Analysis of Thrust Bearings With Faced Pads- PTFE TEHD

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I. INTRODUCTION

Tilting pads of hydrodynamic thrust bearings are usually faced with a thin layer of soft alloy to prevent shaft damage and embed contaminative particles. This alloy, called white metal or babbitt, imposes a temperature limit on safe bearing operation. At elevated temperatures babbitt loses its strength and starts to creep. Another negative effect of temperature is excessive pad thermal crowning. These thermal constraints can be eased by application of polytet-rafluoroethylene (PTFE) as pad facing. This material sustains higher temperatures and by virtue of its excellent insulating prop- erties prevents excessive pad thermal crowning.

Failures of large babbitted bearings in the generators of a hy- droelectric station on the Volga River caused by excessive pad thermal deflection prompted application of a novel PTFE compos- ite coating to thrust bearing pads. The coating developed by aresearch group of Professor A. M. Soifer [1] had a total thickness of 10 mm that included 2 mm of pure PTFE, 1 to 2 mm of combined PTFE-wire mesh and 6 to 7 mm of wire mesh. After extensive laboratory tests the first PTFE-faced thrust bearing was brought to operation in 1974 [2]. Today, 30 years later, all thrust bearings used in Russian hydroelectric stations have pads faced with this coating. Despite such a long and successful operating experience data available in literature on actual bearing perfor-mance is limited [3].

Recently, Ettles et al. [4] presented results of a TEHD analysis of two thrust bearings that had pads faced with a layer of a 8 –10 mm thick PTFE composite. Russian wire mesh technique [1] was used to bond PTFE to the pads. Based on the TEHD analysis and The goal of the present paper is to further analyze and quantify the effect of a thin PTFE layer on bearing thermohydrodynamic performance. For this purpose THD and TEHD models backed by a complete set of measured bearing operating characteristics such as temperature, power loss, oil film thickness and pressure are employed.

Keywords: PTFE, Thrust Pad Bearing, Temperature, Oil Film Thickness, Pressure

II. THEORY

A TEHD analysis similar to the one presented for the first time in Ref. [5] for tilting-pad journal bearings is applied to thrust bearings. In a previous study [6], the influence of the layer of soft material on the pads was not considered. In the present study, two models, THD and TEHD, are used. The temperature field is three- dimensional and the heat transfer in the pads is considered. In the TEHD model, the thermal and mechanical displacements of the pad and the PTFE layer are considered. The finite difference method is employed to solve the THD problem and the finite element method is used to determine pad (steel backing material) deformations due to the temperature and pressure fields. The PTFE layer deformations due to the pressure field are determined using an analytical model.

2.1 Basic Equations. The thrust bearing is assumed to op-erate without misalignment and consequently, the behavior of only one pad is studied. The film thickness is expressed as a function of the film thickness at the level of the pivot hp , of the radial and circumferential tilt angles (a and þ) and of the thermal and mechanical displacements (6d(r,0)). Using cylindrical coordinates, the film thickness is expressed as follows:

$$h(r,0)=h_p+R_p \sin b-r(\sin b\cos 0-\sin a\sin 0)+6_d(r,0) \quad (1)$$

Under steady state operating conditions, for a Newtonian in-compressible fluid, the generalized Reynolds equation is written as follows:

$$\frac{\&}{\&r}\left(rG_1\frac{\&p}{\&r}\right)+\frac{1}{r}\frac{\&}{\&0}\left(rG_1\frac{\&p}{\&0}\right)=ru\frac{\&G_2}{\&0} \quad (2)$$

where G_1 and G_2 are the integrals defined by:

$$G_1=\int_0^{h}\frac{z}{\mu}\left(z-\frac{I_2}{J_2}\right)dz;\quad G_2=\int_0^{h}\frac{z}{\mu}dz \quad (3)$$

In this relation, ε_{ij} and σ_{ij} are strain and stress tensors respec-

$$I_2=\int_0^h \frac{z}{\mu}dz; \quad J_2=\int_0^h \frac{dz}{\mu} \qquad (4)$$

In these relations, r, θ, and z are the cylindrical co-ordinates, $\mu(r,\theta,z)$ is the dynamic viscosity and p is the film pressure, which is also the main unknown of this equation.

The temperature in the fluid film is determined by solving the energy equation:

$$\rho c_p\left(u\frac{\partial T}{\partial r}+\frac{v}{r}\frac{\partial T}{\partial \theta}+w\frac{\partial T}{\partial z}\right)=K\frac{\partial^2 T}{\partial z^2}+\mu\left[\left(\frac{\partial u}{\partial z}\right)^2+\left(\frac{\partial v}{\partial z}\right)^2\right] \qquad (5)$$

u, v, and w are the velocity components and $T(r,\theta,z)$ is the fluid temperature.

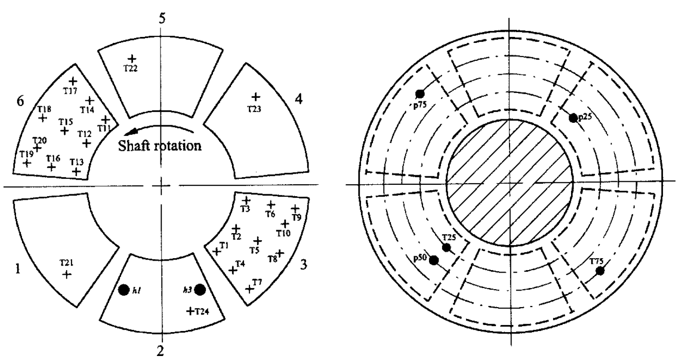
In the pads, i.e., both backing and soft material, the temperature is calculated using the heat transfer equation:

$$\frac{\partial^2 T_p}{\partial r^2}+\frac{1}{r}\frac{\partial T_p}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T_p}{\partial \theta^2}+\frac{\partial^2 T_p}{\partial z^2}=0 \qquad (6)$$

$T_p(r,\theta,z)$ is the pad temperature.

