Effect of Bearing Design Parameters on the Minimum Oil Film Thickness of Hydrodynamic Journal Bearing

Mohammed Ali[1], S.K. Somani[2]

[1]*Department of Mechanical Engineering, Medi-Caps University, Madhya Pradesh, India, 453331.*
[2]*Vice Chancellor, Medi-Caps University, Madhya Pradesh, India, 453331.*

---***---

Abstract - *This paper deals with the effect of various bearing design parameters on the minimum film thickness of hydrodynamic journal bearing. In one of the paper the authors have designed the bearing on the basis of heat balance i.e. heat generated inside the bearing should be nearly equal to the heat dissipated from the bearing housing to the surroundings. As it is an iterative process, this analysis also includes computer aided bearing design. Matlab programming provides easy, accurate and fast results. This paper gives the effect of bearing design parameters such as oil grades, lubrication systems, bearing housing geometry, R/C and L/D ratios on the minimum oil film thickness of hydrodynamic journal bearing.*

***Key Words*: Bearing design, design parameters, minimum oil film thickness, and hydrodynamic journal bearing.**

1. INTRODUCTION

Hydrodynamic journal bearing have wide use in many industrial applications. Due to these industrial applications and severe working condition of hydrodynamic journal bearing, many researchers have tried various approaches for optimum and safe design of bearing [1, 4]. As there is very little clearance between the bearing and the journal, there are greater chances of heat generation. As being the key member of many big machines, the design of bearing should be stable. Stable means the operating temperature of the bearing should remain under safe limit, irrespective of the operating condition of the bearing. Viscosity of the lubricant is the root property which is responsible for the hydrodynamic action in the bearing. Since, the viscosity is mainly dependent on the operating temperature of bearing; it has to be ensured that working of the bearing and its operating temperature should be under safe limit. To maintain this, the paper includes bearing design for stable zone, by considering the bearing characteristic, $\frac{\mu N}{P} \geq 0.362 \times 10^{-6}$ [9] and finding the operating temperature by equating heat generation to heat dissipation for certain values of R/C ratio, L/D ratio and SAE oil grades.

Once the stability in the operating temperature for the bearing is developed, the value of minimum film thickness developed inside the bearing can be obtained. According to the Trumpler [9], the maximum operating temperature should be less than and equal to 120°C and also to avoid the asperities contact, the minimum film thickness should be

$$h_0 = 0.005 + 0.0004D \text{ mm} \quad \ldots (1)$$

Where, D is the journal diameter in mm.

The performance characteristics of bearing are prime things for the bearing operation. This may get affected by many parameters such as length to diameter ratio selected, R/C ratio selected, lubricant used, lubrication scheme used etc. Muzakkir et al [5] experimentally concluded for different R/C ratios of journal bearing used in Sugar mills, that there will be minimum friction, if R/C ratios selected in between 1000 to 1500. Sharma and Nathi [8] have studied the influence of micro polar lubricants on the performance of slotted hybrid journal bearing and concluded that the value of minimum film thickness increases, while the value of coefficient of friction decreases by the use of micro polar lubricant instead of Newtonian lubricant. Kasai et al [3] further investigated the influence of lubricant on plain bearing performance and concluded that the friction coefficient will be reduced by using polymer containing oils over base oil. To test the performance characteristics of hydrodynamic journal bearing, many researchers have analyzed the minimum film thickness criteria and tabulated the influence of various design parameters on the minimum film thickness. Myung et al [6] have studied the effect of circumferential groove on the minimum oil film thickness developed in engine bearing. They have also provided the comparative results of minimum oil film thickness for grooved and ungrooved journal bearing.

Further, Dobrica and Fillon [2] have studied the performance degradation of scratched journal bearings. They have also described the effect of scratches on minimum film thickness.

To study the performance characteristics of bearing, this paper have tabulated the effect of various parameters such as oil grade used, lubricating scheme used, bearing housing geometry selected, also L/D and R/C ratio selected on the minimum film thickness developed inside the bearing.

