Vibration And Tribology Analysis Of Bearing With Experimental Method And Fea

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Abstract:

bearings are vital components in rotating machinery, significantly impacting performance and reliability. As machines operate under increasingly demanding conditions, bearings must withstand higher loads and temperatures. However, these conditions can lead to increased vibration, accelerating fatigue failure. There are two primary types of bearings: radial hydrodynamic bearings, which rely on fluid film lubrication, and hydrostatic bearings, which utilize external pressure. Hydrodynamic bearings operate under elasto-hydrodynamic lubrication conditions, while journal bearings are prone to self-induced vibrations due to radial clearance. To ensure reliable operation, monitoring bearing performance is crucial. Vibration analysis has emerged as a popular method for detecting potential issues and predicting bearing health. This study focuses on the vibration and tribological analysis of various bearings.

Index Terms - Vibration Analysis, Tribological Analysis, FFT Analyzer

I. INTRODUCTION

Vibration is a complex phenomenon characterized by oscillations around an equilibrium point. While often undesirable, vibration analysis can provide valuable insights into machine condition. By measuring vibration signatures on a machine's external surface, it's possible to detect defects such as imbalance, misalignment, and bearing faults.

Bearings play a crucial role in supporting radial, axial, and thrust loads, enabling smooth rotor operation. Hydrodynamic bearings, in particular, rely on lubricant flow through a throttling hole, generating carrying capacity through hydrodynamic pressure in the clearance between the bearing and shaft. Hydrodynamic journal bearings are essential in various machines, including rotors, turbines, and pumps. These bearings operate under elasto-hydrodynamic lubrication (EHL) conditions and are prone to self-induced vibrations due to radial clearance. Researchers have extensively studied these vibrations, their impact on bearing performance, and methods to mitigate them. Recent studies have also explored the effect of surface roughness on hydrodynamic lubrication in journal bearings. By modifying bearing surface geometry, researchers aim to improve bearing performance. Theoretical studies have investigated the impact of rough journal bearing surfaces on lubrication fields, offering potential avenues for optimization.

II. PROBLEM SPECIFICATION

This investigation explores the effects of vibrations on bearing performance and lifespan under diverse operating conditions. It also examines the tribological properties that influence bearing behavior and failure.

Key Findings

Advanced Modeling Technique - A new three-dimensional dynamic model is developed, utilizing Hertz contact theory. This model integrates multiple degrees of freedom, rotational dynamics, and mixed lubrication, enabling accurate simulations for high-speed applications.

Validation Process - Experimental validation confirms the model's reliability and real-world applicability, ensuring its usefulness in practical scenarios.

III. PRINCIPLES OF TRIBOLOGY AND HYDRODYNAMIC BEARING

The analysis of comparative experiments shows that the feature extraction method can extract bearing fault features more accurately, and the fault diagnosis model based on this method has higher accuracy.

3.1 Principles Of Tribology

Tribological property is a very important characteristics of journal bearings. The bearing characteristics are usually evaluated from the orbital path, the oil film thickness, the friction power loss, the hydraulic and the asperity film pressure distribution. For the lubrication performance of the journal bearings, the main factors include the amplitude of the external load, the lubricating oil temperature, the oil supply pressure and the bearing clearance. The lubrication characteristics of the bearings in different conditions will be discussed in this part.

The origins of hydrodynamic lubrication research date back to the 19th century. Pioneering work by F.A. von Pauli and G.A. Hirn explored the behavior of shafts and bearings under hydrodynamic conditions. Later, Nikolay Petroff's findings confirmed that fluid-film bearing friction is a hydrodynamic phenomenon. Lubrication regimes can be categorized into several types, including:

1. Dry contact: Surfaces interact without lubrication.
2. Boundary lubrication: Partial separation with some surface contact.
3. Mixed lubrication: A combination of boundary and fluid-film effects.
4. Fluid-film lubrication: Surfaces are fully separated by a lubricant film.

In certain applications, such as rolling-element bearings, surface elasticity significantly affects lubricant behavior. This phenomenon is known as elasto-hydrodynamic lubrication (EHL), characterized by distinct properties.