The thermoelastic displacements of the pad steel backing are obtained by applying the thermoelastic law:

$$\varepsilon_{ij}=\frac{1+\nu_p}{E_p}\sigma_{ij}-\frac{\nu_p}{E_p}\sigma_{kk}+\alpha_p\Delta T_p\delta_{ij} \qquad (7)$$



The PTFE layer is modeled as a layer of a homogenous material having the characteristics of the pure PTFE. Number of nodes used is as follows: 5 nodes across the layer thickness; 11 nodes in the radial direction; 15 nodes in the circumferential direction. 7 nodes are used in pad thickness direction (in the steel backing).

In the TEHD calculations the soft layer deformation is treated by means of a simple analytical model. This model is known as the column or Winkler model [7] and was used for compliant journal bearings by several authors [8,9]. According to the model, the deformation due to pressure on the layer is determined using the following relationship:

$$u_{PTFE}(r,\theta)=\frac{(1+\nu_{PTFE})(1-2\nu_{PTFE})}{(1-\nu_{PTFE})}\frac{t_{PTFE}}{E_{PTFE}}p(r,\theta) \qquad (8)$$

This relationship is obtained assuming plane strain hypothesis and that the layer is thin compared to the other dimensions of the pad. The local deformation is supposed to be only dependent on the local pressure. A comparison between numerical results (Finite Element Method) and results obtained with Eq. (8) for a given pressure field confirms the accuracy of this relationship.

2.2 Boundary Conditions. Classical Reynolds boundary conditions are used when solving the generalized Reynolds equa- tion. The thrust bearing is fully flooded and the pressure is equal to zero at the pad boundaries.

The film temperature at the leading edge of the pad is deter- mined taking into account the imposed oil flow rate and the total quantity of hot oil carried out from the inner radius and the trailing edge of the pads as described in Ref. [6]. Compared to the classic determination of the inlet temperature, this method does not require a mixing coefficient in the grooves. In this way, the average temperature at the leading edge of the film is obtained by applying conservation equations and taking into account the amount of fresh oil entering into the bearing and the total quantity of hot oil carried out from one pad to next.

The continuity of the heat flux is ensured at the film-layer and layer-pad backing interfaces. Temperature at the collar surface is constant and equal to measured one. On the outer surfaces of the pad, free convection hypothesis is used to take into account the heat exchange with the oil surrounding the pads.

At the back of the pad, axial displacements are equal to zero at the level of the pivot zone. The displacements of the correspond- ing point at the active pad surface are imposed at zero in both radial and circumferential directions. Solids characteristics are given in Table 1.

2.3 Numerical Procedure. The temperature field in both film and pad as well as the hydrodynamic pressure field are ob- tained using iterative techniques. The finite difference and Gauss– Seidel methods with over relaxation are employed to solve the generalized Reynolds equation. The same methods are used to solve the heat transfer equation in the pad, both layer and pad backing. An implicit finite difference method is used to solve the energy equation in the film. The method takes into account the reversed flow as described in Ref. [10]. The number of nodes in the oil film is 11 across the film thickness, 11 in the radial direc- tion, and 15 in the circumferential direction.

Iterative technique is also used to determine the pad deforma- tions. Classical finite element method is employed for pad backing to calculate axial pad displacements due to both pressure and tem- perature fields. For the PTFE layer, only the displacements due to the hydrodynamic pressure are considered: They are calculated analytically applying equation (8). Of course, due to the high non linearity of the TEHD problems, the displacements are under relaxed. An under-relaxation coefficient of 0.01 is applied for the PTFE layer displacements.

The CPU time to obtain a TEHD solution for given operating conditions varies from 1 to 10 min.

III. EXPERIMENTS

The experimental facility used in the tests is described in detail elsewhere [11]. A 6 pad bearing with a mechanical equalizing system was used in the tests. The bearing had sector shaped pads each subtending an arc of 50° and supported on a spherical pivot. Pads of the test bearing were lined with a 1.4 mm layer of a PTFE composite. Bearing main characteristics are tabulated in Table 2. Bearing instrumentation includes temperature and film thick- ness sensors. Their location is depicted in Fig. 1. All pads are equipped with thermocouples at the 75/75 positions (expressed as percent radial and percent circumferential extent). In addition two pads are instrumented with 9 thermocouples each to obtain temperature distributions over the pads. All thermocouples are located approximately 4 mm below the actual PTFE surface. To measure oil–PTFE interface temperature two thermocouples T6, T16 are installed according to the method described in Ref. [12]. Each thermocouple, T6 and T16, is placed at the bottom of a 0.5 mm diameter hole that ‘‘taps’’ oil from the oil film. Oil flows from the film, across the head of the thermocouple and is then expelled through a hole at the pad trailing side. Thus, these thermocouples allow measurement of the interface temperature. Thermocouples T6, T16 are located at the nondimensional radial/circumferential position (%,%) 50/90. Additional thermocouples are used to moni- tor oil supply and drain temperatures.

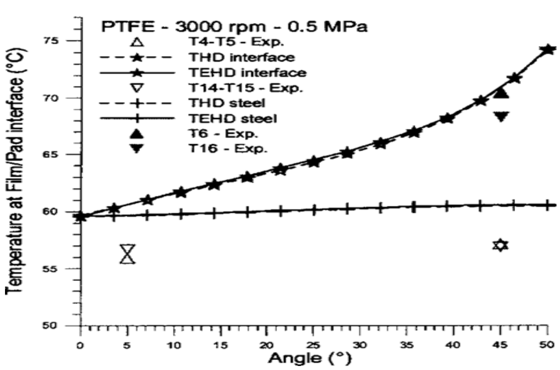


Fig. 2 Temperatures at the film–PTFE interface and in the pad steel backing, 4 mm from the interface

