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Analytical Investigation of Low-speed, Large Overall-dimension Journal Bearings Operating in Textile Machines

Abstract
The subject of this paper is the theoretical investigation of low-speed, mechanically and thermally heavy loaded journal bearings of calendaring and similar type machines operating under identical conditions. A heated press cylinder had hollowed journals of large diameters with a heating system inside. The temperature of the heated cylinders even reaches 300 °C and the temperature of the journal reaches about 250 °C. These large overall dimensions, journal bearings, heavy loaded mechanically and thermally, require a very precise analytical investigation. Application of the developed program of calculations of heavy loaded, pressurized or not journal bearings should allow the durable and reliable design of bearings to be obtained.

Key words: *hydrodynamic lubrication, journal bearings, heated bowl press.*

List of symbols

c_t - *specific heat of oil, J/kgK*
D - *bearing diameter, m*
e - *eccentricity, m*
g - *acceleration of gravity, m/s²*
F_{stat} - *static load of bearing, N*
$\overline{H}$(□) - *dimensionless oil film thickness,* $H = h/(R - r)$
$\overline{H}_{min}$ - *minimum dimensionless oil film thickness at arbitrary position of journal in bearing*
K_T - *thermal coefficient,* $K_T = \omega \cdot \eta_0/(c_t \cdot \rho \cdot g \cdot T_0 \cdot \psi^2)$
L - *bearing length, m*
O_b - *centre of bearing*
O_j - *centre of journal*
O_L - *centre of lobe*
$\overline{p}$ - *dimensionless oil film pressure,* $\overline{p} = p\psi^2/(\eta\omega)$
p - *oil film pressure, MPa*
Pe - *Peclet's number* $\rho c \omega r^2/h$
r, R - *journal and bearing radius, m*
So - *Sommerfeld number,* $So = F_{stat} \cdot \psi^2/(L \cdot D \cdot \eta \cdot \omega)$
t - *time, s*
T - *temperature of oil film, °C*
T_0 - *temperature of supplied oil, °C*
$\overline{T}$ - *dimensionless oil film temperature,* $\overline{T} = T/T_0$
$\overline{z}$ - *dimensionless axial coordinate,*
α - *attitude angle, °*
β - *load angle, °*
γ_i - *angle of lobe centre point, °*
ε - *relative eccentricity,*
φ - *dimensionless time,* $\varphi = \omega t$
η - *dynamic viscosity of oil, Ns/m²*
$\overline{\eta}$ - *dimensionless oil viscosity,* $\overline{\eta} = \eta / \eta_0$
η_0 - *dynamic viscosity of supplied oil, Ns/m²*
□ - *peripheral co-ordinate, °*
ρ - *oil density, kg/m³*
ω - *angular velocity, s⁻¹*
ψ - *bearing relative clearance, ‰*
ψ_{si} - *lobe relative clearance,*
□, □□, *z, zz, t - first and second partial derivative with regard to axial and peripheral coordinates and time*

Introduction

The calendering of textiles (or pressing the moire as well as the flatting) on calendaring machines, consists of the passing of textile between pressed rotary cylinders of which one (for each pair) is heated. These cylinders are heated by means of steam, oil or electric current. As result of heat conduction, the temperature of cylinder journals even reaches the value of 250 °C. The linear pressures between the cylinders can reach the value up to 3 kN/cm, e.g. in a cylinder at an operating length of 2000 mm the force acting on one bearing is about 300 kN.

The operation of the supporting bearings, also in the conditions of mixed friction, causes an increase of temperature in the oil film and in the design elements of the bearing node. However, the conditions of mixed friction operation can be avoided by application of pressurised journal bearings [1 - 5].

The journal bearings of squeezing machines operate in similar conditions as the bearings of calandering machines but the cylinders are not heated inside. Other examples of heavy loaded bearings are those of rolling mills for plastics and rubber (temperature of operation is about 200 °C).

It can be stated that operating conditions in calendaring machines are characterised by:

- low rotational speed (0 ÷ -40 r.p.m., v_p = m/s),
- strong reaction of high temperatures that result from the technology as well as the phenomena in the friction process (oil film),
- large forces applied to the bearings,
- journals misalignments resulting from the deflection of shafts,
- large diameters of bearings.

