

Thermohydrodynamic Lubrication Characteristics of High-Speed Tilting Pad Journal Bearings

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Tilting pad journal bearings are widely used for high-speed rotating machinery because they offer high vibration stability. A high rotational speed causes the bearing temperature to increase, which gives rise to serious concerns about seizure problems. Given this, the ability to precisely predict the bearing temperature is important. This report describes the experimental results for measurements of the bearing temperature at sliding speeds of up to 94 m/s. After that, the experimental results are compared to the simulation results obtained using the thermohydrodynamic lubrication (THL) method. In the THL simulation, a new mixing model in which the mixing ratio depends on the oil flow ratio between the pad inlet and outlet is introduced. Based on the comparison with the experimental results, the authors confirm that the present mixing model is reasonable and that it improves the accuracy with which the bearing temperature can be predicted.

1. Introduction

Tilting pad journal (hereinafter referred to as tilting pad) bearings have the advantage of high vibration stability at high speed, and are therefore widely used for high-speed rotating machinery such as turbo compressors. When designing such high-speed bearings, it is important to accurately estimate static characteristics such as oil film thickness and bearing temperature and dynamic characteristics such as stiffness and damping. This paper focuses in particular on the bearing temperature among the static characteristics.

In recent years, for the tilting pad bearing, a Directed Lubrication (DL) bearing⁽¹⁾ type has been sometimes used in place of a Flood Lubrication (FL) bearing type which has been widely used in the past. **Figure 1** illustrates a schematic diagram of these types of tilting pad bearings. The FL bearing type is one adapted to supply lubrication oil to a bearing surface while holding the lubrication oil inside a casing to some extent by side seals attached on both sides of the casing. The lubrication oil inside the casing is churned by a high-speed rotating shaft, and therefore the FL bearing type has the disadvantages of a large mechanical loss and the resulting rise in bearing temperature. On the other hand, the DL bearing type is one adapted to supply lubrication oil to the vicinity of a bearing surface from oil feed nozzles arranged between adjacent pads without providing any side seal. A churning loss as seen in the FL bearing is small, thus preventing a rise in bearing temperature. Note, however, that supplying appropriate lubrication oil is an important point.

In general, as a bearing material, soft alloys such as tin-based white metals and aluminum-tin alloys are used. These

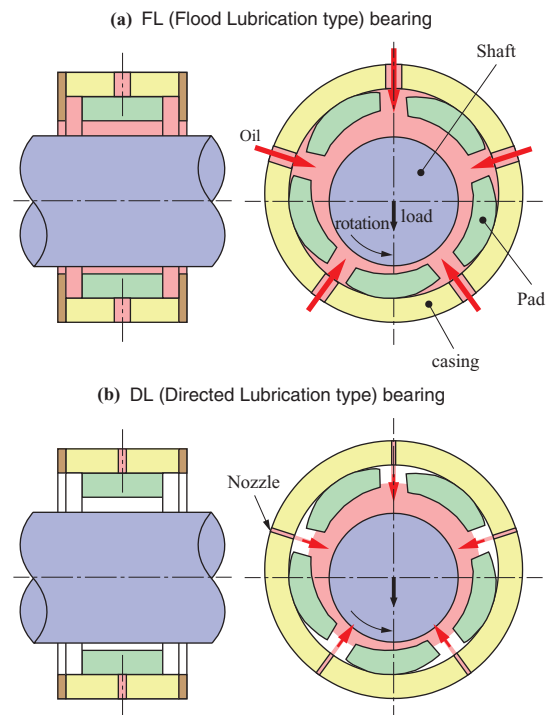


Fig. 1 Schematic diagram of tilting pad bearings

materials are soft and superior in anti-seizure performance, whereas it has the disadvantage that mechanical strength significantly reduces at high temperature, thus causing plastic flow. For example, in the case of tin-based white metal, the practically maximum allowable temperature of a bearing is said to be 120°C. Inside a bearing of high-speed rotating machinery, the rotation of a shaft causes shear heat generation of an oil film to increase the temperature of it,

and therefore it is extremely important to accurately predict the temperature of the bearing to keep the temperature less than the allowable temperature.