3.2 Principles Of Hydrodynamic Bearing

Lubrication reduces friction between moving surfaces by introducing a substance called a lubricant. Hydrodynamic journal bearings operate under thick film lubrication, where a fluid film supports the load due to the shape and motion of the sliding surfaces.

Hydrodynamic Lubrication Principle

1. At rest, the shaft contacts the bearing surface directly under load. 2. As the journal rotates, it draws fluid into the wedge-shaped clearance, generating pressure. 3. This pressure supports the load, creating a self-acting bearing.

Types of Hydrodynamic Journal Bearings

1. Full Journal Bearing: 360-degree contact between the journal and bearing, capable of handling radial loads in any direction.

2. Partial Journal Bearing: Less than 180-degree contact, typically 120 degrees, suitable for radial loads in one direction.

Key Characteristics

- Eccentricity: The distance between the journal and bearing centers, influencing minimum oil film thickness and radial clearance.

- Oil Film Formation: As the journal rotates, it climbs up the bearing surface, creating a lubricating film.

- Design Considerations: Oil holes are strategically placed to minimize side leakage, particularly in high-load areas.
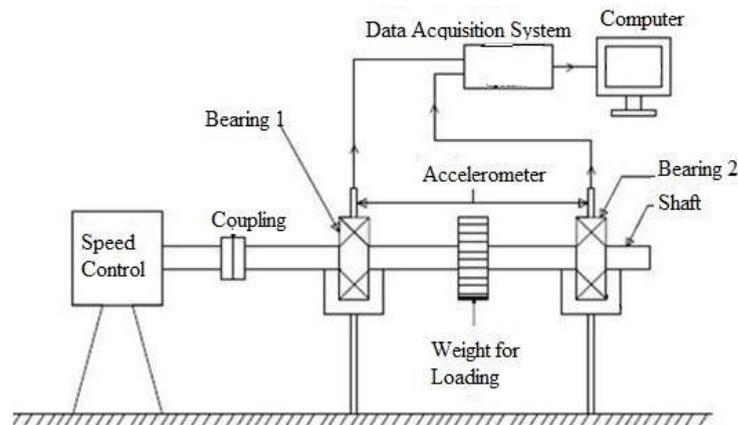
IV. EXPERIMENTATION

Fig.4.1: experimental setup for vibrational analysis of bearing

The bearing testing apparatus consists of a sturdy steel frame supporting a DC motor and a shaft coupled to the motor. The setup includes:

1. A hydrodynamic journal bearing mounting system with oil supply lines. 2. A circular oil groove in the bearing wall for adequate lubrication. 3. A 2.2 kW DC motor operating at up to 2000 rpm.

Testing Procedure

The experiment involves testing three bearing types:

Plain (non-grooved), Two-grooved, Three-grooved

Bearings with specific dimensions (OD: 46 mm, ID: 30 mm, radial clearance: 0.02 mm, length: 15 mm) are used. The testing protocol includes:

1. Varying speeds (600, 900, 1200, and 1500 rpm)
2. Different loads (3 kg and 8 kg)
3. Continuous oil supply
4. Accelerometer measurements for amplitude and frequency analysis

Data Analysis : The experiment generates data on amplitude and frequency, which is used to plot graphs and analyze bearing performance under various operating conditions.

4.1 Tribological Testing of Bearing

The same bearings are subjected to tribological analysis using a Pin on Disc Tribemeter. The test setup involves:

1. Disc: Made of EN24 material, with a diameter of 165 mm and thickness of 12 mm.
2. Pin: Made of Bronze material, with a diameter of 12 mm.

Testing Procedure

The pin and disc are mounted on the tribometer. The load is applied, and the lubricating pump is started. Readings are taken at different loads (3 kg, 5 kg, and 8 kg) and a constant speed of 600 rpm. The tribometer measures frictional force (in Newtons) and wear (in micrometers).

Key Features

1. Maximum loading capacity: 200 N
2. Proximity sensors zero out wear and frictional force readings before testing.

The tribological testing provides valuable insights into the frictional and wear characteristics of the bearing materials under various operating conditions.