The studies on the bearing application in calendering machines show that the rolling bearings (at smaller diameters of cylinder) are widely used. The journal bearings of simple design that are chosen by means of approximation are used too. It such conditions it is useful to apply the journal bearings operating with a pressurized supply of oil to get hydrodynamic lubrication. A Pressurized oil supply allows the control of the bearing operation by means of variation the bearing characteristic [4, 5]; it is particularly important in the transition stage from mixed to hydrodynamic lubrication [1]. The design of journal bearings with pressurized oil supply is more and more popular. Mathematical (numerical programs) and technological tools during the manufacturing process of such bearings are the secret of foreign manufacturers.

The tribological system of supporting bearings affects the durability and reliability of such heavy-loaded textile machines. Theoretical investigation are the basis for the development of calculation methods of heavy loaded mechanically and thermally low speed and large overall dimension journal bearings [2, 3].

In this paper, the results of theoretical investigation of low-speed, large overall-dimension journal bearings operating in calendaring and similar types of machines into the static characteristics were described. These characteristics include: oil film pressure, temperature, viscosity, static equilibrium position angles, Sommerfeld number, minimum oil film thickness, maximum oil film pressure and temperature, power loss and oil flow.

State of the art in the range of heavy loaded journal bearings of textile machinery

Scientific investigation of calendering machines (calenders and flatters) are carried-out by the scientific-research centres of manufacturers. New and very important results of investigation have not been not published. Most of the papers include the technological aspects of flattening, or pressing the spatious effects on textiles.

Such manufacturers as Küsters and Ramisch have special, patented solutions for so called 'floating cylinders'. The level of technology in Poland does not allow to achieve such solutions even avoiding the patents.

There is a lack of literary data that concern the operation of heavy loaded journal bearings of heated bowl presses. There are few data in Poland on the investigation of bearings in very specific conditions of heavy mechanical and thermal loads [6 - 8]. However, investigation on the journal bearings of turbines, turbo-compressors, internal combustion engines operating at high speeds [2, 9, 10] have been carried-out by many scientific centres.

Theoretical and experimental investigation into journal bearing theory and practice are difficult and time absorbing. The advantages of these bearings, particularly with respect to fluid lubrication, is the main reason for its wide application. Many scientific institutions in Poland such as Warsaw, Wroclaw, Cracow, Gdansk, Łódź Technical Universities, Institute of Fluid Flow Machinery in Gdansk investigate journal bearings and bearings systems.

Foreign scientific centres have investigated the journal bearing for for 100 years. These centres are, e.g. Kingsbury Ltd (USA), Bentley Pressurized Bearings (USA), Laboratory of Solid Mechanics in Poitiers (France), Institute of Machine Design in Karlsruhe (Germany) [2], Sartorius Gleitlager Gottingen (Germany) and many others, but the data on the codes of computations of the static, dynamic characteristics and stability of different types of bearings are the property of these centres and kept in the secret.

Hybrid journal bearings

The pressurized bearings which are mostly applied in low-speed machines are generally known as hydrostatic bearings. Hydrodynamic bearings rely on the relative velocity between the rotating and stationary parts of the bearing to develop a self-sustaining oil wedge that supports a machine's shaft.

In the real operation of a journal bearing, boundary conditions in the hydrodynamic lubricating gap, depend on the oil supply pressure [1 - 4]. In some determined conditions, in the lubricating pockets of a multilobe journal bearing, a large static pressure can be generated [4]. This means that such a bearing operates as a hybrid and it combines hydrostatic and hydrodynamic pressures. Controlling of the oil supply pressure allows to obtain the micro-displacements of journal [4, 5], variations of the static and dynamic characteristics of bearing [4, 5] and the operation of bearing without mixed friction regime.

The advantages of hybrid bearings are as follows:

- low resistance to motion and higher efficiency,
- no metal to metal contact during the start-up of machine,
- ability of bearing accommodation to practical requirements ,
- no restriction in the bearing length to diameter ratio L/D.