The bearing temperature can be predicted by Thermohydrodynamic Lubrication (THL) analysis^{(2), (3)}, which has already been introduced to designing. The THL analysis is one adapted to predict the bearing temperature from some factors such as the heat generation of the oil film and heat dissipation to the surroundings. However, since a flow inside the bearing is complicated, advanced modeling is required for the analysis. In particular, in the case of the FL bearing, part of high temperature oil discharged from a previous pad flows to a following pad, and at the same time, surrounding oil heated by churning is taken in. Accordingly, appropriately modeling how these differently heated oils are mixed is an important challenge in order to improve the prediction accuracy of the bearing temperature; however, a definitive model has not been established so far.

This paper describes current prediction accuracy that was examined by measuring pad temperatures of FL and DL bearings using a high-speed bearing test rig and comparing the measured temperatures with prediction results obtained by the THL analysis. Further, as for the FL bearing, an improvement in the prediction accuracy of the bearing temperature was achieved by introducing a new mixing model.

2. Symbols

Symbols used in this paper are as follows.

- h : Oil film thickness
- N : Rotational speed of shaft
- P : Oil film pressure
- r : Radial coordinates of pad
- T : Oil film temperature (lubrication oil temperature)
- T_B : Pad temperature
- U : Sliding velocity
- u : Velocity of lubrication oil in x direction
- v : Velocity of lubrication oil in y direction
- w : Velocity of lubrication oil in z direction
- x : Coordinates in rotational direction

- y : Coordinate in oil film thickness direction
- z : Coordinate in bearing width direction
- κ : Specific heat of lubrication oil
- λ : Thermal conductivity of lubrication oil
- θ : Coordinates in circumferential direction
- μ : Viscosity of lubrication oil
- ρ : Density of lubrication oil

3. Test method

Figure 2 illustrates the high-speed journal bearing test rig. The test rig is capable of rotating a test shaft having a diameter of $\phi 100$ mm to a maximum rotational speed of $25\,000\text{ min}^{-1}$. The test shaft is supported by two support bearings, and in the middle of the two support bearings, a test bearing is arranged. The test bearing can be applied with a static load from below by a load cylinder, and also applied with a dynamic load by a two-axis hydraulic shaker. Note that in this study, in order to focus on the static characteristics of a bearing, a test with the dynamic load applied was not performed.

As the bearing, a five-pads tilting pad bearing was used. The specifications of the test bearing are listed in **Table 1**. Two types of bearings having different preload factors m and clearance ratios ψ were used. The bearing model M00P30 has a preload factor $m = 0.003$ and a clearance ratio $\psi = 0.002\,96$, whereas the bearing model M03P20 has $m = 0.303$ and $\psi = 0.001\,98$. Numbers following M and P of each bearing model respectively represent rounded values of m and ψ .

In this study, pad surface temperature was measured in detail as a bearing characteristic. **Figure 3** illustrates a bearing

Table 1 Bearing specifications

Item	Symbol	Unit	Specification	
Bearing model	—	—	M00P30	M03P20
Bearing diameter	D	mm	99.989	99.989
Pad width	L	mm	50	50
Assembled radial clearance	C_b	mm	0.148	0.099
Machined radial clearance	C_p	mm	0.148 5	0.142
Preload factor	$m (= 1 - C_b/C_p)$	—	0.003	0.303
Clearance ratio	$\psi (= 2C_b/D)$	—	0.002 96	0.001 98

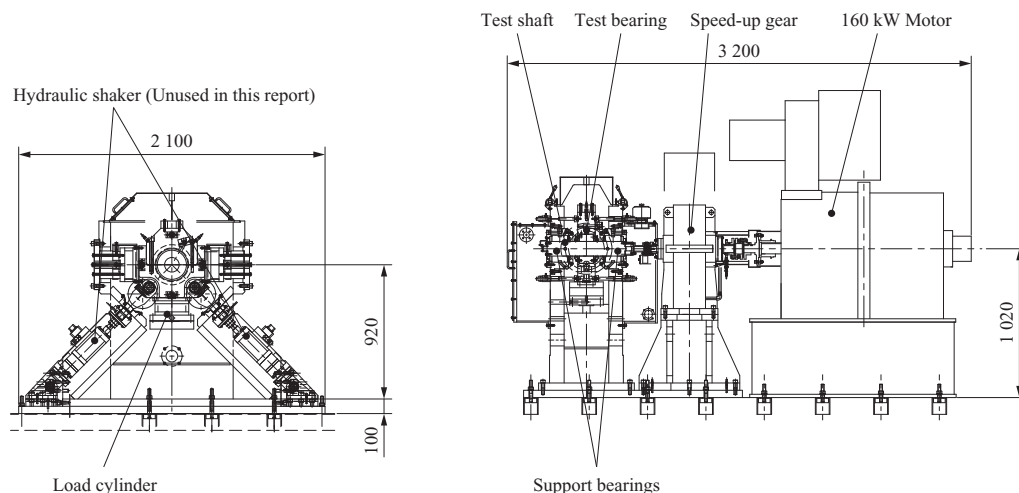


Fig. 2 High-speed journal bearing test rig (unit : mm)

