ANALYSES OF HEAT-PIPE COOLED ISOTHERMAL JOURNAL BEARINGS

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ABSTRACT

Isothermal journal bearings have been developed by incorporating heat-pipe cooling. The heat pipe assists in spreading heat rapidly along the bearing circumference, resulting in a very small temperature gradient, low peak temperature, and stable transient thermal behavior. This paper presents a model for the isothermal journal bearings and the results of analyses. The heat pipe in the isothermal journal bearing is modeled as a heat conductor whose effective thermal conductance is determined through the correlation between the analytical and experimental data. The heat-transfer coefficients at the bearing boundaries are obtained with the assistance of the experimental measurement and calculation using a semi-empirical correlation method. Good agreement is observed between the analytical and experimental results. The effects of bearing geometry and materials on the performance of the isothermal journal bearing are numerically investigated using the model thus developed. The analyses further confirm that the isothermal journal bearings have the ability to battle the heat-induced problems, which can significantly benefit bearing operation and failure prevention.

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Nomenclature:

A: Contact area, or heat-input area, m²
B: Bearing width, m
c: Specific heat, J/kg°C
dₒ, dᵢ, dₑ: Outer, inner, and mean diameters of the bearing, m
h: Heat transfer coefficient, W/m²°C
K, Kₑ, Kₛ: Thermal conductivity, thermal conductivity for steel and copper, W/m°C
Khp: Heat-pipe effective conductivity, W/m°C
Q: Heat input, W
Q, q₀: Heat flux, average heat flux over the contact area, W/m²
Rₒ, Rᵢ, Rₑ: Outer, inner, and mean radius of the bearing, m
T, T₀: Temperature, initial temperature, °C
t: Time, s
z, r, θ: Cylindrical coordinates, m, rad

Greek:

β: Angle of bearing orientation, degree
ΔT: Temperature difference, °C
ψ: Liquid volume angle, degree
ρ: Mass density, kg/m³
φ: "Load" distribution angle, degree

Subscripts:

c: Copper, or calculated temperature
hp: Heat-pipe (heat sink)
In, out: Input, output
m, ∞: Measured temperature, ambient temperature
max, min: Maximum, minimum
s: Surface, or stainless steel
T: Total
z, r, θ: Cylindrical coordinates

Superscripts:

k: Iteration number
1. INTRODUCTION

Today, machines with compact designs and tribological contacts that minimize lubricant consumption are increasingly important. However, heat is a serious barrier and it is anticipated that the accumulation of frictional heat will become a major problem for the development of these tribological systems. The common type of journal-bearing contact failure is seizure of the journal and bearing surfaces (Budinski, 1988 and Wang, 1998; Burton, R. A., 1965; Heckmann and Burton, 1979). It has been found that contact and lubricant film breakdown are responsible for failure initiation, and that temperature can alter the state of contact and lubrication (Khonsari and Kim, 1989; Ting and Winer, 1989; Ni and Cheng, 1995, a and b; Wang et al., 1994). Due to bearing structural distortion caused by frictional heating, the bearing may lose its designed clearance, resulting in multiple-contacts and seizure failure (Hazlett and Khonsari, 1996, Wang et al., 1995, Pascovici et al., 1995, Monmousseau et al., 1997). It is clear that temperature reduction at the contact interface and rapid dissipation of frictional heat from the contact region are crucial to protecting tribological interfaces and retaining the anti-failure capability of bearing surfaces.

A new isothermal journal bearing has been developed by utilizing the heat-pipe cooling technology (Wang and Cao, 1996, and Chen et al., 1998). This bearing has a number of interconnected circumferential heat-pipe grooves that are formed simply with a turning process. Stainless-steel bearings with methanol as the heat-pipe working fluid were fabricated for mechanism verification. Experimental results demonstrate that heat pipes enable the journal bearing to become a nearly isothermal element. The heat pipe uniformly distributes the “frictional heat” along the circumference of the bearing and drastically reduces the maximum temperature of the bearing. Experimental data also indicate that with the assistance of the heat
pipe, the bearing can work at much higher heat input than that the conventional bearings can.

