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A Study of the Lubricant Film Characteristics
of Journal Bearings *
(Part 2, Effects of Various Design Parameters on
Thermal Characteristics of Journal Bearings)

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This paper presents the theoretical results of a study on thermal characteristics of a journal bearing.

Joint calculation is made of the temperature distribution across the oil film thickness and on the bearing bush inner surface, as well as, of the heat flow ratio as a function of the clearance ratio, eccentricity ratio, journal speed, rate of heat diffusion on the bearing bush outer surface, lubricant viscosity and supply temperature. The results show that the maximum temperature and the location vary markedly with a change in the clearance ratio.

Moreover, the theory presented agrees with the experimental results of D. Dowson and others.

Key Words: Bearing, Lubrication, Lubrication Theory,
Thermal Conduction, Friction, Numerical Analysis

1. Introduction

In recent years, with an increase in speeds and working accuracy of rotary machines, the operating conditions of journal bearings, an important machine element, are increasingly severe. It has become difficult to accurately predict bearing performance merely on the basis of a theoretical analysis using the conventional isothermal (isoviscous) theory. The need has arisen for a theoretical analysis that takes viscosity variations in the oil film into account.

The first part of this study (1) reports on a theoretical analysis of the static characteristics of the oil film in light of temperature increases due to heat generation of the oil film.

The present report examines in detail the effects of various design parameters which include bearing clearance ratio, eccentricity ratio, supply temperature, viscosity characteristics of the lubricant, and heat flow radiated from the bearing bush outer surface on the thermal characteristics of journal bearings in order to provide more detailed basic information on bearing design.

Table 1 The properties of lubricants (at 40°C)

Lubricant		Transformer oil	#90 turbine oil	#140 turbine oil	HVI 65 mineral oil
Density	kg/m ³	852	861	868	869
Specific heat	J/(kg·K) [kcal/kgf°C]	1 993 [0.476]	1 951 [0.466]	1 943 [0.464]	1 993 [0.476]
Heat conductivity	W/(m·K) [kcal/m·h·°C]	0.123 [0.106]	0.131 [0.113]	0.130 [0.112]	0.131 [0.113]
Viscosity coefficient $\times 10^{-3}$ Pa·s [$\times 10^{-4}$ kgf·s/m ²]	30°C	18.8 [19.2]	39.2 [40.0]	78.5 [80.0]	38.7 [39.5]
	40°C	14.2 [14.5]	27.3 [27.8]	46.6 [47.5]	25.0 [25.5]
	50°C	9.81 [10.0]	17.7 [18.0]	31.4 [32.0]	17.7 [18.0]
	60°C	7.32 [7.46]	13.1 [13.4]	21.9 [22.3]	11.3 [11.5]
	70°C	5.49 [5.60]	9.81 [10.0]	15.7 [16.0]	8.83 [9.00]
Viscosity index	1/°C	0.028	0.029	0.033	0.029

2. Calculation conditions and model bearing specifications

Except for the model bearing used in the comparison with Dowson's experimental data, the bearing used in calculations in this report has a configuration basically identical to the model bearing of Part 1 of this study, being circular with a diameter of 50 mm, a bush outer diameter of 90 mm, a width of 35 mm and an oil groove of the upper metal in the vertical direction. With regard to bearing clearance, however, a change is made in the bearing clearance ratio ϕ , which in this report ranges from 0.0005 to 0.005. The

lubricants used in this part of the study were transformer oil (3), #90 and #140 turbine oils (4), and HVI 65 mineral oil (5). Table 1 gives values for the density, specific heat at constant pressure, and heat conductivity of these lubricants at 40°C.