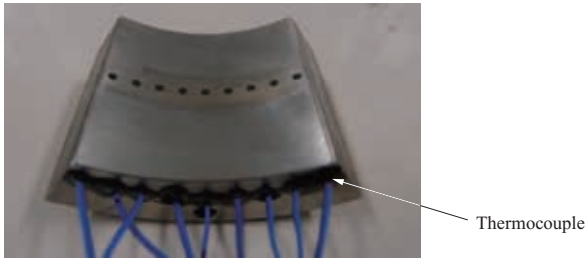


Fig. 3 Bearing pad attached with thermocouples

pad attached with thermocouples. The thermocouples used were type K thermocouples. Also, the thermocouples were passed through holes ($\phi 1.6$ mm) in a bearing surface from sides opposite to the junctions of them and fixed by epoxy resin at positions where the junctions were flush with the bearing surface. When any of the fixed junctions protruded from the bearing surface by $10 \mu\text{m}$ or more, the protrusion was ground using a grinder to eliminate an effect on oil film characteristics. Nine circumferential points in the middle of bearing width were set as measurement position. Five pads of this type were prepared per bearing, and temperatures at the 45 points in total were measured.

Figure 4 illustrates an oil feed nozzle used for the DL bearing type. The oil feed nozzle is one adapted to feed lubrication oil from 15 holes ($\phi 1.6$ mm) per pad, and aim to feed a bearing surface with the oil as cold as possible as

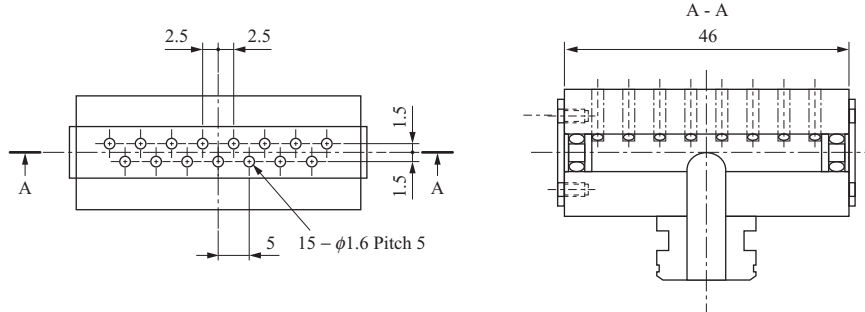


Fig. 4 Oil feed nozzle for directed lubrication bearing (unit : mm)

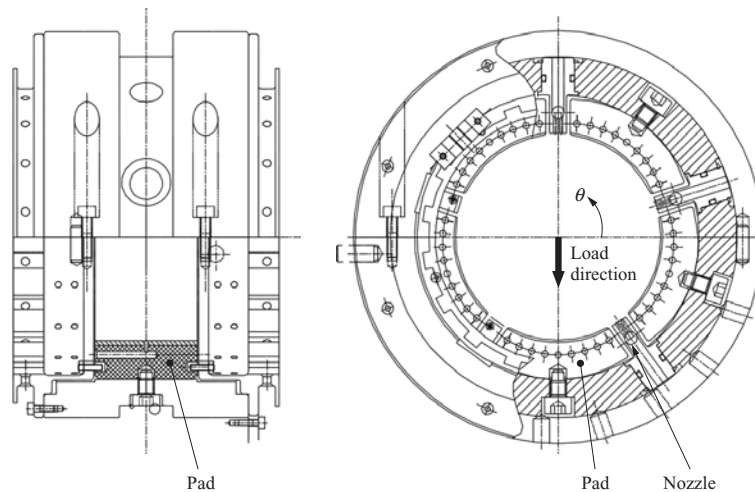


Fig. 5 Directed lubrication bearing equipped with nozzle

direct lubrication. Figure 5 illustrates the DL bearing type attached with nozzles of this type.