Development of the isothermal journal bearing demands extensive investigations of its thermal performance and the effects of design parameters. It would be time consuming and costly if all the investigations were done exclusively through experimentation. A model for analyzing the heat-transfer process of isothermal journal bearings is necessary. This paper aims at developing a heat-transfer model for the isothermal journal bearing and studying the effects of bearing geometry and materials on its performance.

2. MODELING OF AN ISOTHERMAL JOURNAL BEARING

2-1 Description of the Isothermal Journal Bearing

An isothermal journal bearing generally consists of an inner ring, an outer ring, and several circumferential grooves machined on one of the rings. In the current study, the grooves are on the inner ring. These two rings are assembled by welding the edges of the contacting interfaces to form an isothermal journal bearing. After evacuating the space formed between the two rings, an amount of working fluid is charged into the grooves. The inner and outer rings of the bearing sandwich a group of interconnected circumferential heat pipes, as shown in Fig. 1 (a). Assuming that the contact surface area between the shaft and bearing is on the bottom half of the inner ring, the evaporator section of the heat pipes should be in the lower section of the bearing. During the bearing operation, frictional heat is generated in the loading region corresponding to the contact surface area. This heat is absorbed through the vaporization process of the working fluid in the heat-pipe evaporator section. The generated vapor flows circumferentially and condenses in the upper portion of the heat pipe, releasing the latent heat absorbed in the loading region and forming liquid condensate. The condensate flows back to the
evaporation section with the assistance of gravity, thus maintaining a continuous heat-transfer process in the heat pipe. Due to the large latent heat of vaporization, the heat generated in the loading region can be promptly spread over the entire circumferential surface area of the bearing to be dissipated. As a result, the temperature in the loading region can be significantly reduced, and that of the entire bearing can be much more uniform. Because the heat-pipe cooling is completely passive, no additional energy sources are needed to support the cooling process.

2-2. Mathematical Modeling

As a first step to investigate the thermal function of the isothermal journal bearing, heat can be input at a fixed location and achieved by using an electric heater. The heat-transfer process may be described by the heat conduction, Eq. (1), subject to the heat-input condition, Eq. (2), and convective boundary conditions, Eq. (3), in a cylindrical coordinate system:

\[ \frac{\partial}{\partial r} \left( K(r, \theta, z) \frac{1}{r} \frac{\partial T}{\partial r} \right) + \frac{1}{r^2} \frac{\partial}{\partial \theta} \left( K(r, \theta, z) \frac{\partial T}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( K(r, \theta, z) \frac{\partial T}{\partial z} \right) = \rho c \frac{\partial T}{\partial t} \quad (1) \]

\[ -K(r, \theta, z) \frac{\partial T}{\partial r} \bigg|_{r=R_i} = q(R_i, \theta, z), \quad 0 < z < B \quad \theta_i < \theta < \theta_z; \]

\[ -K(r, \theta, z) \frac{\partial T}{\partial r} \bigg|_{r=R_o} = h(T_\infty - T_s), \quad 0 < z < B \quad \theta \leq \theta_i, \theta \geq \theta_z; \]

\[ -K(r, \theta, z) \frac{\partial T}{\partial r} \bigg|_{r=R_p} = h(T_\infty - T_s), \quad 0 < z < B \]

\[ -K(r, \theta, z) \frac{\partial T}{\partial z} \bigg|_{z=0} = h(T_\infty - T_s), \quad R_i < r < R_p \]

\[ -K(r, \theta, z) \frac{\partial T}{\partial z} \bigg|_{z=b} = h(T_s - T_\infty), \quad R_i < r < R_o \]
where $K(r, \theta, z)$ is either the thermal conductivity of the bearing material or the effective thermal conductance of the heat pipe, $K_{hp}$. Because no bearing motion is involved in the analyses, the circumferential boundary condition other than that shown in Eq. 2 is not considered. These equations were solved by a finite difference (FD) method using Harvard Thermal’s PC3D computer code with 2,520 brick elements. The surface convection was discretized into 2,008 convection elements. Figure 1 (b) shows the mesh arrangement. The numerical convergence was controlled as $\Delta T = |T^{k+1} - T^k| / T^{k+1} < 0.1\%$ for temperature, and $\Delta Q = |Q_{\text{in}} - Q_{\text{out}}| / Q_{\text{in}} < 1\%$ for the overall heat balance.

