Investigations on Large Turbine Bearings Working Under Transitional Conditions Between Laminar and Turbulent Flow

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The report deals with investigations on large turbine bearings, a tilting pad bearing, and an elliptical bearing. In special cases, the maximum babbitt temperatures show a sudden leap, caused by a slight change of the operating conditions. According to the test results, the temperatures do not change within the whole bearing, but only in a region near the minimum gap width. As proved by comparative theoretical investigations, the reason for this temperature leap is a local transition between laminar and turbulent flow, changing the thermal conductivity within the oil film to a high degree. Thus, the consideration of thermal diffusion in the energy equation is essential for precalculating this phenomenon. As a consequence of locally high surface temperatures, considerably large thermoelastic deformations are observed at both test bearings. For this reason the comparative calculations are based upon measured gap width profiles.

1 Introduction

The design of journal bearings for high-speed turbomachines requires the application of bearing calculation programs. The computer programs that are available for industrial purposes can be applied to calculate many different types of journal bearings, but they are often restricted to certain simplifications. The most important simplifications are that constant viscosity is supposed over the complete film thickness and that thermoelastic bearing deformations are neglected. The reliability of results that can be attained this way depends on experience in pre-estimating the operational clearances and the gap inlet temperatures. But even if these influences are taken into account, such programs still produce results that are too inaccurate under certain conditions, especially for large turbine bearings. Therefore, a particular interest exists in further experimental investigations to find the physical reasons for these inaccuracies and, based on the results obtained, to improve the calculation programs. Results of these investigations on big turbine bearings that are given in detail in [1] and [2] are summarized here.

2 Test Rig for Turbine Bearings

The tests are carried out at the test rig shown in Fig. 1, which is suitable for examining large turbine bearings with a nominal diameter of 500 mm and a maximum bearing length of 520 mm under practice-orientated operating conditions. The maximum driving power of 1200 kW allows a maximum shaft speed of about 4000 1/min depending on the friction in the test bearing. The lubricant is turbine oil ISO VG 32. Figure 2 shows the technical drawing of the test rig. The test bearing (1) which is attached within the rigid frame (2) is stationarily loaded with a maximum force of 1 MN by a pipe compensator used as a pneumatic bellows (3). The force acting on the shaft (4) is transferred to two symmetrically aligned support bearings (5). These as well as the pneumatic bellows are supported by the test rig body (6), which is also very rigid. For measurements of oil film pressures, oil film thicknesses, and shaft temperatures, the shaft contains two piezoelectric pressure measuring systems, two capacitive distance measuring systems and several thermocouples. A special device allows the rotating shaft to be axially shifted. Thus, pressures, film thicknesses and shaft temperatures can be measured not only in the circumferential direction but also along the bearing length (due to the use of a capacitive measuring system, however, the film thicknesses can only be measured in those areas where the gap is completely filled with oil). Up to now, an elliptical bearing and a tilting pad bearing have been examined; the data of these test bearings are given in Fig. 3. In order to measure the bearing temperatures, both have been equipped with numerous thermocouples on their surfaces and on the back of the bearing or the pads, respectively.

In addition to the stationary force, the test bearing can be loaded by sinusoidal forces. For this purpose two unbalance-vibration generators (7 in Fig. 2) are attached to the frame of the test bearing. The continuously changing oil film forces and the relative movements between shaft and bearing are both measured during the test procedure in order to find out the spring- and damping coefficients. The results of these investigations which have been summarized in [3] shall not be discussed here.

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3 Results

3.1 Bearing Temperatures Under Transitional Conditions Between Laminar and Turbulent Flow. Earlier bearing tests carried out by some manufacturers of large turbines (e.g., by the firms Asea Brown Boveri and Siemens) and by Gardner and Ulschmid [4] as well as theoretical investigations by Suganami and Szeri [5] have already shown a queer phenomenon which we call the "temperature leap": In the case of constant load and increasing shaft speed, this phenomenon corresponds to the fact that the maximum bearing temperature is rising at first, as was to be expected, but afterwards suddenly drops by about 10 to 15 K at a certain rotational speed depending on the bearing load. Figure 4 shows bearing surface temperatures measured in the center plane of the tilting pad bearing under a constant specific load of $\bar{p}$ = 2,0 MPa for different rotational speeds. Figure 5 shows the corresponding results for the elliptical bearing. This demonstrates that bearing surface temperatures are not reduced within the whole bearing when this phenomenon occurs, but that this change in the temperature profile is kept confined to the section near to the minimum gap width. As demonstrated in Fig. 6, the phenomenon can also be observed if shaft speed is kept constant and load is varied (for the test bearings with a nominal diameter of 500 mm this occurs only above 2000 1/min). When the load is increased, the bearing surface temperatures practically do not change up to a certain load limit (depending on shaft speed). Above that, the surface temperatures jump upwards—at first only in the direct surrounding of the minimum gap width—and have a marked maximum which grows approximately in proportion to loading with further load increase.