The rotational speed of a shaft was set to $N = 15\,000 \text{ min}^{-1}$ (sliding speed $U = 79 \text{ m/s}$) or $N = 18\,000 \text{ min}^{-1}$ ($U = 94 \text{ m/s}$). As the lubrication oil, Daphne Super Turbine Oil (VG32) was used, and oil feed temperature was set to $58 \pm 2^\circ\text{C}$. The lubrication oil was supplied to the bearing model M00P30 at 78 l/min and to the bearing model M03P20 at 61 l/min .

4. THL analysis

4.1 Basic equations

When performing the THL analysis, convergent calculation is performed by simultaneously solving the Reynolds equation and energy equation for an oil film, the heat conduction equation for a pad, etc.^{(2), (3)}.

First, the pressure distribution of the oil film is obtained in accordance with the Reynolds equation given by Equation (1). The boundary condition is that pressure at the outer circumference of the pad is $P = 0$.

$$\frac{\partial}{\partial x} \left(f \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial z} \left(f \frac{\partial P}{\partial z} \right) = U \frac{\partial g}{\partial x} \quad \dots\dots\dots(1)$$

where f and g are functions of viscosity, and using

$$\Phi_k(y) = \int_0^y \frac{y^{k-1}}{\mu} dy \quad (k = 1, 2, 3) \quad \dots\dots\dots(2)$$

respectively defined by

$$f = \varphi_3(h) - \frac{\{\varphi_2(h)\}^2}{\varphi_1(h)} \quad \dots\dots\dots(3)$$

$$g = h - \frac{\varphi_2(h)}{\varphi_1(h)} \quad \dots\dots\dots(4)$$

The temperature distributions of the oil film and the pad are obtained in accordance with the energy equation given by Equation (5) and the thermal conduction equation given by Equation (6).

$$\rho\kappa \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = \lambda \frac{\partial^2 T}{\partial y^2} + \mu \left\{ \left(\frac{\partial u}{\partial y} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 \right\} \quad \dots\dots\dots(5)$$

$$\frac{\partial^2 T_B}{\partial r^2} + \frac{1}{r} \frac{\partial T_B}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_B}{\partial \theta^2} + \frac{\partial^2 T_B}{\partial z^2} = 0 \quad \dots\dots\dots(6)$$

The boundary condition is that at the boundary of the oil film and the pad, temperature and heat flux are continuous. The rest of the pad surface is applied with heat transfer based on Newton's law of cooling stating that a heat dissipation amount is proportional to the difference between surface temperature and ambient temperature. At the boundary of the oil film and a shaft, the heat balance between the shaft and the oil film is set to zero in accordance with Equation (7). Note that an area in Equation (7) is assumed to be the area of the entire sliding surface.

$$\iint \lambda \frac{\partial T}{\partial y} dS = 0 \quad \dots\dots\dots(7)$$

The temperature dependence of the viscosity of the lubrication oil is defined by an exponential function given by Equation (8). Here, μ_0 represents viscosity at temperature T_0 and β represents a constant determined by the lubrication oil.

$$\mu = \mu_0 e^{-\beta(T-T_0)} \quad \dots\dots\dots(8)$$

4.2 Mixing model

Equation (5) giving the energy equation for the oil film should be provided with the temperature of the lubrication oil at a pad inlet (mixing temperature) as the boundary condition. **Figure 6** illustrates the mixing of lubrication oil at the pad inlet. As illustrated in the figure, the temperature T_1 of the lubrication oil flowing from the inlet of some pad B to a bearing surface is that of the mixture of part of high temperature (T_2) oil discharged from a previous pad A and low temperature (T_m) oil taken in from the surroundings.

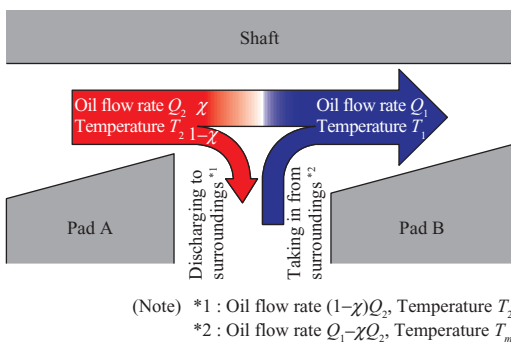


Fig. 6 Mixing of lubrication oil at pad inlet

Here, it is assumed that only a certain ratio χ of a lubrication oil amount Q_2 discharged from the previous pad A is fed to the following pad B. χ is referred to as a mixing ratio.

