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Selection of Appropriate Range of Operating Parameters for Tribological Simulation of Journal Bearings on a Test Rig.

Jai Prakash sharma[1] , M R Tyagi[2], G D Thakre[3]

[1,2]Department of Mechanical Engineering, Manav Rachna University, Faridabad, (Haryana), India

[3]Tribology and Combustion Division, CSIR-Indian Institute of Petroleum, Dehradun, (Uttarakhand) India

Abstract

Hydrodynamic bearings are most commonly used entities in process plants. These bearings are huge structures designed according to the load and speed requirements. The testing and evaluation of these bearing is a tedious and complicated process. Hence, lab level testing on a scale down model can be of significant help for bearing designers. Moreover, these bearings operate under severe conditions which are difficult to replicate on a laboratory test rig. It is therefore imperative to select operating parameters that can best simulate the operating conditions of real bearings on a test rig. The present paper therefore provides an experimental methodology to select the operating parameters for testing of bearings on a test rig. A parameter based approach has been utilized with the help of Taguchi design of experiment technique. The statistical techniques of analysis of variance and signal to noise ratio were used to study the tribological characteristics of journal bearing. The combinations of experimental design parameters were related with the Sommerfeld Number to ascertain the lubrication regime. On the basis of the investigations it has been observed that the bearing performance is significantly influenced by the operating parameters. The methodology adopted provides the information on selection of operating parameters for bearing testing on test rig.

Keywords: Hydrodynamic lubrication, Journal Bearing, Wear, Taguchi Method, Regression analysis, ANOVA.

1.0 Introduction

Bearings are critical component for rotating machinery system. Load bearing capacity of such bearing is good but not sufficient to bear higher load. For such condition Hydrodynamic bearings are used .Hydrodynamic performance gained significant importance among design engineers. Approches were proposed by various researcher for design and performance evaluation. [1, 2]. A way ahead by Reddecliff and Vohr [3] predicted the steady state and time-dependent performance of bearings. The authors presented analysis for hydrostatic bearings by considering the effect of turbulence, inertia and compressibility in fluid film. The theoretical analysis was validated experimentally and the bearings were observed to be stable in all conditions. However, in practical applications it has been observed that the bearings are susceptible to high friction and wear during the starting and stopping of machinery. In case of plain hydrodynamic journal bearing at start, the shaft move in a spiral whirling locus and gets separated from the bearing. Prior to separation the shaft slides over the bearing. Similarly at stopping, it follows a hydrodynamic locus and then rest on the bearing [4]. However, during the start up, the acceleration and surface roughness significantly affect the tribo-dynamic performance of the bearings [5]. Sinhasan et al. [6] investigated the influence of elastic deformation of bearing shell and presented the effects of bearing flexibility on the performance characteristics. On the similar lines Poulios et al [7] determined the evolution of lubricant film thickness and pressure in hydrodynamic contacts. In order to achieve enhanced performance at higher loads, the use of hybrid journal bearing with different hole-entry configurations too have been proposed [8]. In hybrid bearing configurations, the use of external pressure source help in enhancing the start-up performance by reducing the wear and improving stability of the bearings.

At starting and stopping of HDB causes wear which reduces life of bearing with geometrical changes but it was found that minor wear enhances lubrication at low speed cases[9]. However, Hashimoto et al [10] through their experimental and theoretical studies concluded that the geometric changes due to wear significantly affect the steady-state characteristics of the bearings. Scharrer et al. [11] reported that the bearing performance i.e. the stiffness, damping and leakage, degrades steadily for wear greater than 5 percent of radial clearance. The wear within the bearings is caused by the sliding motion during starting and stopping. However, stopping do not have significant impact on the wearing process [12-13]. Wear due to contaminant particles in oil was investigated by Ronen et al. [14]. It was reported that contaminated oil increases wear by a factor of 20 at minimum oil film thickness location. The wear due to abrasive particles can be mitigated by incorporating grooves on the shaft surfaces [15]. Machado and Cavalca [16] presented a mathematical model to represent wear in bearings considering the maximum depth, angular span and the angular position. The authors evaluated and identified the wear parameters from the unbalanced frequency response of the rotor-bearing system in directional coordinates. Konig et al. [17] developed a wear prediction model considering the bearing wear on macroscopic and asperity contact scale. Wear due to contaminant abrasive particles is mainly dependent on the hardness of shaft and liner materials used. Numerical simulations have revealed that smaller the shaft-to-liner hardness ratio higher is the liner wear and lesser is the shaft wear [18].

