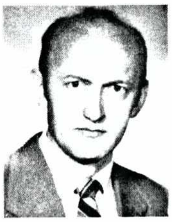

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# Theoretical research on sliding radial seals applicable to ship CP propellers

SUMMARY

*Assumptions of a physical model, equations describing it, flow diagram of a relevant computer calculation program as well as results of the static calculations of the sliding radial seals usually applied in ship CP propeller systems are presented in the paper. The model takes into account the diathermic description of oil flow through the bearing interspace. The calculation results of an exemplary seal are compared with those obtained from a simpler, adiabatic oil flow model.*

## INTRODUCTION

The sliding radial seals whose working principle is shown in Fig.1 are usually applied to ship CP propellers and Kaplan's water turbines. They make it possible to deliver oil under pressure into rotating shaft interior in order to change pitch of a propeller or turbine hydraulically.

Service experience revealed that even similar bearings in nearly the same working conditions were able to behave in a qualitatively quite different way. For a long time it was the main reason for applying the oil pressure limit of less than 10 MPa.

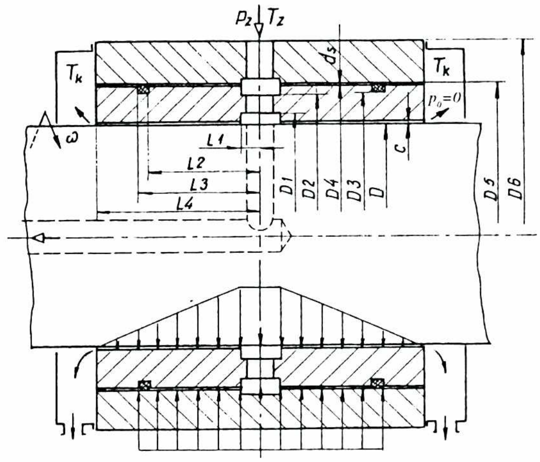


*Fig. 1. Axial cross-section of the sliding radial sleeve*

Research works carried out by the team of the Faculty of Ocean Engineering and Ship Technology, Technical University of Gdańsk, made it clear that the inadequate work of the seals of certain design parameters was caused by unfavourable dynamic phenomena appearing within the oil film between the rotating shaft and bearing sleeve. A detail description of the phenomena is presented in [1]. Their occurence was demonstrated by an unstable work of the seal accompanied by a dozen or so times higher oil leakage and power consumption than that at its stable work. The bearing interspace shape which was changing during the seal work in result of elastic and thermal deformations of the flexible bearing sleeve (first of all), and also of the shaft (to a lower extent), was found the main factor responsible for occurrence of the phenomena.

The elastic deformations depend mainly on the distribution of the pressure acting on the axial cross-section of the sleeve, shown in Fig.1, but the thermal ones on the temperature distribution.

It is very important to have at disposal a computation program which would make possible to determine with a sufficient accuracy the deformations and other magnitudes which influence the seal work. The task is rather complicated as it requires solving several mutually coupled equations which describe the course of the phenomena accompanying the seal work, such as flow, energy, thermal conductivity, oil viscosity and deformation equations. A number of the equations and their complexity depends on a physical model assumed.

Two models of adiabatic oil flow through the bearing interspace for calculating the sliding radial seals are presented in [2,3]. The models differ from each other by a way in which thermoelastic deformations of the sleeve and shape of the bearing interspace are calculated.

A more complex model describing the phenomena based on the diathermic oil flow through the bearing interspace with two-directional,i.e. axial and radial, heat flow through the seal sleeve taken into account is presented in [6].

This paper is a continuation of the above mentioned work as well as its extension by a computer calculation program and research results of an exemplary seal.

## PHYSICAL MODEL ASSUMPTIONS

The physical model which describes the phenomena occurring in the sliding radial seals is based on the following assumptions :

- Two-directional, i.e. axial and circumferential ,oil flow through the bearing interspace.
- Oil is incompressible, therefore $\rho$ = const.
- Oil viscosity is a pressure and temperature function : $\mu = f(p, T)$.
- The heat generated in the oil flow through the bearing interspace is carried away by the flowing oil as well as due to the heat conductivity of the shaft and sleeve.
- Oil temperature changes in the interspace are three-directional, i.e. circumferential, axial and radial – across the oil film thickness.
- Sleeve temperature changes along axial and radial directions. This is connected with accounting for the heat exchange between the oil and sleeve on its surfaces (cylindrical and face ones).
- Shaft surface temperature is assumed equal to that calculated on the sleeve internal surface.
- Thermoelastic deformations of the sleeve are axial-symmetric. It means that the height of the bearing interspace changes in the axial direction in dependence on sleeve deformations, and in the circumferential direction in dependence on mutual orientation of the shaft and sleeve.