The results indicated that tribological performance increased significantly in turbulent flow in smooth and multistep journal bearings.

4.2 Analysis Of Bearing

The vibration analysis results for a two-groove bearing, obtained through FEA using Ansys software, are presented in tables for various load conditions:

- Load: 0 kg, 3 kg, and 8 kg
- Speed: Different operating speeds

Analysis Details :Boundary Conditions: The end-supported bearings are fixed, with load applied to the center bearing. Lubrication: A thick fluid film is used for lubrication.Vibration Measurement: Amplitude is measured in displacement form (mm). Validation: FEA results validate experimental findings for different load and speed conditions.

Two probes are used to capture vibration readings, providing a comprehensive understanding of bearing behavior under various operating conditions.

Table 4.2.1: Two Grooves Bearing at Zero kg Load

Speed (rpm)	Drive Bearing Amplitude (µm)
600	1.98
900	2.60
1200	3.11
1500	6.75

From the Table 4.2.1 and experimental results, it is seen that both results are nearly same. Hence the experimental method used for experimentation is validated by FEA.

Table 4.2.2: Two Grooves Bearing at 3 kg Load

Speed (rpm)	Drive Bearing Amplitude (µm)
600	2.98
900	3.01
1200	4.45
1500	7.98

The Table 4.2.2 shows results obtained by FEA for the two grooves bearing at 3 kg load

Table 4.2.3: Two Grooves Bearing At 8 kg Load

Speed (rpm)	Drive Bearing Amplitude (µm)
600	4.40
900	5.23
1200	8.41
1500	10.12

The results are shown for two grooves bearing only. The fixed supports are provided at two end bearings. The end bearings are fixed with housing. The number of nodes and elements used for meshing are 8150 and 1389. The coarse meshing was used for the meshing. the load applied on the centre bearing for testing. The load was varied for each testing as zero kg, 3 kg and 8 kg. The FEA results for three groove and non-groove bearings are shown in Table 4 and Table 5. It gives the details about the values of amplitude of vibrations of both bearings at respective speed.

Table 4.2.4 Three Groove Bearing At Zero kg, 3 kg, 8 kg Load

Speed (rpm)	Drive Bearing Amplitude (µm)		
	0 kg	3 kg	8 kg
600	1.63	2.45	3.67
900	2.45	6.05	6.89
1200	2.98	8.23	7.98
1500	2.90	11.8	8.40

Table 4.2.5 Plain Bearing at Zero kg, 3 kg, 8 kg Load

Speed (rpm)	Drive Bearing Amplitude (μm)		
	0 kg	3 kg	8 kg
600	8.23	13.64	20.55
900	9.23	11.47	17.25
1200	8.64	10.62	16.58
1500	6.78	8.08	15.79

V. RESULTS AND DISCUSSION

The results were found out for vibrational characteristics of hydrodynamic journal bearings with two grooves and three grooves bearing also for plain bearing. Also the results for tribological results were obtained for asperities of bearing and shaft are represented in tabular form are explained.

5.1 Comparing Both Methode Experimental And FEA

Using FFT analyzer and FEA, vibration readings were captured for hydrodynamic journal bearings under various speed and load conditions. The results are presented in graphical form, showing amplitude vs. speed for each load condition.

Load Conditions- No load, 3 kg loading, 8 kg loading
Speed Conditions- 600 rpm, 900 rpm, 1200 rpm,1500 rpm
Bearing Types- Non-groove, Two-groove, Three-groove

Plain Bearing

Table 5.1.1: Plain Bearing at Zero Load

Speed (rpm)	Drive			Drive End	
	Amplitude (μm)	FEA Amplitude (μm)	Frequency (Hz)	Amplitude (μm)	Frequency (Hz)
600	10.40	8.23	5.05	4.11	4.10
900	11.70	9.23	7.05	5.98	6.06
1200	10.50	8.64	3.98	4.72	5.05
1500	6.65	6.78	3.09	3.76	3.88