Oil film pressure and temperature distributions

The oil film pressure, temperature and viscosity fields were obtained from the geometry Reynold's, energy, and viscosity equations on the assumption of laminar adiabatic oil flow through the bearing gap [9-10]. The oil film thickness of the cylindrical journal bearing is determined by equation (1).

$$H_c = 1 - \varepsilon \cdot \cos(\varphi - \alpha) \qquad (1)$$

The oil film pressure was computed from Reynold's equation in the following form:

$$\frac{\partial}{\partial\varphi}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\ \overline{p}}{\partial\ \varphi}\right) + \left(\frac{D}{L}\right)^2\frac{\partial}{\partial\ \overline{z}}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\ \overline{p}}{\partial\ \overline{z}}\right) = 6\frac{\partial\ \overline{H}}{\partial\ \varphi} + 12\frac{\partial\ \overline{H}}{\partial\ \phi} \qquad (2)$$

The transformation by means of pressure function u expressed by equation (3) was applied [4, 6].

$$u = (H^3\ \eta^{-1})^{0.5}\ p \qquad (3)$$

The partial differential equation (2) obtained the following form in the pressure function u:

$$u_{\varphi\varphi} + \left(\frac{D}{L}\right)^2 u_{zz} + d(\varphi, \alpha, \overline{\eta}, \overline{z}) = 6\overline{H}^{\frac{3}{2}}\overline{\eta}^{\frac{1}{2}}(\overline{H}_\varphi + \tfrac{2}{\omega}\overline{H}_t) \qquad (4)$$

The third member of equation (4) is expressed by equation (5).

It was assumed that on the bearing edges and in the regions of negative pressures the oil film pressure is nil, i.e. $p(\varphi,z) = 0$. The oil film pressure that was obtained from equation (4) was put into transformed energy equation (6) allowing the calculation of the oil film temperature and viscosity distributions [9].

The viscosity was described by exponential equation [6 - 10]. Oil film temperature and viscosity distributions were found by an iterative solution of equations (1) through (6) [6]. Temperature values $T(\varphi, z)$ on the boundaries ($z = \pm L/2$) were determined by means of parabolic approximation [6]. The boundary conditions for pressure and temperature took into account the inlet pressure and temperature.

$$d(\varphi,\alpha,\overline{\eta},\overline{z}) = \frac{1}{2}\frac{\overline{\eta}_{\varphi\varphi}}{\overline{\eta}} - \frac{3}{2}\frac{\overline{\eta}_\varphi}{\overline{\eta}^2}(\overline{\eta}_\varphi - \frac{\overline{H}_\varphi}{\overline{H}}\overline{\eta}) - \frac{3}{2\overline{H}^2}\left(\frac{\overline{H}_\varphi}{2} + \overline{H}\cdot\overline{H}_\varphi\right) - \frac{3}{4}\left(\frac{D}{L}\right)^2\left[\frac{\overline{H}_z}{\overline{H}} - \frac{2}{3}\frac{\overline{\eta}_{zz}}{\overline{\eta}} + 2\frac{\overline{\eta}_z}{\overline{\eta}^2}(\overline{\eta}_z - \frac{\overline{H}_z}{\overline{\eta}}\overline{\eta})\right] \qquad (5)$$

$$\frac{\overline{H}}{Pe}\left[\frac{\partial^2\overline{T}}{\partial\ \varphi^2} + \left(\frac{D}{L}\right)^2\frac{\partial^2\overline{T}}{\partial\ \overline{z}^2}\right] + \left[\frac{\overline{H}^3}{12\overline{\eta}}\frac{\partial\ \overline{p}}{\partial\ \varphi} - \frac{\overline{H}}{2}\right]\frac{\partial\ \overline{T}}{\partial\ \varphi} + \left(\frac{D}{L}\right)^2\frac{\overline{H}^3}{12\overline{\eta}}\frac{\partial\ \overline{p}}{\partial\ \overline{z}}\frac{\partial\ \overline{T}}{\partial\ \overline{z}} = -\frac{\overline{H}^3}{12\overline{\eta}}\left[\left(\frac{\partial\ \overline{p}}{\partial\ \varphi}\right)^2 + \left(\frac{D}{L}\right)^2\left(\frac{\partial\ \overline{p}}{\partial\ \overline{z}}\right)^2\right] - \frac{\overline{\eta}}{\overline{H}} \qquad (6)$$

Equations 5 and 6.