In the case of the FL bearing type, since the lubrication oil is churned inside the casing of the bearing, the temperature T_m of the oil taken in from the surroundings is higher than the oil feed temperature T_{in} because of the heat generation of the bearing. Given that the heat generation amount of the whole of the bearing is represented by E , T_m can be obtained from the energy balance of the whole of the bearing in accordance with Equation (9).

$$E = \rho\kappa Q_{in} (T_m - T_{in}) \quad \dots\dots\dots(9)$$

Mixing temperature T_1 in this case is given by

$$T_1 = \{ \chi T_2 Q_2 + T_m (Q_1 - \chi Q_2) \} / Q_1 \quad \dots\dots\dots(10)$$

In addition, in the case of the DL bearing type, the cold oil (T_{in}) from the outside is directly fed to the bearing surface, and therefore $T_m = T_{in}$.

In general, the mixing ratio χ is said to be approximately 0.7 to 0.8, and usually treated as a constant⁽⁴⁾. However, it is considered that the value of χ changes depending on the magnitude relationship between an oil amount at the inlet of the pad B and that at the inlet of the previous pad A. For example, when the clearance of the pad B inlet is smaller than that of the pad A outlet, i.e., the oil amount Q_1 at the pad B inlet is smaller than the oil amount Q_2 at the pad A outlet, χ decreases, whereas Q_2 is smaller, χ increases and is expected to come close to 1. For this reason, an oil amount ratio $Q_{1/2} = Q_1/Q_2$ is defined, and a function representing that as $Q_{1/2}$ increases, χ increases and comes close to 1 is defined.

$$\chi = 1 - (1 - \chi_0)^{Q_{1/2}} \quad \dots\dots\dots(11)$$

Here, the parameter χ_0 represents a mixing ratio when $Q_{1/2} = 1$. The relationship between the oil amount ratio $Q_{1/2}$ and the mixing ratio χ is illustrated in **Fig. 7**. In this paper,

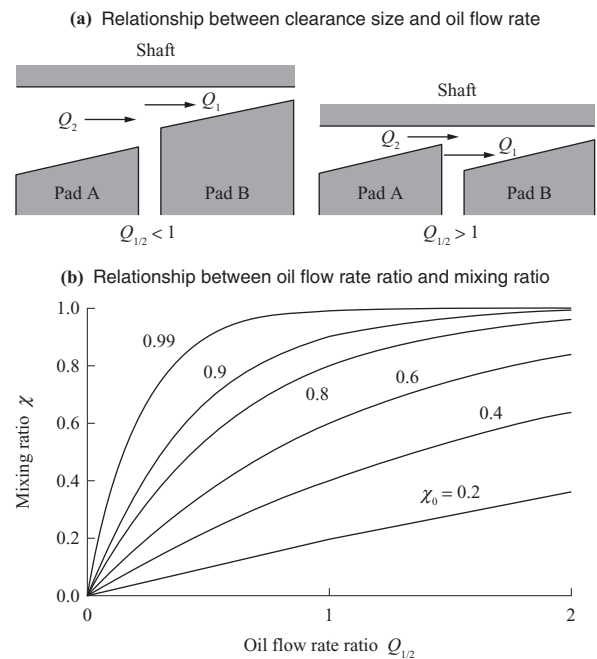


Fig. 7 Relationship between flow rate ratio $Q_{1/2}$ and mixing ratio χ

from a result of calibration based on test results, the parameter χ_0 was set as $\chi_0 = 0.8$.

5. Results and discussion

First, the bearing temperatures of the bearing models M00P30 and M03P20 used as the DL bearing type at the rotational speed $N = 18\,000\text{ min}^{-1}$ ($U = 94\text{ m/s}$) are illustrated in **Fig. 8**. A load was applied on the center of a pad 1 ($\theta = -90^\circ$), i.e., a Load on Pad (LOP) condition. The pad 1 carrying the load exhibits the highest temperature, and a temperature rise of 50 to 60°C relative to the oil feed temperature is observed. In addition, measured values and corresponding calculated values by the present analysis well

