International Research Journal of Innovations in Engineering and Technology (IRJIET)

ISSN (online): 2581-3048

Volume 7, Issue 11, pp 654-660, November-2023

https://doi.org/10.47001/IRJIET/2023.711086

Performance Test Study TM280 Journal Bearing Testing Equipment

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***Abstract* - The TM 280 unit is used to visualize the pressure curve in journal bearings with hydrodynamic lubrication. There were problems related to several tests on this tool. The author wanted to carry out tests again by improving the parameters that had been carried out during previous tests. From this test, comparative data can be taken regarding load variations and resulting speed variations which will later be useful in future use of testing equipment. The test variables for lubricating oil pressure distribution are load, lubricating oil viscosity and journal shaft rotation speed. In this test, 3 variations of rotation speed were carried out, namely: 1500 Rpm, 2000 Rpm, and 2500 Rpm with load variations of 6N, 7N, 8N, 9N, and 10 N. The test was carried out at the Vibration and Machine Diagnosis Laboratory, Department of Mechanical Engineering, Diponegoro University. The results obtained from the test are readings of pressure distribution data on the manometer tube with various variations in loading and variations in rotation speed from the tool data and data obtained from the test. After that, look for Sommerfeld numbers which are useful for reading Raimondi and Boyd graphs to analyze the thickness of the lubricant layer, pressure. maximum, and shaft rotation. Produces a comparison of maximum pressure and position at maximum pressure from theory with testing. Comparison of theoretical data with test data for sliding bearing devices can only be carried out on maximum pressure distribution data and film position at maximum pressure. The performance of the tool for reading pressure distribution is still good but for comparative analysis of theoretical data with test data it still produces large errors. the effect of variations in load and speed is directly proportional.**

Keywords: Pressure Distribution, Journal Bearings, Sommerfeld, Reynolds Equation.

I. INTRODUCTION

In a piece of equipment/machine, it can be ascertained that there are many components that move in the form of either angular movement or linear movement. There are two types of mechanisms used by bearings to overcome friction, namely the sliding mechanism and the rolling mechanism. For sliding mechanisms, where there is relative movement between surfaces, the use of lubricants plays a very important role. Meanwhile, for the rolling mechanism, where there is no relative movement between the surfaces in contact, the role of the lubricant is smaller. [1] In Sliding Bearings, the choice of material plays a major role because the mechanism is in direct contact between surfaces. Therefore, properties such as anti-friction, anti-welding, being able to withstand lubricating oil, having high ductility, being able to absorb fine dirt and good rust resistance are properties that must be looked for in accordance with the use of sliding bearings [2].

Bearings are components in machines that are able to support a load-bearing shaft, so that rotation occurs smoothly, safely and has a long life. In general, bearings have two types, namely non-friction bearings and plain bearings. According to the direction of the load on the element (none friction), bearings can be divided into three types, namely radial bearings, axial bearings and special bearings. Meanwhile, according to the friction of the bearing against the shaft (plain bearing) there are two types, namely bullet bearings and journal bearings. The TM 280 unit is used to visualize the pressure curve on journal bearings with hydrodynamic lubrication [3].

In this study the author used the TM280 test equipment where the tool used was a journal bearing test equipment. The TM280 testing equipment has acrylic bearings and a large manometer board, so that the lubricating oil pressure in the bearing can be clearly observed. TM280 to investigate pressure distribution in slide bearings depicts hydrodynamic lubrication. There were problems related to several tests on the TM280 tool. The author wanted to carry out tests again by improving the parameters that had been carried out during previous tests. By carrying out the tests, it is hoped that the tool will obtain accurate and actual proof results so that the results can then be concluded whether the TM280 journal bearing test tool is still in good condition or has deficiencies. The tests that will be carried out by the author are in the form of cartesian coordinate graphs of pressure distribution, polar graphs of pressure distribution, observation results of the minimum thickness of the lubricant layer that occurs on the

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TM280 journal bearing test equipment. From this test, comparative data can be taken regarding the variations in load and resulting speed which will later be useful in future use of the tool.