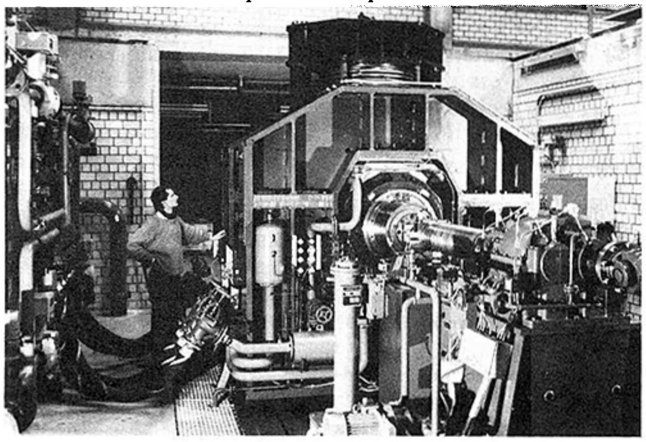


Fig. 1 Test rig for large turbine bearings

The sudden local change of bearing surface temperatures in this region is caused by a flow transition, which may happen either directly between laminar and turbulent flow or passing through an intermediate range with laminar Taylor Vortices.

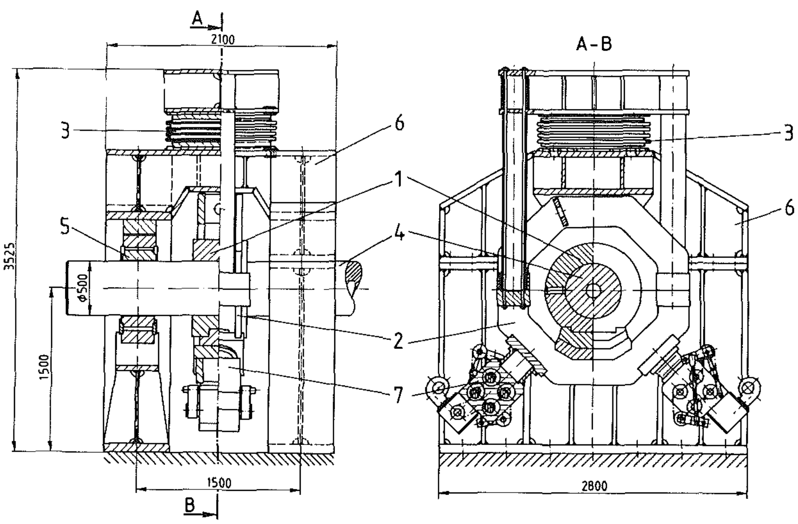
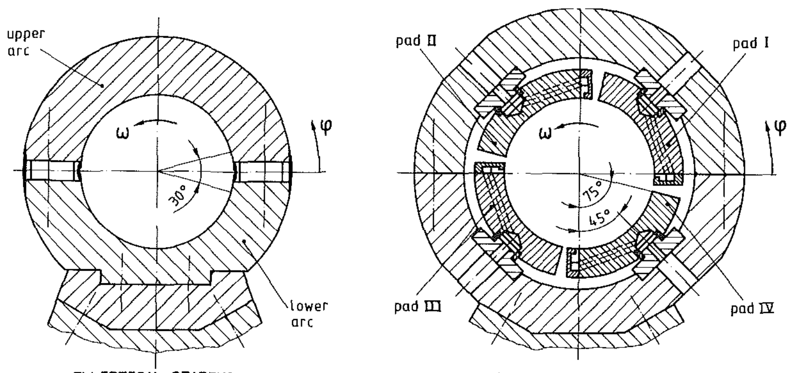


Fig. 2 Technical drawing of the test rig



ELLIPTICAL BEARING

nominal diameter	d = 500 mm
bearing axial length	b = 500 mm
bearing outer diameter	D = 880 mm
dimensionless partial arc clearance	ψ_S = 3 ‰
dimensionless minimum clearance	ψ_B = 1,2 ‰

TILTING PAD BEARING

nominal diameter	d = 500 mm
bearing axial length	b = 450 mm
pad outer diameter	D = 695 mm
dimensionless pad clearance	ψ_S = 1,6 ‰
dimensionless pivot circle clearance	ψ_B = 1,35 ‰