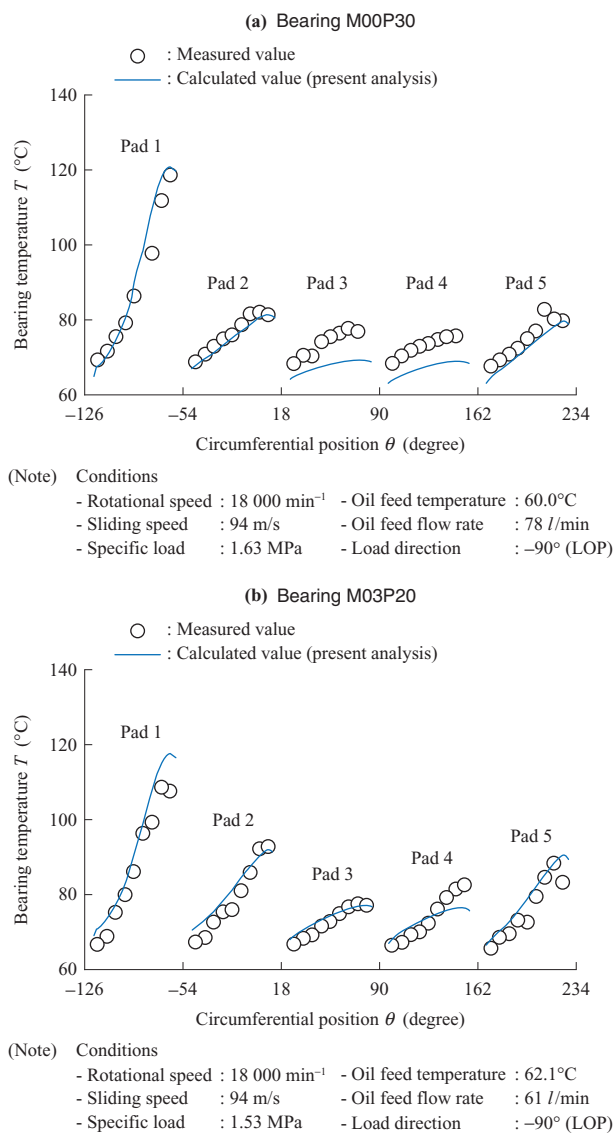


Fig. 8 Bearing temperature of directed lubrication bearings

coincides with each other, and the prediction error of the maximum temperature is approximately 5°C.

Next, the bearing temperature of the bearing model M00P30 used as the FL bearing type at the rotational speed $N = 15\,000\text{ min}^{-1}$ ($U = 79\text{ m/s}$) and a specific load of 1.61 MPa is illustrated in **Fig. 9**. When the loaded pad 1 was at the maximum temperature, there appeared an error of approximately 10°C between the measured value and the calculated value by the present analysis. In addition, in order to confirm the validity of the mixing model described in **Section 4.2**, this figure also illustrates calculated values of bearing clearance by the present analysis and results obtained by conventional analysis with the mixing ratio fixed at $\chi = 0.8$. Further, the flow rate ratio and mixing ratio between adjacent pads (analysis results) in this case are listed in **Table 2**. As for the loaded pad 1 expected to have the highest temperature, the pad inlet temperature obtained by the present