The assumption on axial-symmetric distribution of pressure and temperature in the bearing interspace for calculating the sleeve deformations makes the calculation model much simpler and is reasonable in view of correct sleeve operation as the favourable, stable operation of the sleeve is obtainable at the characteristic, almost co-axial location of the shaft and sleeve. Pressure and temperature differences over the sleeve circumference are therefore small and their influence on the sleeve deformation can be neglected.

## PHYSICAL MODEL EQUATIONS

Particular equations together with short comments on their solving methods are presented below.

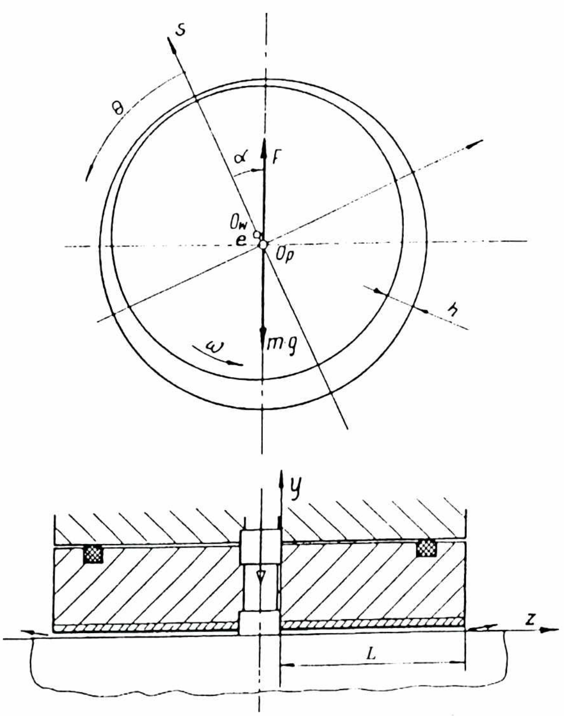


***Fig.2.** Scheme of the sleeve assumed for theoretical calculations*
*m - loading mass, g - gravity acceleration*

### Equation of oil flow through the bearing interspace

Reynolds flow equation takes the following form in view of the accepted assumptions :

$$\frac{\partial^2 p}{\partial x^2}\int_0^h C_1 dy + \frac{\partial p}{\partial x}\frac{\partial}{\partial x}\int_0^h C_1 dy + \frac{\partial^2 p}{\partial z^2}\int_0^h C_1 dy + \frac{\partial p}{\partial z}\frac{\partial}{\partial z}\int_0^h C_1 dy = \omega R \int_0^h \frac{\partial}{\partial x} C_2 dy \qquad (1)$$

where :

$$C_1 = \int_0^h \frac{y}{\mu} dy - \frac{\int_0^h \frac{y}{\mu} dy}{\int_0^h \frac{1}{\mu} dy} \cdot \int_0^y \frac{1}{\mu} dy \qquad (2a)$$

$$C_2 = \frac{\int_0^y \frac{1}{\mu} dy}{\int_0^h \frac{1}{\mu} dy} \qquad (2b)$$

$R = \frac{1}{2}D$ - shaft radius (Fig.1)
$p$ - oil pressure
$\omega$ - shaft angular speed
$x, y, z$ - coordinate in the circumferential, radial and axial direction, respectively
$h$ - bearing interspace height (Fig.2) defined by (3) :

$$h = c + e \cdot \cos\theta + \Delta h(z) \qquad (3)$$

where :
$c$ - design radial clearance of the seal (Fig.1)
$e$ - eccentricity of the sleeve location in respect to the shaft (Fig.2)
$\Delta h(z)$ - sleeve thermoelastic deformations
$\theta$ - angular coordinate
$\mu$ - oil dynamic viscosity calculated from (4) :

$$\mu = \mu_o \exp[a \cdot p + b(T_{od} - T_o)] \qquad (4)$$

where :
$\mu_o$ - oil dynamic viscosity in the reference temperature $T_{od}$ at the atmospheric pressure
$T_{od}$ - reference temperature
$T_o$ - oil temperature
$a, b$ - characteristic coefficients of oil properties.

The reference system of the cylindrical coordinates $r$, $\theta$, $z$ and the following boundary conditions were assumed to solve the equation (1) :

$$p(\theta, z = 0) = p_z \qquad \text{(Fig.1,2)}$$

$$p(\theta, z = L) = p_o \qquad \text{(Fig.1,2)}$$

$$p(\theta, z) = p(\theta + 2\pi, z) \qquad (5)$$

$$\frac{\partial p}{\partial \theta}(\theta, z) = \frac{\partial p}{\partial \theta}(\theta + 2\pi, z)$$

where : $L = L_4 - L_1$ (see Fig.1).