1.1 Method

This pressure distribution test is used to determine the distribution of lubricating oil in the sliding bearing test equipment. In this test, 3 variations of rotation speed were carried out, namely: 1500 Rpm, 2000 Rpm, and 2500 Rpm with load variations of 6N, 7N, 8N, 9N, and 10 N. The duration of the test for each rotation speed variation was 20 minutes. After that, pressure readings can be taken on the manometer panel. Testing was carried out at the Vibration and Machine Diagnosis Laboratory, Department of Mechanical Engineering, Diponegoro University. The tool used is the TM280 Gliding Bearing Test Equipment made by TecQuipment Ltd, England. The journal bearing testing tool can be seen in Figure 1 below.

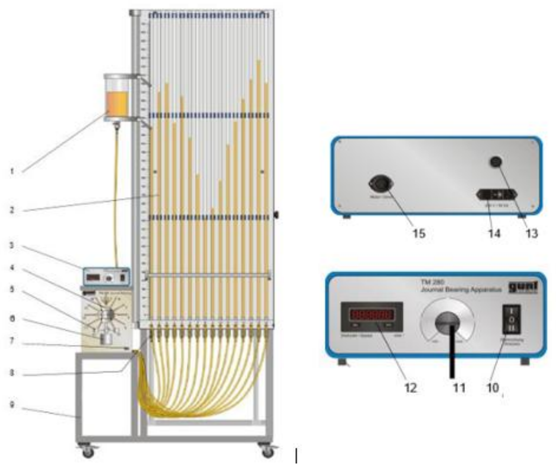

Figure 1: TM280 Journal Bearing Testing Tool

Figure 1 Caption:

1. Oil tank
2. Manometer tube panel
3. Control unit
4. Bearing housing
5. Shear load
6. Load variation device
7. Thermocouple connector
8. Ball valve
9. Frame
10. Rotation direction button
11. Potentiometer
12. Speed display
13. Speed sensor
14. Motor power button
15. Power source connector

1.2 Theoretics

Journal Bearings This type of sliding bearing (journal bearings) is very widely used in machines that have rotating elements (rotating machines), such as steam turbines, generators, blowers, compressors, combustion motors, ship shafts, even as bearings on elements that are supposed to use rolling bearings (rolling elements bearings). [4]

1.3 Lubrication

There are several types of lubrication that occur in a bearing and five forms of lubrication that can be identified in a bearing:

- Hydrodynamic Lubrication.
- Hydrostatic Lubrication
- Elastohydrodynamic Lubrication
- Boundary Field Lubrication
- Solid Lubrication

1.4 Viscosity

In a bearing there must be lubrication and in this research a sliding bearing is used which uses a fluid to hold the load. So a depiction of the viscosity that occurs in a bearing can be seen in Figure 2.

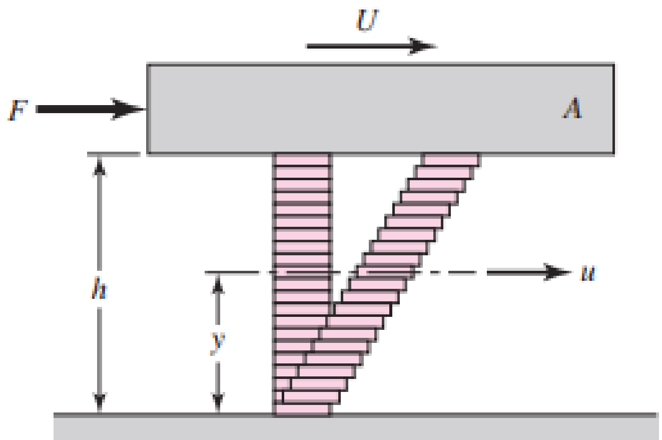

Figure 2: Newton's definition of dynamic resilience regarding viscous flow [5]

From figure 2 mathematically it can be written:

$$\tau = \mu \frac{du}{dy} = \mu \frac{u}{h} \qquad \text{(a)}$$

Where: τ = fluid shear stress $\left(\frac{N}{m^2}\right)$

μ = dynamic viscosity (Poise,P)

u = relative surface speed (m/sec)

h = thickness of lubrication layer (m)

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1.5 Petroff's Equation Theory