Table 5.1.2: Plain Bearing at 3 kg Load

Speed (rpm)	Drive			Drive End	
	Amplitude (μm)	FEA Amplitude (μm)	Frequency (Hz)	Amplitude (μm)	Frequency (Hz)
600	12.5	13.64	5.00	7.85	4.00
900	10.64	11.47	7.00	7.65	6.00
1200	9.18	10.62	4.00	5.84	5.10
1500	8.94	8.08	3.10	4.95	4.12

Table 5.1.2: Plain Bearing at 3 kg Load

Speed (rpm)	Drive			Drive End	
	Amplitude (μm)	FEA Amplitude (μm)	Frequency (Hz)	Amplitude (μm)	Frequency (Hz)
600	20.10	20.55	5.00	8.10	4.00
900	19.00	17.25	6.00	6.00	6.10
1200	18.20	16.58	4.00	5.10	5.00
1500	17.00	15.79	3.00	4.10	3.90

Two Groove Bearing

Table 5.1.3: Two Grooves Bearing at Zero Load

Speed (rpm)	Drive			Drive End	
	Amplitude (µm)	FEA Amplitude (µm)	Frequency (Hz)	Amplitude (µm)	Frequency (Hz)
600	2.97	1.98	7.10	1.58	10
900	3.66	2.60	8.25	1.52	8.98
1200	4.80	3.11	7.10	1.17	12.20
1500	5.82	6.75	7.65	1.27	13.20

Table 5.1.4: Two Grooves Bearing at 3 kg Load

Speed (rpm)	Drive			Drive End	
	Amplitude (µm)	FEA Amplitude (µm)	Frequency (Hz)	Amplitude (µm)	Frequency (Hz)
600	2.70	2.98	7.05	1.16	10
900	2.80	3.01	8.11	1.92	9
1200	4.31	4.45	6.05	1.65	14
1500	5.65	7.98	8.10	2.57	22

Table 5.1.5: Two Grooves Bearing at 8 kg Load

Speed (rpm)	Drive			Drive End	
	Amplitude (µm)	FEA Amplitude (µm)	Frequency (Hz)	Amplitude (µm)	Frequency (Hz)
600	3.32	4.40	7.00	1.80	10
900	4.84	5.23	7.08	2.20	9.10
1200	5.79	8.41	5.98	1.95	13.95
1500	8.11	10.12	7.10	2.30	17.20

Three Groove Bearing

Table 5.1.6: Three Grooves Bearing at Zero Load

Speed (rpm)	Drive			Drive End	
	Amplitude (µm)	FEA Amplitude (µm)	Frequency (Hz)	Amplitude (µm)	Frequency (Hz)
600	1.46	1.63	16.00	1.05	17.10
900	2.55	2.45	19.98	1.20	19.99
1200	2.89	2.98	18.00	1.50	23.20
1500	2.70	2.90	21.20	1.10	25.35

Table 5.1.7: Three Grooves Bearing at 3kg Load

Speed (rpm)	Drive			Drive End	
	Amplitude (µm)	FEA Amplitude (µm)	Frequency (Hz)	Amplitude (µm)	Frequency (Hz)
600	2.27	2.45	16	1.4	40
900	5.90	6.05	20	2.2	20
1200	8.00	8.23	18	2.4	25
1500	9.00	11.8	24	3.0	24

Table 5.1.8: Three Grooves Bearing at 8kg Load

Speed (rpm)	Drive			Drive End	
	Amplitude (μm)	FEA Amplitude (μm)	Frequency (Hz)	Amplitude (μm)	Frequency (Hz)
600	4.16	3.67	15.10	1.78	28.00
900	7.19	6.89	18.00	1.88	22.10
1200	7.72	7.98	18.20	3.82	25.5
1500	8.40	8.40	23.20	3.92	24.00