The flow equation was solved by means of the five-point, finite difference method which consists in the covering of the entire integration area by a mesh of calculation points in which the derivatives were replaced by the relevant finite differences. The equation system obtained in this way was solved with the use of an iteration method. The calculated oil pressure values in every mesh point make it possible to calculate hydrodynamic force components by applying the following relationships :

$$F_s = R\int_0^{2\pi}\int_0^{L} p(\theta,z)\cdot cos\theta d\theta dz \qquad (6a)$$

$$F_n = R\int_0^{2\pi}\int_0^{L} p(\theta,z)\cdot sin\theta d\theta dz \qquad (6b)$$

The resultant hydrodynamic force F was then calculated :

$$F = \sqrt{F_s^2 + F_n^2} \qquad \text{(Fig.2)} \qquad (7)$$

and the angle between the direction of the resultant force and the line of centres of the shaft and sleeve, $\alpha$ :

$$\alpha = arc\, tg \frac{F_n}{F_s} \qquad (8)$$

The output of the oil leaking through the interspace was calculated from the following expression :

$$Q_z = \frac{R}{12}\int_0^{2\pi}\frac{h^3}{\mu}\frac{dp}{dz}d\theta \qquad (9)$$

## Energy equation

The oil temperature distribution in the bearing interspace was derived by solving the energy equation of the following form corresponding to the assumed model :

$$\rho c_v\left(u\frac{\partial T_o}{\partial x} + w\frac{\partial T_o}{\partial z}\right) - k_o\frac{\partial^2 T_o}{\partial y^2} = \mu\left[\left(\frac{\partial u}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial y}\right)^2\right] \qquad (10)$$

where :

- $\rho$ - oil density
- $c_v$ - oil specific heat
- $k_o$ - oil heat conduction coefficient
- $u$ - circumferential component of oil flow velocity, expressed by (11) :

$$u = \frac{\partial p}{\partial x}C_1 - \omega R(1 - C_2) \qquad (11)$$

and „$w$" - axial component of oil flow velocity defined by (12) :

$$w = \frac{\partial p}{\partial z}C_1 \qquad (12)$$

After transformation of (10) by means of cylindrical coordinates and replacement of the temperature derivatives by finite difference expressions the equation was solved with the use of the iterative method and the following boundary conditions :

$$\begin{aligned} &T_o(\theta, r, z=0) = T_z \\ &T_o(\theta, r=R, z) = T_w(z) \\ &T_o(\theta, r=R+h, z) = T_p(z) \\ &T_o(\theta, r, z) = T_o(\theta+2\pi, r, z) \\ &\frac{\partial T_o(\theta, r, z)}{\partial \theta} = \frac{\partial T_o(\theta+2\pi, r, z)}{\partial \theta} \\ &for\ R < r < R+h \end{aligned} \qquad (13)$$

where :

- $T_z$ - seal input oil temperature (environmental temperature, Fig.1)
- $T_w(z)$ - temperature distribution over the shaft surface
- $T_p(z)$ - temperature distribution over the internal sleeve surface.

The first calculation step is carried out for the temperature assumed on the sliding surface of the sleeve and shaft. In result a temperture distribution in the oil film is obtained that makes it possible to calculate circumferentially averaged temperature gradients on the sleeve sliding surface as well as the average temperature at the bearing interspace outlet.

These parameters serve as the boundary conditions for the heat conduction equation presented further on ; in result of solving it the temperature distribution over the sleeve is achieved. It makes determining a new temperature distribution on the sliding surfaces of the sleeve and shaft possible that forms the boundary conditions for the energy equation in the consecutive iteration step of calculation.

The temperature on the sleeve sliding surface is determined from the following relationship :

$$T_o^{i+1}(\theta, r=R+h, z) = (1-m)\cdot T_o^i(\theta, r=R+h, z) + m\cdot T^i(r=R+h, z) \qquad (14)$$

where :

- $i$ - iteration step number
- $m$ - relaxation factor.

The shaft surface temperature is assumed equal to that calculated on the sleeve internal surface. The cycle of computation is repeated until results of the subsequent steps differ sufficiently little.

## Thermoelastic deformation equation

The finite element method was applied to calculate thermoelastic deformations of the sleeve. It was possible to reduce the three-dimensional problem to two-dimensional one due to assuming the axial-symmetric loading on the sleeve. In view of expected small deformations of the sleeve the pressure was assumed to form the static load of a constant distribution calculated from the flow equation and circumferentially averaged, and of a steady direction.

The searched displacement vector $\overline{u}$ is obtained from the global solution (for the entire sleeve) of the equilibrium equation as follows :

$$\overline{K}\cdot\overline{u} = \overline{r} \qquad (15)$$

where :

- $\overline{K} = \sum_{e=1}^{N}\overline{K}_e$ - stiffness matrix obtained in result of agregation process of the stiffness matrices $\overline{K}_e$ of all $N$ sleeve elements
- $\overline{r} = \sum_{e=1}^{N}(\overline{P}_e + \overline{P}_e^t)$ - load matrix obtained in result of aggregation process of the vectorial sums of nodal loadings due to the pressure, $p_e$ , and temperature, $p_e^t$.