Fig. 3 Test bearings

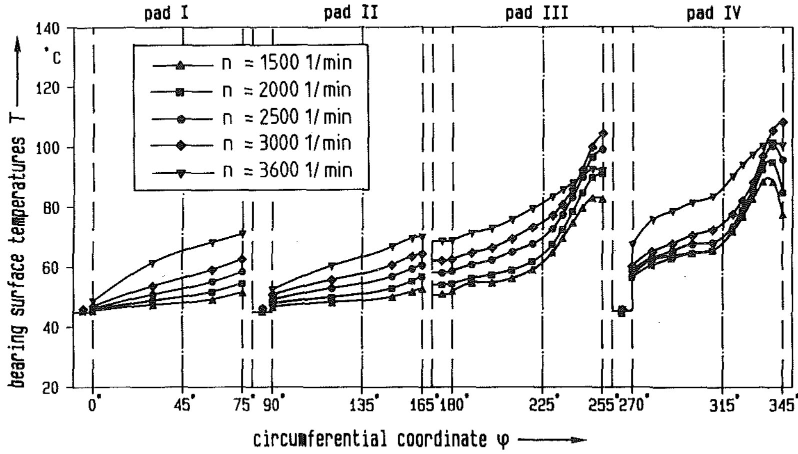


Fig. 4 Surface temperatures in the center plane (z = 0 mm) of the tilting pad bearing for the specific load 2.0 MPa and different rotational speeds

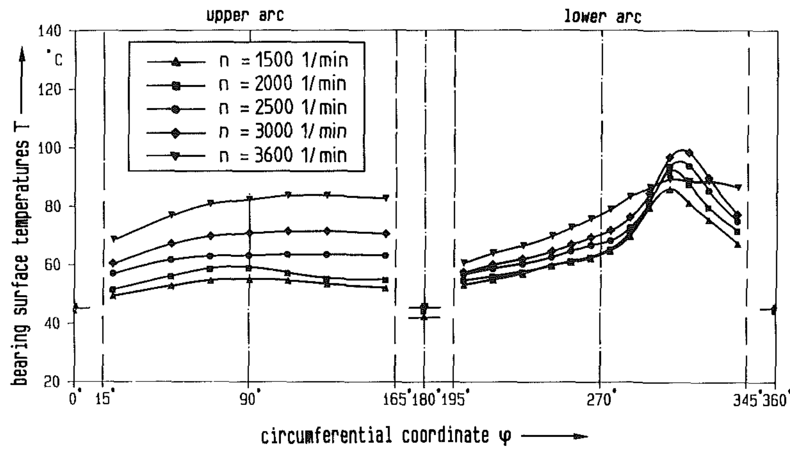


Fig. 5 Surface temperatures in the center plane (z = 0 mm) of the elliptical bearing for the specific load 2.0 MPa and different rotational speeds

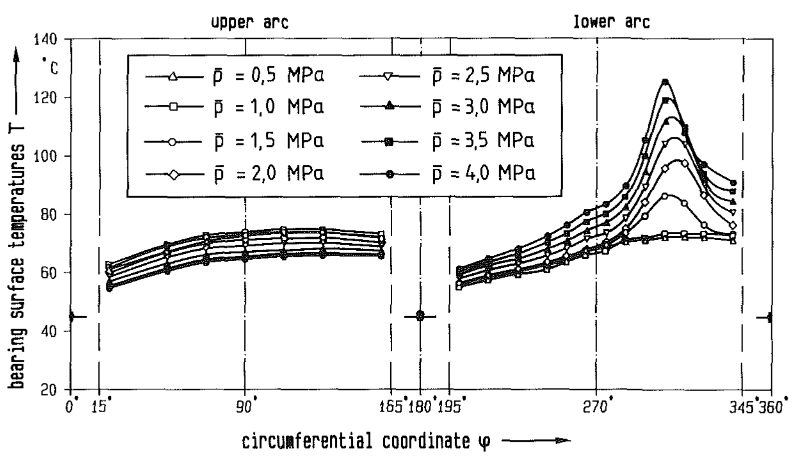


Fig. 6 Surface temperatures in the center plane (z = 0 mm) of the elliptical bearing for the rotational speed 3000 1/min and different specific loads

As the physical effects resulting from Taylor Vortices are similar to those resulting from turbulence, the term "turbulence" in the following explanation is used for both kinds of instability. The increase of temperatures occurs at a transition from turbulent flow—which can still be dominant at the oil inlet and at the beginning of the gap—into laminar flow in the vicinity of the minimum film thickness. In the reverse case, going from laminar to turbulent, an increase of the rotational speed under constant load leads to turbulence even at the minimum gap width and, thus, to the observed temperature drop.

