of frequencies, making it more suitable for high-load or dynamic environments.
3. The radial spherical plain bearing offers the best damping performance, reducing vibrations across all frequencies.
4. For applications prioritizing minimal vibration, the radial spherical plain bearing is preferable.
5. For high-speed or dynamic environments, cylindrical roller bearings might be better suited despite their higher vibration response.
6. Deep groove ball bearings are optimized for general use where low-frequency vibrations are not critical.

b- With load condition

1. The peaks at low frequencies became more pronounced, indicating that the load amplifies low-frequency vibration components in Deep Groove Ball Bearing.
2. In Cylindrical Roller Bearings, Peaks appear across a broader frequency range, especially at mid-range frequencies, likely due to the rollers adapting to load distribution.
3. The vibration peaks in Radial Spherical Plain Bearing remain minimal, confirming its ability to dampen vibrations effectively even under load.
4. The load amplifies vibrations in all bearings, but the impact is more pronounced in the Deep Groove Ball Bearing and Cylindrical Roller Bearing.
5. The Radial Spherical plane bearing maintains relatively low vibration levels even with the load.
6. The Radial Spherical Plain Bearing shows the least change in frequency response.

The following conclusions can be drawn from these findings:

- The deep groove ball bearing represents the best choice for the low loading condition at the range of rotational speed.
- The cylindrical roller bearing is more suitable for high-loading operation due to its high stiffness.
- The radial spherical plain bearing operates with high performance at high rotational speeds.

Accordingly, the results of this study directly affect industrial applications, especially predictive mainte–nance and bearing selection. Engineers can make well-informed decisions to improve machinery performance by establishing a correlation between vibration features and operating conditions. From a scientific standpoint, this work provides a strong foundation for future dynamic systems research by bridging the gap between theoretical vibration analysis and real-world engineering requirements.

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УПОРЕДНА ЕКСПЕРИМЕНТАЛНА СТУДИЈА ВИБРАЦИЈА РАЗЛИЧИТИХ ЛЕЖАЈЕВА КОРИШЋЕЊЕМ FFT-А И СТАТИСТИЧКЕ ЕКСТРАКЦИЈЕ КАРАКТЕРИСТИКА

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У овој студији, проучавани су обрасци вибрација различитих типова лежајева, као што су куглични лежајеви са дубоким жлебом, цилиндрични ваљкасти лежајеви (једноредни) и радијални сферични клизни лежајеви. Они се најчешће користе у механичким деловима ротационих машина под дејством оптерећења и различитих угаоних брзина: 1488 о/мин, 1900 о/мин и 2450 о/мин. Поред два случаја, првог без оптерећења и другог са оптерећењем од 29,43 N, на вратилу је монтиран и три диска. Техника праћења и анализе вибрација, која је поуздана и тачна метода процене за избор одговарајућег елемента лежаја, анализа брзе Фуријеове трансформације, коришћена је за обраду сигнала вибрација који је прикупљен из три различита типа лежајева за три различите брзине рада са два нивоа оптерећења. Поред тога, неколико индикативних параметара, укључујући средњи квадратни корен, куртозис, однос од врха до врха и асиметрију, издвојени су из података о вибрацијама коришћењем анализе сигнала у временском домену. Из експерименталних резултата, примећено је да куглични лежај са дубоким жлебом даје најмању амплитуду вибрација међу сва два лежаја када је систем без ефекта оптерећења, док цилиндрични ваљкасти лежај даје најмању амплитуду под условима рада са оптерећењем; коначно, радијални сферични клизни лежај је најбоље радио при високим брзинама ротације.