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The viscosity coefficients approximate, an exponential function of temperature, as in Part 1 of the study. For reference, Table 1 also gives values for the viscosity coefficient at various temperatures and the concomitant viscosity index β . As for the other physical values and running conditions, unless otherwise specified, the bearing bush heat conductivity $\lambda_B = 53 \text{ W/(m}\cdot\text{k)}$ ($\lambda = 46 \text{ kcal/(m}\cdot\text{h}\cdot^\circ\text{C)}$), the heat transfer coefficient α for ambient air on the outer surface of the bearing bush $= 58 \text{ W/(m}\cdot\text{k)}$ ($= 50 \text{ kcal/(m}\cdot\text{h}\cdot^\circ\text{C)}$), ambient temperature $T_a = 20^\circ\text{C}$, and supply temperature of the lubricant $T_l = 40^\circ\text{C}$. The effect of these values and running conditions on the thermal characteristics of journal bearings was studied by varying the respective independent parameters in different ways. As in Part 1 of the study, a parameter η_{fj} is used to indicate the heat flow ratio to the rotating journal and a parameter η_{mix} to indicate the mixing coefficient between supply oil and recirculating oil at the lubricant supply inlet. In this report, the range used in the calculations is between 0.02 and 0.1 for η_{fj} and between 0 and 1 for η_{mix} .

3. Effects of the various parameters on the thermal characteristics of journal bearings

3.1 Effect of bearing clearance ratio

Figure 1-(a), (b) and (c) give the calculated effect of bearing clearance ratio ψ on temperature distribution $\bar{\theta} = (T - T_l)/(T_j - T_l)$ against the non-dimensional coordinate across the oil film η . Where, T_l is the supply temperature of the lubricant and T_j the journal surface temperature, the values $\eta = 0$ and $\eta = 1$ shown on the abscissa corresponding respectively to the positions of the bearing bush inner surface and the journal surface. Parameter φ in the figure expresses the circumferential angle from the lubricant supply location in the direction of rotation. The bearing clearance ratio ψ is 0.00075 in Fig. 1-(a), 0.0015 in Fig. 1-(b), and 0.002 in Fig. 1-(c) for eccentricity ratio $\epsilon = 0.5$ in each case. Thus as seen from these figures, in the case of a constant eccentricity ratio, there is a tendency for the overall temperature of the oil film to rise uniformly with a decrease in the bearing clearance ratio. The reason for this is examined below.

Viscosity dissipation of the oil film is represented by $\{\phi = \bar{\mu}(\partial u/\partial r)^2 + (\partial w/\partial r)^2\}$ from Part 1 of this study; when pressure distribution in the oil film is used, it can be qualitatively expressed as $\phi = \phi(\bar{\mu}^{-1}, H, \partial A/\partial \varphi, \partial A/\partial \xi, \eta)$ where $\bar{\mu}$ is the nondimensional viscosity coefficient, A is the nondimensional film pressure, and H is the nondimensional film thickness. Since the smaller the bearing clearance ratio, the greater the rise in temperature due to heat generation in the oil film, at a constant eccentricity ratio, the nondimensional pressure will

Fig. 1 Temperature distribution across the film thickness

drop and the extent of change in the nondimensional pressure gradient in the direction of rotation will become smaller in the vicinity of maximum pressure position. Furthermore, the nondimensional viscosity coefficient approaches an exponential function of nondimensional temperature and, in the high temperature region, the extent of changes in the viscosity coefficient with changes in temperature is small. From the above discussion, the variation in viscosity dissipation ϕ in the direction of rotation becomes smaller as the bearing clearance ratio decreases.

(a)

(b)

(c)

Fig. 2 Isotherms of the bearing bush

On the other hand, heat transfer to the bearing bush from the oil film is determined by the equation for thermal boundary conditions, Eq. (9) in Part 1 of this study. When the eccentricity ratio remains constant, the local nondimensional heat transfer coefficient in the direction of rotation

$\bar{\lambda}(1 + \phi^{-1})H^{-1}$ changes according to the bearing clearance ratio ϕ but there is no change in the degree of relative variation in the direction of rotation. Therefore, at a constant eccentricity ratio, differences in oil film temperature distribution for different circumferential positions caused by changes in the bearing clearance ratio seem to be influenced primarily by variations in oil film viscosity dissipation ϕ for different circumferential positions. At the same time, the smaller the bearing clearance ratio the smaller the difference in oil film temperature distribution for different circumferential positions and the greater the uniformity of oil film temperature in the direction of rotation from approximately the area of maximum oil film pressure position to the vicinity of minimum oil film thickness. It is considered that the rise in oil film temperature is delayed in the direction of rotation by the effect of the heat convection caused by the oil flow in that direction with an increasing bearing clearance ratio; and this in turn causes a shift in the same direction of the location of maximum temperature on the bearing bush inner surface.