NOMENCLATURE

b	*Constant*	R	*Radius of journal (m)*
C	*radial clearance (m)*	S	*Sommerfeld number*
D	*diameter of the bearing (m)*	T_f	*Operating temperature (^0C)*
e	*eccentricity (m)*	T_∞	*Atmospheric temperature (^0C)*
f	*Coefficient of friction*	U	*Combined overall coefficient of radiation and convection heat transfer W/(m^2 ^0C)*
h	*lubricant film thickness (m)*	W	*Load (N)*
H_g	*Heat generation*	α	*Karelitzs constant*
H_d	*Heat dissipation*	$\in$	*Eccentricity ratio $\in$ = e/C*
L	*length of bearing (m)*	λ	*Slenderness ratio (L/D)*
N	*journal rotation speed (rpm)*	ω	*Journal angular speed (rad/s)*
P	*film pressure (Pa)*	μ	*Viscosity of lubricant film (Pa s)*

2. BEARING DESIGN PROCEDURE

For stable bearing operation, the bearing temperature should be inside the limitations of the temperature zone. There should be thermal balance in bearing operation i.e. the heat generated inside the bearing should fully get dissipated to the surroundings. To design the bearing for a particular load and journal speed, let us assume some initial values for the parameters like L/D ratio and R/C ratio, and standard SAE oil grade. For the bearing design, select the value of overall heat transfer coefficient (U) which depends upon the bearing material, geometry of the bearing, roughness and the temperature difference between housing and the surrounding objects and air velocity. Also, select the Karelitz's constant (α) according to lubrication scheme and the bearing housing geometry.

Table-1: Overall Coefficient for Different Conditions [9]

U W/ (m^2 ^0C)	Conditions
11.3	For still air
15.3	For shaft stirred air
33.4	For air moving at 2.5 m/s

Table-2: Karelitz's Constant (α) for Different Lubrication Scheme and Conditions [9]

Lubrication System	Conditions	Range of α
Oil ring	Moving air	1-2
	Still air	½ -1
Oil bath	Moving air	½ -1
	Still air	1/5 – ½

Initially assume some operating temperature of the bearing. Now, for this operating temperature and assumed SAE oil grade, the viscosity can be calculated by using the chart given in reference [9] or using the formula,

$$\mu = \mu_0 \exp [b / (1.8 T + 127)] \quad \text{... (2)}$$

The values for μ_0 and b for different SAE oil grade can be obtained from the Table-3.

Table-3: Values of μ_0 and b for Different SAE Oil Grade [9]

SAE Oil Grade	Viscosity μ_0, mPas	Constant b, ^oC
10	0.1089	1157.5
20	0.0937	1271.6
30	0.0971	1360.0
40	0.0827	1474.4
50	0.1171	1509.6
60	0.1288	1564.0

Calculate the value of length of bearing and corrected pressure, For thick film lubrication inside the bearing, the bearing characteristic $\frac{\mu N}{P}$ should be greater than or equal to 0.362 x 10^{-6}.

$$\frac{\mu N}{P} \geq 0.362 \times 10^{-6} \quad \text{... (3)}$$

The overall pressure inside the fluid film can be calculated as,

$$P = \frac{\mu N}{0.362 \times 10^{-6}} \quad \text{... (4)}$$

This pressure can also be defined as the load per unit projected area,

$$P = \frac{W}{L*D} \quad \text{... (5)}$$

Comparing equation (4) and equation (5), the length and diameter of the bearing can be calculated. The obtained values of length and diameter may be used to calculate the corrected pressure.

Calculate the value of Sommerfeld number as,

$$S = \frac{\mu N}{P}\left(\frac{R}{C}\right)^2 \quad \text{... (6)}$$

From Raimondi and Boyd [7] chart for coefficient of friction variable Vs Sommerfeld number, find the value of coefficient of friction variable for an assumed L/D ratio. As we have assumed the R/C ratio initially, the coefficient of friction can be calculated.

When journal rotates inside the bearing, the shearing action takes place between the layers of lubricant, which causes the heat generation. Heat generation can be calculated mathematically as,

$$H_g = f \text{ x W x } \pi \text{ x D x N} \quad \text{... (7)}$$

Heat dissipation is a phenomena in which heat dissipates from the bearing housing to the surroundings, can be calculated mathematically as,

$$H_d = \frac{U \text{ x } A}{1+\alpha}\left(T_f - T_\infty\right) \quad \text{... (8)}$$

If $H_d \geq H_g$ means there is no heat accumulation inside the bearing and hence the design of the bearing is stable and if it is not so, then iterate the procedure by changing the operating temperature till the required condition is obtained. In this paper the computer aided design procedure is adopted which helps to iterate the process for small step of temperature variation. Also the computer aided design process allows us to vary the assumed L/D ratios; R/C ratios and SAE oil grades easily and can be compared for the different conditions effectively.