At first sight, the explanation that a transition from turbulent to laminar flow causes the sudden temperature increase, seems to be contradictory, because friction and, consequently, the dissipated energy are certainly lower in laminar than in turbulent flow. But one should pay attention to the fact that the measured temperatures are surface temperatures. Figure 7 shows that even at maximum temperatures of more than 120 °C the shaft temperature did not exceed 70°C. Accordingly, in spite of the minimum film thickness of only about 100 μm, the temperature difference within the oil film may amount to 50 K. That means, the temperature gradient can reach the magnitude of 500 K/mm. Temperature gradients of this size are—if extremely high heat fluxes are excluded—practically possible only in a medium with a low heat conductivity. This basically applies to oil, but only in the case of laminar flow. Turbulence, however, increases the thermal conductivity of fluids to a high degree. For oil, more precisely for fluids with high Prandtl Numbers $\mathrm{Pr} = \mu \cdot c_p/\lambda$, this increase is much greater than the increase of friction. Approximately it can be stated that turbulence causes a Pr-fold higher increase of thermal conductivity than of friction. Accordingly in such cases, when a high temperature gradient exists between shaft and bearing in laminar flow close to the transition, the transition to turbulence can lead to a lower temperature gradient and, consequently, to lower bearing surface temperatures.

With the intention of finding out which kind of flow instability really exists at the transition to purely laminar flow, local Reynolds and Taylor Numbers were determined from the test results (cf., Frêne and Godet [6]). Following Constantinescu, Pan, and Hsing [7], the influence of a superimposed pressure flow on the occurrence of Taylor Vortices has been taken into account by calculating local Taylor numbers with the mean local velocity:

$$\mathrm{Ta}_{Um} = \frac{U \cdot h}{\nu} \cdot \sqrt{\frac{h}{R}} \cdot \left[1 + 2 \cdot \frac{U_{mp}}{U}\right]$$

with

$$U_{mp} = -\frac{h^2}{12 \cdot \mu \cdot R} \cdot \frac{dp}{d\varphi}$$

According to this, Taylor Vortices are to be expected for Taylor numbers greater than 41. This assumption takes into account that the occurrence of Taylor Vortices can be suppressed by a pressure rise in the direction of shaft rotation and that it can be released by a decrease of pressure (cf., Brewster et al. [8] and DiPrima [9]).

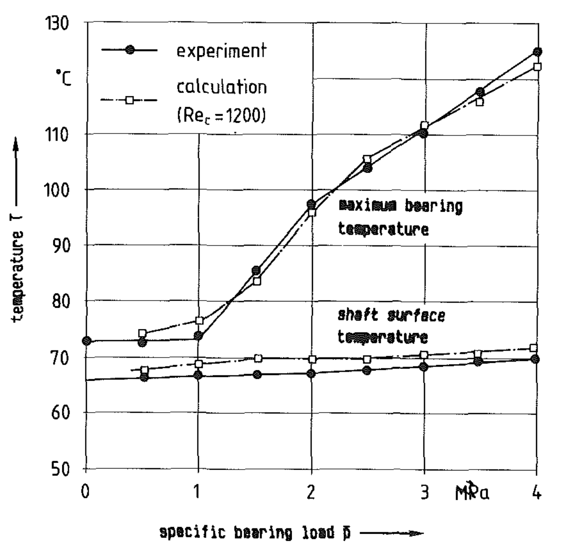


Fig. 7 Increase of the maximum bearing temperature and the shaft surface temperature with the specific load (n = 3000 1/min)

On the other hand, proceeding on the proof of Elrod and Ng [10] that the influence of a pressure gradient on turbulence is independent on its direction, calculating local Reynolds Numbers analogously to the Taylor numbers with a mean local velocity seems to make little sense. As no reliable findings about the influence of the pressure gradient on the onset of turbulence are available, the local Reynolds numbers have simply been calculated with the circumferential speed. Therefore the criterion for the transition between laminar and turbulent flow reads as follows:

$$\frac{U \cdot h}{\nu} > \mathrm{Re}_c$$

For the critical Reynolds Number Re_c quite different values between 500 and 2000 are given in literature. Taylor and Dowson [11] quote a typical critical value to be $\mathrm{Re}_c = 1000$.

Evaluating the present test results, the local viscosities ν or μ, respectively, that are included in both the Reynolds and Taylor Numbers were calculated with the arithmetical mean between surface and shaft temperatures. In some cases, at the point of the temperature leap both numbers fell below their critical values, so that it was impossible to determine the kind of flow before the transition to laminar flow. In other cases, however, especially those at a rotational speed of 3600 1/min, the temperature leap occurred at an almost constant local Reynolds number of about Re = 1200—which is a plausible value for the transition between laminar and turbulent flow—whereas the local Taylor numbers Ta_{Um} were considerably smaller than 41.