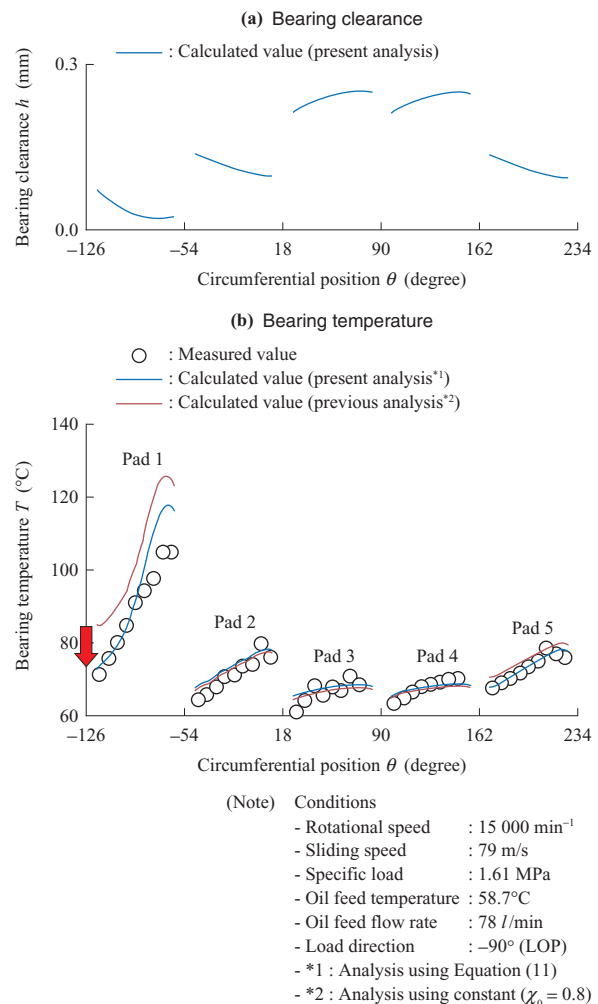


Fig. 9 Bearing temperature of flood lubrication bearing M00P30

Table 2 Flow rate ratio and mixing ratio between adjacent pads (analysis results)

Adjacent pad numbers		5 - 1	1 - 2	2 - 3	3 - 4	4 - 5
Flow rate ratio $Q_{1/2}$		0.465	3.92	1.97	0.852	0.502
Mixing ratio χ	THL (Present analysis)	0.527	0.998	0.958	0.746	0.554
	THL (Previous analysis)	0.8				

analysis was lower than that obtained by the conventional analysis, and well coincides with the measured value. In the conventional analysis, 80% ($\chi = 0.8$) of the heat amount of the oil discharged from the previous pad 5 is fed to the following pad 1. However, as compared with a calculated value of an oil amount by the present analysis, since the flow rate ratio is $Q_{1/2} = 0.465 < 1$, in other words, since the oil amount Q_2 discharged from the previous pad 5 is considerably larger than the oil amount Q_1 flowing in from the pad 1, 80% is never fed, and therefore the mixing ratio decreases. In fact, the present analysis estimated the mixing ratio as $\chi = 0.527$. On the other hand, as for the inlet of the pad 2, $Q_{1/2} = 3.92 > 1$, and the heat amount of the oil discharged from the previous pad is mostly fed to the following pad ($\chi = 0.998$).

As described above, it has been confirmed that the present analysis model using χ as a function of the flow rate ratio $Q_{1/2}$ between the inlet and outlet of a pad can represent appropriate results over a wide range of flow rate ratio $Q_{1/2}$. In addition, as a result, the prediction accuracy of the maximum bearing temperature has been improved by 8°C.

From the above results, it has turned out that the THL analysis tends to make an estimate higher than a measured value by approximately 5 to 10°C although it can predict bearing temperature with high accuracy. As one of the causes for the misestimate, the effect of thermal deformation of a bearing is conceivable. When the temperature of the sliding surface of a pad rises, the pad is thermally deformed in a direction to open because of the temperature difference with the back surface. In addition, the casing itself of the bearing is also thermally expanded along with the temperature rise of the lubrication oil. From the above, it can be supposed that a bearing clearance is enlarged to suppress heat generation. The present analysis takes account of the thermal deformation of a pad on the basis of the average temperature of the front surface of the pad; however, the accuracy of the deformation may be insufficient. We will examine this matter to clarify it in the future by accurately measuring the bearing clearance in operation. In addition, other causes such as heat transfer at a pad outlet, and the effect of adiabatic expansion of oil due to the quick release of oil film pressure are conceivable, and in the future, it is necessary to figure out them.

6. Conclusion

In this paper, the bearing temperatures of the high-speed tilting pad bearings were measured at sliding speeds up to 94 m/s, and the measurement results were compared with the THL analysis results. The THL-based prediction accuracy of the bearing temperatures was approximately 5 to 10°C. In addition, for the mixing ratio important for predicting the temperature of flood lubrication bearing, the new model adapted to determine the mixing ratio on the basis of the ratio in flow rate between the inlet of some pad and the outlet of the previous pad was introduced. Using this model allowed the prediction accuracy of the maximum bearing temperature to be improved by 8°C as compared with the conventional analysis using a fixed mixing ratio. We will aim to decrease the size of and increase the efficiency of high-speed rotating machinery using the analysis technology established this time.

We will also improve the analysis technique to decrease the temperature prediction error examined in this paper, as well as examining the prediction accuracy of dynamic characteristics (the stiffness and damping characteristics of the oil film) not described in this paper to achieve the stable operation of rotating machinery.

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