From the equation (6), the Sommerfeld number can be obtained and with the use of Raimondi and Boyd charts and graphs, the corresponding values of minimum film thickness for assumed L/D ratio can be determined. To calculate the value of minimum film thickness for any L/D ratio other than the L/D ratio given in the graph, the formula can be used,

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

... (9)

Where, y – Minimum film thickness variable to be calculated at any L/D ratio

y_∞ - Minimum film thickness variable at L/D ratio equal to infinity

y_1 - Minimum film thickness variable at L/D ratio equal to 1

$y_{\frac{1}{2}}$ - Minimum film thickness variable at L/D ratio equal to ½

$y_{\frac{1}{4}}$ - Minimum film thickness variable at L/D ratio equal to ¼

From this obtained minimum film thickness variable, the actual value of minimum film thickness can be determined as,

Minimum film thickness = (minimum film thickness variable) x (clearance) ... (10)

The Matlab program is used for the calculation and iteration. The graphs are plotted from the obtained results. Matlab program is parametric, therefore, the changes in the initial assumed condition shows the corresponding changes in the final obtained graphs.

3. METHOD TO USE GRAPHICAL RESULTS

The bearing is designed for particular condition of load and speed of shaft. Let us assume, the bearing is working at a load of 2500N and speed of 900RPM. The required minimum film thickness for any particular L/D ratio can be observed from the graphs shown in Chart-1 to Chart-18. From these charts., for a particular value of L/D ratio and minimum film thickness, we can obtain sets of SAE oil grade, R/C, U and α. One of such set is shown in Table-4. From such set of values, choose optimum set according to the SAE oil grade available, setting mode of heat transfer, lubrication scheme and bearing housing geometry. Another set of differ L/D values, SAE oil grade, U and α for particular value of minimum oil film thickness and R/C value are shown in Table-5, Table-6 and Table-7. Same type of tables can be obtained for different values of minimum film thickness, L/D, R/C, SAE oil grade, U and α.

Table-4: Values of SAE Oil Grade, U and α for Particular Value of h_{min} and L/D

h_{min} (mm)	L/D	SAE oil grade	R/C	U W/ (m^2 ^0C)	α
0.025	1	30	800	11.3	0.3
		60	1000	15.3	1.5
		40	1000	15.3	0.75
		60	1200	11.3	0.75
		20	1200	33.4	0.75

Table-5: Values of L/D, SAE Oil Grade, U and α for Particular Value of h_{min} and R/C=800

h_{min} (mm)	R/C	L/D	SAE oil grade	U W/ (m^2 ^0C)	α
0.025	800	1.2	40	11.3	0.75
		1.4	30	11.3	0.75
		1.6	20	11.3	0.75
		0.8	40	11.3	0.3
		1	30	11.3	0.3
		1.6	20	11.3	0.3
		1.8	20	15.3	1.5

		0.8	40	15.3	0.75
		1.6	20	15.3	0.75
		0.4	20	33.4	1.5
		1.6	20	33.4	1.5
		1.6	20	33.4	0.75

Table-6: Values of L/D, SAE Oil Grade, U and α for Particular Value of h_{min} and R/C=1000

h_{min} (mm)	R/C	L/D	SAE oil grade	U W/ (m² ⁰C)	α
0.025	1000	2	10	11.3	0.75
		0.4	60	11.3	0.3
		2	10	11.3	0.3
		2	10	11.3	0.75
		0.6	30	33.4	1.5
		1.2	20	33.4	1.5
		2	10	33.4	1.5
		0.6	20	33.4	0.75
		1.2	20	33.4	0.75
		2	10	33.4	0.75

Table-7: Values of L/D, SAE Oil Grade, U and α for Particular Value of h_{min} and R/C=1200

h_{min} (mm)	R/C	L/D	SAE oil grade	U W/ (m² ⁰C)	α
0.025	1200	1	60	11.3	0.75
		1.2	30	11.3	0.3
		1.4	40	15.3	1.5

These graphical results can be reproduced in reverse direction also i.e. for given condition of L/D, R/C, SAE oil grade, U and α, the minimum film thickness developed can be obtained. Such a graphical database can be developed for different loading conditions and speed variations also.

