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CALCULATION OF NONISOTHERMAL CHARACTERISTICS OF SLIDE JOURNAL BEARINGS BY MEANS OF APPROXIMATION FORMULAE

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Abstract

The results of the computer simulation concerning the generation of approximation formulae for calculations of static and dynamic characteristics of a slide journal bearing with a cylindrical bush with a full angle of contact are presented. An adiabatic flow of the oil has been assumed, taking into account the mixing of the oil in the vicinity of the supply groove. The approximation formulae have been generated using the least square method.

1. Introduction

In the dynamics of mechanical systems, a slide bearing is a complex, nonlinear viscoelastic element which can exert a strong influence on the behaviour of the system supported in this bearing [4, 5]. The value of the amplitude of forced vibrations depends on the value of excitation forces and on the relation between rotor rotations and critical rotations. Maintaining the difference between the above mentioned numbers large enough is a more effective way to decrease the forced vibrations than decreasing the excitation forces [2]. The kinematic accuracy and a lack of shaft vibrations depend mainly on the design of bearings.

With slide bearings, the specific dynamic properties originate from the presence of a lubricating medium and from oil film characteristics. The displacement of the journal centre around the equilibrium position, caused by an external disturbance, results in a change in the geometry of the oil gap and, consequently, brings about a change in the resultant hydrodynamic force. An increase in the hydrodynamic force is a resultant of the oil film stiffness force which maintains the journal centre motion and the oil film damping force which inhibits this motion. An analysis of oil film dynamic properties is reduced thus to the investigations of low vibrations around the static equilibrium position. At such an assumption, the nonlinear reaction of the oil film [3] can be approximated by means of the linear function of displacement versus journal centre displacement velocity. The threshold velocity values are calculated through the investigations of a proper stability criterion of the rotating system described by means of equations of motion, employing linear stiffness and damping coefficients.

The development in the field of design of machines and devices is characterised by an increase in their dynamic activity manifested by vibrations. Therefore, the determination of oil film dynamic properties is so necessary in design of shaft bearing systems.

Calculation of slide bearing characteristics is connected with a solution of a conjugate system of partial differential equations describing the complex phenomena of mass and energy transfer which occur in the bearing. Finding a solution to this system is possible only by means of approximation methods, especially numerical ones, for the determined geometry of the oil gap, according to the scheme:

$$F = F(\lambda, \alpha) \tag{1}$$

where: F - bearing load, λ - relative eccentricity, α - angle of the position of centre lines.

In design processes, the procedure is different - the position of the journal centre, i.e. the oil gap geometry, at which the oil film load capacity balances the given external bearing load, is sought:

$$(\lambda, \alpha) = g(F, \ldots) \tag{2}$$

A solution is obtained through numerous repetitions of the calculation procedure. Next, for the already known position of the journal centre, the bearing characteristics are calculated. The analysis of design solutions is simplified if the bearing properties are presented in the form of analytical formulae or tables. So far, such formulae have been developed only for an isothermal model for calculations of static characteristics [11, 12, 13] and tables have been prepared for calculations of damping and stiffness coefficients [9].

In the design process of shaft slide bearing systems, the following stages can be distinguished [6]:

1. initial design consisting in determination of main geometrical parameters of the bearing, employing the simplest, isothermal model of flow;
2. detailed design, including corrective calculations, in which as exact a nonisothermal flow model as possible is assumed. As the effect of this design stage, the bearing element is finally designed and the values of design limitations, e.g. the maximum oil temperature, are checked and the oil film stiffness and damping coefficients are calculated.

On the basis of analytical and numerical investigations [1, 10], the following computational procedure for the detailed design of the shaft bearing system, taking into account dynamic properties of the nonisothermal oil film, is proposed:

1. determination of the relation for calculation of the threshold velocity for the shaft loading scheme under consideration and for the position of the bearings with respect to the load, e.g.

$$\Omega^2 = \left(\frac{\omega_{gr}}{\omega_0}\right)^2 = \Omega\{[S],[B],S_0\} \tag{3}$$

where: ω_{gr} - threshold angular velocity, ω_0 - reference velocity, [S] - stiffness matrix, [B]- damping matrix, S_0 - relative bearing load;

2. qualitative analysis of the effect of the bearing design parameters (relative bearing width, relative bearing clearance, dynamic viscosity of the oil) on the threshold velocity for the given value of the external load and the angular velocity of the shaft;
3. initial determination of the bearing design parameters;
4. exact calculation of the bearing static and dynamic characteristics for a nonisothermal oil flow.