The Fig. 5.1.1 shows the graph of amplitude of vibration versus speed for zero kg loads at speeds of 600 rpm, 900 rpm, 1200 rpm and 1500 rpm respectively
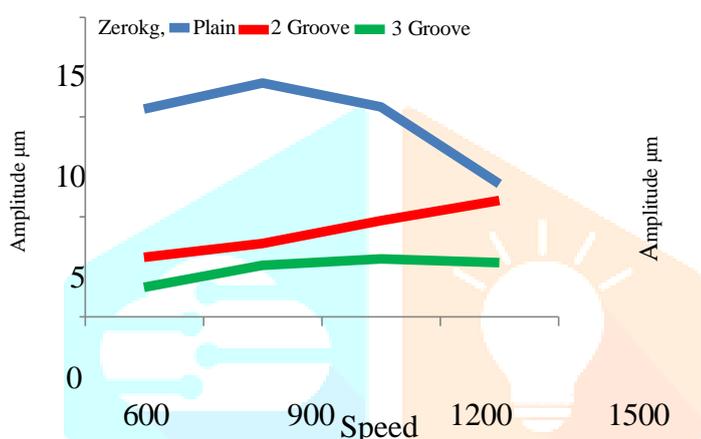
Fig. 5.1.1: Speed Vs Amplitude for zero kg Load, Drive
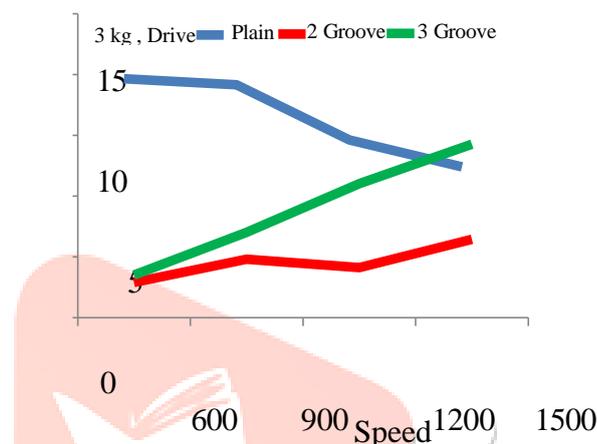
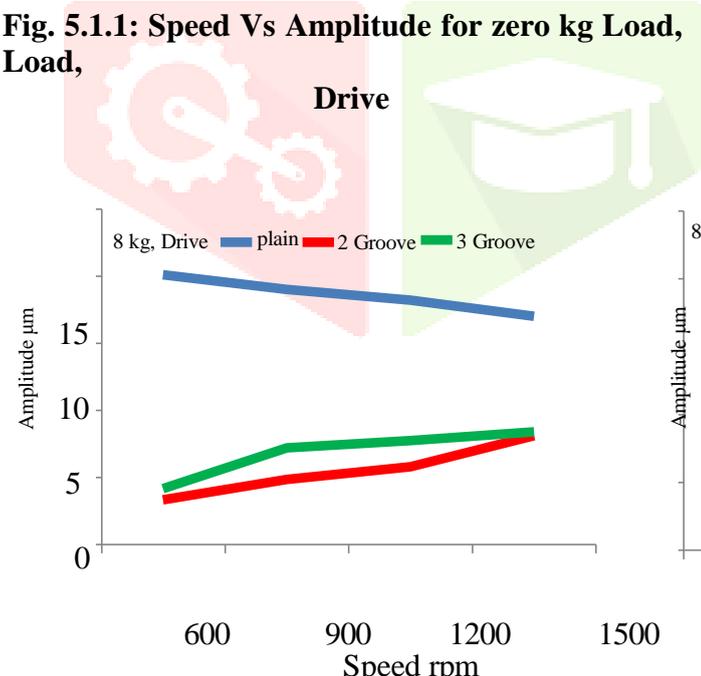**Fig.5.1.2: Speed Vs Amplitude for 3 kg Load, Drive End**

