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DEFORMACJE TERMICZNE WAHLIWYCH PŁYTEK W 5-SEGMENTOWYM ŚLIZGOWYM ŁOŻYSKU POPRZECZNYM

THERMAL DEFORMATION OF PADS IN TILTING 5-PAD JOURNAL BEARING

Ślizgowe łożyska poprzeczne z segmentami wahliwymi stosowane są w wysokoobrotowych sprężarkach oraz turbinach. Łożyska poprzeczne z płytkami wahliwymi stanowią poprawne rozwiązanie, ponieważ zapewniają bardzo dobrą stateczność hydrodynamiczną w zakresie dużych prędkości oraz są mniej wrażliwe na zmiany kierunku obciążenia oraz niewspółosiowość wału. W artykule przedstawiono deformacje segmentów wahliwego 5-segmentowego ślizgowego łożyska poprzecznego z podparciem symetrycznym wywołane polem ciśnienia oraz rozkładem temperatury w warunkach adiabatycznego filmu olejowego. Deformacje segmentów wyznaczone zostały na podstawie rozkładu ciśnienia oraz temperatury w filmie olejowym. Równania Reynoldsa, bilansu energii, geometrii szczeliny olejowej oraz lepkości oleju zostały rozwiązane metodą numeryczną przy założeniu współosiowości wału oraz panewki łożyska w stanie równowagi położenia wału.

Słowa kluczowe: *ślizgowe łożyska poprzeczne z segmentami wahliwymi, obliczenia symulacyjne, MES.*

In high speed compressors and turbine drive trains, the tilting 5-pad journal bearings are applied. Tilting-pad journal bearings are good option because they have very good hydrodynamic stability at high speed and are less sensitive to load direction and shaft misalignment. The paper introduces thermo-elastic deformations of tilting 5-pad journal bearing with symmetric support of pads and operating at the conditions of adiabatic oil film. The deformations of pads were obtained based on the oil film pressure and temperature distributions. Reynolds, energy, geometry and viscosity equations have been solved numerically on the assumption of aligned orientation of bearing and journal axis and at static equilibrium position of journal.

Keywords: *tilting-pad journal bearings, numerical simulation, FEM.*

1. Introduction

Performances of turbomachines and compressors tilting-pad journal bearings are affected by thermal and elastic deformation of pads[1,2,5,6]. These deformation affect the static and dynamic characteristics and the stability of the rotor. For the failure-free operation of bearing, more information regarding thermo-elastic deformation should be available at the early design stage of the bearing. Thermo-elastic effects are important in the evaluation of oil film thickness, maximum oil film pressure and temperature i.e. bearing characteristics deciding about reliable operation of single bearing and the bearing system.

Design variations of tilting-pad journal bearings [3] allow operation in modern, high speed, high output power turbounits or turbine gear trains. High thermal loads that are caused by operating conditions change the geometry of oil film and they generate the thermo-elastic deformations of pads. The pad deformation affects the bearing temperature and reduces the damping of bearing.

The paper describes the procedure of the calculations of thermo-elastic deformation of the tilting 5-pad journal bearing. The ground of deformation calculation are the oil film pressure and temperature distributions that were obtained from the numerical solution of Reynolds, energy and viscosity equations. Incompressible laminar and adiabatic flow of oil in the bearing gap of finite length bearing was assumed. Aligned orientation of bush and journal axis without deflections of pads and journal was considered too. Calculation have been performed at the condition of static equilibrium position of the journal.

2. Nomenclature

D bush diameter (m),
e journal eccentricity (m),
h oil film thickness (µm),
$\bar{H}$ dimensionless oil film thickness,
K_T heat number, $K_T = \omega \cdot \eta_o / c \cdot \rho \cdot T_o \psi^2$,
L bearing length,
$\bar{p}$ dimensionless oil film pressure, $\bar{p} = p \cdot \psi^2 / \eta \cdot \omega$,
Pe Peclet's number,
r journal radius (m),
R_b bush radius (m),
s number of pads,
So Sommerfeld number,
T temperature of oil film (ºC),
T_o temperature of supplied oil (ºC),
$\bar{T}$ dimensionless oil film temperature, $\bar{T} = T/T_o$,
t time,
$\bar{z}$ dimensionless axial co-ordinate,
τ_o angle of the lobe centre line (º),
τ_l angular orientation of centre point of pad (º),
η dynamic viscosity of oil, (Ns/m^2),
η_o dynamic viscosity at ambient temperature (Ns/m^2),
$\bar{\eta}$ dimensionless viscosity of oil, $\bar{\eta} = \eta/\eta_o$,
φ circumferential co-ordinate (º),

ε relative eccentricity, $\varepsilon = e/\left(R_B - r\right)$,
ψ bearing clearance (‰),
ψ_s pad relative clearance,
ω angular velocity (1/s),
ϕ dimensionless time, $\phi = \omega t$.