The equation (15) is modified during solution process by taking into account constraints applied on the structure. In result the global displacement vector is obtained :

$$\bar{u} = \bar{K}^{-1} \cdot \bar{r} \tag{16}$$

After identification : $\bar{u} \rightarrow identification \rightarrow \hat{\bar{u}}_c$ the searched radial displacements of the nodes located on the generatrix of the internal cylindrical surface can be determined for each element as well as deformations at an arbitrary point of the sleeve :

$$\bar{\varepsilon}(y,z) = \bar{B}(y,z) \cdot \hat{\bar{u}}_c \tag{17}$$

and then the stresses :

$$\sigma = \bar{C}^t \cdot (\bar{\varepsilon} - \bar{\varepsilon}^t) \tag{18}$$

where :

$\bar{B}(y,z)$ - deformation (strain) function matrix
$\bar{C}^t$ - constitutive matrix
$\bar{\varepsilon}^t$ - thermal deformation (strain) vector.

## Lost power equation

Power losses in a correct working seal are caused by shearing the oil layer due to shaft rotation and resistance of the oil flow in result of pressure difference. The lost power value was calculated from (19) for one half of the seal :

$$N_t = \omega^2 R^2 \int_0^{2\pi} \int_0^{L} \frac{\mu}{h} d\theta dz + Q_z(p_z - p_o) \tag{19}$$

## Heat conduction equation

The Laplace equation was applied to calculate the temperature distribution in the sleeve in compliance with the earlier defined assumptions, which takes the following form when assuming the cylindrical coordinates $r$, $\theta$, $z$ and axial symmetry :

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0 \tag{20}$$

where: $T$ - temperature of the sleeve material.

The sleeve and bearing casing were assumed to contact each other on their cylindrical surfaces therefore they may be considered to form a uniform element like the sleeve shown in Fig.3.

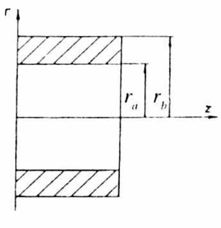


*Fig.3. Scheme of one half of the sleeve together with bearing casing, applied in the theoretical calculation model*

The conduction equation (20) was solved by using the analytical method for two sets of boundary conditions ($d_1$,$d_2$)which differed from each other by the assumed model of heat flow through the middle part of the circumferential ducts of the sleeve (that is highlighted further on) :

if $z = 0$ $$-\frac{\partial T}{\partial z} + h_1 T = h_1 T_z \quad (d_1 \text{ model}) \tag{21}$$

$$\frac{\partial T}{\partial z} = 0 \quad (d_2 \text{ model}) \tag{22}$$

if $z = L$ $$\frac{\partial T}{\partial z} + h_2 T = h_2 T_k \tag{23}$$

if $r = r_a$ $$\frac{\partial T}{\partial r} = g(z) \quad (d_1 \text{ model}) \tag{24}$$

if $r = r_a$ and $z = 0 \div (-L_l)$ $$\frac{\partial T}{\partial r} + h_4 T = h_4 T_z$$

and $z = 0 \div L$ $$\frac{\partial T}{\partial r} = g(z)$$

$(d_2$ model) (25)

if $r = r_b$ $$\frac{\partial T}{\partial r} + h_3 T = h_3 T_{ot}(z) \tag{26}$$

where :

$$h_1 = \frac{\alpha_1}{k} \quad h_2 = \frac{\alpha_2}{k} \quad h_3 = \frac{\alpha_3}{k} \quad h_4 = \frac{\alpha_4}{k} \tag{27}$$

$\alpha_1 \div \alpha_4$ - heat absorption coefficients on the surfaces of $z = 0$, $z = L$, $r = r_b$ and $r = r_a$ , respectively
$k$ - heat conduction coefficient of the sleeve material
$T_k$ - environmental temperature (Fig.1)assumed at the surface $z = L$ equal to the temperature of oil outflow from the bearing interspace, calculated from the eq. (10)
$T_{ot}(z)$ - environmental temperature assumed at the surface $r = r_b$
$g(z)$ - temperature gradient [°C/m] on the surface $r = r_a$.

Moreover the functions $T_{ot}(z)$ and $g(z)$ are assumed of the discrete form of pairs: [ $z_i$ , $g(z_i)$], [$z_i$ ,$T_{ot}(z_i)$] , $i$ = 1, 2,...j.

## COMPUTER CALCULATION PROGRAM

The calculation program presented in Fig.4 in the form of block diagram and written in the FORTRAN language, is intended for carrying out the calculations of :

- pressure and temperature distributions of oil in the bearing interspace of the seal
- temperature distribution on the external and internal surfaces of the sleeve
- distribution of thermoelastic deformations of the sleeve
- stress distribution within the sleeve
- height distribution of the bearing interspace
- hydrodynamic force components
- eccentricity of the mutual sleeve and shaft location
- leakage output and friction power
- static characteristics of the seal.

The program contains the input data check block which signals the possible errors connected with introducing inadequate units or out-of-range numbers. A FEM input data generator is also included to form the data on the basis of a characteristic input information and mesh parameter. A mesh of FEs and nodes as well as the temperature is generated for an assumed sleeve subdivision in compliance with the regularity principle and solution convergence conditions for a given type- and-size class of the seal in question.