Fig. 2-(a), (b) and (c) represent the isotherms of the bearing bush in accordance with the calculating conditions of Fig.

Fig. 3 Changes in bearing bush inner surface temperature according to clearance ratio

1-(a), (b) and (c), respectively. They show that with an increasing bearing clearance ratio, for a nearly identical journal attitude angle, the position of maximum temperature on the bearing bush inner surface changes considerably from upper to lower in the direction of rotation with respect to the location of attitude angle (or location of minimum film thickness).

Figure 3 shows the temperature distribution of the bearing bush inner surface in the direction of rotation for different bearing clearance ratios. Angle from inlet $\varphi = 0^\circ$ on the abscissa corresponds to the location of the lubricant inlet: the supply temperature $T_1 = 40^\circ\text{C}$ and the ambient temperature $T_\infty = 20^\circ\text{C}$. In this figure's case, journal speed is 10 m/s (3,800 rpm). While the maximum temperature of the bearing bush inner surface is around 50°C in the case of ordinary bearing clearance ratios ranging between 0.0015 and 0.002, as the clearance ratio decreases the maximum temperature begins to rise and reaches 88°C for $\psi = 0.0005$. Furthermore, the smaller the bearing clearance ratio, the higher the overall temperature of the bearing bush inner surface; for $\psi = 0.0005$ the temperature of almost the whole part of the bearing bush inner surface reaches the same level of the maximum one.

Figure 4 shows a heat balance of the bearing in the situation described above. The heat generated is divided into portions transferred to the lubricant, the bearing bush, and the journal. It shows the ratio of heat flow to each part against the eccentricity ratio, the parameter ψ being the bearing clearance ratio. From the figures it is found that the ratio of heat flow to the bearing bush becomes larger as the clearance ratio decreases. In every case the value becomes minimum when a given eccentricity

Fig. 4 Heat flow ratio to each part (effect of clearance ratio)

ratio is reached, and it increases with an increasing eccentricity ratio from that value. The smaller the bearing clearance ratio, the smaller the eccentricity ratio giving the minimum value.

3.2 Effect of eccentricity ratio

Figure 5-(a) and (b) represent temperature distribution on the bearing bush inner surface in the direction of rotation for different eccentricity ratios. Figure 5-(a) shows a curve for a bearing clearance ratio $\psi = 0.0015$ and Fig. 5-(b) one for $\psi = 0.002$.

From these figures, it is seen that the maximum temperatures on the bearing bush inner surface in the direction of rotation do not rise in the order of an increasing eccentricity ratio, and that the maximum temperature values increase for high and low eccentricity ratios. In other words, viscosity dissipation in the overall bearing oil film increases with higher eccentricity ratios, but the values for maximum temperatures do not always increase in this order. This seems to be due to the effect of a rising oil film temperature at the location of the oil supply groove ($\varphi = 0^\circ\text{C}$) due to

Fig. 5 Temperature distribution of the bearing bush inner surface according to eccentricity ratio

mixing between recirculating oil and supply oil. For a fixed η_{mix} (mixing coefficient), oil film temperature at $\varphi = 0^\circ$ is determined by recirculating oil flow and recirculating oil temperature. In the case of the calculating conditions in Fig. 5 ($\eta_{\text{mix}} = 0.5$), initial film temperature at the inlet decreases with an increasing eccentricity ratio until at a certain eccentricity ratio where it reaches a minimum and then increases. In the range of low eccentricity ratios, since there is no large local heat generation, it appears that the effect of the initial film temperature level extends in the direction of rotation. In the range of high eccentricity ratios, the initial film temperature level also rises and, in addition, there is a pronounced local heat generation within the oil

Fig. 6 Effect of heat radiation from the bearing bush outer surface on bearing temperature distribution

Fig. 7 Effect of heat transfer coefficient on heat flow ratio

film roughly from the region of maximum film pressure position to the vicinity of minimum film thickness position. Therefore, temperature rises abruptly in this area, resulting in a sharp temperature distribution peak seen in the figure. Here, the lower the value for η_{mix} , the lower the initial film temperature level for low eccentricity ratios. With $\eta_{\text{mix}} < 0.005$, maximum oil film temperature in the direction of rotation increases in nearly the same order as the eccentricity ratio.