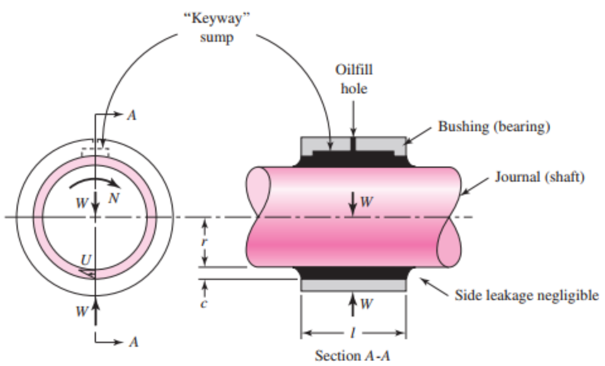


Figure 3: Petroff light loaded journal bearings consist of a shaft journal and bushing with an axial groove internal lubricant reservoir [5]

$$\boldsymbol{f} = 2\boldsymbol{\pi}^2 \frac{\boldsymbol{\mu N}}{\boldsymbol{P}} \frac{\boldsymbol{r}}{\boldsymbol{c}} \qquad \text{(b)}$$

Equation (b) is called the Petroff equation and was first published in 1883. Substituting the appropriate dimensions in each parameter will show that they are dimensionless.

The bearing characteristic number, or Sommerfeld number, is determined by the equation

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

Where:

c = free radial distance (in)
r = bearing journal radius (in)
μ = lubricant viscosity (reyn)
N = rotational speed (rev/s)
P = average load (Psi)

The Sommerfeld number is very important in lubrication analysis because it contains many parameters determined by the designer. Note that it is also dimensionless. The quantity r/c is called the radial clearance ratio. If multiplying both sides of Eq. (b) with this ratio, we obtain an interesting relationship

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

1.6 Variable Relations

Raimondi and Boyd's paper is published in three parts and contains 45 detailed charts and 6 tables of numerical information. In all three sections, charts are used to determine variables for length-to-diameter (l/d) ratios of 1:4, 1:2, and 1 and for beta angles of 60 to 360°.

a) Minimum Film Thickness

The variables minimum film thickness $\boldsymbol{h_0/c}$ and eccentricity ratio = e/c are plotted against the Sommerfeld number S with contours for various l/ d values. The corresponding angular position of the minimum film thickness is found in Fig 4.

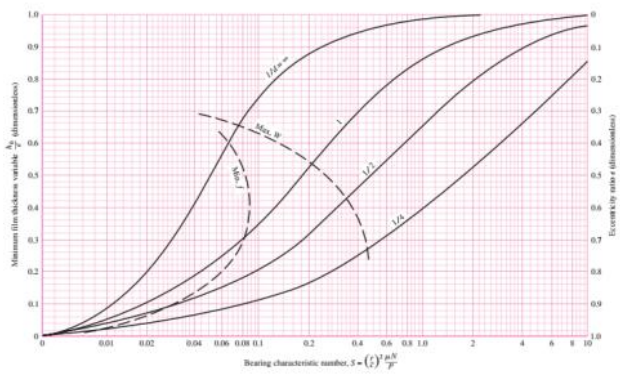
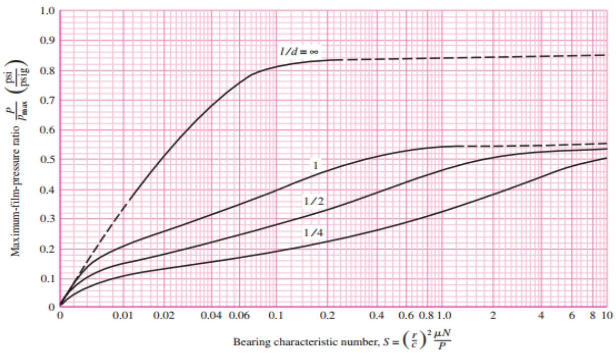


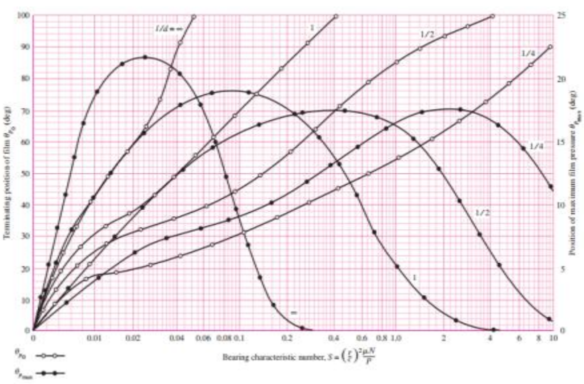
Figure 4: Chart for variables minimum film thickness and eccentricity ratio. The left boundary of the zone determines the optimal h0 for minimum friction (Raimonodi and Bod graph) [5]

b) Lubricating film pressure

The maximum pressure developed in the film can be estimated by finding the pressure ratio P/Pmax from the chart in Figure 5. The location where termination and maximum pressure occur as defined in Figure 6, is determined from Fig.