4. RESULTS AND DISCUSSIONS

The different values of minimum film thickness are achieved, according to the different L/D and R/C ratios selected. It changes according to the lubricating oil and the lubrication scheme adopted. The graphs for minimum film thickness versus L/D ratio has been plotted for different SAE oil grades considering values of U, α and R/C ratios as shown in Chart- 1 to Chart- 18. It has been observed from all these graphs that as if SAE grade of oil increases, the value of minimum film thickness decreases for any particular values of length to diameter ratio (L/D), Overall Heat Transfer Coefficient (U) and Lubrication scheme (α). As the value of L/D increases to 1000 or 1200, little change in h_{min} starts to appear i.e. h_{min} decreases according to decrease in α. For all R/C, U and α values, except for U=33.4, the graphs show the pattern of initial stable and then gradual decrease in the value of h_{min} of SAE oil grade lines (Chart- 1 to Chart- 4, Chart- 7 to Chart- 10, Chart- 13 to Chart- 16). For U=33.4 and any value of R/C and α, the graphs show the initial increase, then stable and finally decrease in h_{min} values for lower SAE grade oils (i.e. SAE 10 and SAE 20 and somewhere SAE 30). And for higher SAE grade oils, it shows initial stable and then gradually decreasing pattern of SAE oil grade lines (Chart- 5, Chart- 6, Chart- 11, Chart- 12, Chart- 17 and Chart- 18).

It has been observed that, the nature of all the graphs is nearly same but as the value of SAE grade of oil increases from SAE 10 to SAE 60, the values of minimum film thickness decreases for any particular value of R/C, U and α (Chart- 1 to Chart- 18).

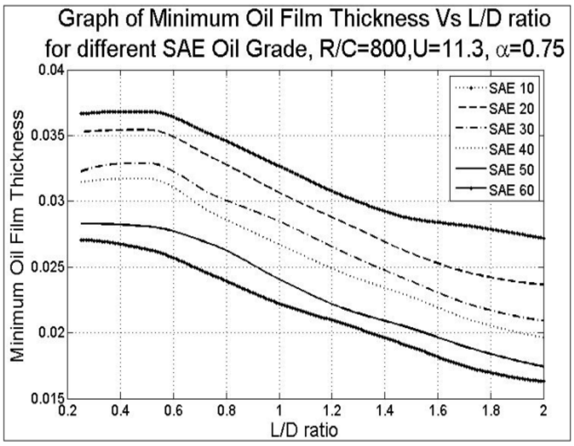

Chart-1: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=800, U=11.3, α=0.75)

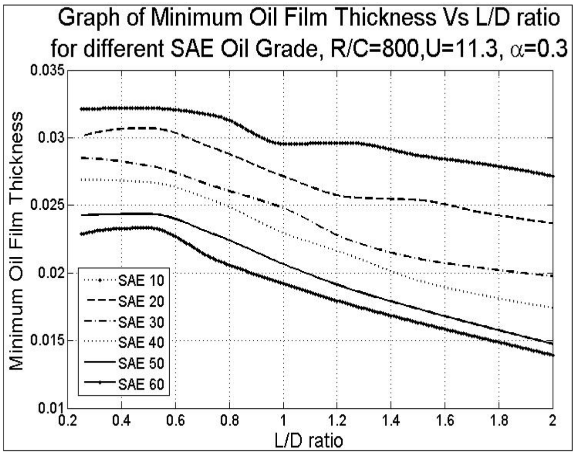

Chart-2: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=800, U=11.3, α=0.3)

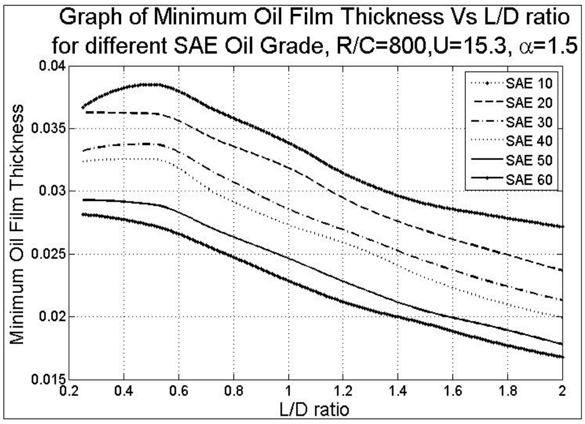

Chart- 3 Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=800, U=15.3, α=1.5)

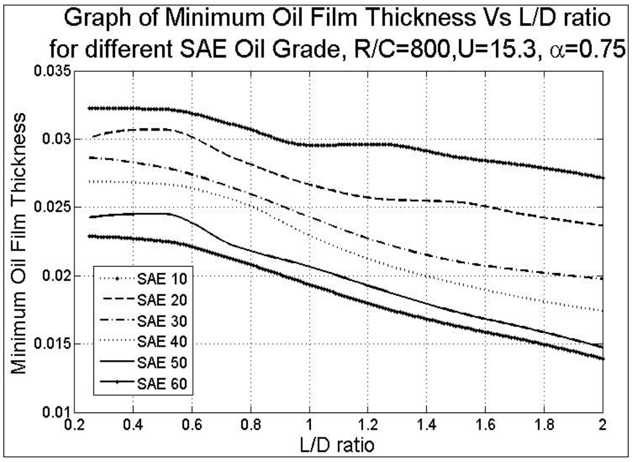

Chart- 4: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=800, U=15.3, α=0.75)