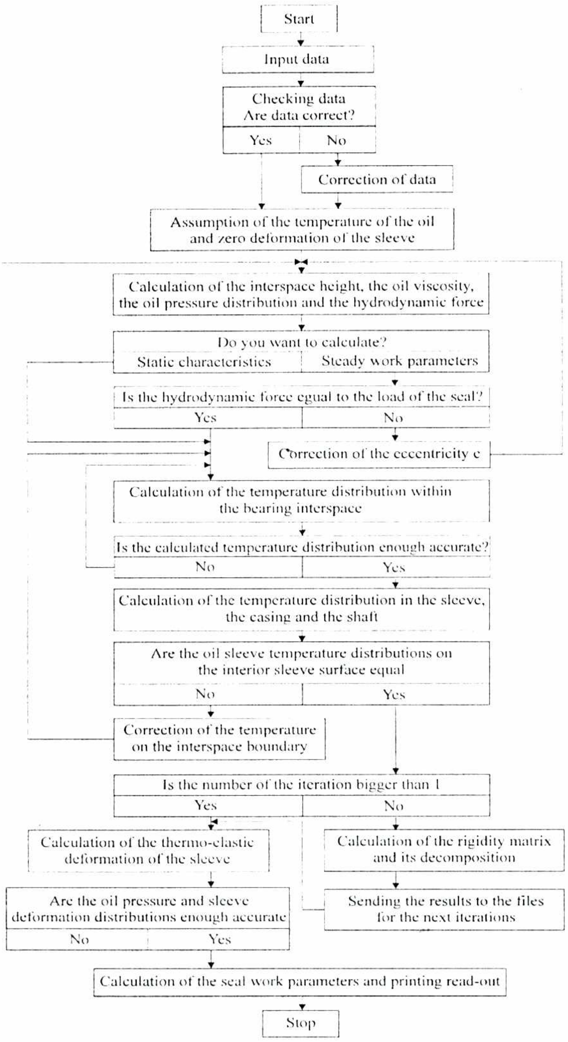


*Fig.4. Block diagram of the calculation program*

## CALCULATION RESULTS

The performed calculations were aimed at assessment of quality of the presented method. It was achieved by comparing the actual results with those obtained by applying the earlier elaborated and experimentally verified method based on the adiabatic model of oil flow through the bearing interspace [3]. The calculations were carried out for the following values of the design and operation parameters (see Fig.1) :

$D = 0.6$ m $\quad D_1 = 0.64$ m $\quad D_2 = 0.68$ m $\quad D_3 = 0.702$ m

$D_4 = 0.716$ m $\quad D_6 = 0.856$ m $\quad L_1 = 0.01$ m $\quad L_2 = 0.103$ m

$L_3 = 0.113$ m $\quad L_4 = 0.130$ m $\quad C = 0.000105$ m

- mass applied to the seal : MASA = 600 kg
- linear heat expansion coefficient of the sleeve material (steel) : ALF = $0.11 \cdot 10^{-4}$ 1/K
- heat conduction coefficient of steel : $k_0 = 52$ W/m/K
- heat absorption coefficients within the middle part of the circumferential ducts of the sleeve : $\alpha_1 = \alpha_4 = 253$ W/m²/K
- heat absorption coeffcient of the external face surfaces of the sleeve and casing : $\alpha_2 = 200$ W/m²/K
- heat absorption coefficient of the external cylindrical surface of the casing : $\alpha_3 = 10$ W/m²/K
- oil dynamic viscosity at the temperature $T_{od} = 50°C$ $\mu_0 = 0.04$ N·s/m²
- oil specific heat : $c_v = 2000$ J/kg/K
- oil density : $\rho = 900$ kg/m³.

Subsequent calculations in accordance with the presented method were carried out for the same input data and two different models of heat flow through the sleeve and casing mutually contacted.

In the first of them the axial heat flow towards the cold oil in the middle part of the sleeve and casing is assumed to be effected through the entire face surface of the elements. This model better reflects heat exchange conditions in the typical, unloaded radial seals whose circumferential ducts are of a large cross-section area, and in consequence their heat exchange area is also relatively large.

In the second model the middle part of the above mentioned elements is assumed free of the circumferential ducts and heat flow from the sleeve to the cold oil to be effected along the circumferential duct in the radial direction only. The axial heat flow between oil and the sleeve and casing is effected on the external side-face surfaces only. This model is more suitable for calculating the not-unloaded seals applicable to small diameter shafts, or unloaded ones and having the middle circumferential ducts of relatively small lateral dimensions.

The calculation results are graphically presented in Fig.5 where the two models are distinguished by $d_1$ and $d_2$ respectively. Additionally the calculation results, marked by „a", are included which are obtained from the earlier mentioned method based on the model of adiabatic oil flow through the bearing interspace and one-dimensional axial heat flow through the sleeve.