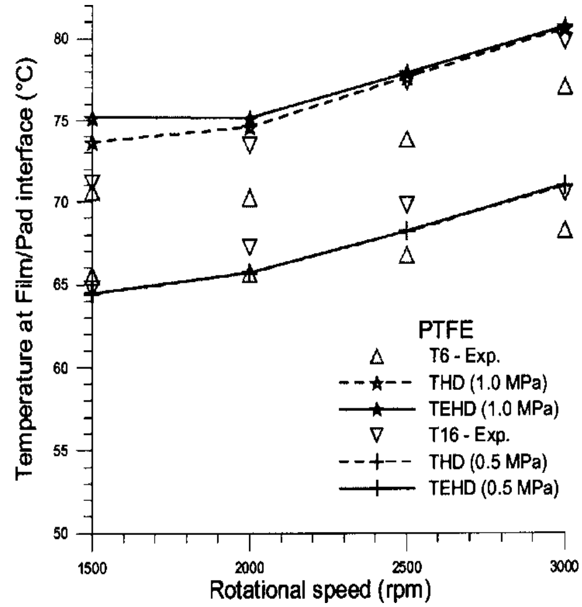


Fig. 3 Temperatures at the film–PTFE interface along the pad mean radius

One of the bearing pads is instrumented with oil film thickness sensors. A capacitive sensor is mounted at the 50/10 position close to the leading edge. Another capacitive sensor is located at the 50/90 position close to the trailing edge as indicated in Fig. 1.

Figure 1 also shows details of sensors mounted in the shaft for measuring temperature and pressure at various points across the shaft collar as it passes across the bearing pads. Numbers in each sensor labeling indicate its nondimensional position relative to the pad inner radius.

Two thermistors, T25 and T75, are installed 1.5 mm beneath the surface of the collar face. Measurements obtained from the ther- mistors, apart from providing information about temperature dis- tribution in the collar, are used for thermal compensation of the pressure transducers.

Three pressure transducers, p25, p50, and p75, are mounted in the collar. Their paths across the pads are indicated in Fig. 1. The measuring area in this case is 0.5 mm in diameter. A maximum sampling rate of 20 kHz allows for 55 pressure readings to be obtained across a pad at 3000 rpm shaft speed.

More details on bearing and shaft instrumentation can be found elsewhere [11]. Uncertainties of the key measured parameters are tabulated in Table 3. All data were logged by a PC-based high-speed data acquisition system.

The lubricant used is a mineral oil ISO VG 68 having viscosity 60.73 at 40°C and 7.36 mPas at 100°C. Oil density is 861 kg/m3 at 40°C. Supplied oil temperature of 50°C and flow rate of 15 l/min were held constant within ±0.2°C and ±0.1 l/min, respec- tively, for all load/speed combinations.

IV. RESULTS AND DISCUSSION

In the analysis the outputs of the THD and TEHD models are compared with the measured bearing characteristics. Results are presented for the PTFE-faced and babbitted bearings. The effect of the PTFE layer thickness on temperature, oil film thickness, pad tilt and pressure is also considered. Results are presented for a PTFE layer thickness of 1.4 mm unless otherwise noted.

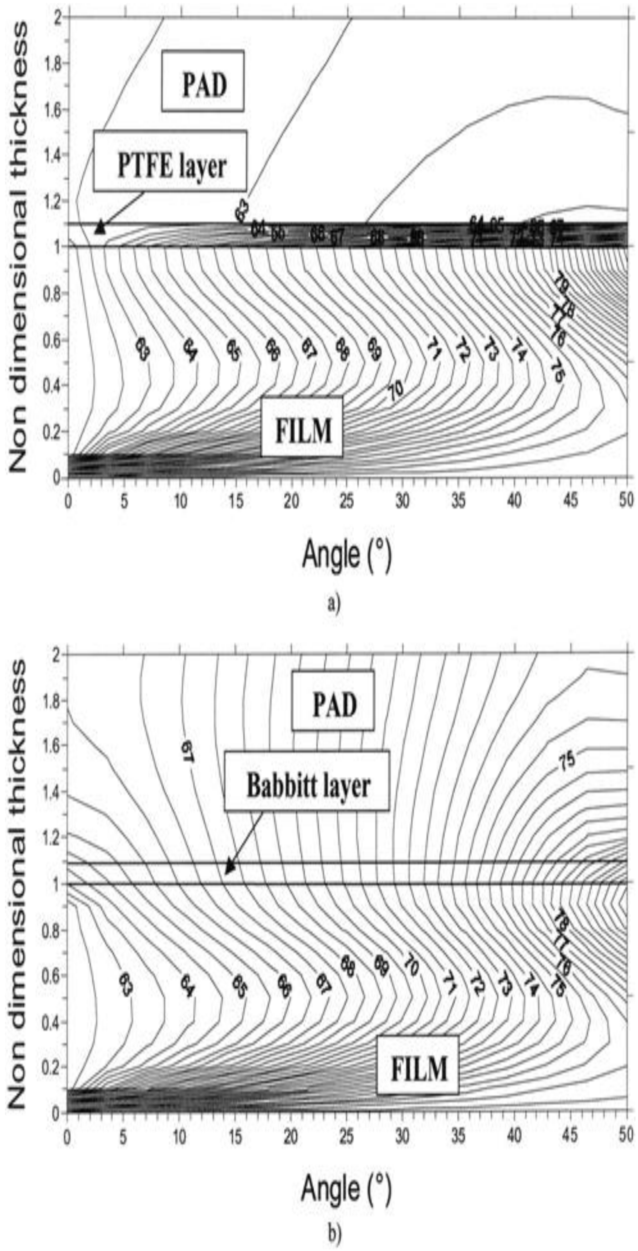
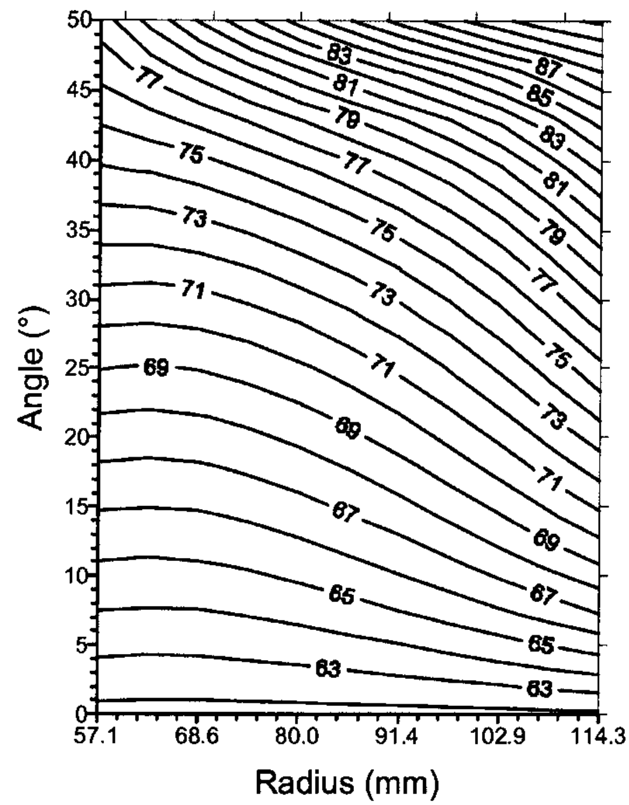