Efforts are made to develop new material for greater performance in HDB. The brass and tin-based materials are among the common bearing materials used in bearings [19]. Mathavan and Patnaik [20] investigated the friction and wear behaviour of Aluminum based bearing materials. The study demonstrated how the tribological performance change with the concentration of alloying elements (Si, Cr, Ni etc) in the base metal. Bajwa et al. [21] studied the wear performance of Sn-based coatings. The study revealed that the Sn-based coatings were able to significantly reduce the friction and wear under mixed and boundary lubrication regimes. Silicon carbide (SiC) when used as bearing material is able to provide higher load carrying capacity. This superior performance is attributed to its high hardness and good thermal conductivity. SiC has proved to be useful in applications involving water as lubricant [22].

Water is also used in various marine and nuclear power application as lubricant. When water is used as lubricant, power and temperature loss are not major issue but bearing material surface finish and tolerance need more attention. [23]. Appropriate viscosity of lubricant increases load capacity but due to friction it decreases. Heshmat and Pinkus [24] studied the effect of hot and cold lubricant mixing at the bearing inlet. They presented empirical relations that are able to determine the inlet temperatures as a function of operating conditions and bearing sizes. It has been observed that high VI lubricants generate slightly lower surface temperatures with reduction in power loss upto 10% as compared to their low VI counterparts. [25]. In recent times ionic liquids are emerging as new generation lubricants. They behave as Newtonian fluids in hydrodynamic domain even at high shear rates. The low viscosity ionic liquids are able to significantly reduce the friction within the bearings [26]. The friction in bearings can also be minimized with the concept of composite-film bearings (CFB). The CFB uses double layer lubricant film that supports the applied load. This approach not only helps in maintaining safe film dimension but also reduces the contact friction [27]. The surface textures too have been extensively studied to investigate their efficacy in reducing the friction and wear in bearings [28].

On the basis of literature survey carried out, it has been observed that the tribological performance of bearings poses a major challenge for the Tribology community and need more investigation. The present study attempts to present empirical relations for the tribological performance of bearings as a function of operating parameters. The experimental study has been performed with the aim to provide insight on the selection of effective operating parameters for investigating the friction and wear behaviour of the bearings. The experiments were planned using Taguchi method for Design of Experiments concept. The levels of operating parameters used in the study were selected based on the literature review undertaken.

2.0 Materials and Method

In the present investigation the test journal bearings shown in figure 1 were fabricated using EN-31 steel. The journal used too was fabricated using EN-31 steel. The physical properties of the bearing material and the geometric dimensions of journal and bearing are given in Table 1 and 2 respectively. EN-31 steel was chosen as the bearing material owing to its excellent wear resistance.

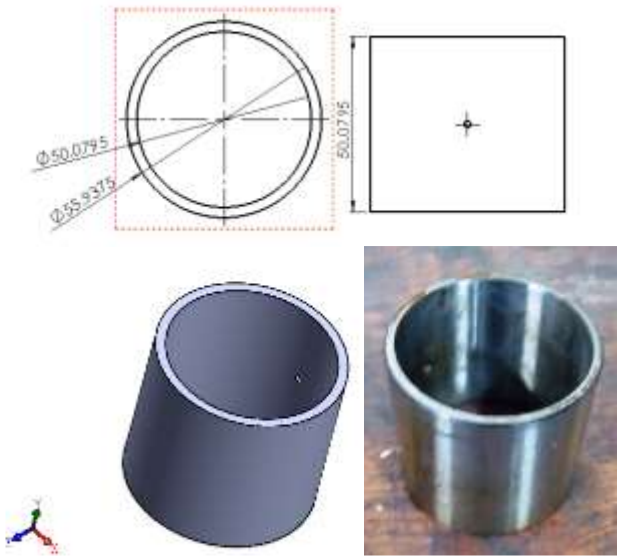

Fig. 1 Fluid film bearing

Table 1: Thermo Physical Properties of bearing material

Sr. No.	Work material	EN-31 steel
1.	Thermal conductivity (W/mk)	46.6
2.	Density(g/cc)	7.81
3.	Electrical resistivity ((Ω-cm)	0.0000218
4.	Specific heat capacity (J/g °C)	0.475