(a)



(b)

Figure 5: Chart for determining the maximum film pressure (a) and for finding the final position of the lubricant film and the position of the maximum film pressure (b) (Raimonodi and Bod chart) [5]

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II. PRESSURE DISTRIBUTION

Testing of pressure distribution in sliding bearings is carried out in the Vibration and Machine Diagnostics laboratory. Lubrication Engineering Department of Mechanical Engineering, Faculty of Engineering, Diponegoro University. It should be noted that points 1, 2, 3, 4 and 5 are in the axial direction (bearing width), while the pressure distribution around the circle (the main object of this research) is shown by test points 3, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15 and 16.

The data obtained from the testing equipment can be seen in the tables below:

Table 1: Pressure Distribution Data at 1500 Rpm Speed in Psi Units

	No	Weight (N)				
		6	7	8	9	10
Pressure P (psi)	1	1,578	1,536	1,578	1,486	1,465
	2	1,777	1,763	1,806	1,792	1,749
	3	1,948	1,934	1,991	1,934	1,905
	4	1,991	1,948	2,012	1,955	1,955
	5	1,678	1,607	1,692	1,571	1,564
	6	2,062	1,962	2,112	2,126	2,225
	7	1,92	1,905	2,069	2,005	2,268
	8	1,223	1,137	1,18	1,066	0,952
	9	0,753	0,64	0,64	0,398	0,284
	10	0,796	0,696	0,732	0,583	0,59
	11	1,016	0,938	1,024	0,881	0,896
	12	1,152	1,095	1,152	1,066	1,066
	13	1,28	1,223	1,265	1,173	1,163
	14	1,393	1,322	1,372	1,322	1,294
	15	1,521	1,479	1,507	1,465	1,422
	16	1,657	1,593	1,614	1,6345	1,561

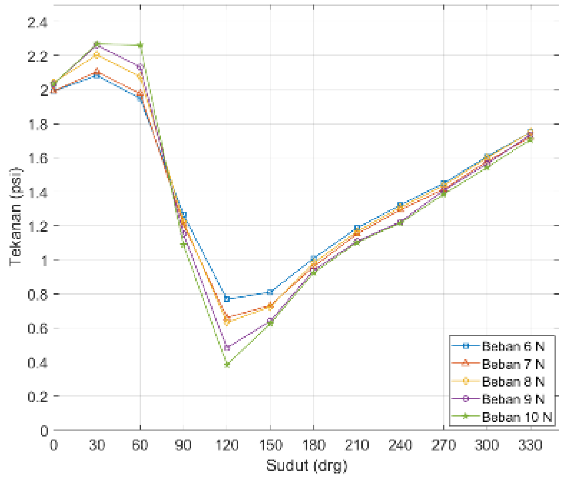

Figure 6: Graph plot of pressure distribution (Psi) at 1500 Rpm rotation variations

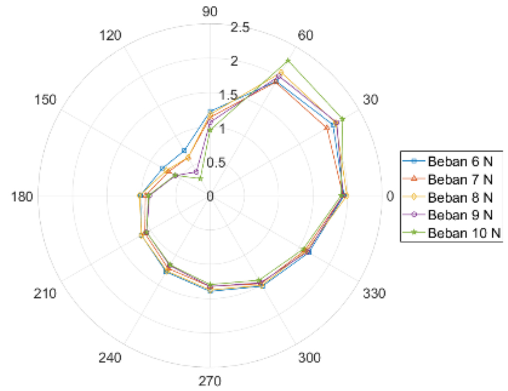

Figure 7: Polar graph of pressure distribution (Psi) at a rotation variation of 1500 Rpm