As observed from the comparisons with the experimental results obtained by Dowson et al. as discussed later, maximum temperatures on the bearing bush inner surface in the direction of rotation rise, in experiments, for low and high eccentricity ratios. This seems to show that, actually, a certain amount of mixing takes place between the recirculating oil and the supply oil.

3.3 Effect of heat dissipated from the bearing bush outer surface

Figure 6 shows the effect of the amount of heat dissipated from the bearing bush outer surface on temperature distribution in the direction of rotation of the bearing bush inner surface and the amount of heat is changed by the heat transfer coefficient on the outer surface of the bearing bush.

The parameter $Bi = 2\pi r_{BO} / \lambda_B$ in the figure is a nondimensional quantity expressing the amount of heat dissipated from the outer surface of the bearing bush, where α : the heat transfer coefficient to the atmosphere, λ_B : the heat conductivity of the bearing bush, $2r_{BO}$: the bearing bush outer diameter. An increase in the heat transfer coefficient brings about an increase in the heat flow dissipated from the outer surface of the bearing bush and a reduction in the temperature of the inner surface of the bearing bush. The extent of these changes in temperature is not so pronounced as compared with the extent of the increase in the heat transfer coefficient. This,

Fig. 8 Temperature increase of the bearing bush due to supply temperature

as shown in the heat flow ratio against the heat transfer coefficient in Fig. 7, is considered to be because the ratio of heat flow through the bearing bush does not strongly depend on the coefficient.

3.4 Effect of lubricant supply temperature

Figure 8 shows a temperature increase of bearing bush inner surface from the supply temperature, $\Delta T = T_{Bi} - T_1$, using the lubricant supply temperature T_1 as a parameter. Figure 9 shows variations in the ratio of heat flow to different parts against lubricant supply temperature. As shown in Fig. 8, the temperature increment ΔT diminishes with a rise in supply temperature. As illustrated by Fig. 9, this is apparently due to the increase in the ratio of heat flow dissipated from the bearing bush with an increased supply temperature; the increased heat flow ratio reduces the contribution of the heat

Fig. 9 Effect of supply temperature on heat flow ratio

Fig. 10 Variations in the bearing bush inner surface temperature according to lubricant

generated to a temperature rise of the oil film. It is even conceivable that in extreme cases film temperature could be lower than supply temperature.

3.5 Bearing thermal characteristics for various lubricants

Figure 10 shows bearing bush inner surface temperatures when different lubricants are supplied under identical operating conditions. It shows the temperature distribution for three of the oils in Table 1: #140 turbine oil, #90 turbine oil, and transformer oil. The figure reveals that the temperature increase when a highly viscous #140 turbine oil is used, is approximately twice that under use of transformer oil.

Figure 11 shows the heat flow ratio when the same three lubricants are used

Fig. 11 Heat flow ratio according to lubricant

Fig. 12 Effect of film viscosity variation on the friction coefficient

and they are represented by the doublehead arrows $\leftrightarrow$. Since these lubricants differ in their physical properties such as specific heat, specific weight, heat conductivity and so on, it is not possible to obtain a continuous curve against the viscosity index on the abscissa. However, the dissimilarity between their properties is not so great that it is possible to predict the heat flow ratio against the viscosity index, and the dotted line is obtained in the figure. It is predicted from the result in Fig. 11 that lowering the viscosity index should result in a lower ratio of heat flow carried away by the lubricant.

3.6 Investigation of the bearing friction coefficient

Figure 12 represents the bearing friction coefficient f_{THD} as the ratio to the friction coefficient f_p calculated using Petroff's equation. The dotted lines in the figure indicate values calculated using isothermal theory (6). The figure shows that in the range of clearance ratios ordinarily used ($\psi = 0.0015$ or more) the difference from the values obtained with isothermal theory is small, but with ratios smaller than or equal to 0.001 the deviation from isothermal theory may no longer be negligible.