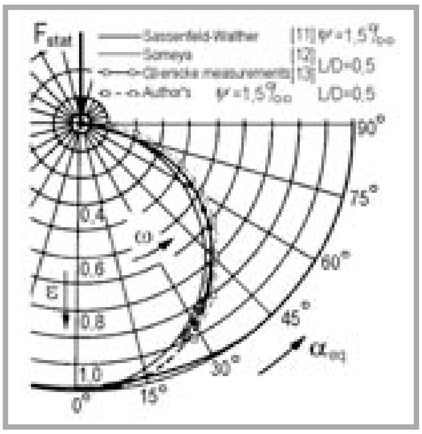

Figure 1. Comparison of static equilibrium position lines that were obtained by different authors.

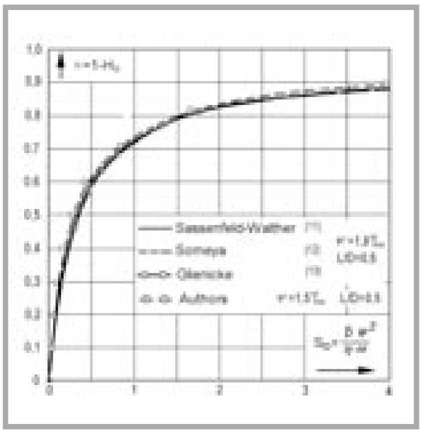

Figure 2. Comparison of relative eccentricities that were obtained by different authors.

Results of calculations

The calculations included oil film pressure and temperature distributions, static equilibrium position angles α_{eq} , Sommerfeld numbers So, minimum oil film thickness H_{min} and maximum oil film temperature T_{max}. Bearing length to diameter ratio L/D = 0.5 and L/D = 1.0, relative clearance ψ =1.5 ‰ and ψ = 1.9 ‰ as well as vertical load direction were assumed. At a rotational speed of journal n = 27.3 r.p.m. the heat number K_T = 0.0071 at the oil supply temperature T = 80 °C; and K_T = 0.00044 at the one T = 120 °C.

The values of journal displacements ε and the static equilibrium position angles α_{eq} that were obtained for the bearing length to diameter ratio L/D = 0.5 by the authors and other researchers [11 - 13] are given in Figure 1 and Figure 2; the run of static equilibrium position angle curves is very close but the differences in the values result from different lobe relative clearances. The values of both parameters, i.e. static equilibrium position angles and journal displacements are a little larger in the author's calculations than in the other researchers' ones.

The operation of heavy loaded journal bearings is characterized by large values of journal displacement in the bearing clearance. It means that in such a case the relative eccentricity is within the range of ε = 0.5 up to ε = 0.95. The oil film pressure and temperature distributions that are shown in Figure 3 through Figure 6, respectively, were obtained at the relative eccentricities of journal ε = 0.9, i.e. for the heavy load of bearing. Oil film pressure curves were determined for the

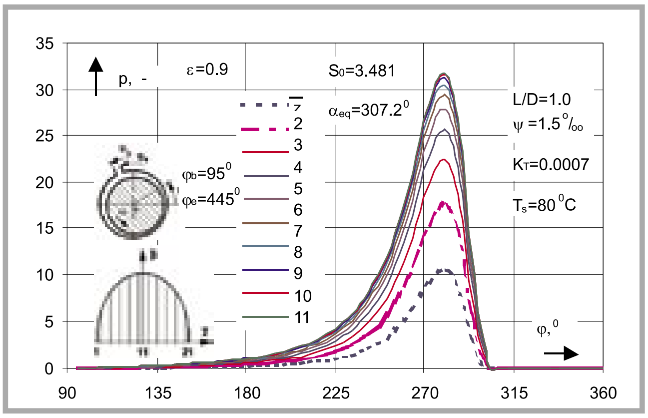

Figure 3. Oil film pressure distribution at low speed, heavy loaded cylindrical journal bearing.