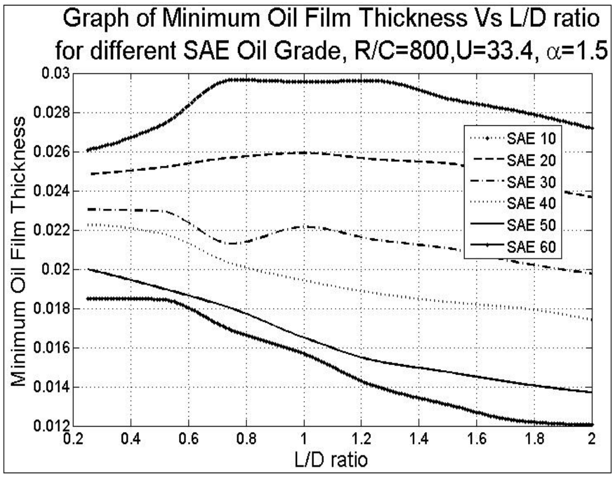

Chart- 5: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=800, U=33.4, α=1.5)

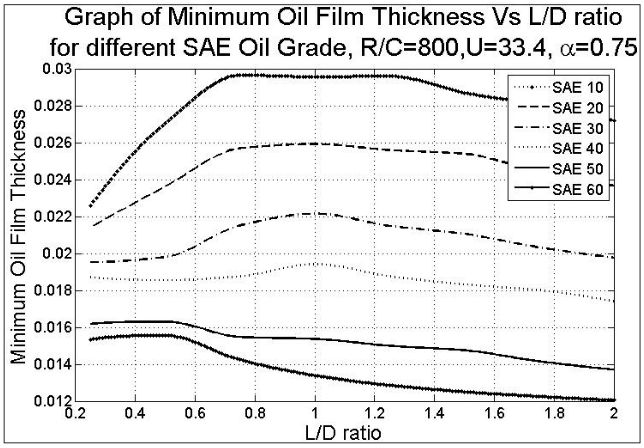

Chart- 6: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=800, U=33.4, α=0.75)

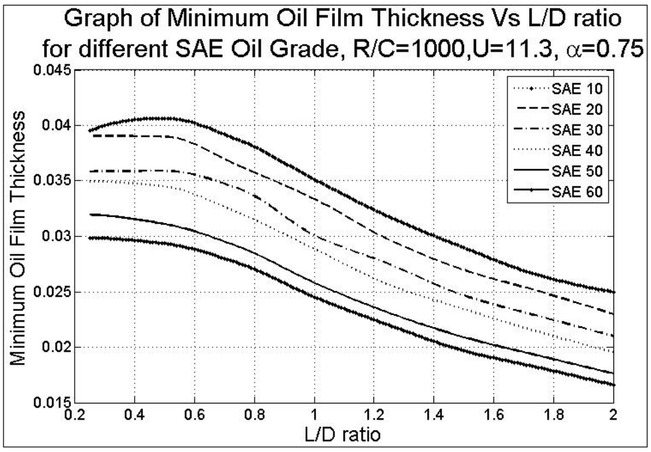

Chart- 7: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1000, U=11.3, α=0.75)

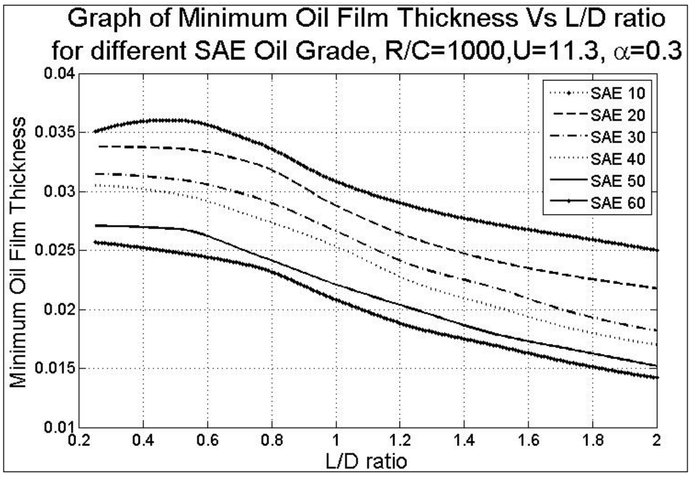

Chart- 8: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1000, U=11.3, α=0.3)