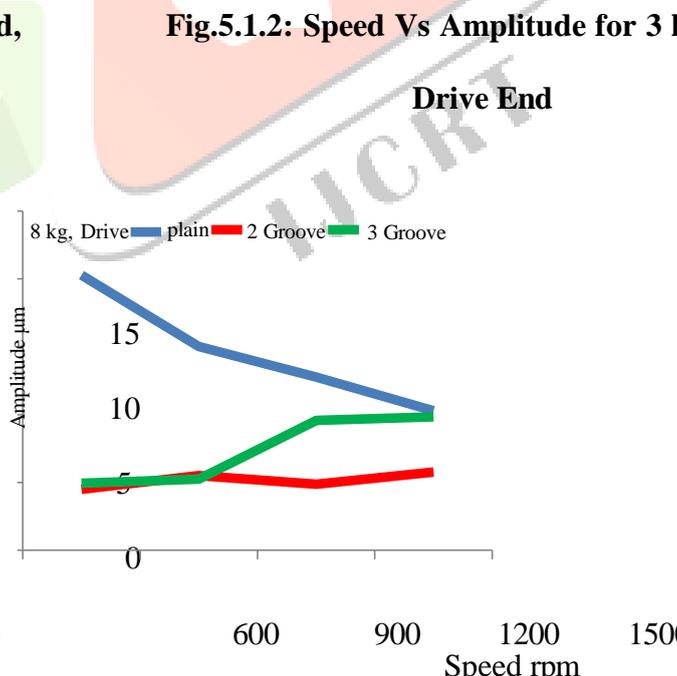**Fig. 5.1.3: Speed Vs Amplitude for 8 kg Load, Drive End**

Fig. 5.1.4:Speed Vs Amplitude for 8 kg Load, Drive

From the above graphs fig.5.1.1,fig 5.1.2,fig5.1.3, fig 5.1.4 it is seen that the plain bearings shows higher value of amplitude of vibration than two groove loading conditions.

5.2 Tribologycal Analysis

This study investigates the tribological behavior of EN24 disc material and bronze pin material under various operating conditions.

Test Parameters

- Speed: Varied
- Load: Varied
- Test Duration: 10 minutes
- Lubricant: SAE 20W40 engine oil, continuously supplied via pump
- Wear Track Diameter: 40 mm (constant)

Data Acquisition: The test setup captures data on: Wear, Coefficient of Friction

The acquired data provides insights into the frictional and wear characteristics of the material pair under different operating conditions.

The value of frictional force is directly taken from display and coefficient of friction is calculated as follows,

$$COF = \frac{FF}{Load}$$

The coefficient of friction reduces slightly as load goes on increasing. The highest coefficient of friction was obtained for a load 29.43 N. The value of coefficient of friction was 0.088 at 600 rpm. The results are shown in Table 5.2.1