It is possible to carry out the qualitative analysis (see point 2) in the case when a sufficiently developed database or approximation formulae allowing for calculation of individual bearing characteristics are available. An influence of the bearing design parameters on the threshold angular velocity of the shaft is shown in Figs. 1 - 2.

The aim of the investigations presented here is to develop approximation formulae for calculations of oil film static and dynamic characteristics.

2. Model of the bearing

The presented investigations concern a slide bearing with a cylindrical bush, supplied by one supply groove, at the parallel-axial position of the journal axis with respect to the bush with a full angle of contact. The bearing geometry is presented in Fig. 3.

$$h = \Psi R [1 + \lambda \cos (\varphi - \alpha)] \tag{4}$$

where: h - thickness of the oil film, φ - circumferential co-ordinate.

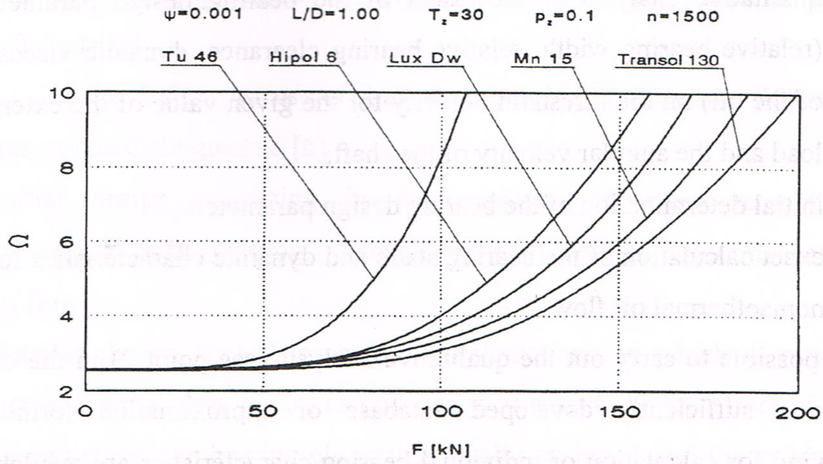

Fig. 1. Influence of the oil viscosity on the threshold velocity

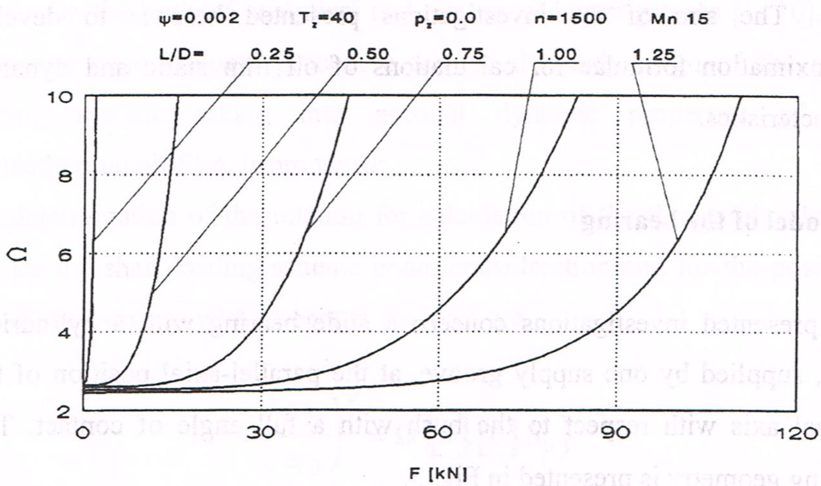

Fig. 2. Influence of the relative bearing width on the threshold velocity

The geometrical form under analysis is a designing standard assumed in analytical investigations, in national standards [11, 12] and international standards [13] devoted to calculations of slide bearings.

The investigations of bearing properties were carried out assuming an adiabatic model of the statically loaded infinite stiffness bearing which operates under fluid friction conditions, with a laminar flow of the oil, and taking into account the mixing of oil streams within the oil groove [1, 10]. The oil streams within the supply groove are shown in Fig. 4.

In the assumed model of flow, the pressure field in the oil film generated as a result of any plane motion of the journal, including both a rotary and plane oscillating motion, is described by means of the equation:

$$\frac{\partial}{R\partial\varphi}\left(\frac{h^3}{\eta}\frac{\partial p}{R\partial\varphi}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=6U\frac{dh}{Rd\varphi}+12\frac{\partial h}{\partial t} \tag{5}$$

where: p - pressure, η - oil viscosity, φ, z - co-ordinates, R- journal radius,

U - circumferential velocity of the journal.