3. Pad thermal deformation

Geometry of considered tilting 5-pad journal bearings with the load on the pad (LOP) and the load between pads (LBP), is shown in Fig. 1. The geometry of lubricating gap determines equation 1:

$$\bar{H}(\varphi) = \psi_s + \frac{\psi_s - 1}{cos(\tau_1 - \tau_o)} \cdot cos(\varphi - \tau_1) - \varepsilon \cdot cos(\varphi - \alpha) \quad (1)$$

Pressure, temperature and viscosity distributions of the oil film were determined on the basis of Reynolds, energy and viscosity equations [5]. The generated oil film pressure field is described by the following non dimensional form of Reynolds equation:

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

The oil film pressure distribution computed from equation (2) to obtain the oil film temperature and viscosity fields [2,3,6]:

$$\begin{aligned} &\frac{\bar{H}}{Pe}\left[\frac{\partial^2 \bar{T}}{\partial \varphi^2} + \left(\frac{D}{L}\right)^2 \frac{\partial^2 \bar{T}}{\partial \bar{z}^2}\right] + \left[\frac{\bar{H}^3}{12\bar{\eta}}\frac{\partial \bar{p}}{\partial \varphi} - \frac{\bar{H}}{2}\right]\frac{\partial \bar{T}}{\partial \varphi} + \\ &+\left(\frac{D}{L}\right)^2 \frac{\bar{H}^3}{12\bar{\eta}}\frac{\partial \bar{p}}{\partial \bar{z}}\frac{\partial \bar{T}}{\partial \bar{z}} = -\frac{\bar{H}^3}{12\bar{\eta}}\left[\left(\frac{\partial \bar{p}}{\partial \varphi}\right)^2 + \left(\frac{D}{L}\right)^2\left(\frac{\partial \bar{p}}{\partial \bar{z}}\right)^2\right] - \frac{\bar{\eta}}{\bar{H}} \end{aligned} \quad (3)$$

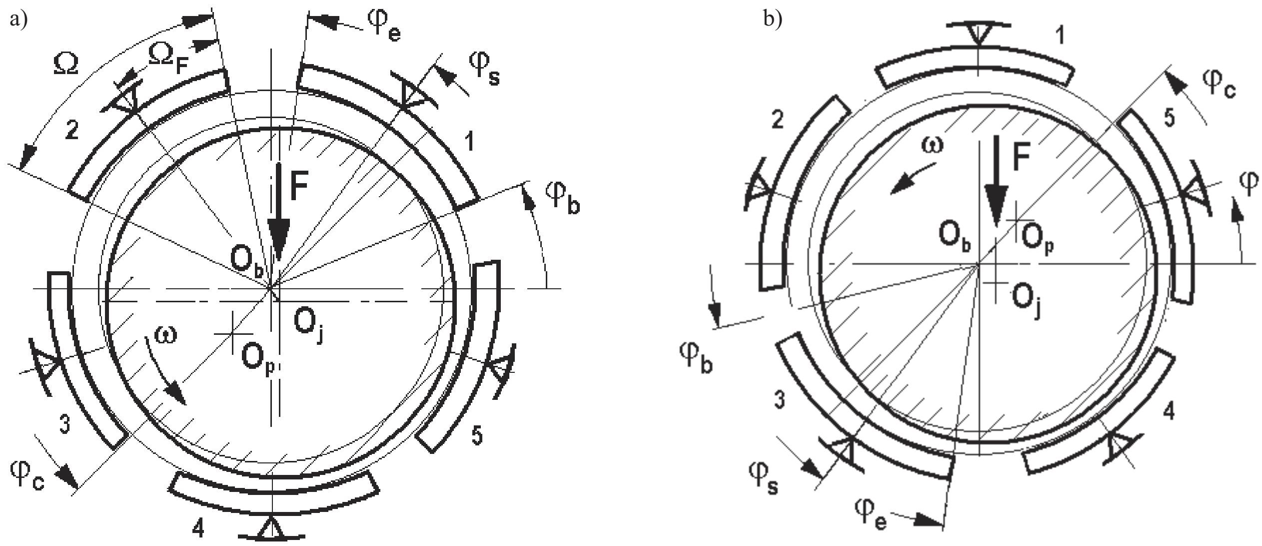
The viscosity was described by exponential equation [3,4,5]. Oil film temperature and viscosity distributions have been found by iterative solution of equations (1), (6) and (8) [3]. Temperature values $T(\varphi, z)$ on the boundaries ($z = \pm L/2$) have been determined by means of the parabolic approximation [3,6]. The boundary conditions for pressure and temperature take into account the inlet pressure and temperature.