Fig. 4 Film-layer-pad isotherms at the mean radius for the PTFE-faced a and babbitted b bearings. Rotational speed— 3000 rpm, specific load—1.0 MPa



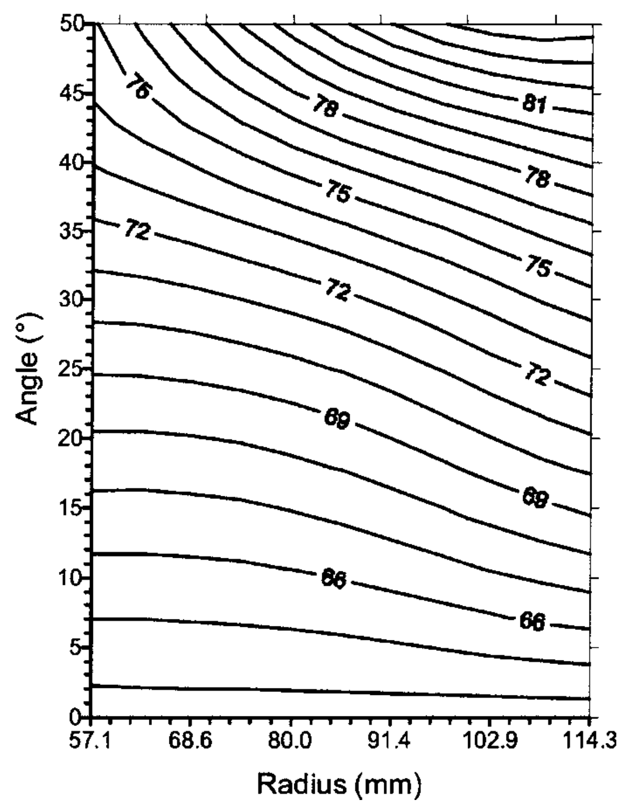


Fig. 5 Temperatures at the film-layer interface over the pad face for the PTFE-faced a and babbitted b bearings. TEHD, 3000 rpm, 1.0 MPa.

4.1 Temperature. Temperatures at the film–PTFE interface obtained by the THD and TEHD models are shown in Fig. 2 as a function of the angular position. It can be seen that both models

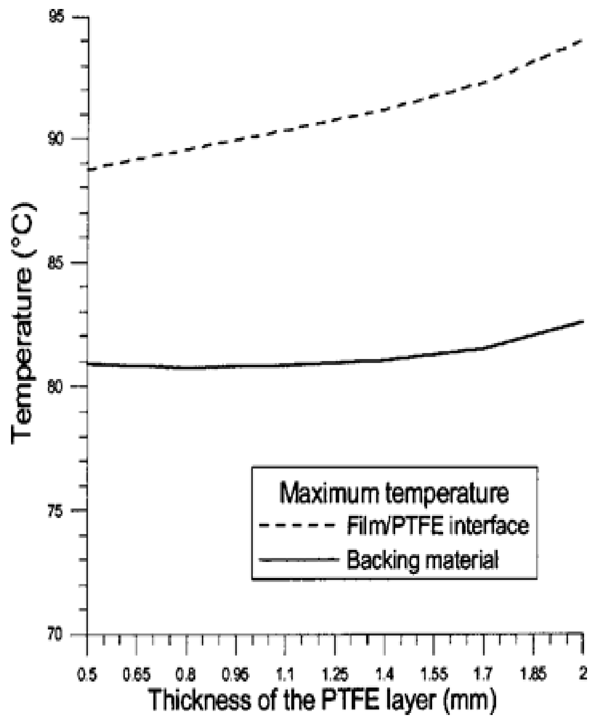


Fig. 6 Maximum temperature at the film–PTFE interface and in the pad backing 4 mm from the interface as a function of the PTFE layer thickness

give very similar temperatures. The interface temperature in- creases gradually within the first three quarters of the circumfer- ential pad length with a subsequent rapid rise close to the trailing edge. Two filled triangles represent temperature measured by thermocouples in the feed-through holes in pads 3 and 6 (see Fig. 1).