According to these results, the reason for the sudden rise of temperature is definitely a transition to pure laminar flow. In addition to that, a direct transition between turbulent and laminar flow without an intermediate range of laminar Taylor Vortices seems to be certain in some cases. Further experiments are intended to find more detailed criteria for the transition.

3.2 Bearing Deformations and Journal Displacements. Figures 8 and 9 show measured gap widths and oil film pressures for both test bearings. By analyzing the measured gap width distributions it is possible to determine the journal displacements and the radial bearing deformations in relation to any point on the bearing surface. For both bearings

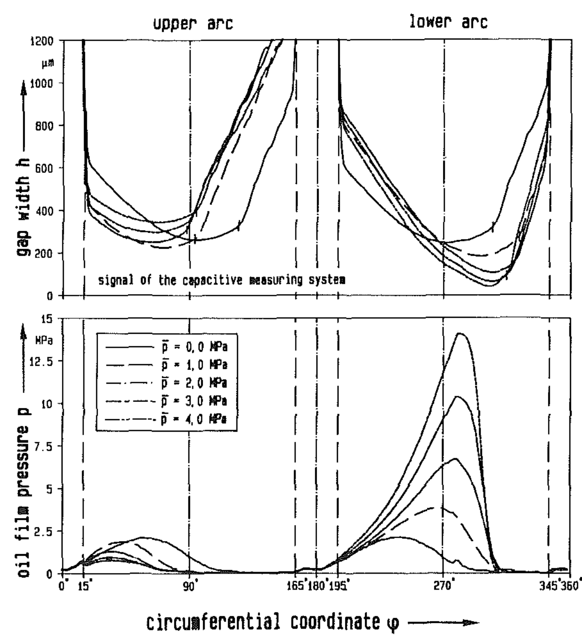


Fig. 8 Oil film pressure and gap width profiles measured in the center plane of the elliptical bearing for different specific loads (n = 3000 1/min)

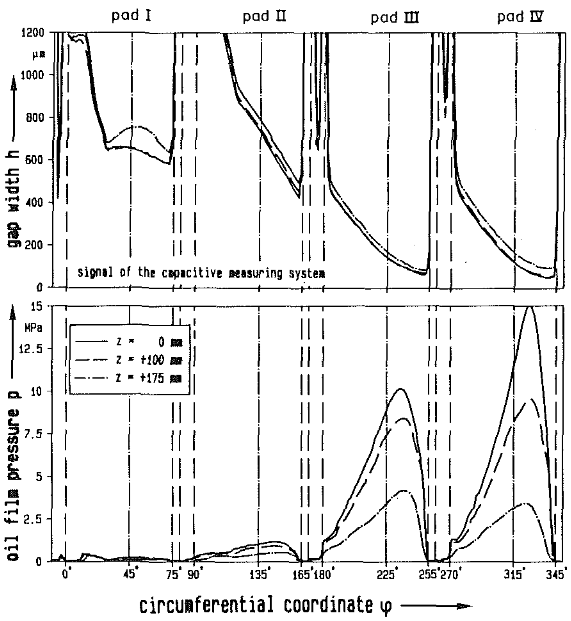


Fig. 9 Oil film pressure and gap width profiles measured at different axial positions of the tilting pad bearing (n = 3000 1/min, $\bar{p}$ = 3 MPa)

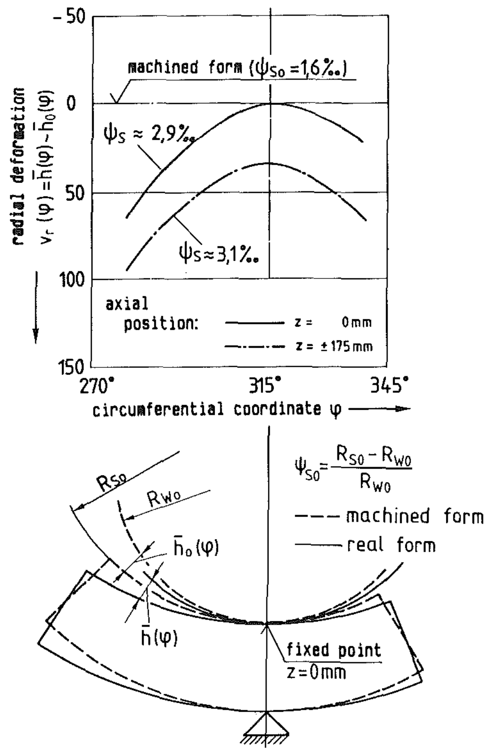


Fig. 10 Measured radial deformation $v_r(\varphi)$ of tilting pad IV at the specific load 3 MPa and the rotational speed 3000 1/min