Examples of the results of oil film temperature calculations are given in Fig.2 and Fig.3. Turbine oil of viscosity η_o = 0,0487 Ns/m^2 at 40ºC was used in this calculations. The pressure boundary condition assumes the positive values only and the ambient pressure on the sides of the pad. Calculation of the temperature on the sides of the pad was determined by means of parabolic approximation [2].

Elastic deformation of the tilting-pads of bearing loaded uniformly by oil film pressure and temperature can be obtained by analytical or by finite element method (FEM) [6]. The system I-DEAS NX Series [7] was applied in this work. The load of pad sliding surface in the form of oil film pressure and temperature distribution in the bearing gap was modeled based on the results obtained from the numerical calculation [2,5]. The model of pad takes into consideration full length of pad. It was assumed that: the temperature of bearing metal surface is equal to the temperature of oil film, outer surface of pad contacts the air, temperature state of the pad structure is steady (state thermal analysis), the mixed boundary condition of heat exchange according to Newton's law (mixed boundary condition of Hankels) for the temperature calculation on the outer surface of pad. Examples of deformation calculation LBP variant are shown in Fig.4 (by pressure only) and Fig.5 (by temperature only).

In case of bearing with the length to diameter ratio L/D=0,4 and oil supplied temperature 50ºC the results of displacements calculations can be observed in Fig.6 and Fig.7. There are larger total displacement of pad No.4 but this pad is the most loaded mechanically and thermally pad of the bearing. The radial displacements are larger for the pad No.4, too.

The oil film temperature distribution on the pad can be observed in Fig.8 and Fig.9, too. These temperatures were assumed for the displacements calculations. The maximum displacements in the middle plain of pads are shown in Fig.9 and Fig.10.



Rys. 1. Schemat 5-segmentowego ślizgowego łożyska poprzecznego a)obciążenie na segmencie (LOP), b)obciążenie pomiędzy segmentami (LBP)
Fig. 1. Layout of tilting 5-pad journal bearing: a) load on the pad (LOP), b) load between pads (LBP)

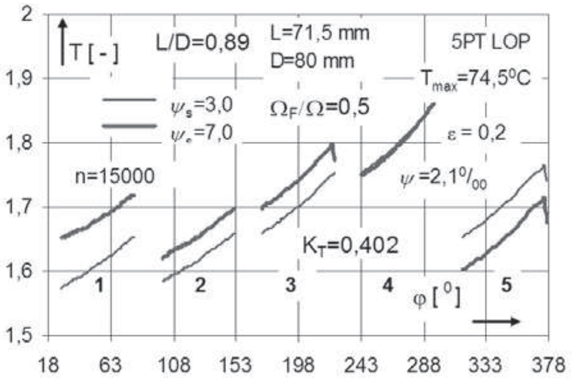


Fig. 2. Oil film temperature distribution of 5-pad bearing LOP (dimensionless)