Table 2: Geometric dimension of journal bearing

Sr. No.	Geometric Property	Value (mm)
1.	Inner diameter	+0.075 $50^{+0.084}$
2.	Outer diameter	−0.10 $56^{-0.025}$
3.	Axial length	50

The bearings were lubricated using commercial bearing oil used for lubrication of bearings in steam turbines. The physico-chemical characteristics of the lubricant used are given in Table 3.

Table 3: Physico-chemical properties of test lubricant

Sl. No.	Parameter	Characteristic value
01	Density at 15 °C (g/cm³)	0.871
02	Flash Point, °C	200
03	Kinematic Viscosity, cSt at 40°C at 100°C	 46.3 6.94
04	V.I.	106
05	Pour point, °C	-30

2.1Experimental

The experiments were performed on DUCOM India make Journal Bearing Test Rig, shown in Figure 2. The test rig used is fully computer controlled and consists of loading system, lubrication system and the test bearing. Load is applied though pneumatic bellows which receive compressed air from the compressor. The lubrication system comprises of lubricant reservoir, pump and heater for continuous supply of lubricant to the bearing with temperature control. The Journal is rotated with the help of electric motor coupled with variable frequency drive for the desired rotational speed which are connected through belt drive. Frictional torque is measured by used of load cell and bearing wear is measured through weight difference before and after each experiment run.

Test setup was cleaned with solvent and test lubricant prior to actual test to remove any trace of dirt/dust. Also test bearing is thoroughly cleaned with solvent and fried in an oven at 50° C and measured before test upto four decimal place of gram (g) then assembled in the bearing housing. After this the lubricant pump and heater were switched on and as it reaches the desired temperature, journal bearing was start rotating.

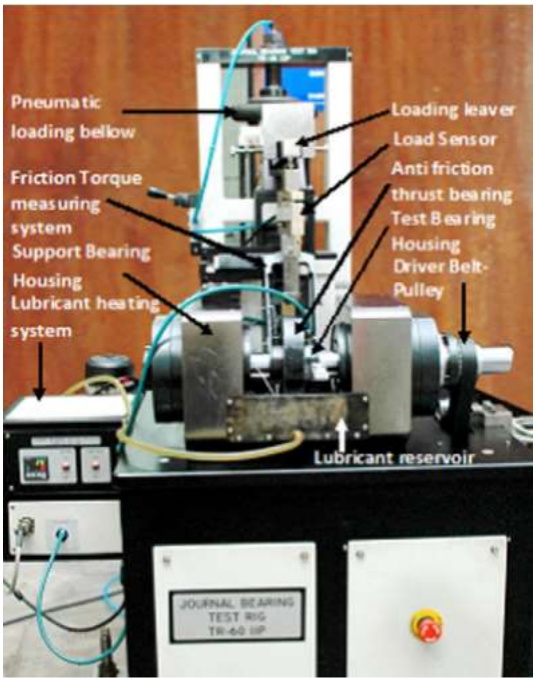

Fig. 2: Journal bearing test rig

As desired speed was achieved by journal, load was applied for four hours. After test bearing was removed and cleaned thoroughly for weight loss due to wear. The contact friction in terms of frictional torque was continuously monitored and converted into coefficient of friction using empirical relation. Each of the experiment was repeated twice in order to ascertain the repeatability and reproducibility of the results.