In order to solve Eq. (5), the following boundary conditions have been assumed:

- the oil pressure in the region confined by the supply groove edges is equal to the supply pressure p_z,
- the oil pressure at the bushing outlet edges along the axial direction is equal to the ambient pressure,
- a non-negative pressure occurs in the oil gap,

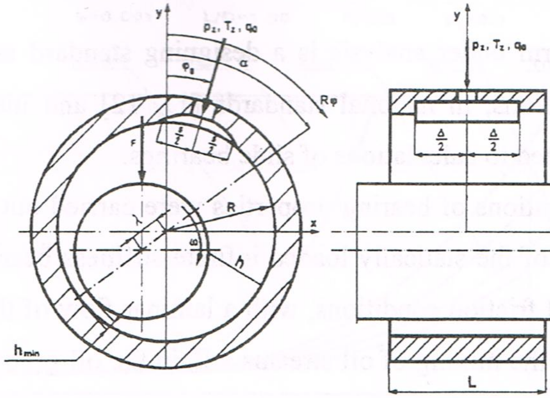

Fig. 3. Bearing geometry

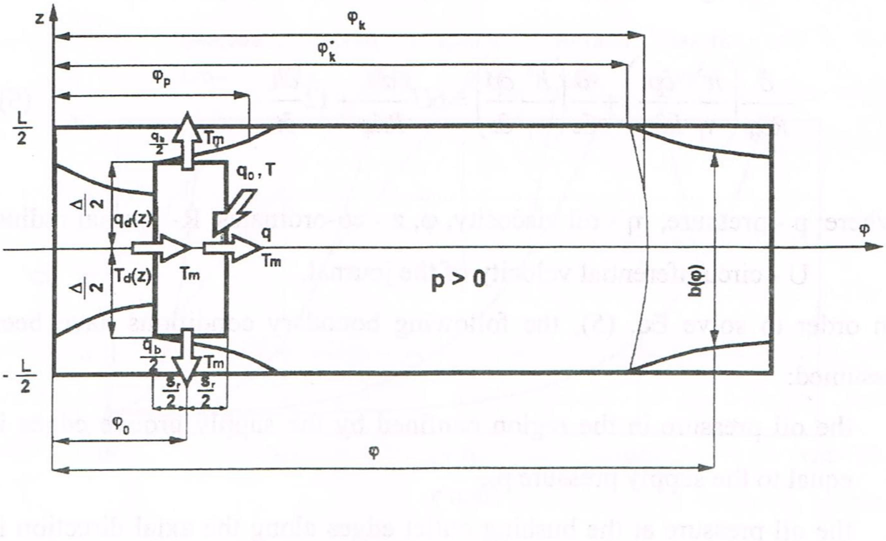

Fig. 4. Oil streams within supply grooves

- the end of the oil film is determined by the geometrical position of the points in which the film pressure is equal to the ambient pressure, whereas the pressure gradient is equal to zero.

Starting with the equation of energy [8], the equation of the plane temperature field of the oil film $T(\varphi, z)$ has been obtained:

$$\rho c\left[\left(\frac{U}{2}-\frac{h^2}{12\eta}\frac{\partial p}{R\partial\varphi}\right)\frac{\partial T}{R\partial\varphi}-\left(\frac{h^2}{12\eta}\frac{\partial p}{\partial z}\right)\frac{\partial T}{\partial z}\right]=\frac{h^2}{12\eta}\left[\left(\frac{\partial p}{R\partial\varphi}\right)^2+\left(\frac{\partial p}{\partial z}\right)^2\right]+\frac{\eta U^2}{h^2}+ \\ +\kappa\left(\frac{\partial^2 T}{R^2\partial\varphi^2}+\frac{\partial^2 T}{\partial z^2}\right) \quad \mathbf{(6)}$$

where: ρ - oil density, c - oil specific heat, κ - thermal conductivity coefficient.

Equation (6) has been solved at the assumption that the oil fulfils the whole height of the oil gap. To solve the equation of the temperature field, the following boundary conditions have been assumed:

- the oil temperature in the supply groove is equal to the mixing temperature which will settle as a result of the complete and full mixing, assuming that the supply region is adiabatic,
- the oil temperature at the oil film boundary $(\varphi, z = -L/2)$, $(\varphi, z = L/2)$, $(\varphi, z = b(\varphi)/2)$, $(\varphi, z = -b(\varphi)/2)$ has been approximated with a parabola.