2-3 Determining the Heat-Transfer Coefficient and Heat Pipe’s Thermal Conductance

The average convective heat-transfer coefficient, $h$, and the effective thermal $K_{hp}$, for the heat pipe in the journal bearing are unknown and should be determined using experimental data or a correlation. Once $h$ and $K_{hp}$ are obtained, the model can be applied to study the thermal performance of the isothermal journal bearing. Stainless steel bearings, with and without the heat pipes, fabricated with the structures shown in Figs. 2 (a) and (b) were tested using the apparatus shown in Fig. 3. Group 1 is named for a pair of larger bearings with and without the heat pipe. Similarly, group 2 is for a pair of smaller bearings. An electric heater was attached on the inner surface covered by angle $\phi$ shown in Figure 1 (a). A heat isolator was applied on the upper surface of the heater to minimize the heat loss. Chen et al. (1998) reported details of the experiments. The model shown in Fig. 1 (b) was first applied without the heat pipe to simulate the conventional bearing. The measured temperatures from this conventional bearing were used to determine the heat-transfer coefficient, $h$, by minimizing the root-mean-square $T$, relative to the measured ones, $T_m$. 
\[
\min\{Error(q,h)\} = \min\left\{ \frac{\sqrt{\frac{1}{n} \sum_{i=1}^{n} (T_{ci} - T_{mi})^2}}{\sqrt{\frac{1}{n} \sum_{i=1}^{n} T_{mi}^2}} \times 100\% \right\}
\]  

(4)

where \( n \) is the number of data records compared.

Two cooling boundary conditions were used in the present study. The first cooling boundary condition is that of natural convection with the environment that may be suitable for journal-bearing surfaces subjected to dry contact. A value of 12.5 W/(m\(^2\)°C) was obtained for the heat-transfer coefficient for bearing group 1 with RMS errors less than 11%. The heat-transfer coefficient for bearing group 2 obtained was 15 W/(m\(^2\)°C) and the corresponding RMS errors to be less than 20%. This larger error might be due to the neglect of the heat loss through the isolator.

On the other hand, the heat transfer coefficient can be calculated by using existing correlation. Theoretically, the heat-transfer coefficient for natural convection should include convection and radiation, because the radiate and convective heat transfer may have the same order of magnitude:

\[
h = h_c + h_r.
\]

(5)

\[
h_c = \frac{N_{aD} K_{air}}{D}
\]

(6)

Based on a heated horizontal cylinder the Nusselt number, \( N_{aD} \), may be calculated as follows (Incropera and DeWitt, 1996):

\[
N_{aD} = \left\{ 0.60 + \frac{0.387 R_{aD}^{1/6}}{1 + (0.559/P_r)^{9/16}} \right\}^{2/7}
\]

(7)

with
where $R_{aD}$ is the Rayleigh number, $g$ is gravitational acceleration, $\beta = 2.86 \times 10^{-3}$ W/mK is the volumetric thermal expansion coefficient, $D = 0.107$ m (for bearing group 1), and $D = 0.057$ m (for bearing group 2) are the outer diameters of the journal bearings, $\nu = 20.92 \times 10^{-6}$ m$^2$/s is the kinematics viscosity of air, $\alpha = 29.9 \times 10^{-5}$ K$^{-1}$ is the thermal diffusivity, $Pr = 0.7$ is the Prandtl number. For the thermal radiation of a cylinder, the equivalent heat transfer coefficient can be obtained as:

$$h_r = \varepsilon \sigma (T_s + T_w)(T_s^2 + T_w^2)$$

(9)

where $\varepsilon = 0.8$ is a radiative property of the surface emissivity and $\sigma = 5.67 \times 10^{-8}$ W/m$^2$K$^4$ is the Stefan-Boltzmann constant. Using Eqs. (9-13), the heat transfer coefficient was found to be $h = 11.2 \sim 12.3$ W/m$^2$C with the smaller value for the larger bearing. It was noticed that $h_r$ is about 40% of the total value of $h$. Comparison with the calculated heat-transfer coefficient reveals that the values determined through experimental correlation are sufficiently accurate for modeling.