The specific wear rate (W_s) was calculated using the formula:

$$W_s = \frac{V}{P*L} \qquad (1)$$

Where V = wear volume (m^3) = $\frac{(w_1 - w_2)*1000}{\rho}$

P = normal load (N)

L = Sliding distance (m)

Where, w_1 and w_2 are the specimen weights before and after the test. ρ is experimental density of the specimen. For a better understanding of the lubrication regime prevailing in the experiments performed, Sommerfeld number was calculated using:

$$S = \left(\frac{r}{c}\right)^2 \mu \frac{N}{P} \qquad (2)$$

Where S = Sommerfeld Number or bearing characteristic number (dimensionless), r = shaft radius (m), c = radial clearance (m), μ = absolute viscosity of the lubricant (Pascal*s), N = speed of the rotating shaft (1/s) and P = load per unit of projected bearing area (N/m^2)

2.2Experimental Design

The experiments were planed using the Taguchi design of experiments technique. In the present study three factors at three levels have been considered for the experimental design [29]. The three level Taguchi design was therefore selected for the three input parameters (Load, Speed and Temperature). The tribological characteristics namely contact friction and wear were selected as the output or response. The parameters to be studied and the attributes of respective levels are indicated in Table 4.

Table 4: Operating parameters

Operating parameters	Level 1	Level 2	Level 3
Applied Load (N)	100	1000	2000
Journal speed (RPM)	10	100	1000
Lubricant temperature °C	30	50	75

The levels (low, medium and high) of operating parameters were selected on the basis of the bearing application and the operating limits of the test rig. The corresponding L9 orthogonal array is given in table 5.

Table 5: L9 Orthogonal array for the experimental design

Experiment No.	Applied Load (N)	Speed (RPM)	Temperature (°C)
1	100	10	30
2	100	100	50
3	100	1000	75
4	1000	10	50
5	1000	100	75
6	1000	1000	30
7	2000	10	75
8	2000	100	30
9	2000	1000	50

The signal/noise ratio (S/N) was used to optimize the quality characteristics of experimental data. In the present work for the contact friction and bearing wear, "Smaller the better", approach has been followed. The S/N ratio is given by the equation;

$$S/N = -10\log_{10}\frac{1}{a}\sum_{i=1}^{a} b^2 \tag{4}$$

Where a – number of experiments, b - number of response value.

The ANOVA was performed to find out the operating parameters that significantly affected the tribo-performance of the bearing. The experimental outcomes were further employed to establish the quadratic model for friction and bearing wear using Response Surface technique. This model can be expressed as:

$$Z = c_0 + \sum_{i=1}^{k} c_i Y_i + \sum_{i=1}^{k} c_{ii} Y_i^2 + \sum_{i=1}^{k} c_{ij} Y_i Y_j \tag{5}$$

Where, Z is the response variable namely friction and bearing wear in the present case. c_0, is the constant, c_i, c_{ii} and c_{ij} are the coefficients of linear, quadratic and cross product terms, respectively. Y_i is the coded variable. Finally the tests for significance of the regression and individual model coefficients were carried out. The complete experimental design has been obtained using Minitab v17 software.

3.0 Results and discussion

3.1Experimental observations

The frictional force generated over the entire test duration for the different combinations of operating conditions is shown in figure 3.

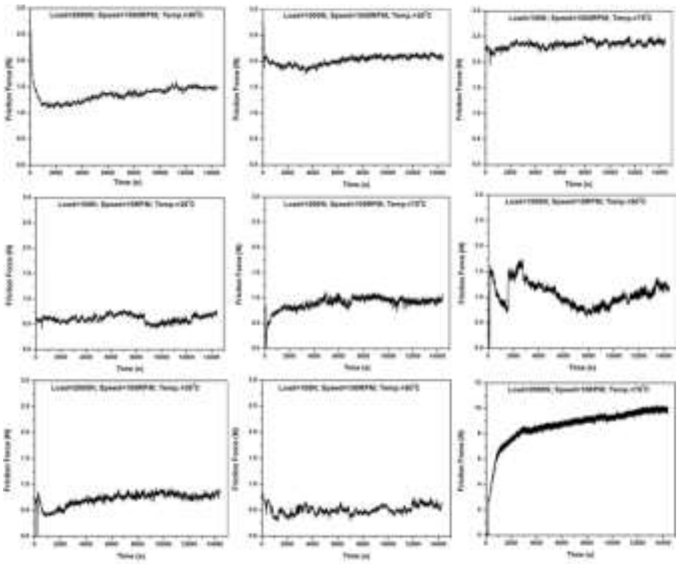

Fig. 3: Friction vs. time for different test run.