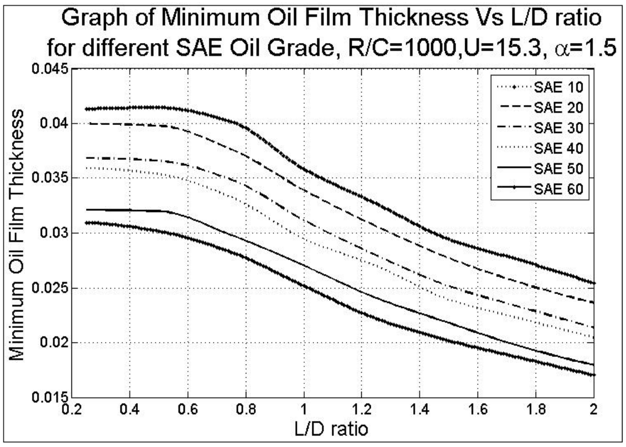

Chart- 9: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1000, U=15.3, α=1.5)

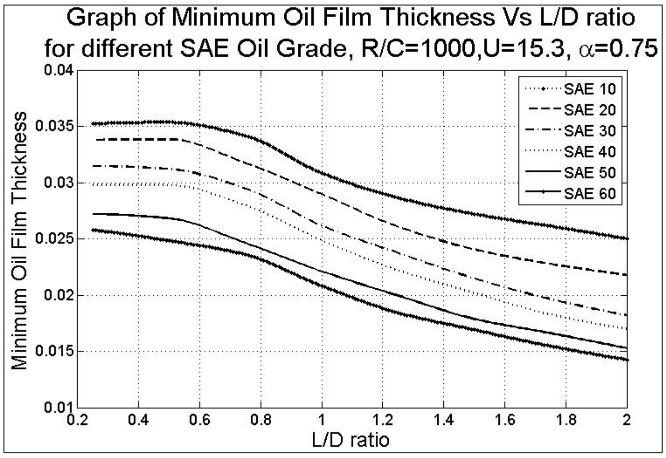

Chart- 10: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1000, U=15.3, α=0.75)

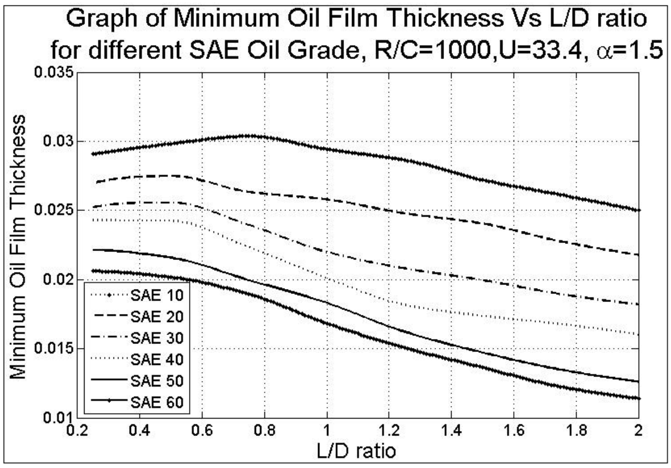

Chart- 11: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1000, U=33.4, α=1.5)

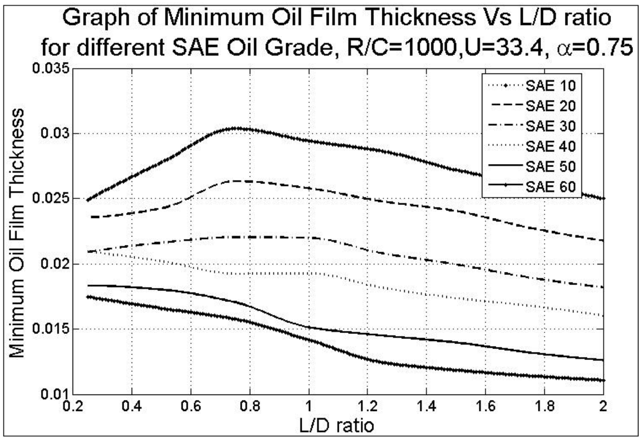

Chart- 12: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1000, U=33.4, α=0.75)