The agreement between predicted and measured values is good. This also confirms the efficiency of the ‘‘feed-through’’ monitor- ing technique as it gives actual interface temperature. A thermo- couple installed in the pad backing shows much lower tempera- ture compared with a thermocouple in the feed-through hole, unfilled triangles in Fig. 2. The temperature difference increases as the angle increases. Both theoretical models underestimate the

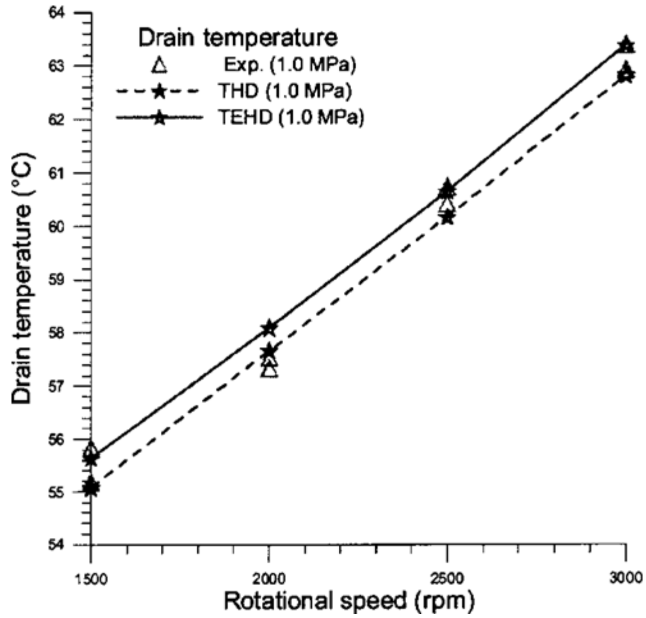


Fig. 7 Bearing drain temperatures

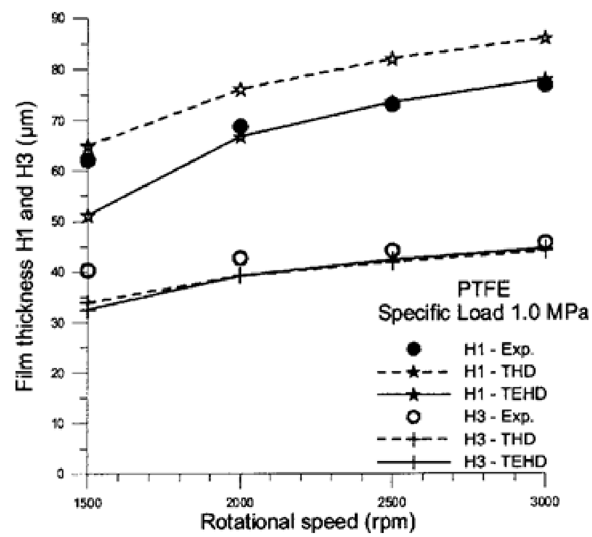


Fig. 8 Comparison of the predicted and measured oil film thicknesses

difference giving higher pad backing temperatures. At the same time, constancy of pad backing temperature with angular position is correctly predicted.

The interface temperature increases with speed as shown in Fig.

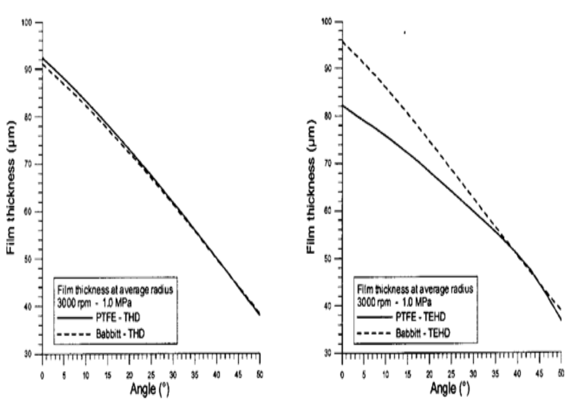
3. The theoretical results obtained with both models are similar except for the lowest speed, 1500 rpm, and 1.0 MPa specific load. Under these conditions, the hydrodynamic effect is low and the influence of deformations on the bearing behavior becomes pre- dominant.

Once again, the agreement between predicted and measured interface temperatures is good. Some variation in measured temperature values is due to machining tolerances and unequal pad loading (friction between leveling links in the equalising system hinders complete load equalisation).

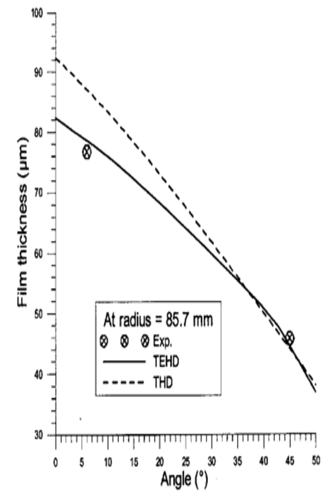
The effect of the PTFE layer on temperature fields in the oil film and pad backing is shown in Fig. 4. The isotherms are ob- tained with the TEHD model for the babbitted and PTFE-faced pads. A thin layer of babbitt does not prevent heat generated in the oil film from flowing into the pad. This results in a significant temperature variation in the pad backing. The variation is about 20°C and similar to the one in the oil film. On the other hand, a thin layer of PTFE serves as an effective thermal barrier to the heat flux from the oil film. As a result, temperature field in the backing of the PTFE-faced pad is uniform with a marginal varia- tion in temperature.

With the exception of the trailing edge, temperature fields in the oil film are similar for both bearings. In the vicinity of the trailing edge much higher temperature gradient can be observed near the oil–PTFE interface. This results in 3.5°C higher oil film tempera- ture compared to the babbitted pad.

Isotherms at the oil–babbitt and oil–PTFE interface are shown in Fig. 5. Comparing babbitted and PTFE pads, two important differences can be observed. First, temperatures in the inlet zone are lower at the PTFE surface than those obtained for the babbitt. Temperature near the leading edge is 2°C higher for the babbitted pad. Second, PTFE trailing edge temperatures are higher than those determined for the babbitt. This behavior is attributed to the fact that PTFE is a good thermal insulator. Heat flux from the oil into the pad is retarded leading to elevated oil film temperatures at the trailing edge. At the same time, the PTFE layer prevents oil heating in the inlet zone which results in lower inlet temperatures. For the case of 3000 rpm and 1.0 MPa, maximum oil film temperature for the PTFE bearing is 91.2°C while it is 85.5°C for



a) b)



c)

Fig. 9 Predicted and measured oil film thicknesses along the pad mean radius

Fig. 9 Predicted and measured oil film thicknesses along the pad mean radius

the babbitted one. This difference increases with an increase in the PTFE layer thickness. Figure 6 shows maximum oil film tempera- ture as a function of the layer thickness. It can be seen that chang- ing the thickness from 0.5 to 2.0 mm increases maximum oil film temperature from 89 to 94°C. The same Figure also depicts maxi- mum backing temperature that remains nearly constant for up to a 1.7 mm layer thickness.