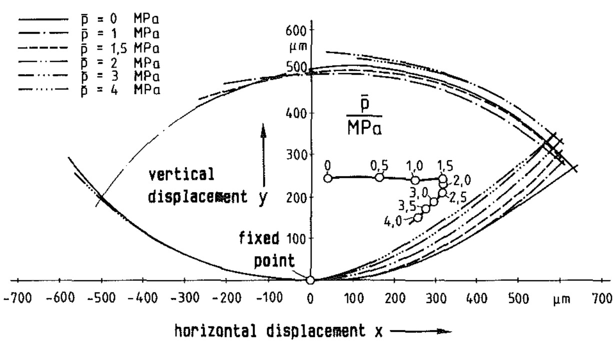


Fig. 11 Measured clearances of the elliptical bearing and journal displacements for different specific loads and a rotational speed of 3000 1/min

a remarkable change in the shape of the lubricating gap profile has been measured simultaneously with the sudden appearance of high bearing surface temperatures. For the tilting pad bearing the gap width evaluation yielded distinct pad deflections in both circumferential and axial direction, which are shown in Fig. 10 in relation to the surface at the pivot position. In this figure the machined shape (ψ_s = 1,6 ‰) has already been eliminated, thus $v_r(\varphi) = 0$ would be the result for a nondeformed pad.

A remarkable result of the gap width evaluation for the elliptical bearing is an inward local expansion of the surface, which increases approximately in proportion to the maximum surface temperature. In relation to the lowest point of the bearing surface Fig. 11 shows the dependence of the clearance profile upon load. Obviously there is a strong influence of the surface expansions on the shaft displacements.

3.3 Comparative Calculations. The computer program used for the comparative calculations allows the assumption of either turbulent or laminar flow within the gap, depending on the magnitude of the local Reynolds number. The change of flow type is carried out automatically in the course of the iteration process whenever the local Reynolds number values fall below or exceed the critical value. The Reynolds equation for the two-dimensional pressure distribution and the energy equation for the three-dimensional lubricant temperature distribution are solved simultaneously by means of a finite difference method. Heat conduction into the bearing is included by the simultaneous solution of the Laplace equation. For the shaft an applicable boundary condition is that the shaft surface temperature does not vary in the circumferential direction. The value of this temperature is not invariable, but—dependent on thermal conduction in the whole gap area and in the oil grooves—is calculated in such a way that, in the sum, a given heat flux between shaft and oil is achieved. The gap inlet temperatures are calculated by assuming a mixing model between hot oil carried over and fresh supply oil within the oil grooves.

In the case of turbulent flow calculation, a viscous sublayer with a negligible influence of turbulence on the friction is assumed at each of the two surfaces. The thicknesses of these sublayers, which are varying in circumferential as well as in axial direction, are calculated by means of the formula $\delta_1 = K \cdot \nu/\sqrt{\tau_0/\rho}$, taken from [12], where K is an empirical constant, ρ the fluid density, τ_0 the surface shear stress and ν the kinematic viscosity at surface temperature. In the inner part of the gap (i.e., between the viscous sublayers) an eddy viscosity is computed according to Prandtl's Mixing Length Approach. The variation of the mixing length with the gap width coordinate is supposed as shown in Fig. 12. In the inner part it is similar to the parabolic approach proposed by Constantinescu [13]. Here, the constant κ indicates the gradients of the mixing length at the boundaries to the viscous sublayers.

For calculating oil temperatures, the thermal conductivity within the viscous sublayers is supposed to be constant in radial direction and to be increased by a constant factor in relation to its molecular value. The factor $Pr^{0,3}$, which has been taken from well-known formulas for the heat transfer in pipes, has proved to be applicable. Beyond the viscous sublayers the thermal conductivity of fluids with high Prandtl numbers increases to such an extent that a locally constant temperature between the two boundaries can be assumed. For calculating this inner temperature the energy equation is integrated within these boundaries. In those areas of the gap, where the local Reynolds numbers fall below a given limit, purely laminar flow is supposed and, therefore, the energy equation is solved three-dimensionally.

The following passages describe the results of comparative calculations for the elliptical bearing with a rotational speed of $n = 3000$ 1/min and different loads. In default of exact criteria for the flow transition and with reference to the test results laminar flow was assumed for local Reynolds numbers lower than Re = 1200. As, up to now, the program does not allow the computation of thermo-elastic deformations the comparative calculations are based upon the measured gap widths shown in Fig. 8 (in cavitation regions these were determined by extrapolation). Additionally, the measured feeding rates were taken as fixed data. The shaft temperature was calculated with the assumption that heat, in the sum, is neither added nor subtracted through the shaft. The results can be influenced by changing the empirical constants κ and K in the applied turbulence model. The constants $\kappa = 0{,}4$ and $K = 7$ yielded the optimum accordance of the calculated lubricating film pressures and bearing surface temperatures with the measured values.