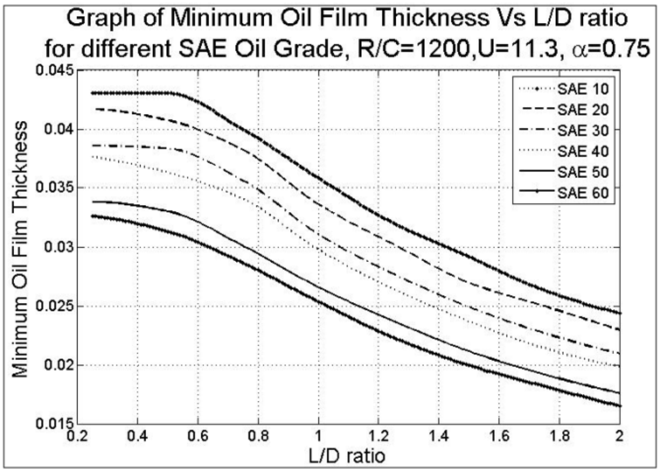

Chart- 13: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1200, U=11.3, α=0.75)

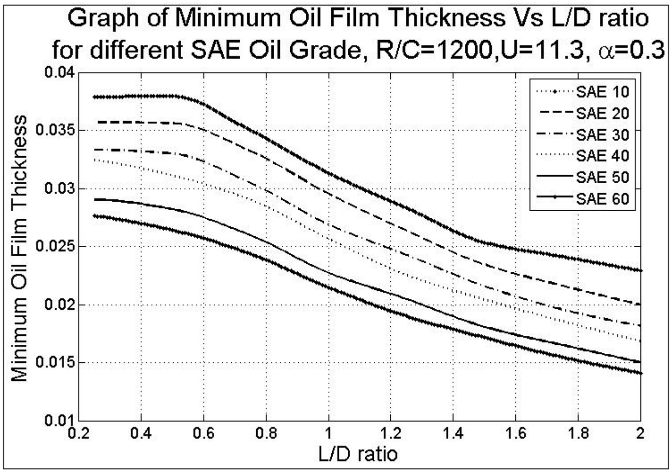

Chart- 14: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1200, U=11.3, α=0.3)

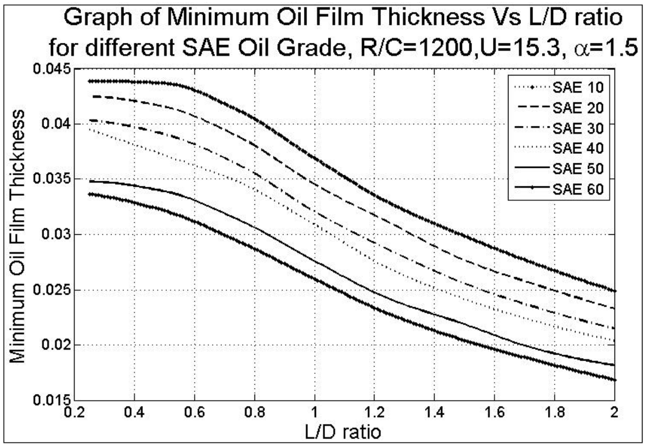

Chart- 15: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1200, U=15.3, α=1.5)

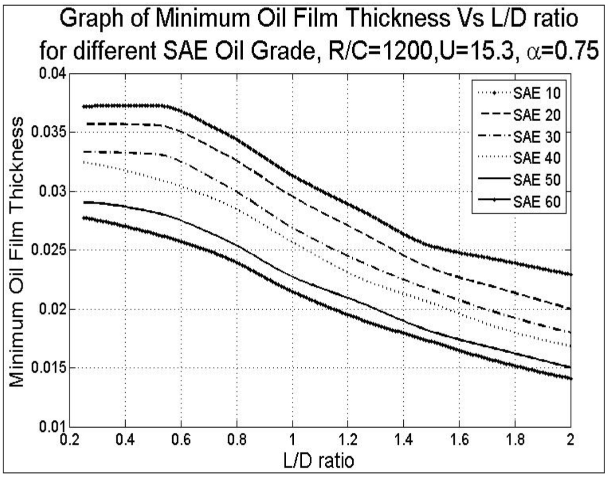

Chart- 16: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1200, U=15.3, α=0.75)

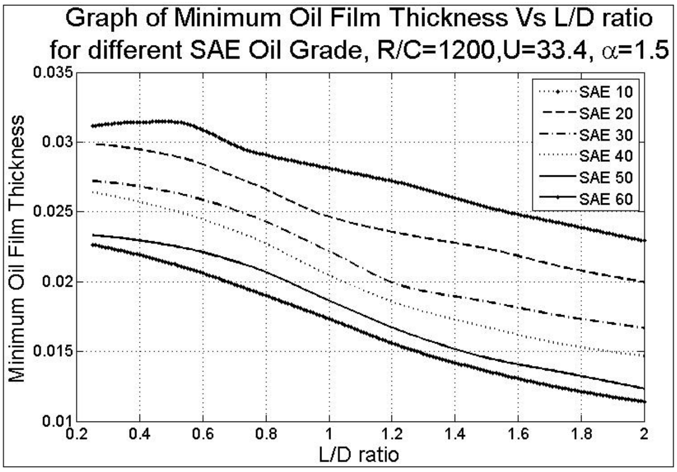

Chart- 17: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1200, U=33.4, α=1.5)