A good indicator of bearing power loss is oil temperature rise from supply to discharge. In the test oil supply temperature was held constant and equal to 50°C. Measured and predicted drain temperatures are shown in Fig. 7. Once again, the agreement be- tween predicted and measured values is good.

4.2 Oil Film Thickness. Variation in film thickness, h1 and h3 (see Fig. 1 for the sensor location), with speed is shown in Fig.

8. On the whole, both models give an accurate prediction of h3, film thickness near the trailing edge. However, the agreement for the h1 values is poor if the THD model is used. The film thickness is apparently greatly affected by the deformation of the PTFE layer.

This influence is illustrated in Fig. 9 that shows THD and TEHD film thickness profiles for the PTFE-faced and babbitted

pads. As can be seen, Fig. 9(a), the THD model gives similar film thicknesses for both materials. But when the deformations are

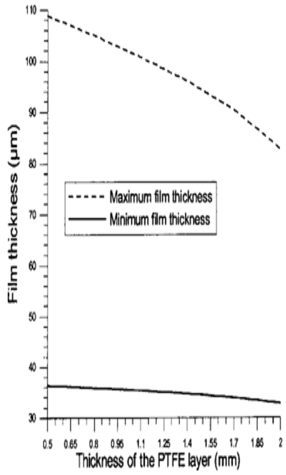


Fig. 10 The effect of the PTFE layer thickness on oil film thickness

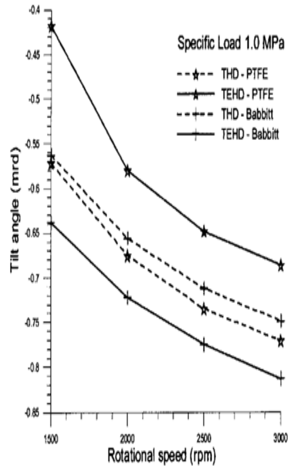


Fig. 12 Pad inclination predicted by the THD and TEHD models for the babitted and PTFE bearings

taken into account, Fig. 9(b), it can be noted that the deformation of the pads with babbitt leads to an increase in film thickness at the inlet zone while the opposite tendency is observed for the PTFE-faced pad. The difference is attributed to the compliant na- ture of the PTFE material. Deformation of the PTFE layer com- pensates crowning of the steel pad due to the pressure and temperature fields.

Comparison of measured and predicted values, Fig. 9(b), shows that THD film h1 differs significantly from the measured values.

At the same time, TEHD results show a very good agreement with the measured values of oil film thickness.

Inlet and outlet films will be affected if the thickness of the PTFE layer is changed. Figure 10 shows how maximum and mini- mum film thicknesses, predicted by the TEHD model, vary with an increase in the layer thickness. As can be seen both films are reduced. The reduction is much greater for the maximum film.

When the thickness of the layer is changed, geometry of the oil film is also affected as depicted in Fig. 11. As the layer thickness increases deformation of the PTFE layer leads to a concave sur- face with a constriction formed at the trailing edge. The inlet film thickness decreases faster than the outlet one so that collar and pad surfaces become more and more parallel. This is especially clearly visible for the thickest layer. A similar result was also reported by Ettles et al. [4]. Some differences in the shape of the film between our study and the one of Ref. [4] are due to the lower specific loads and much thinner layer of the PTFE. Never-

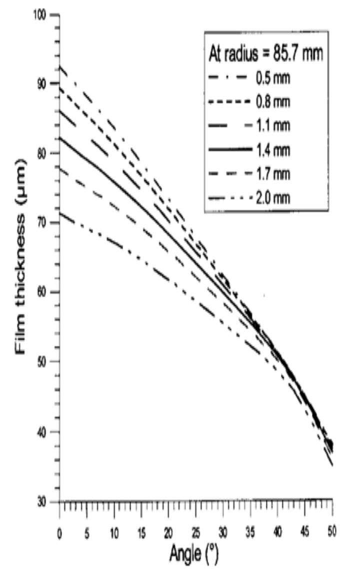


Fig. 11 The effect of the PTFE layer thickness on oil film thickness along the pad center line

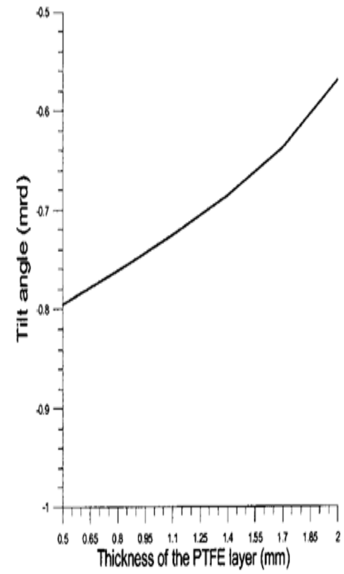


Fig. 13 The effect of the PTFE layer thickness on pad inclination

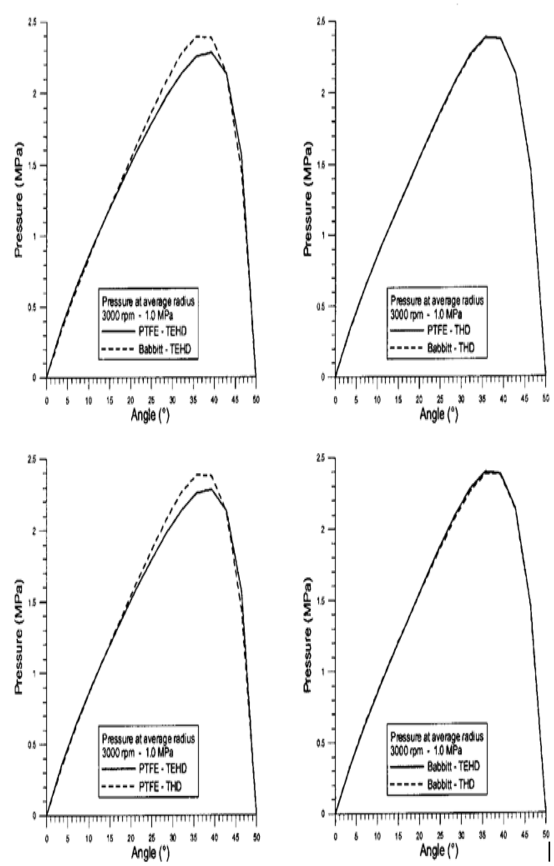


Fig. 14 THD and TEHD pressure profiles for the PTFE and babbitted bearings along mean radius

theless, formation of a ‘‘pocket’’ in the zone of maximum pressure has a favorable effect on pressure distribution resulting in lower peak pressure (see the next section).