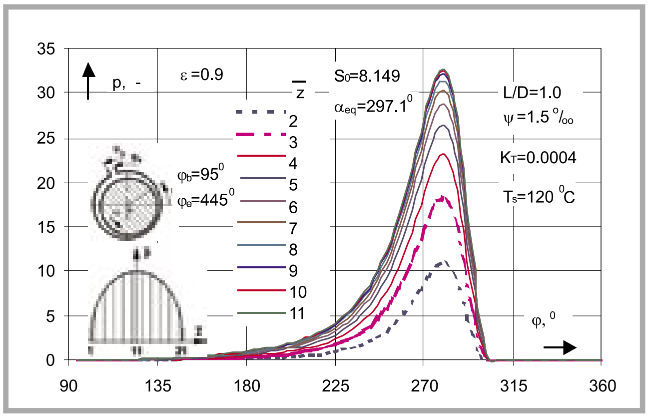

Figure 4. Oil film pressure distribution at low speed, heavy loaded cylindrical journal bearing.

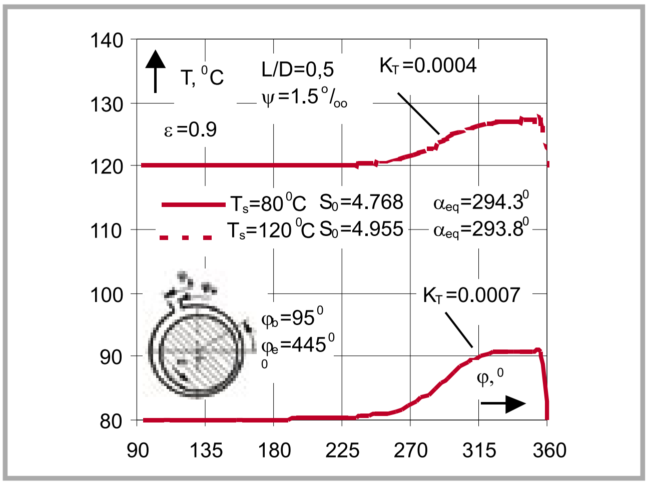

Figure 5. Oil film temperature distribution of low speed, heavy loaded cylindrical journal bearing.

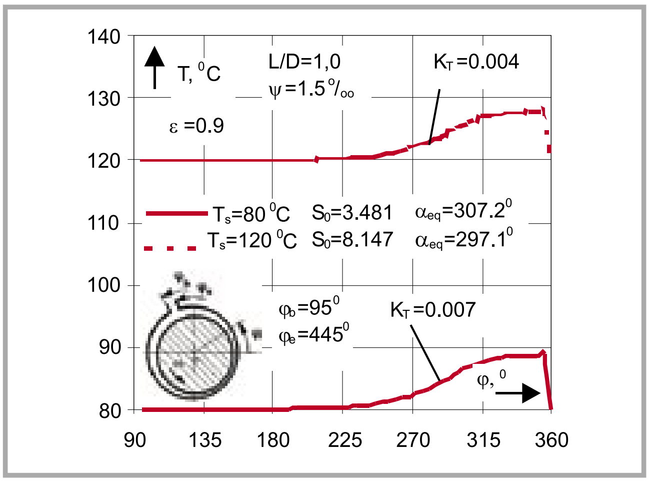

Figure 6. Oil film temperature distribution of low speed, heavy loaded cylindrical journal bearing.