The second boundary condition, the forced convection cooling, was achieved through a fan. This cooling condition may be used to simulate an oil-cooling condition. For bearing group 1, the heat-transfer coefficient was obtained as $54.5$ W/(m$^2$C) through correlation with the experimental data with RMS errors less than 15%. Estimation can also be made with the empirical relation for the forced convection of a rectangular bar, which is similar to the shape of the bearing cross section:

$$h_c = CR_{eD}^m P_r^{1/3} \frac{K}{D}$$

(10)

where $C = 0.102$, $m = 0.675$, and $D$ is the characteristic length that was taken to be the width of the bearing ring. The air speed, $V$, for the bearing experiment was estimated to be 2~5m/s. The
heat transfer coefficient of the bearing for the forced convection was determined to be 35–65 W/m²°C.

The values of \( h \) thus obtained were then used in the model to determine the effective heat conductance, \( K_{hp} \), for the heat pipe. The same methods of correlating the modeling with the experimental data and minimizing the RMS errors at the same conditions were adopted:

\[
\min \{\text{Error}(q, K_{hp})\} = \min \left\{ \frac{\sqrt{\frac{1}{n} \sum_{i=1}^{n} (T_{ci} - T_{mi})^2}}{\sqrt{\frac{1}{n} \sum_{i=1}^{n} T_{mi}^2}} \times 100\% \right\}
\]  

(11)

The value of \( K_{hp} \) was then found to be a function of the heat input, as shown in Fig. 4, with both natural and forced cooling conditions. The data reveal that \( K_{hp} \) generally increases with the heat input and that its value is higher when the heat transfer coefficient is lower. These characteristics of the heat pipe in the journal bearing suggest that the isothermal journal bearing can perform efficiently at high heat input and severe heat-transfer conditions. For convenience of calculation, \( K_{hp} \) as a function of heat input are empirically expressed as follows:

\[
\frac{K_{hp}}{K_s} = 651.6 \left( 10^{0.039 \left( \frac{qR_j}{T_sK_s} \right)} \right) \text{ for natural convection, and } \frac{qR_j}{T_sK_s} > 8.0
\]

\[
\frac{K_{hp}}{K_s} = 190.2 \left( 10^{0.17 \left( \frac{qR_j}{T_sK_s} \right)} \right) \text{ for forced convection, and } \frac{qR_j}{T_sK_s} > 3.0
\]

(12)

3. RESULTS VERIFICATION AND BEARING PERFORMANCE

Figures 5 (a) and (b) present the circumferential temperature distributions of the isothermal journal bearing and comparisons with that of a conventional journal bearing under the
same conditions for both experimental numerical data with an average error less than 10%. The maximum error occurs for the maximum outer-surface temperature of the cases without the heat pipe due to the fact that it is difficult to measure the large temperature gradient with a limited number of thermocouples. Similar conclusions can be drawn for the smaller bearings with an average error of 10%-15%.

The results indicate that the heat pipe efficiently spreads the heat circumferentially to facilitate highly effective heat convection with the surrounding media, leading to temperature uniformity and a sharp reduction of the peak temperature. Figure 6 further compares the isotherms of bearings with and without the heat pipe. The maximum temperature difference for the conventional journal bearing could be as high as 270°C. For the isothermal journal bearing, this difference reduces to as low as 18°C, only 6.7% of the former. The peak temperature is reduced about 50%. Temperature uniformity and low maximum temperature can significantly benefit bearing operation and failure prevention.

Figure 7 shows the comparison of the transient performance of the journal bearings with and without heat pipes. The maximum temperature difference for the bearing without the heat pipe increases with time until it reaches its steady state, resulting in a very high peak temperature. On the other hand the maximum temperature difference for the isothermal journal bearing is negligibly small. The temperature values rise with time but with a much smaller increment. The superb heat-transfer ability of the heat pipe enables the isothermal journal bearing to work at a much more stable condition.