Changes in film geometry inevitably affect pad tilt. This can already be seen in Fig. 10. A further proof is provided by Fig. 12 that shows the computed results of pad inclination for the babbit- ted and PTFE-faced pads as a function of speed. The THD model predicts lower pad tilt for the babbitted pad. If thermal and me- chanical deformations are taken into account, the situation is reverse: The tilt angle is always higher for the babbitted pad. For example, for the case of 3000 rpm speed (specific load 1.0 MPa) the tilt angle is 18% higher for the babbitted pad compared to the PTFE pad.

For the babbitted pad deformations increase the tilt angle be- cause of the crowning of the pad. The peculiar behavior of the PTFE-faced pad is due to the formation of the concave surface.

Concavity of the PTFE surface depends on the magnitude of the thermo-mechanical deformation. The latter increases with loading and layer thickness. Figure 13 shows how the tilt angle is affected as the PTFE layer thickness increases. It can be noted that increas- ing the layer thickness from 0.5 to 2.0 mm results in a 28% lower pad tilt.

4.3 Oil Film Pressure. Deformations of the layer have a positive effect on pressure distribution. Figure 14 shows THD and TEHD pressure profiles for the babbitted and PTFE-faced pads. The pressure profiles at the mean radius are similar when the deformations are not taken into account (THD model). For the babbitted bearing even the TEHD model gives almost the same pressure profile as the THD model. Greater compliance of the PTFE layer results in a different TEHD pressure profile compared to the one for the babbited pad. Lower peak pressure and a slight shift to the pad outlet zone can be observed. A thicker layer will result in a further decrease of the peak pressure as shown in Fig. 15.

A comparison of the TEHD and measured pressure profiles is presented in Fig. 16. Measured pressures are for a rotational speed of 2000 rpm. This speed is chosen as at higher speeds, 1.0 MPa specific bearing load, shape of p50 profile indicates oil starvation at the pad leading edge. In such conditions only a marginal pres- sure rise is observed in the inlet part of the pad with a subsequent sharp increase in pressure at a distance halfway to the pad center. As can be seen in Fig. 16 all theoretical profiles are similar in shape with the measured ones. There are, however, some discrep- ancies in the position and magnitude of the peak pressure for each profile. Peak pressures are underestimated by the TEHD model. This can be attributed to unequal pad loading caused by machin- ing tolerances and friction between the leveling plates in the equalising system. A shift in the position of the theoretical peak pressure in the direction to the trailing edge can be explained by a deviation of actual PTFE composite properties from those used in the TEHD model.

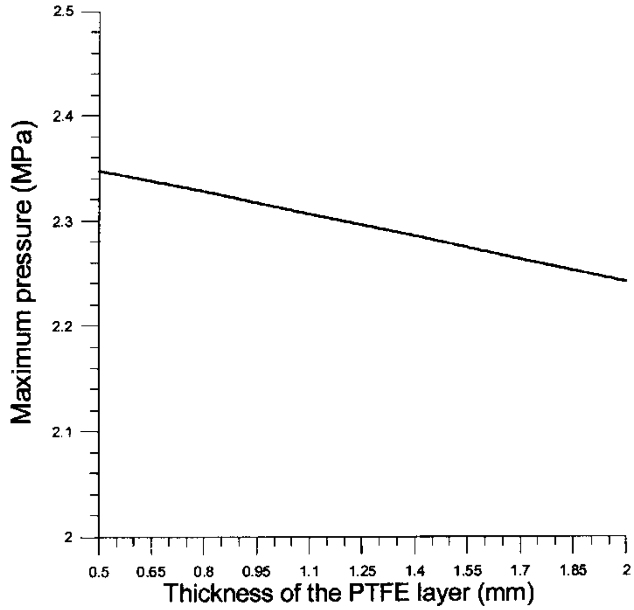


Fig. 15 The effect of the PTFE layer thickness on maximum oil film pressure

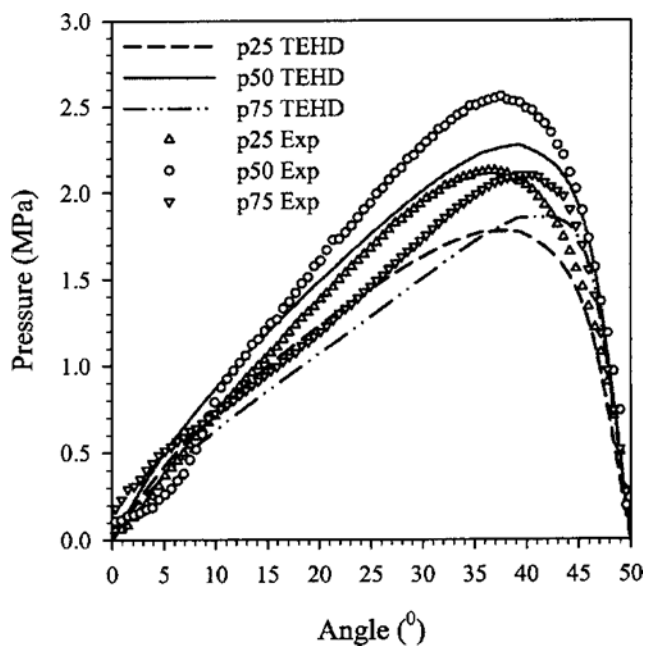


Fig. 16 Comparison of the predicted and measured pressure profiles. TEHD, 2000 rpm, 1.0 MPa.