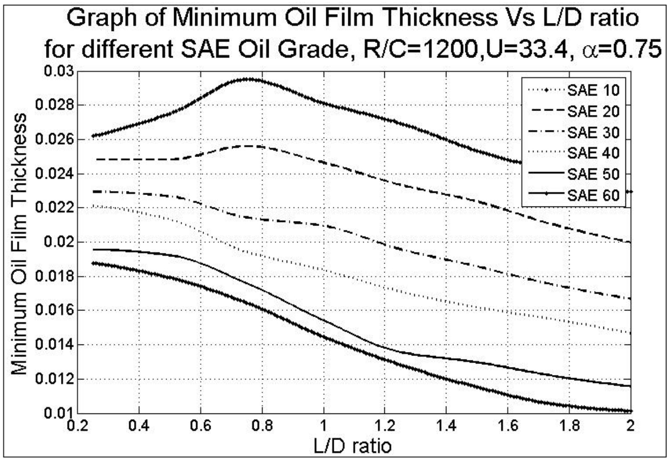

Chart- 18: Graph of minimum film thickness vs L/D ratio for different SAE oil grades (R/C=1200, U=33.4, α=0.75)

5. CONCLUSIONS

1. For higher SAE grade of oil, the value of minimum film thickness is lower.
2. As R/C ratio increases, the minimum film thickness decreases for any particular value of U and α.
3. As L/D ratio increases, the minimum film thickness decreases.
4. This CAD application helps in selecting proper bearing geometry, lubrication scheme, SAE oil grade, L/D and R/C ratios according to the required optimum film thickness ratio.
5. As these are Matlab programming based results, they have more accuracy.
6. This application helps in preparing a graphical reference data base that can be helpful in selecting a proper L/D ratio, R/C ratio, U and α, to the required optimum film thickness ratio.

REFERENCES

[1] Cameron A., "Basic Lubrication Theory", 3rd edition, Ellis Horwood Ltd., Publishing; 1981, pp 48-50.

[2] Dobrica M. B., Fillon M., "Performance Degradation in Scratched Journal Bearings", Tribology International, Volume 51, July 2012, Pages 1-10.

[3] Kasai Moritsugu, Fillon Michel, Bouyer Jean, Jarny Sebastien, "Influence of Lubricants on Plain Bearing Performance: Evaluation of Bearing Performance with Polymer-Containing Oils",

Tribology International, Volume 46, Issue 1, February 2012, Pages 190-199.

[4] Majumdar B. C., "Introduction to Tribology of Bearings" S. Chand and Company Ltd., *Publishing*; 1986, pp 47-56.

[5] Muzakkir S.M., Hirani Harish, Thakre G.D., Tyagi M.R., "Tribological Failure Analysis of Journal Bearings Used in Sugar Mills", Engineering Failure Analysis, Volume 18, Issue 8, December 2011, Pages 2093-2103.

[6] Cho. M.R., Shin. H.J. and Han. D.C., "A Study on the Circumferential Groove Effects on the Minimum Oil Film Thickness in Engine Bearings", KSME International Journal, Vol. 14, No. 7, 2000, Page 737- 743.

[7] Raimondi A, Boyd J., "A Solution for the Finite Journal Bearing and its Application to Analysis and Design – Part I, II, III" Trans. ASLE, vol. 1, No 1, April, 1958; pp. 159-203.

[8] Sharma S. C., Ram N., "Influence of Micropolar Lubricants on the Performance of Slot-Entry Hybrid Journal Bearing", Tribology

International, Volume 44, Issue 12, November 2011, Pages 1852-1863.

[9] Shigley J. E. and Mischke C. R., "Mechanical Engineering Design(in SI units)", sixth edition, Tata McGraw Hill Publishing; 2003.

[10] K. Tripathi, S.K.Somani, M.Ali, S.Gadakh, S.Rajput, "Study of Pressure Distribution and Attitude Angle (Bearing Performance Characteristics) by Using FDM", An International Journal of Engineerig & Technology, Suresh Gyan Vihar University, Vol-1, Issue-1, Jan 2013, Pages 77-83, ISSN:2277-6915.