On the basis of the balance of the heat streams within feedholes and assuming that (Fig. 3):

- the oil stream comes from the cavitation zone of the oil film $q_d(z) \equiv q_z T[(\varphi_0 - s_r/2)_{-0}, z]$ at the temperature $T_d(z) \equiv T[(\varphi_0 - s_r/2)_{-0}, z]$,

- the oil stream comes form the groove into the pressure zone of the oil film q at the temperature T_m,
- the oil stream comes out along the side edges of the groove along the axial direction q_b,

the following formula for calculation of the mixing temperature in the supply groove has been generated:

$$T_m = \frac{q_0 T_z + \int_{z_1}^{z_2} \left(T\left[\left(\varphi_0 - \frac{s_r}{2} \right)_{-0}, z \right] q_d \left[\left(\varphi_0 - \frac{s_r}{2} \right)_{-0}, z \right] \right) dz}{q_b + q}, \qquad (7)$$

where: $T[(\varphi_0 - s_r/2)_{-0}, z]$ - temperature of the oil film in front of the supply groove edge, calculated from the equation of energy, z_1, z_2 - limits of integration.

The set of Eqs. (5, 6) has been solved numerically by means of the finite difference method, using a five-point differential operator. The efficient Simultaneous Over-Relaxation (SOR) method with the Chebyshev acceleration has been applied. A numerical computation procedure and the CHALADYN computer program written in NPD FORTRAN, implemented into the UNIX SCO operating system, have been developed. The program enables investigations of static and dynamic characteristics of slide journal bearings with circular bushes. The following characteristic properties have been assumed as the static characteristics: load capacity, maximum oil film pressure, maximum oil film temperature, oil stream flowing through the bearing, coefficient of the resistance to motion, static equilibrium position curves. The oil film linear stiffness and damping coefficients are the dynamic characteristics.

3. Scope of investigations

The numerical simulation has been carried out, assuming the following ranges of variability in the values of bearing design parameters:

$$L/D \in \{0.25, 0.5, 0.75, 1.0, 1.25\}$$

$$\psi \in \{0.003, 0.0025, 0.002, 0.0015, 0.0001\}$$

$$\text{oil} \in \{\text{Tu 46, Hipol 6, LuxDW, Mn 15, Transol 130}\}$$

$$n \in \{1000, 1500, 2000, 3000, 4000, 5000\}$$

$$Tz \in \{20, 30, 40, 50\}$$

4. Approximation formulae

The search for the approximation formula Y = f(x) has been carried out according to the following procedure:

1. determination of the formula form (polynomial; an exponential, power or logarithmic function or a combination of the above mentioned functions);
2. calculation of the constant coefficients occurring in the formula;
3. a degree of the conformity of the function with the values of y(x) points, to which we adjust the curve, has been assumed as a correctness criterion of the selection of the formula and coefficients.

In the investigations the least square method has been applied, in which a degree of the conformity is measured by means of the sum of square deviations of y from Y.

$$S=\sum_{i=1}^{n}\left(Y_i-y(x_i;a_0,a_1,a_2,\ldots,a_n)\right)^2 \rightarrow \min \tag{8}$$

The task consists in finding the values of the coefficients at which the sum of the square deviations is the smallest.

In the present study, the approximation of the properties of slide journal bearings has been investigated by means of the following formulae:

$$y=a_0+a_1x+a_2x^2+a_3x^3+a_4x^4+a_5x^5+a_6x^6 \tag{9a}$$

$$y^{-1}=a_0+a_1x+a_2x^2+a_3x^3+a_4x^4+a_5x^5+a_6x^6 \tag{9b}$$

$$y^2=a_0+a_1x+a_2x^2+a_3x^3+a_4x^4+a_5x^5+a_6x^6 \tag{9c}$$

$$\ln(y)=a_0+a_1x+a_2x^2+a_3x^3+a_4x^4+a_5x^5+a_6x^6 \tag{9d}$$

$$\sqrt{y}=a_0+a_1x+a_2x^2+a_3x^3+a_4x^4+a_5x^5+a_6x^6 \tag{9e}$$

As a result of the investigations carried out, the relations presented by the following formulae have been obtained:

- static characteristics:

$$\alpha = \left(m_1\lambda^4 + m_2\lambda^2 + m_3\lambda + 1\right) arctg\left(\pi\frac{\sqrt{1-\lambda^2}}{2\lambda}\right) \tag{10a}$$

$$\left.\begin{matrix} F \\ p_{max} \\ \ln(q) \\ \ln(\mu) \\ T_{max}^2 \end{matrix}\right\} = W_6(\lambda) \tag{10b}$$

where: p_{max} - maximum oil film pressure, q - oil stream, μ - friction coefficient, T_{max} - maximum oil film temperature, m_1, m_2, m_3 - coefficients depending on the bearing design parameters.