Table 2: Pressure Distribution Data at 2000 Rpm Speed in Psi Units

	No	Weight (N)				
		6	7	8	9	10
Pressure P (psi)	1	1,635	1,891	1,593	1,578	1,557
	2	1,834	1,834	1,877	1,891	1,87
	3	1,991	1,991	2,041	2,033	2,033
	4	2,005	2,092	2,119	2,119	2,133
	5	1,721	1,678	1,735	1,721	1,713
	6	2,083	2,105	2,204	2,261	2,268
	7	1,948	1,977	2,076	2,133	2,261
	8	1,265	1,208	1,223	1,152	1,088
	9	0,768	0,661	0,632	0,483	0,384
	10	0,81	0,732	0,725	0,64	0,625
	11	1,009	0,96	0,981	0,938	0,924
	12	1,187	1,152	1,166	1,109	1,102
	13	1,322	1,294	1,308	1,223	1,216
	14	1,45	1,415	1,436	1,408	1,386
	15	1,607	1,578	1,6	1,564	1,543
	16	1,749	1,721	1,749	1,735	1,706

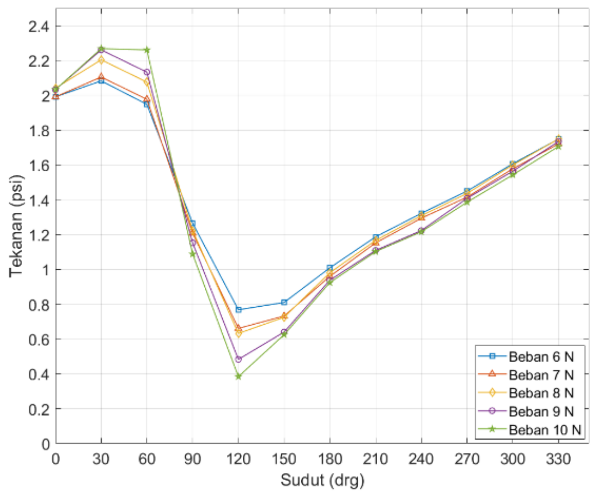

Figure 8: Graph plot of pressure distribution (Psi) at 2000 Rpm rotation variations

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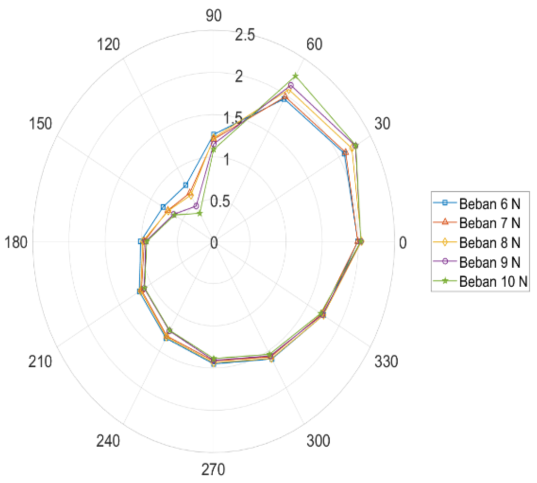

Figure 9: Polar graph of pressure distribution (Psi) at a rotation variation of 2000 Rpm

Table 3: Pressure Distribution Data at 2500 Rpm Speed in Psi Units

	No	Weight (N)				
		6	7	8	9	10
Pressure P (psi)	1	1,777	1,735	1,756	1,692	1,692
	2	2,005	1,991	2,033	2,062	2,055
	3	2,09	2,09	2,133	2,154	2,169
	4	2,176	2,176	2,233	2,254	2,289
	5	1,806	1,777	1,806	1,863	1,849
	6	2,119	2,147	2,225	2,275	2,297
	7	1,962	1,991	2,062	2,126	2,154
	8	1,422	1,265	1,351	1,308	1,294
	9	0,931	0,803	0,768	0,632	0,583
	10	0,888	0,81	0,796	0,739	0,711
	11	1,045	0,995	1,009	1,024	0,995
	12	1,258	1,208	1,223	1,194	1,18
	13	1,465	1,408	1,4	1,308	1,308
	14	1,621	1,564	1,585	1,422	1,543
	15	1,82	1,777	1,806	1,749	1,763
	16	1,927	1,891	1,927	1,934	1,941

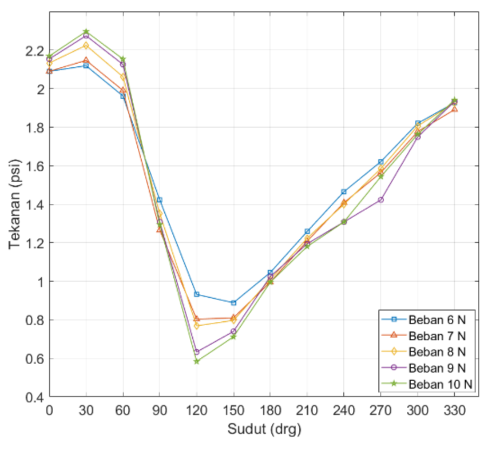

Figure 10: Graph plot of pressure distribution (Psi) at 2500 Rpm rotation variations