4. PARAMETRIC INVESTIGATION

4-1 Geometric Design of the Isothermal Journal Bearing
Bearing width, $B$, and diameter, $d$, are two of the most important design parameters that affect the heat transfer and temperature distributions. For a given bearing width, increasing the bearing diameter increases the heat pipes length in the bearing. The effects of the diameter change on the bearing thermal performance for a constant heat input rate $Q$ and a constant heat flux $q$ that is applied within a given heat-application angle, $\phi$, are shown in Fig. 8 in terms of $T/T_\infty$ and $B/d$. Isothermal bearings with all sizes can work well and for both heat-loading cases. The reduction of the temperature difference is about 88-93%. The maximum reduction (about 92-93%) occurs when $B/d = 0.14 \sim 0.29$.

The cross-sectional area of the heat pipe, $A_{hp}$, strongly affects the bearing heat-transfer capacity. A ratio of the surface heating area to the cross-sectional area of the heat pipe, $A_{in}/A_{hp}$, is defined and varied by increasing the heat-input area, $A_{in}$. The effect of this ratio on the performance of the isothermal journal bearing was analyzed and the results are plotted in Fig. 9 (a). The results suggest that the isothermal journal bearing maintains a very small and nearly constant maximum-temperature difference for all of the area ratios considered. However, the isothermal journal bearing’s ability to reduce the maximum temperature difference, which is very high for the case without heat pipes, becomes more pronounced for larger heat-input areas.

The ratio of the cross-sectional area of the bearing over that of the heat pipe, $A_T/A_{hp}$, represents another key factor for the bearing performance. The $A_T/A_{hp}$ variation was numerically achieved by increasing the bearing outer diameter while keeping its inner diameter and the heat-pipe cross-sectional area the same. Figure 9 (b) shows the calculated results, which indicate that the isothermal bearings in the entire calculation range can work efficiently although the reduction in the maximum temperature difference is slightly enhanced as $A_T/A_{hp}$ decreases. This result suggests that heat pipes with a small cross-sectional area can still handle the cooling of the
bearings that have thicker walls very well as long as the heat-pipe function is properly maintained.

4-2  Effect of the Bearing Thermal Conductivity

The thermal conductivity of the bearing materials determines the heat conduction in conventional plain bearings. The superb heat conductance of the heat pipe greatly enhances the heat conduction in isothermal journal bearings. The effect of this enhancement becomes more pronounced for bearing materials of poorer heat conductivity. The heat-conductivity effect may be considered in terms of the relation between the bearing thermal performance and the ratio of the heat-pipe thermal conductance to the material heat conductivity, $K_{hp}/K_m$. Figure 10 presents the results of this analysis. The temperature control by the heat pipe is not very significant if the bearing material’s thermal conductivity is close to the value of the heat pipe’s thermal conductance. However, for most of the engineering materials, whose heat conductivity is below 200 ~ 300 W/mK, the heat pipe cooling has a strong control on both the maximum temperature difference and the peak temperature. It can be deduced from Fig. 10 that embedded solid heat-pipe rings and thick surface coatings made of highly heat-conductive materials may partially achieve the heat-pipe cooling functions.

5. CONCLUSION

This paper presents the modeling of a new isothermal journal bearing. The heat pipe in the isothermal journal bearing is modeled as a heat conductor whose thermal conductance is found to be very high and a function of the heat input. The heat-transfer coefficients at the bearing boundary were obtained with the assistance of experimental measurements and calculations using existing correlations. Good agreement was observed between the numerical
results and experimental data.

The performance of the isothermal journal bearings is characterized with significant reduction in the maximum temperature difference, depression of the peak temperature, and stability of transient temperature control. These advantages suggest that the isothermal journal bearing can perform efficiently at high heat input and under adverse heat-transfer conditions. Although designs with $B/d_l = 0.14\sim0.29$, a higher $A_T/A_{hp}$, and a lower $A_T/A_{hp}$ could provide a maximum reduction of the temperature, a journal bearing with other sizes can also work satisfactorily. For most of the engineering materials, whose heat conductivity is below 200 – 300 W/m$^\circ$C, the heat pipe cooling provides strong temperature control. The superb heat-transfer ability of the heat pipe enables the isothermal journal bearing to work at a much more desired thermal condition that significantly benefits bearing operation and failure prevention.

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REFERENCES