The results of calculations for the above specified design parameters indicated a stable performance of the seal within the entire investigated range of supply oil pressure changes and of shaft rotation speed.

The calculated value of the relative eccentricity defined as the ratio of the shaft-to-sleeve distance and the design radial clearance was very small : not exceeding 0.025 even at the relatively low supply pressure of 6 MPa. It means that the mutual location of the sleeve and shaft was practically co-axial, in compliance with the experimental results.

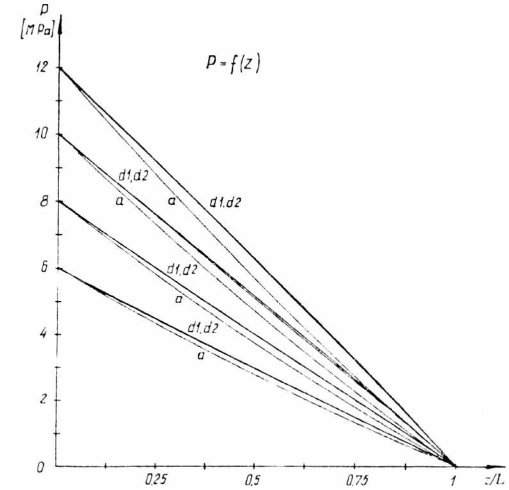


*Fig.5. Pressure distribution of oil in the bearing interspace, calculated according to three oil flow models ($d_1$, $d_2$, a) and the following data: $n_r = 2$ rps, $T_z = 45°C$ and $p_z = 6, 8, 10$ and 12 MPa*

In Fig.5 the distribution of the average (circumferentially) pressure of oil in the bearing interspace is presented. The curves calculated in compliance with both models $d_1$, $d_2$ of the diathermal flow practically cover each other. It means that the way of modelling the heat flow through the sleeve in the circumferential duct region is almost of no importance in this case. Distinct differences in shape of the curves, which grow along with the supply oil pressure increase, are observed only for the different models of oil flow through the bearing interspace. It results mainly from that the oil temperature in the bearing interspace, calculated by means of the diathermal model, grows, in consequence of the heat transference to the sleeve and shaft being accounted for, to a smaller extent than in the case of the adiabatic model, that can be observed in Fig.6. It causes a far less drop of oil viscosity and somewhat smaller thermal deformations of the sleeve and in result a changed course of pressures.

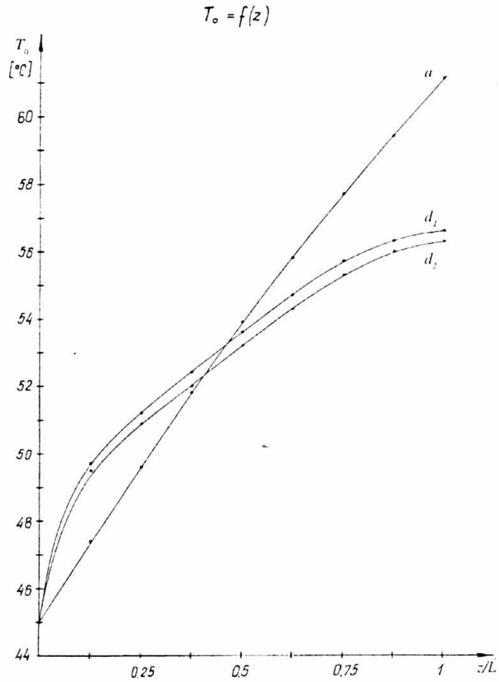


*Fig.6. Distribution of the average oil temperature in the bearing interspace, calculated according to three oil flow models ($d_1$, $d_2$, a) and the following data: $n_r$ = 2 rps, $T_z$ = 45°C and $p_z$ = 12 MPa*

The oil temperature distribution in the bearing interspace, similarly to the case of the pressure distribution, depends first of all on whether the heat exchange is or is not taken into account, and -to a very small, however noticeable extent - on the assumed conditions of heat flow through the sleeve. As expected, accounting for the middle sleeve ducts in the calculations leads to obtaining a somewhat lower oil temperature.

A greater influence of the assumed model of heat flow through the sleeve and casing can be observed on the temperature distributions over the sleeve cylindrical surfaces, shown in Fig.7.

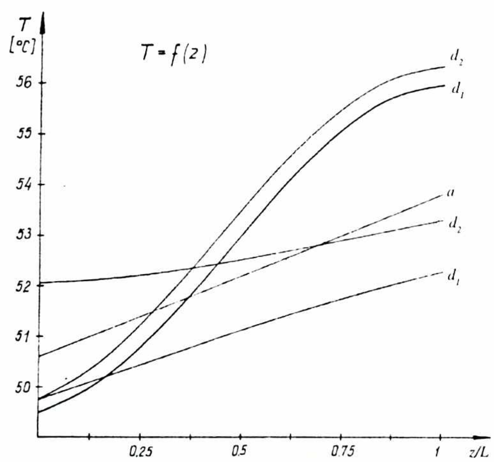


*Fig.7. Temperature distribution on the sleeve surfaces, calculated according to three oil flow models ($d_1$, $d_2$, a) and the following data: $n_r$ = 2 rps, $T_z$ = 45°C and $p_z$ = 12 MPa*

The temperature on the sleeve internal surface, calculated for both diathermal models still differs only slightly, but the temperature differences on the external surface are of even 2°C.