V. CONCLUSIONS

The effect of the PTFE layer on bearing operation has been studied both numerically and experimentally. THD and

TEHD models were used to predict bearing characteristics such as tem- perature, oil film thickness and pressure. Predicted values were compared with the test data.

The conclusions are as follows.

1. Temperature.

Oil film temperature is significantly affected by the PTFE layer due to its powerful insulating properties. Inlet film temperature is reduced as there is no heat coming from the pad, while outlet film temperature is increased as the heat flow into the pad is reduced by the PTFE layer. The maximum temperature of the oil film increases as the PTFE layer becomes thicker. The agreement be- tween the TEHD data and measured oil–PTFE interface tempera- tures is good.

2. Oil film thickness and pad tilt.

Oil film thickness and geometry are strongly affected by the compliant nature of the PTFE layer. Film thicknesses at the inlet and outlet are reduced. A concave surface with a constriction near the trailing edge is formed. This tendency becomes more pro- nounce with an increase in thickness of the PTFE layer. Pad tilt is significantly reduced compared to the babbitt-faced pad. The agreement between measured and predicted, by the TEHD model, oil film thickness is good.

3. Oil film pressure.

Peak pressures are lower for the PTFE-faced pad compared to the babbitted one. Increasing the PTFE thickness leads to a gradual reduction in maximum oil film pressure. A comparison with measured pressure profiles shows that the TEHD model un- derestimates oil film pressures.

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I. NOMENCLATURE

c_p = specific heat of the lubricant (J/kg•K)

E_p , E_{PTFE} = Young modulus of pad backing, of PTFE layer (Pa)

h = film thickness (m)

h_p = film thickness at the level of the pivot (m)

K = thermal conductivity of the lubricant (W/m K)

p = pressure (Pa)

$r, 0, z$ = cylindrical coordinates (m,rd,m)

R_p = radial coordinate of the pivot (m)

$T(r,0,z)$ = film temperature (°C)

$T_p(r,0,z)$ = pad temperature (°C)

t_{PTFE} = thickness of the PTFE layer (m)

u, v, w = velocity components in the film (m/s)

$u_{PTFE}(r,0)$ = axial displacement of PTFE layer (m)

a = radial tilt angle of the pad (rd)

a_p , a_{PTFE} = coefficient of thermal expansion of pad backing, of PTFE layer (1/K)

b = circumferential tilt angle of the pad (rd)

$6_d(r,0)$ = total axial displacement of the pad (m)

AT_p = local temperature variation in the pad (K) s_{ij} = strain tensor

μ = dynamic viscosity of the lubricant (Pa s)

U_p , U_{PTFE} = Poisson coefficient of pad backing, of PTFE layer

p = density of the lubricant (kg/m^3)

o_{ij} = stress tensor (Pa)

m = angular speed of the runner (rd/s)

REFERENCES

[1] Soifer, A. M., Kodnir, D. S., and Baiborodov, Yu. I., 1965, ‘‘Elastic Sliding Bearing on a Base of Resilient Deformable Material Combined With Fluoro- plastic,’’ Izv. VUZov, Mashinostroenie, 7, pp. 67– 69.

[2] Baiborodov, Yu. I. et al., 1977, ‘‘Operating Experience With Elastic Metal- Plastic Pads in a Thrust Bearing Unit N 9 of Volga Hydropower Station Named After V. I. Lenin,’’ Gidrotekh. Stroit., 10, pp. 28 –31.

[3] Glavatskih, S. B., 2003, ‘‘Evaluating Thermal Performance of a PTFE-Faced Tilting Pad Thrust Bearing,’’ J. Tribol., 125, pp. 319–324.

[4] Ettles, C. M., Knox, R. T., Ferguson, J. H., and Horner, D., 2003, ‘‘Test Results for PTFE-Faced Thrust Pads, With Direct Comparison Against Babbitt-Faced Pads and Correlation With Analysis,’’ J. Tribol., 125, pp. 814 – 823.

[5] Fillon, M., Souchet, D., and Freˆne, J., 1990, ‘‘Influence of Bearing Element Displacements on Thermohydrodynamic Characteristics of Tilting-Pad Journal Bearings,’’ Proceedings of the 1990 International Tribology Conference, Nagoya, JAST, pp. 635– 640.

[6] Glavatskih, S. B., Fillon, M., and Larsson, R., 2002, ‘‘The Significance of Oil Thermal Properties on the Performance of a Tilting-Pad Thrust Bearing,’’ ASME J. Tribol., 124, pp. 377–385.

[7] Rades, M., 1972, ‘‘Dynamic Analysis of an Inertial Foundation Model,’’ Int. J. Solids Struct., 8, pp. 1353–1372.
[8] Pascovici, M. D., and Suciu, C. V., 1992, ‘‘Analytical Investigation of a Narrow Compliant Journal Bearing,’’ Acta Tribologyca, 1, pp. 89–93.
[9] Lin, C. R., and Rylander, H. G., 1991, ‘‘Performance Characteristics of Com- pliant Journal Bearings,’’ ASME J. Tribol., 113, pp. 639– 644.
[10] Boncompain, R., Fillon, M., and Freˆne, J., 1986, ‘‘Analysis of Thermal Effects in Hydrodynamic Bearings,’’ ASME J. Tribol., 108, pp. 219–224.
[11] Glavatskih, S. B., 2002, ‘‘Laboratory Research Facility for Testing Hydrody- namic Thrust Bearings,’’ Proc. Inst. Mech. Eng., Part J: J. Eng. Tribol., 216,pp. 105–116.
[12] Glavatskih, S. B., 2004, ‘‘A Method of Temperature Monitoring in Fluid Film Bearings,’’ Tribol. Int., 37, pp. 134 –148.