It is observed from the figure that the friction encountered is dependent on the operating parameters. Lower friction force to the order of 0.5N is observed in case of (i) low load, low speed and temperature, (ii) high load, moderate speed and low temperature, (iii) low load, moderate speed and temperature [6]. The friction force increased to 1-2.5N with operating conditions of (i) moderate load, moderate speed, high temperature, (ii) moderate load, high speed, low temperature, (iii) high load, high speed, moderate temperature, (iv) moderate load, low speed and moderate temperature and (v) low load, moderate speed and high temperature. Excessive high friction force to the order of 10N is observed in case of high load, low speed and high temperature.

The Sommerfeld Numbers calculated for different combinations of operating parameters is shown in table 6. It is observed that in 0.5N of friction force, has the Sommerfeld Number in the range of 2.08E-6 to 4.17E-6, 1-2.5N has range of 4.17E-6 to 4.1E-4. Furthermore highest friction force is encountered when the Sommerfeld Number is of the order of 2.08E-7. This suggests that the selected combinations of operating conditions are able to simulate lubrication regimes in the range of mixed/partial lubrication to hydrodynamic lubrication. Higher friction is recorded when the bearing operates in the mixed/partial lubrication regime represented by High Load, Low Speed and High Temperature (i.e. Experiment No. 7). On the contrary lowest friction is reported when the lubrication regime is in close proximity of elastohydrodynamic lubrication represented by the Experiment No. 1, 2 and 8. The other operating parameter combinations prevail in Hydrodynamic lubrication regime and the friction force increases with increase in the Sommerfeld Number.

Experimental result shows minimum wear is 0.0001g and max is .0081 g. However, the absolute value of weight loss is unable to provide a clear picture on the bearing wear, hence, the specific wear rate has been taken into consideration. The smallest value of specific wear rate is observed in case of Experiment No. 9 followed by Experiment No. 6. The combinations of operating parameters for these experiments represents hydrodynamic lubrication regime and hence lowest value of specific wear rate is observed. On the contrary highest specific wear rate is observed in case of Experiment No. 4, where a smaller value of Sommerfeld Number corresponds to mixed/partial lubrication regime.

Table 6: Wear behaviour of test bearings

Experiment No.	Applied Load (N)	Speed (RPM)	Temperature (°C)	Bearing Wt. loss (gm)	S/N Ratio	Specific wear rate (mm^3/Nm)	Sommerfeld no.
1	100	10	30	0.0001	80	4.16E-4	4.17E-6
2	100	100	50	0.0003	70.4576	2.75E-4	4.17E-5
3	100	1000	75	0.0013	57.7211	3.77E-4	4.17E-4
4	1000	10	50	0.0081	41.8303	2.35E-3	4.17E-7
5	1000	100	75	0.0018	54.8945	1.65E-4	4.17E-6
6	1000	1000	30	0.0026	51.7005	7.53E-5	4.17E-5
7	2000	10	75	0.0018	54.8945	2.61E-4	2.08E-7
8	2000	100	30	0.0032	49.8970	1.47E-4	2.08E-6
9	2000	1000	50	0.0012	58.4164	1.74E-5	2.08E-5

3.2Statistical analysis

The experimentally observed results have been statistically investigated by determining the S/N ratio for different levels of operating parameters. The main effects plot for S/N ratios is shown in figure 4 (a, b). It is observed from figure 4a that the bearing wear decreases with increase in speed and temperature. The lowest bearing wear has been observed at 1000 RPM and 75°C temperature. However, in case of load it is observed that initially the bearing wear decreases with increase in load, but it increases with further increase in load. Lowest bearing wear is obtained at a load of 1000N. Similarly in case of friction, the friction decreased with increase in speed. The lowest friction is observed at 1000 RPM. The friction initially increased with increase in load and temperature, but subsequently decreased with further increase in load and temperature. The lowest value of friction was observed at 100N load and 75°C. The optimum level of input parameters considering the smaller the better criterion for bearing wear is Load 2-Speed 3-Temperature 3, and for the friction Load 1- Speed 3- Temperature 3.