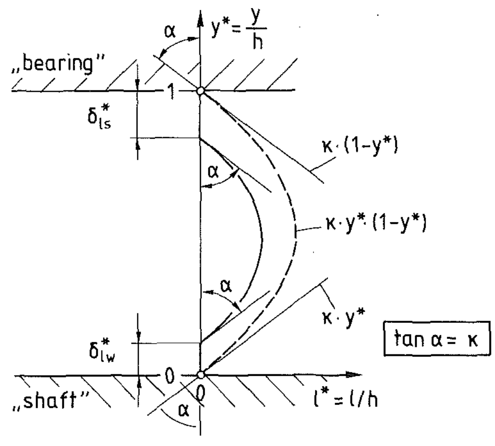


Fig. 12 Assumed variation of the mixing length with the gap width coordinate

Figure 13 shows calculated bearing surface temperatures for these constants in circumferential direction which are directly comparable with the measured temperatures in Fig. 6. Here the specific load calculated for the measured gap widths serves as a curve parameter. Due to the use of appropriate constants, this, in most cases, corresponds well with the real load. Merely in the range where the transition to laminar flow occurs first there is a slightly increased deviation ($\bar{p}_{cal} = 1.66$ MPa compared with $\bar{p} = 1.5$ MPa). A comparison of the measured and calculated maximum bearing temperatures and of the shaft temperature as a function of the experimental load is given in Fig. 7.

For the specific load $\bar{p} = 3.0$ MPa, Fig. 14 shows the gap width profile taken as the basis for the calculation, the viscous

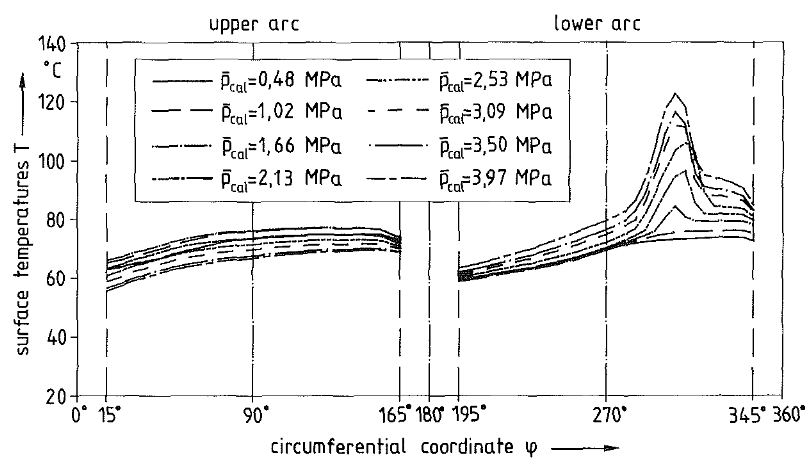


Fig. 13 Calculated surface temperatures in the center plane of the elliptical bearing (comparable with Fig. 6)

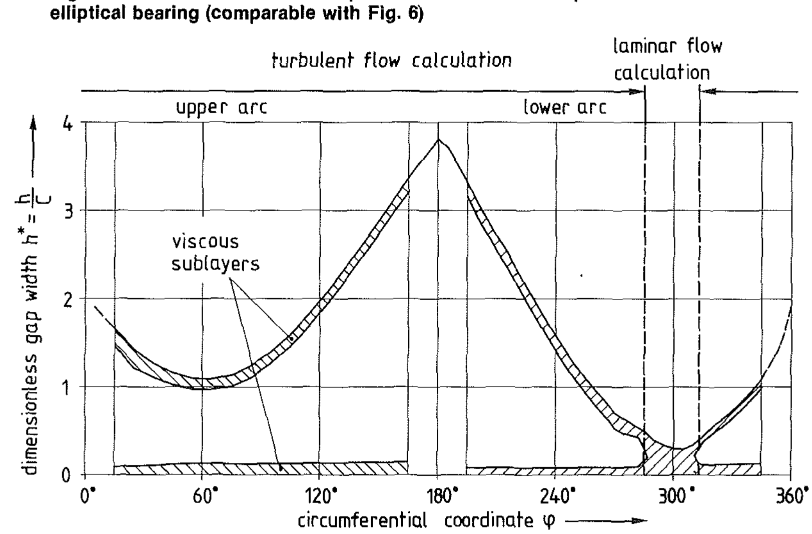


Fig. 14 Gap width profile of the elliptical bearing with laminar and turbulent flow regions (n = 3000 1/min, $\bar{p}$ = 3 MPa)