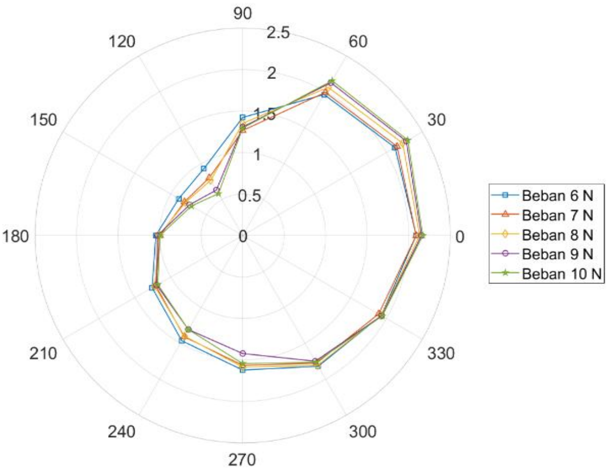

Figure 11: Polar graph of pressure distribution (Psi) at a rotation variation of 2000 Rpm

III. RESULTS AND DISCUSSIONS

3.1 Sommereld Number Calculation Results

The following are the results of Sommerfeld's number calculations:

Table 4: Sommerfeld Number

No	Weight (N)	Rotation Speed (Rpm)		
		1500	2000	2500
1	6	0,044	0,059	0,074
2	7	0,038	0,051	0,064
3	8	0,033	0,044	0,056
4	9	0,029	0,039	0,049
5	10	0,026	0,035	0,044

To be able to read the Raimondi and Boyd graph it is necessary to determine the l/d of the sliding bearing, then:

$$\frac{l}{d} = \frac{2,9}{2} = 1,4$$

Because l/d is not in the graph, l/d is found using the following equation:

$$y = \frac{1}{(\frac{l}{d})^3}\Big[\frac{1}{8}\Big(1-\frac{l}{d}\Big)\Big(1-2\frac{l}{d}\Big)\Big(1-4\frac{l}{d}\Big)y_\infty$$
$$+\frac{1}{3}\Big(1-2\frac{l}{d}\Big)\Big(1-4\frac{l}{d}\Big)y_1$$
$$-\frac{1}{4}\Big(1-\frac{l}{d}\Big)\Big(1-4\frac{l}{d}\Big)y_{\frac{1}{2}}$$
$$+\frac{1}{24}\Big(1-\frac{l}{d}\Big)\Big(1-2\frac{l}{d}\Big)y_{1/4}\Big]$$

International Research Journal of Innovations in Engineering and Technology (IRJIET)

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Volume 7, Issue 11, pp 654-660, November-2023

https://doi.org/10.47001/IRJIET/2023.711086

So,

$$y=\frac{1}{(\frac{2,9}{2})^3}\left[\frac{1}{8}\left(1-\frac{2,9}{2}\right)\left(1-2\frac{2,9}{2}\right)\left(1-4\frac{2,9}{2}\right)y\infty\right.$$
$$+\frac{1}{3}\left(1-2\frac{2,9}{2}\right)\left(1-4\frac{2,9}{2}\right)y_1$$
$$-\frac{1}{4}\left(1-\frac{2,9}{2}\right)\left(1-4\frac{2,9}{2}\right)y_{\frac{1}{2}}$$
$$\left.+\frac{1}{24}\left(1-\frac{2,9}{2}\right)\left(1-2\frac{2,9}{2}\right)y_{1/4}\right]=1$$

The y value is the desired variable in the interval$\infty > \frac{l}{d} > \frac{1}{4} > dan\ y_\infty, y_1, y_{1/2}$ and $y_{1/4}$are variables related to the ratio ∞, 1, ½, and ¼. So the value of l/d in this study is 1.