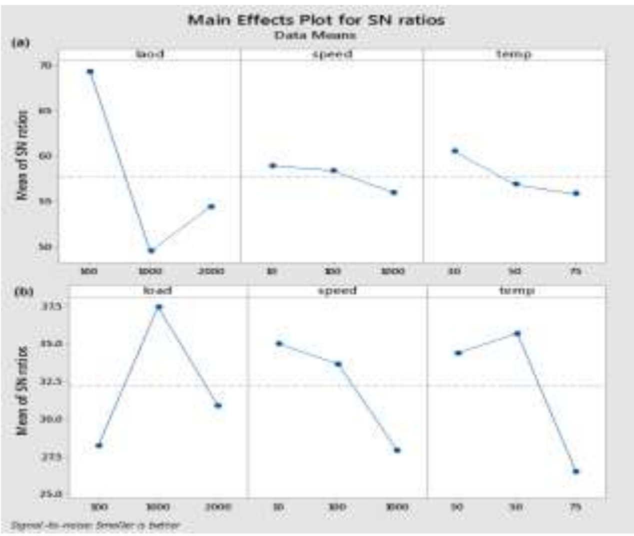

Fig. 4: Main effect plot for (a) Bearing wear and (b) friction.

Subsequently, the analysis of variance (ANOVA) was performed to recognize the factors that significantly affect the bearing's tribo-performance. The ANOVA with the response measurements for bearing wear and friction are given in Table 7 and Table 8 respectively. The percentage contribution as observed from Table 7 reveals that the bearing wear is significantly influenced by the applied load followed by rotational speed and then the lubricant temperature. The applied load has a maximum contribution of 42%, followed by rotation speed of 13% and lubricant temperature of 8% on the bearing wear. Similarly, it is clear from Table 7, that the friction is largely influenced by Lubricant temperature, followed by applied load and then the speed. The percentage contributions are 30%, 24% and 22% for temperature, load and speed respectively.

Table 7: ANOVA for bearing Wear

Source	DoF	Adj SS	Adj MS (Variance)	F-Value	P-Value	Percentage contribution
Load	2	0.00002	0.00001	1.14	0.468	41.67
Speed	2	0.000005	0.000003	0.3	0.771	12.50
Temp.	2	0.000004	0.000002	0.24	0.808	8.33
Error	2	0.000017	0.000009			37.50
Total	8	0.000046				100

Table 8: ANOVA for bearing Friction

Source	DF	Adj SS	Adj MS (Variance)	F-Value	P-Value	Percentage contribution
Load	2	0.006144	0.003072	0.98	0.504	23.52
Speed	2	0.005868	0.002934	0.94	0.516	22.46
Temp.	2	0.007865	0.003932	1.26	0.443	30.10

Error	2	0.006247	0.003123			23.91
Total	8	0.026124				100

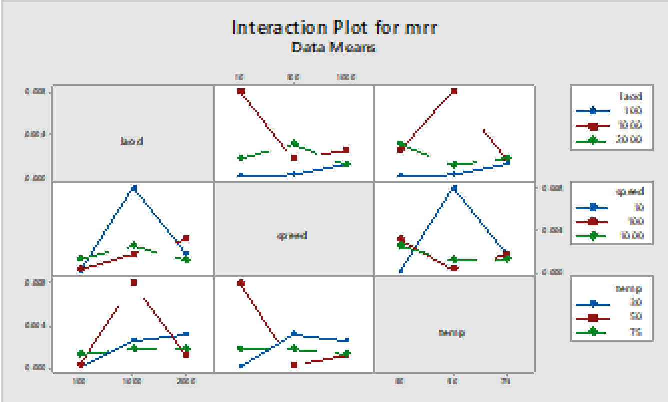

Fig. 5: Interaction of improved bearing performance

It is observed from the interaction plot that at lower speed (10 rpm), initially mean value for wear increases with increase in load and then decreases. At 50°C mean value of wear increases and then decreases with increase in load. At 1000 N load mean value of wear drastically decreases and then increases slowly with respect to speed. At 50°C mean value of wear initially decreases with respect to increase in speed. At 1000 N load and 10 rpm, mean value increases and then decreases with respect to temperature.