Rys. 2. Rozkład temperatury w szczelinie olejowej 5-segmentowego łożyska LOP

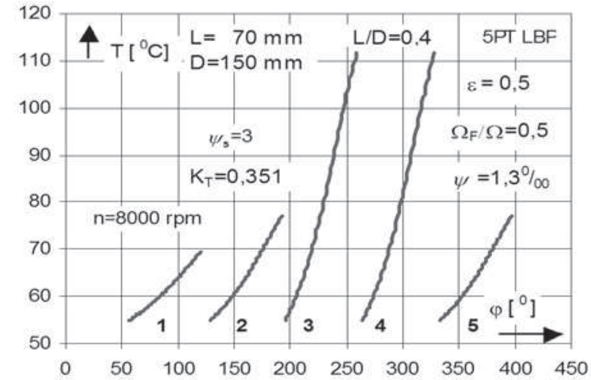


Fig. 3. Oil film temperature distribution of 5-pad bearing LBP

Rys. 3. Rozkład temperatury w szczelinie olejowej 5-segmentowego łożyska LBP

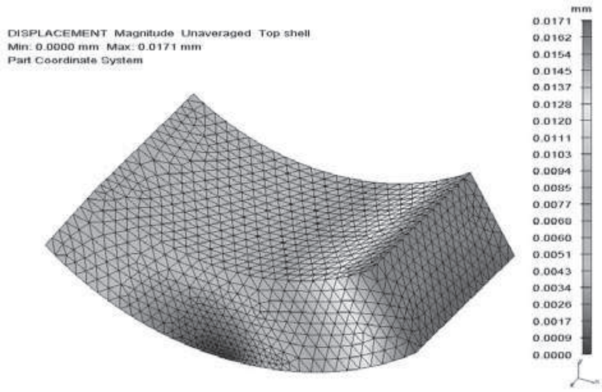


Fig. 4. Deformation of pad No.4 caused by pressure

Rys. 4. Deformacje segmentu Nr4 wywołane ciśnieniem

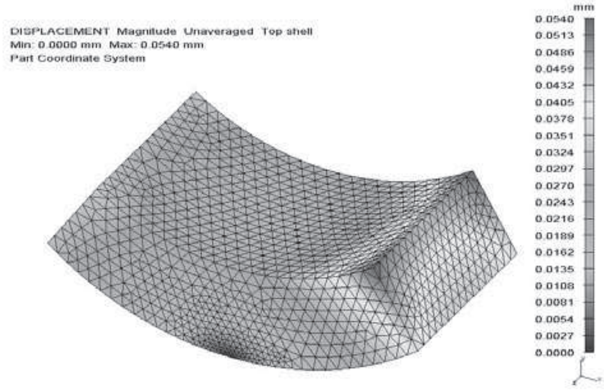


Fig. 5. Deformation of pad No.4 caused by temperature

Rys. 5. Deformacje segmentu Nr4 wywołane polem temperatury

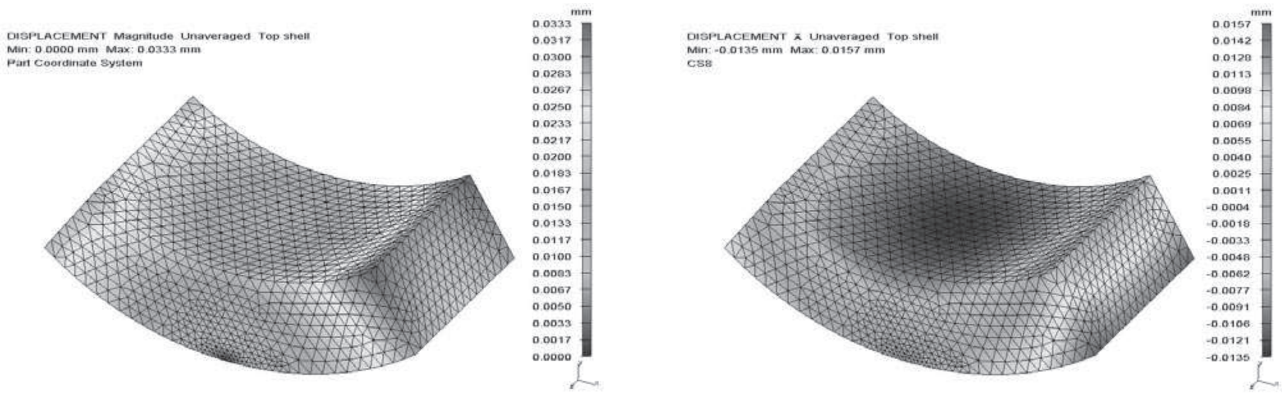


Fig. 6. Maximum displacements (0,0333 mm) and radial one (0,0157 mm) of the 1st pad (least loaded one) generated by the oil film pressure and temperature

Rys. 6. Przemieszczenia (0,0333 mm) oraz przemieszczenia promieniowe (0,0157 mm) segmentu Nr1 (najmniej obciążonego) wywołane przez film olejowy i temperaturę