The temperature values calculated from the adiabatic model (concering the full sleeve thickness) fit within the temperature range obtained from the diathermal models and are close to their average (thickness-wise) temperature values. Attention should be paid however to a smaller increment of the sleeve temperature based on the adiabatic model in comparison with the average ( thickness- wise) sleeve temperature based on both diathermal models. It results from neglecting the heat exchange in the bearing interspace, namely : heat abstraction at the interspace entrance and heat absorption along the remaining part of it. The phenomenon is illustrated in Fig.8 by the oil temperature distribution within the bearing interspace between the sleeve and rotating shaft.

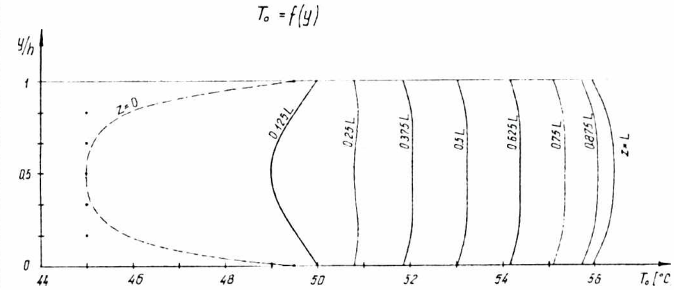


*Fig.8. Distribution of the oil temperature in the bearing interspace, calculated according to the oil flow model $d_1$ and the following data: $n_r$ = 2 rps, $T_z$ = 45°C and $p_z$ = 12 MPa*

Each of the curves deals with a successive, equally spaced cross-section of the bearing interspace, counting from its entrance towards outlet. To simplify the drawing, the interspace walls are depicted parallel. The first curve of $z = 0$ is depicted without any accurate fitting of points to make it more like its real course. The points put on the drawing present the temperatures which form the boundary condition for calculating the remaining temperature values. The temperature gradients observed at the inlet of the cold oil flow into the narrow interspace are the greatest, therefore cooling of this part of the sleeve and shaft is the greatest too. Changing the curve slope directions can be observed beginning from third curve, which means that heat is transferred from the oil to the sleeve and shaft. The close-to-symmetrical shape of the curves should be stressed because it is connected with the temperature on the shaft surface, equal to that on the internal sleeve surface, as it is assumed in the calculation model.

The amount of heat generated within the bearing interspace depends mainly on shaft speed. In Fig.9 the distribution curves are presented of the average (circumferentially) oil temperature in the interspace at five shaft speed values.

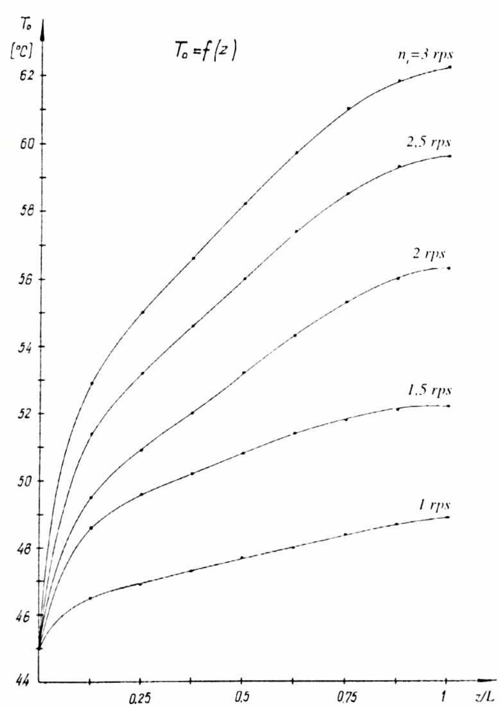


*Fig.9. Distribution of the oil temperature in the bearing interspace, calculated according to the oil flow model $d_1$ and the following data: $n_r$ = 1, 1.5, 2, 2.5 and 3 rps, $T_z$ = 45°C and $p_z$ = 12 MPa*

A distinct shaft speed influence on oil temperature increase in the interspace can be observed. Taking into account that calculations in compliance with the adiabatic model provide much higher values of the temperatures, one can confirm that elaboration of diathermal models is necessary for the water turbine seals where shaft speed can reach even 4 rps.

The influence of the supply oil pressure on the oil temperature in the interspace is low, as it can be observed in Fig.10.