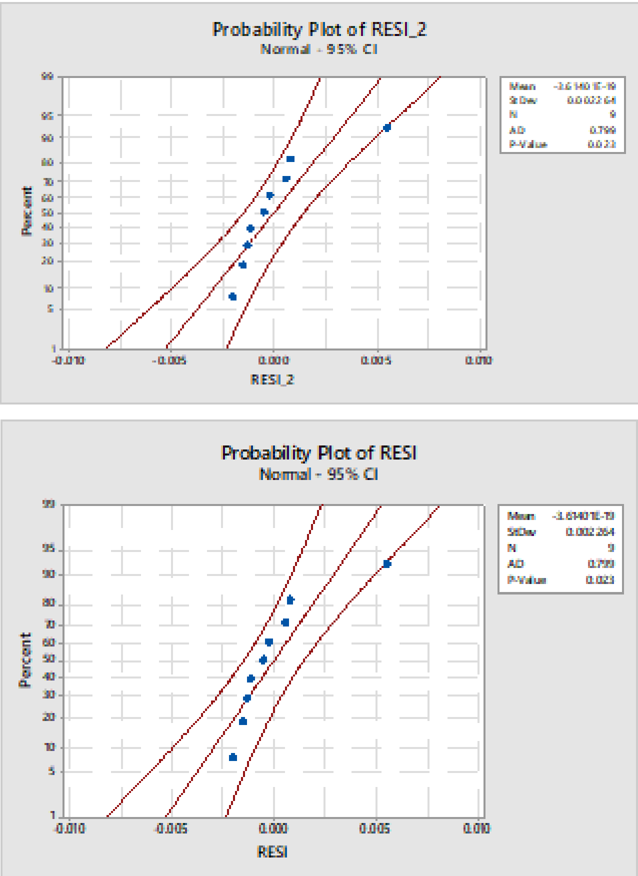

Fig. 6: Normal probability plots

Figure 6 shows the normal probability plots for the experimental data obtained. It is observed that all the experimental findings are in close agreement with the statistical analysis. The experimental data obtained are close to the curve which shows that there is a 95% confidence interval. All the points fall close to the fitted line, so that the assumption that the residuals are normal is reasonable. The Anderson-Darling statistic (0.799) and p-value (0.023) confirm the conclusion.

The regression equations for determining the anticipated value of bearing wear and friction response for any given combination of operating parameters are;

$$\text{Bearing Wear (g)} = 0.00238 + 0.000001 * Load - 0.000001 * Speed - 0.00001 * Temperature \quad (3)$$

$$\text{Friction} = -0.0304 - 0.000023 * Load + 0.000057 * Speed + 0.001412 * Temperature \quad (4)$$

Conclusion

In this work, the concept of design of experiments has been applied to select the operating parameters for testing and evaluating the tribological performance of journal bearing on a laboratory test rig. The principle of lower the better has been adopted to find out the combinations of operating parameters that can result into lower friction and wear in the bearing. Empirical relations for the friction and wear have been determined as a function of operating parameters. On the basis of the study undertaken following salient conclusions are made:

1. Tribological properties of EN-31 journal bearing are significantly influenced by the operating parameters i.e. applied load, rotational speed and lubricant temperature. Results indicate that bearing wear decreases with increase in speed and temperature. The combination of operating parameters corresponds to different lubrication regimes as evident from the Sommerfeld Number.

2. The optimum operating conditions for lower bearing wear is Load 2-Speed 3-Temperature 3, and for the friction Load 1-Speed 3- Temperature 3, as confirmed from the main effect plot and S/N Ratio. The optimum parameter combination corresponds to the lubrication regime in close proximity to the elastohydrodynamic lubrication as evident from the Sommerfeld Number.

3. The regression equation for bearing friction and wear as a function of load, speed and temperature has been developed.

4. Load was the most influencing parameter between all there parameters i.e. load, speed & temperature for wear and coefficient of friction. Hence, by following this approach the parameter testing and performance evaluation of bearings on test rigs can be achieved.

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