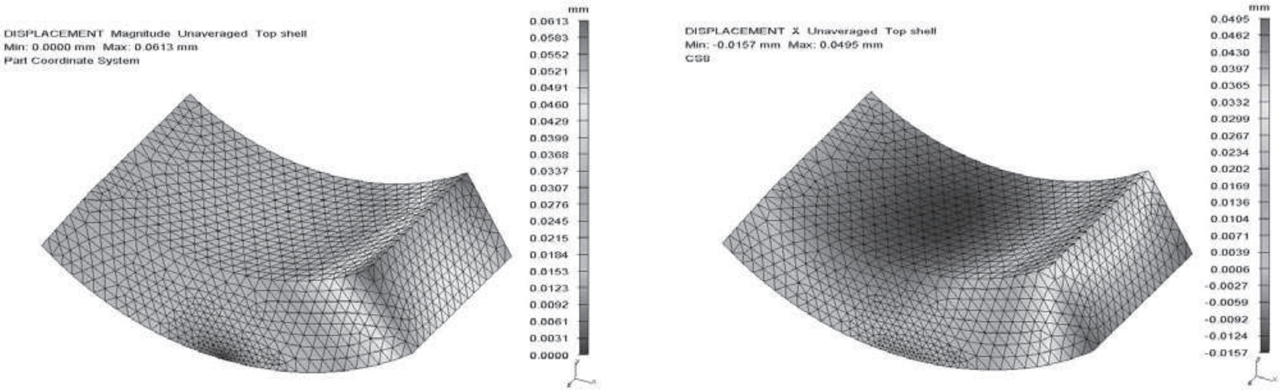


Fig. 7. Maximum displacements (0,0613 mm) and radial one (0,0495 mm) of the 4th pad (most loaded one) generated by by the oil film pressure and temperature

Rys. 7. Przemieszczenia (0,0613 mm) oraz przemieszczenia promieniowe (0,0495 mm) segmentu Nr4 (najbardziej obciążonego) wywołane przez film olejowy i temperaturę

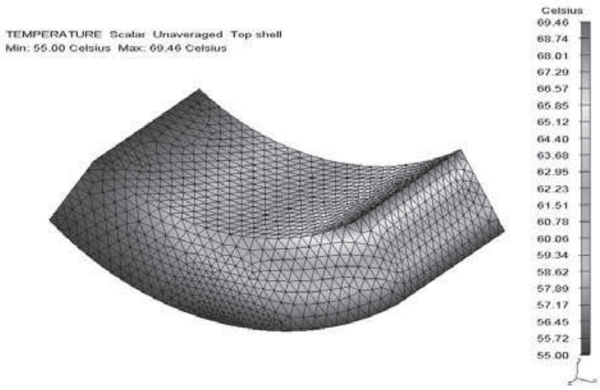


Fig.8. Oil film temperature on the 1st pad (T_{max}=69,49ºC)

Rys. 8. Rozkład temperatury w segmencie Nr1 (T_{max}=69,49ºC)

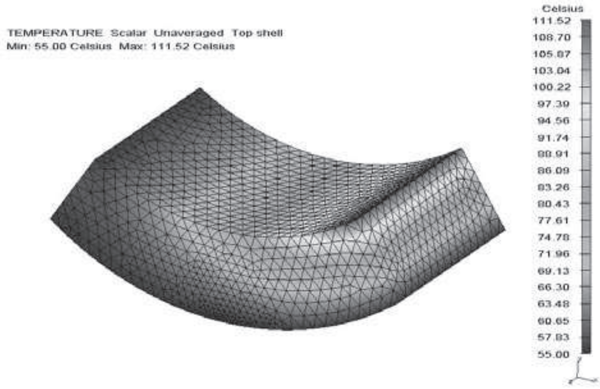


Fig.9. Oil film temperature on the 3rd pad (T_{max}= 111,51ºC)

Rys. 9. Rozkład temperatury w segmencie Nr3 (T_{max}=111,51ºC)

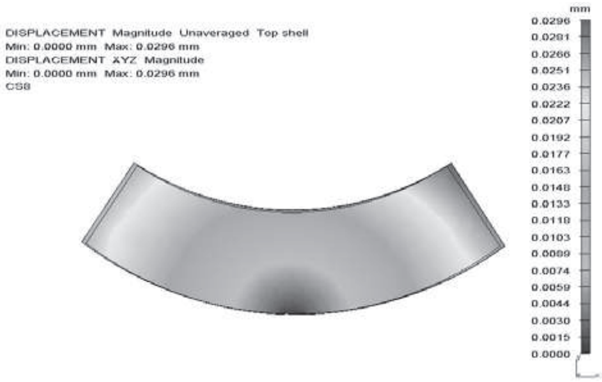


Fig. 10. Maximum displacement (0,0296 mm) in the middle cross-section of 1st pad

Rys. 10. Przemieszczenia (0,0296 mm) w płaszczyźnie środkowej segmentu Nr1

Fig. 11. Maximum displacement (0,0518 mm) in the middle cross-section of 4th pad

Rys. 11. Przemieszczenia (0,0518 mm) w płaszczyźnie środkowej segmentu Nr 4

4. Conclusions

Program developed for calculation of tilting-pad bearing static characteristics at the conditions of static equilibrium position of journal gives correct input data for determination of the thermo-elastic deformation of pad.

The procedure of calculation consisting in application of own developed thermo-hydrodynamic lubrication program and finite-element one have been proved successfully.

Developed procedure of static characteristics and elastic deformation calculation can be used for determination of deformation of any realistic bearing form.

5. References

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