3.2 Analysis of Film Thickness in Journal Bearing Test Equipment

After reading the graph using Sommerfeld numbers and carrying out calculations, the minimum layer thickness analysis results are obtained as follows:

Table 5: Minimum Lubrication Film Thickness

	No	Weight (N)	Rotation Speed(Rpm)		
			1500	2000	2500
Minimum Film Thicknes (in)	1	6	0,031	0,039	0,043
	2	7	0,028	0,034	0,04
	3	8	0,025	0,031	0,037
	4	9	0,021	0.026	0,031
	5	10	0,02	0,025	0,034

3.3 Maximum Pressure Analysis in Gliding Bearing Test Equipment

After reading the graph using Sommerfeld numbers and calculating itThe minimum layer thickness analysis results obtained are as follows:

Table 6: Position of Maximum Film Pressure

	No	Weight (N)	Rotation Speed (Rpm)		
			1500	2000	2500
Position of Maximum Film Pressure (deg)	1	6	18	18,5	19
	2	7	17,5	18,5	18,7
	3	8	17	18	18,5
	4	9	16,5	17,5	18
	5	10	16	17,5	18

Table 7: Maximum Film Pressure

	NO	Weight (N)	Rotation speed (Rpm)		
			1500	2000	2500
Maksimum Film Pressure (Psi)	1	6	0,709	0,648	0,613
	2	7	0,854	0,803	0,757
	3	8	1,01	0,946	0,891
	4	9	1,175	1,1	1,033
	5	10	1,353	1,263	0,728

3.4 Comparison of Data from Theoretical Analysis with Test Results Data

Comparison of data from the results of theoretical data analysis with test data can be seen in the following table:

Table 8: Comparison Data of Theoretical Calculations with Test Data

no	Data	Weight (N)	Rotation Speed(Rpm) and Position Pmax(deg)					
			1500	deg	2000	deg	2500	deg
1	testing (Psi)	6	2,062	30	2,082	30	2,119	30
	Teoritis (Psi)		0,709	18	0,648	18,5	0,613	19
	error(%)		65	40	68	38	71	36
2	testing (Psi)	7	1,962	30	2,105	30	2,147	30
	Teoritis(Psi)		0,854	17,5	0,803	18,5	0,757	18,7
	error(%)		56	41	61	38	64	37
3	testing (Psi)	8	2,112	30	2,204	30	2,225	30
	Teoritis (Psi)		1,01	17	0,946	18	0,891	18,5
	error(%)		52	43	56	40	59	38
4	testing (Psi)	9	2,126	30	2,261	30	2,275	30
	Teoritis (Psi)		1,175	16,5	1,1	17,5	1,033	18
	error(%)		44	45	51	43	54	40
5	testing (Psi)	10	2,225	30	2,268	30	2,297	30
	Teoritis (Psi)		1,353	16	1,263	17,5	0,728	18
	error(%)		39	46	44	41	68	40

ISSN (online): 2581-3048
Volume 7, Issue 11, pp 654-660, November-2023
https://doi.org/10.47001/IRJIET/2023.711086

IV. CONCLUSION

From research on the comparison of test data for sliding bearing test equipment with data from the theoretical analysis that has been carried out, it can be concluded that:

a) The performance of the sliding bearing test equipment is still good for pressure distribution readings.
b) Comparison of theoretical data with test data for sliding bearing devices can only be carried out on maximum pressure distribution data and determining the position of the maximum pressure film produces very large errors. The theoretical equation used for data comparison does not produce a satisfactory comparison, so the equation used considered not suitable for use in pressure distribution comparisons. The pressure distribution in sliding bearings has different pressures at each point of the sliding bearing section as depicted in the polar graphic image, some experience very high pressure and some experience low pressure.
c) In testing the sliding bearing, it can be seen that the effect of load and rotation on pressure is that the greater the load given and the faster the rotation given, the greater the pressure that occurs.
d) Because in the sliding bearing testing equipment there is no tool to test the thickness of the lubrication layer that occurs, the layer thickness can be determined through calculations in Chapter IV and results in the conclusion that at the same rotational speed the greater the load used, the smaller the minimum layer thickness will be.

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Citation of this Article:

Ojo Kurdi, Djoeli Satrijo, Toni Prahasto, Kevin Ardi Kusuma, "Performance Test Study TM280 Journal Bearing Testing Equipment" Published in *International Research Journal of Innovations in Engineering and Technology - IRJIET*, Volume 7, Issue 11, pp 654-660, November 2023. Article DOI https://doi.org/10.47001/IRJIET/2023.711086

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