- dynamic characteristics:

$$\left.\begin{matrix} B_{xx}^{-1} \\ B_{yy}^{-1} \end{matrix}\right\} = W_6(\lambda) \tag{11a}$$

$$\left.\begin{matrix} \ln(S_{xx}) \\ \ln(S_{xy}) \\ \ln(S_{yy}) \\ \ln(B_{xy}) \\ \ln(B_{yx}) \end{matrix}\right\} = W_6(\lambda) \tag{11b}$$

where: $W_6(\lambda)$ - polynomial of the sixth order.

The constant coefficients occurring in approximation formulae (10, 11) are functions of the following bearing design characteristics: relative width, relative bearing clearance, viscosity, temperature and pressure of the supply oil, and angular velocity of the journal, and they have been determined by means of numerical computations performed with the CHALADYN program developed by the author of the present paper.

In Figs. 6 - 9 some exemplary approximation characteristics and the coefficients used in their computation are shown.

5. Conclusions

1. The generated formulae approximate the bearing characteristics with the accuracy ≤ 5% in the investigated range of the bearing design parameter variability.
2. The developed approximation formulae can be used in design processes to analyse an influence of the bearing design parameters on the bearing static and dynamic characteristics.
3. It is expected that further investigations will be carried out and that these formulae will be used in design processes of slide bearing systems, especially in software for engineering calculations.