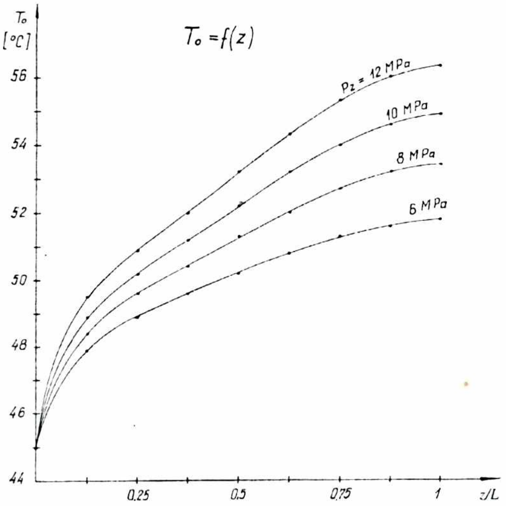


*Fig.10. Distribution of the oil temperature in the bearing interspace, calculated according to the oil flow model $d_1$ and the following data: $n_r$ = 2 rps, $T_z$ = 45°C and $p_z$ = 6, 8, 10 and 12 MPa*

The oil supply pressure more distinctly influences the sleeve elastic deformations and thus the bearing interspace height. In Fig.11 the interspace shapes are shown at four oil pressure values.

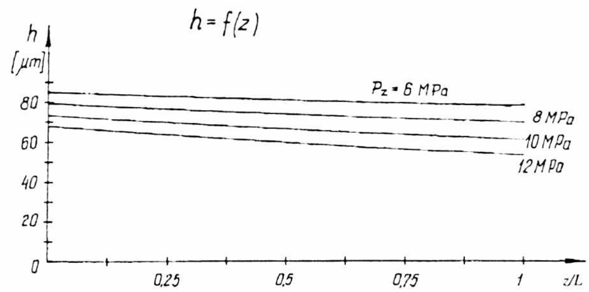


*Fig.11. Axial shape of the bearing interspace, calculated in accordance with the oil flow model $d_1$ and the following data: $n_r$ = 2 rps, $T_z$ = 45°C and $p_z$ = 6, 8, 10 and 12 MPa*

Increasing the supply oil pressure in the calculated seal results in lowering the height of the interspace and changing its shape simultaneously. Elastic deformations of the sleeve are very small at a low supply oil pressure therefore the interspace shape is close to the design parallel clearance based on the assumed fit of the sleeve against shaft.

Increasing the pressure results in increasing the sleeve elastic deformations which means decreasing the interspace height with distinct growing of the convergence of shape. In this case the shape depends on the location of the sealing rings that decide about distribution of the pressure applied to the axial cross-section of the sleeve (see Fig.1).

It should be mentioned however that the interspace shape will be probably somewhat different if the calculations are performed at different, particularly higher, shaft speeds. Increasing the speed will cause the interspace convergence to drop.

In Fig.12 the axial shapes of the bearing interspace are presented, calculated in compliance with the three flow models. As expected, taking into account the heat exchange within the interspace and middle circumferential ducts of the sleeve leads to obtaining the lowest height of the interspace with the relatively high shape convergence maintained.



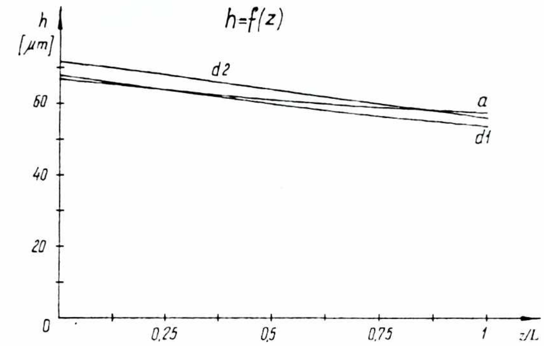


*Fig.12. Axial shape of the bearing interspace, calculated in compliance with three oil flow models ($d_1$, $d_2$, a) and the following data: $n_r$ = 2 rps, $T_z$ = 45°C and $p_z$ = 12 MPa*

# FINAL REMARKS

♦ The analytical solution of the conduction equation of the two-dimensional heat flow through the sleeve and seal casing, described in [6], made it possible to put into operation the complete calculation program based on the diathermal model of oil flow through the bearing interspace, with three-dimensional temperature distribution and oil viscosity accounted for.

♦ The above presented results of calculations makes positive assessment of the physical model and calculation method applied to the considered phenomena justified, and in consequence practical applications of the elaborated calculation program to designing the CP propellers and Kaplan water turbines possible.

♦ The program is especially suitable for calculating the operation parameters of the seals under high thermal loading, e.g. in such cases as :

- in high-load propellers of naval vessels
- at high shaft speeds, typical of water turbines
- during emergency operation of the CP propeller with blocked pitch control when the supply oil pressure is as low as to effect fading the axial oil flow through the bearing interspace and thus the heat abstraction too
- the seals with thin-walled sleeves described in [5].

The thermal phenomena occurring in the above mentioned cases affect performance of the seal to a greater extent than other ones and therefore application of the diathermal model to calculating such seals is fully justified.

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