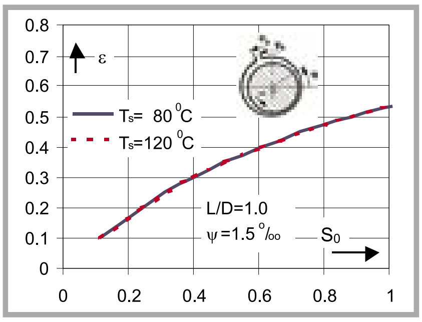

***Figure 7.** Relative eccentricity versus Sommerfeld number.*

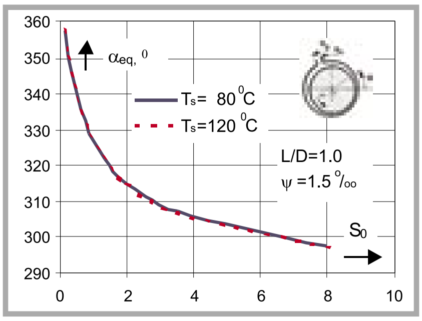

***Figure 8.** Static equilibrium position angle versus Sommerfeld number.*

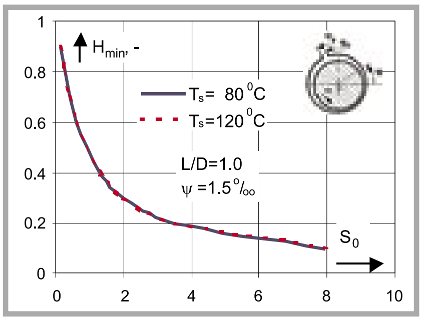

***Figure 9.** Minimum oil film thickness versus Sommerfeld number.*

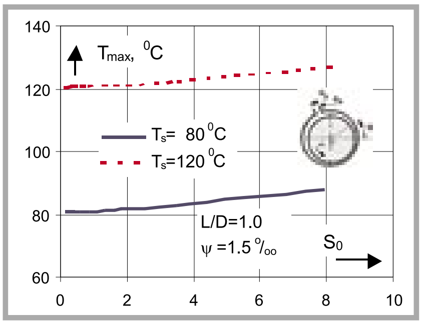

***Figure 10.** Maximum oil film temperature versus Sommerfeld number.*

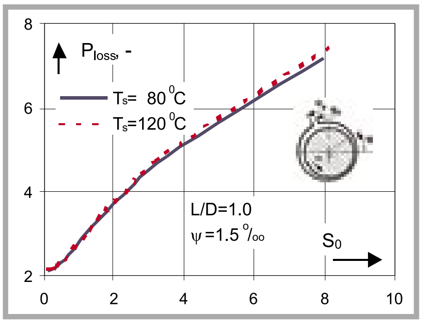

***Figure 11.** Power loss versus Sommerfeld number.*

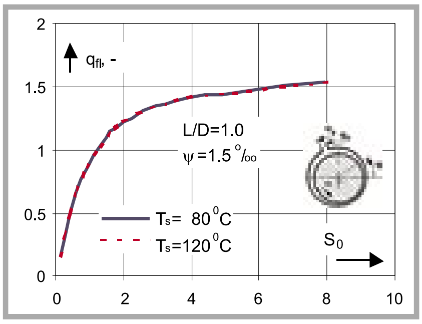

***Figure 12.** Oil flow versus Sommrfeld number.*

axial cross-sections of bearing, i.e. from the plain close to the edge of the bearing (e.g. Figure 3 and Figure 4 at $z = 2$) to the middle plain (Figure 3 and Figure 4 at $z = 11$). Oil film temperature distributions are shown in Figure 5 and Figure 6 for different values of length to diameter ratios. An increase in the temperature of the oil supplied causes the increase in the temperatures of oil film temperature distributions.

The journal displacements ε, static equilibrium position angles α_{eq}, Sommerfeld number *So*, minimum oil film thickness H_{min}, maximum oil film temperature T_{max}, power loss P_{loss} and oil flow q_{fl} are shown in Figure 7 through Figure 12 for different values of supplied oil temperature. Within the range of assumed oil supplied temperatures there is a very small effect of these temperatures on the considered static characteristics of bearings.

Conclusions

The results of analytical investigation of low-speed, large overall-dimension and heavy loaded journal bearings operating in, e.g. calendaring machines into the static characteristics allow to present the following conclusions:

1. The developed program of computation is capable of obtaining the static characteristics of heavy loaded, low speed journal bearings.
2. The results obtained by the authors are comparable to the results of other researchers.
3. An increase in the supplied oil temperature causes an increase in the temperature of oil film and its maximum temperature too.
4. The static characteristics as a Sommerfeld number, static equilibrium position angle, power loss, oil flow and minimum oil film thickness show very small variations with an increase in the supplied oil temperature.

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Received 28.07.2006 Reviewed 02.10.2006