4. Comparison of theoretical values with experimental results

Figure 13 compares temperature distributions of the bearing bush using isotherms representing experimental values and theoretical values; Fig. 13-(a) and (b) show the experimental result by Dowson et al. and the calculated result by this study, respectively. From these isotherms, temperature distribution of the bearing bush obtained theoretically is found in good accordance with the experimental values. Though the heat transfer coefficient on the outer surface of bearing bush is constant, it can be considered a quite suitable approximation judging from the results shown in the figure.

Fig. 13 Comparison of experimental and theoretical results

Dowson also measured temperatures of different bearing parts under a changing bearing load. Figure 14 gives both measured values and the corresponding calculated values. The round symbols with dots in the middle $\odot$ represent the measured maximum temperatures of the bearing bush inner surface, the square symbols with dots in the middle $\boxdot$ represent the measured minimum temperature of the bearing bush outer surface, and the dot-and-dash line represents the measured journal surface temperature. The solid lines represent the corresponding calculated values. The same figure also includes calculated values for the

Fig. 14 Effect of bearing load on temperatures (comparison of theoretical and experimental results)

Fig. 15 Variation of heat ratio according to revolutional speed (comparison of theoretical and experimental results)

maximum film temperature and the average temperature of the bearing bush inner surface. It is apparent from the figure that the minimum temperatures for each part are obtained in the approximate region of bearing load = 9800 N (1000 kg) (and in the vicinity of eccentricity ratio $\epsilon = 0.4$). As explained in detail in section 3.2, this seems to be due to a remarkable effect, under low eccentricity ratios, of the rise in initial temperature at the inlet resulting from mixing of recirculating oil and supply oil. The theoretical values for these temperatures show the same tendency as the measured values and a relatively good correlation. This would seem to corroborate the suitability in general terms of the predictions concerning bearing thermal characteristics resulting from theoretical calculations, including the suitability of the mixing coefficient of recirculating oil flow to the supply oil flow, η_{mix} that is postulated theoretically.

Dowson et al. also measured the heat flow ratio to various parts under different journal rotational speeds and bearing loads. Figures 15 and 16 compare the theoretical and measured values of heat flow ratios against these parameters. The circles with dots in the center $\odot$ in these two figures represent Dowson's measured values and the solid lines the theoretical values. In the theoretical values, heat flow ratio to the rotating journal is constant, $\eta_{fJ} = 0.04$, as the calculating condition. The theoretical values show a good correlation to the measured values, but a slight discrepancy is observed in the low and high eccentricity ratio areas. This is thought to be due to the fact that the heat flow ratio to the journal in these areas is a constant in the theory while in reality it increases in these areas. However, in

Fig. 16 Variation of heat flow ratio according to eccentricity ratio (comparison of theoretical and experimental results)

the normal operating range it appears to be permissible to calculate η_{fJ} as a constant having a value between $\eta_{fJ} = 0.04 \sim 0.05$.

5. Conclusions

This report is a continuation of part 1 in which a detailed study is made into the effects on bearing thermal characteristics of various bearing designs and operational parameters such as bearing clearance ratio, eccentricity ratio, lubricant viscosity characteristics, supply temperature, heat transfer coefficient on the bearing bush outer surface, etc. It is elucidated from the study that the bearing clearance ratio has a great effect on thermal characteristics and that changes in its values could cause a remarkable rise in bearing temperature. Such a rise in temperature suggests that 'locking' and 'rubbing' between the bearing and journal may occur. Therefore, careful consideration must be given to this fact in designing and manufacturing the bearing. Furthermore, it is found that, when the eccentricity ratio is changed under otherwise identical conditions, the bearing temperature rises in low and high eccentricity ratio areas. This is due apparently to the effect of initial oil film temperature rising at the inlet resulting from mixing between recirculating oil and supply oil.

In this study, good correlations were obtained in comparisons between the experimental results of Dowson et al. and the calculated results obtained using the theory. This is thought to lend credibility to the thermal calculations for journal bearings based on the theoretical analysis of this study.

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