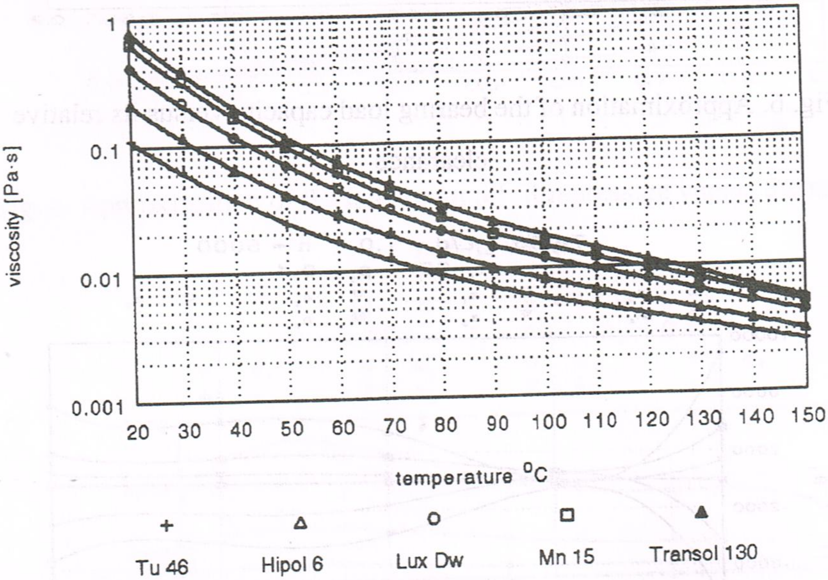

Fig. 5. Characteristics of the oils under consideration

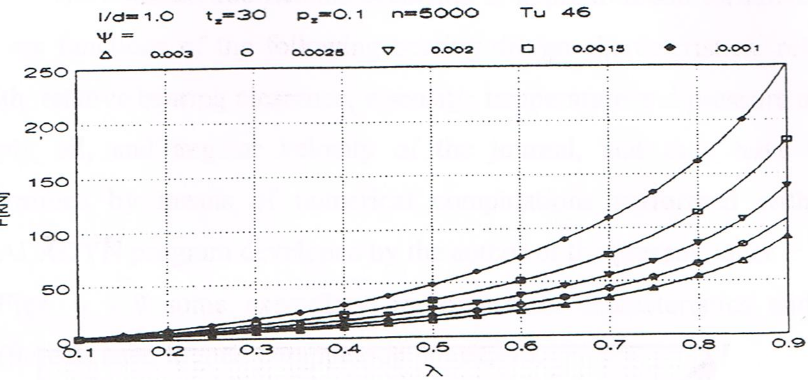

Fig. 6. Approximation of the bearing load capacity versus its relative clearance

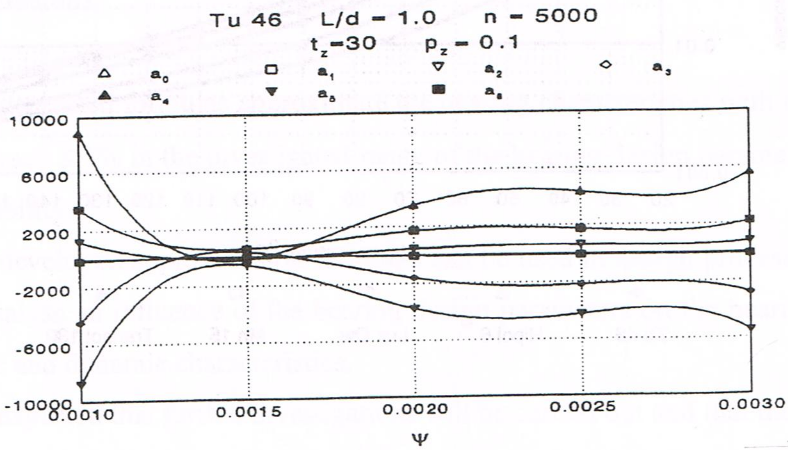

Fig. 7. Coefficients for bearing load capacity computations

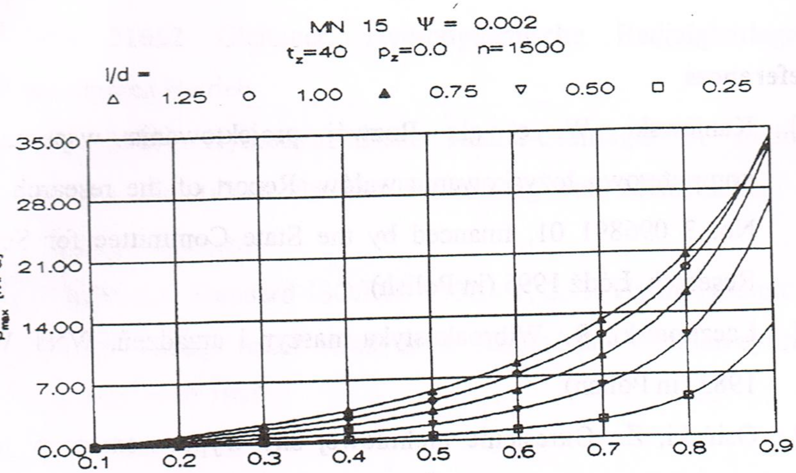

Fig. 8. Approximation of the maximum oil film pressure versus the relative bearing width

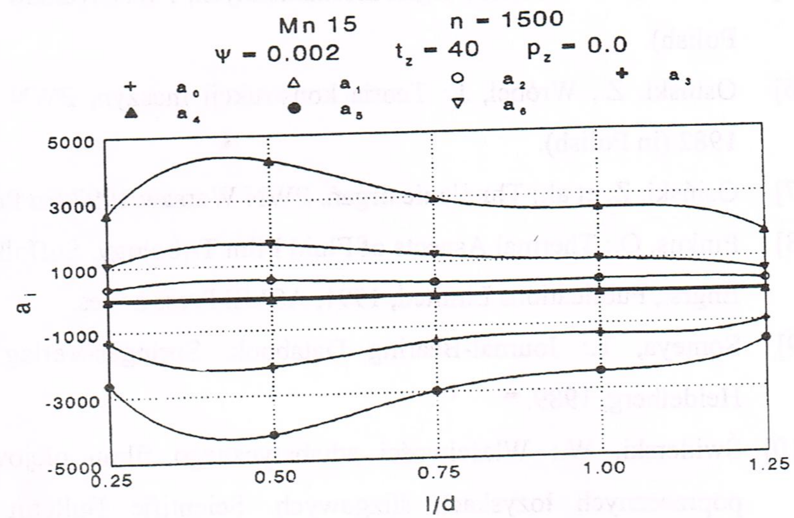

Fig. 9. Coefficients for maximum oil film pressure computations

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