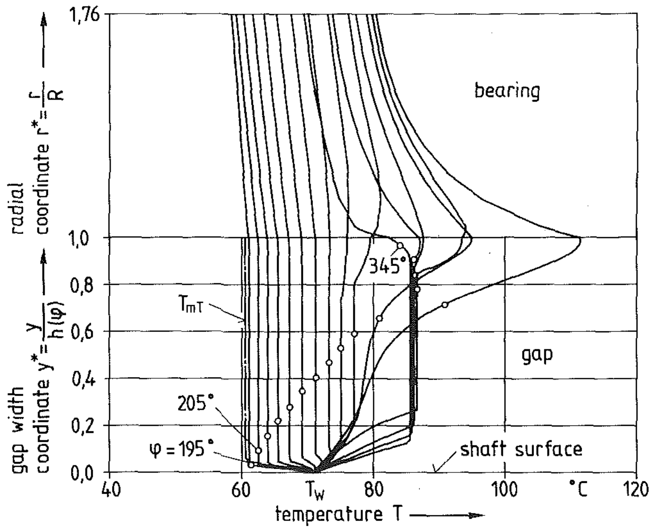


Fig. 15 Calculated shaft, oil film and bearing temperatures in the center plane of the elliptical bearing (n = 3000 1/min, $\bar{p}$ = 3 MPa)

sublayers which are determined during the calculation process and the area where laminar flow was assumed. Figure 15 shows, at several grid lines, oil film and bush temperatures which have been calculated in the center plane of the lower arc. In radial direction two quite different scales have been applied for the combined presentation of gap and bush. The curve parameter is the circumferential coordinate φ. Curves are given in distances of $\Delta\varphi = 10$ deg. Each individual curve parameter is marked by a symbol, the altitude of which corresponds to the circumferential position. The curve with the symbol at about $y^* = 0.1$ reproduces the temperature profile $T(y^*)$ at the leading edge ($\varphi = 195$ deg), the curve with the symbol at about $y^* = 0.9$ shows this profile at the trailing edge ($\varphi = 345$ deg) of the gap.

The first ten curves ($\varphi = 195$ up to 285 deg) are located in an area of turbulent flow where the energy equation, including the term for thermal diffusion, is solved three-dimensionally only within the viscous sublayers, while a constant temperature is assumed in the inner part of the gap. Therefore, taking into account the thermal diffusion in y^*-direction leads to an angle in the temperature profile $T(y^*)$ at the boundaries between these parts. The curves at $\varphi = 295$ and $\varphi = 305$ deg represent the area of laminar flow. Here, the profiles show the enormous temperature differences within the oil film which is in accordance to the experimental results. The four last profiles ($\varphi = 315$ up to 345 deg) are calculated for the cavitation region where, according to the local Reynolds numbers, turbulent flow was supposed to occur again.

Conclusion

Distinctly high surface temperatures with a marked maximum in circumferential direction and large temperature differences within the oil film which can be observed at bearings for high-speed turbomachines under certain operating conditions are due to laminar flow in the region of minimum gap widths. A transition to turbulence or Taylor Vortices, which can be achieved by an increase of the rotational speed as well as by a reduction of load or viscosity, causes an improved heat exchange across the gap width and, accordingly, a reduction of the surface temperatures in such cases. This effect can only be precalculated if the energy equation is solved under consideration of thermal diffusion, i.e., if a varying temperature is supposed to appear across the gap width. Naturally it is not calculable by means of the often-used two-dimensional programs, in which a mean local temperature across the gap width is assumed varying only in circumferential and axial direction.

The test results indicate a number of negative consequences being connected with the occurrence of high bearing surface temperatures: The tilting pads bend up extremely, especially in axial direction and, thus, take a more unfavorable shape in regard to their load carrying capacity. At the elliptical bearing a bulge-shaped inward expansion of the bearing surface was observed, owing to the strong obstruction of thermal deformations in other directions. This presumably leads to a hard strain on the connection between the babbitt metal and the bearing body.

Meanwhile a further verification for the fact that a transition between laminar flow on the one hand and turbulent or vortex flow on the other hand is responsible for the observed temperature leap has been obtained experimentally: The babbit surfaces of the tilting pad bearing were provided with numerous whirl-grooves [14] in those regions of the gap, where the flow became laminar with smooth surfaces. With appropriate shapes and dimensions of the grooves, disturbances can be created artificially which—like vortex flow or turbulence—cause an improved heat exchange across the oil film. As shown in [2] these whirl grooves, in special cases, produce a considerable reduction of the maximum surface temperatures for constant carrying capacity.

Acknowledgments

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