The graph shown in Fig. 6.7 was plotted on software, from the figure it is seen that the value of wear goes on increasing at with start. The value of coefficient of friction and frictional force goes on reducing with respect to time.
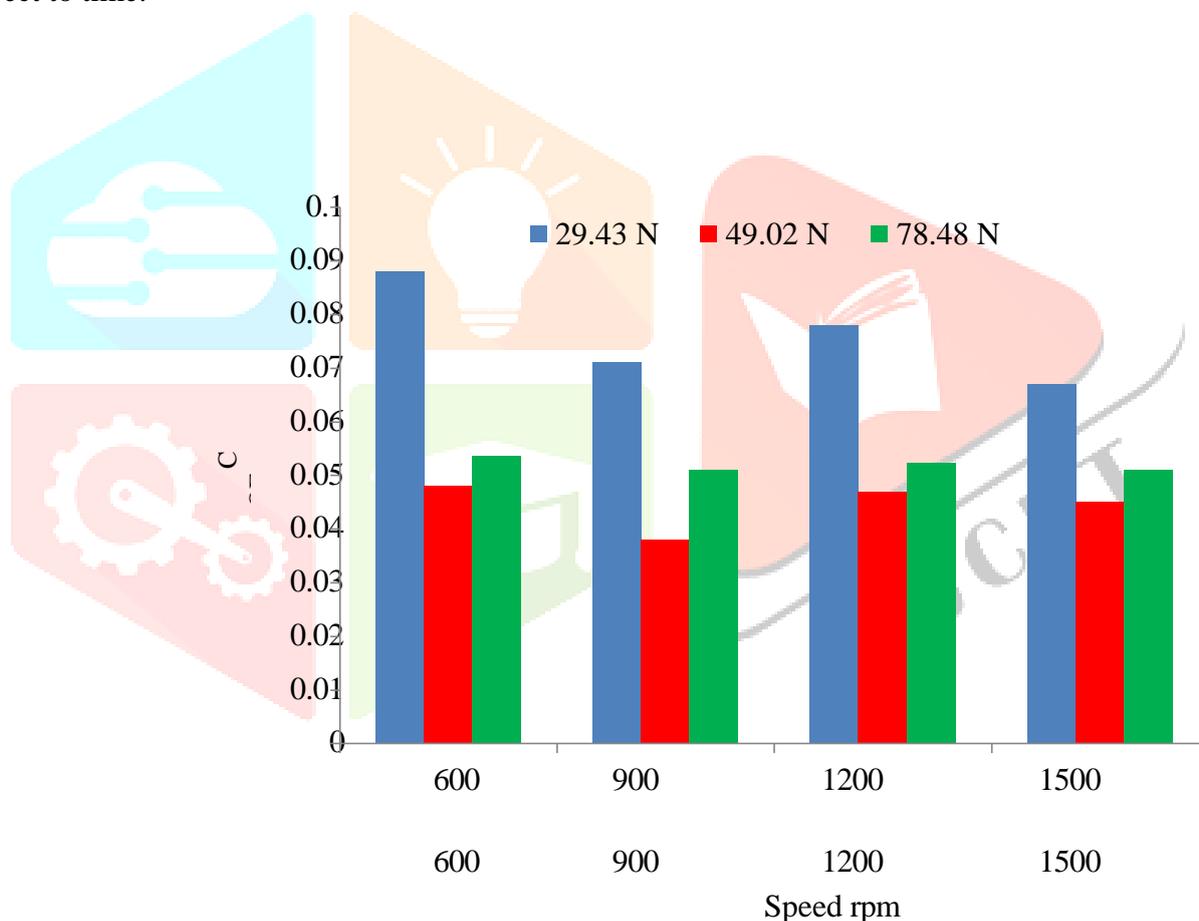

Fig. 5.2.1: Coefficient of friction Vs Speed

For the load 29.43 N, the value of coefficient of friction are higher than other two loads. The fictional force is not much affected by the groove numbers. It is seen that value of coefficient of friction for the load 78.48 N are lower than that observed for 29.43 N.

Table 5.2.1: Coefficient Of Friction

Load (N)	Speed(rpm)	Frictional Force (N)	Coefficient of friction	Wear (μm)
29.43 N	600	2.6	0.088	34
	900	2.1	0.071	27
	1200	2.3	0.078	25
	1500	1.98	0.067	22
49.025 N	600	2.2	0.048	35
	900	1.9	0.038	29
	1200	2.3	0.047	28
	1500	2.2	0.045	26
78.48 N	600	4.1	0.05352	50
	900	4.0	0.05097	47
	1200	4.2	0.05224	45
	1500	4.0	0.0510	40

VI. CONCLUSION AND FUTURE SCOPE

6.1 Conclusion

This study Demonstrate and compares the performance of plain (non-grooved), two-grooved, and three-grooved hydrodynamic journal bearings. The effects of grooves, load, and speed on tribological characteristics and vibrations are evaluated. At no load and low speed, plain bearings show higher vibration amplitude than grooved bearings. As speed increases, vibration amplitude decreases in plain bearings but increases in grooved bearings. Higher at low speeds, decreasing as speed increases. Two-groove and three-groove bearings exhibit lower vibration amplitudes. Grooved bearings generally show lower vibrations compared to non-grooved bearings, with varying performance depending on speed and load conditions. Bearing design parameters, such as groove configuration, load, and speed, significantly impact tribological characteristics and vibrations. further work should explore advancements in bearing design, performance, and reliability.

6.2 Future Scope

Further Study should investigate potential applications in various industries and significant benefits for efficiency, reliability, and sustainability. Improved bearing performance can contribute to the development of more efficient and reliable electric vehicles and renewable energy systems.

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