INTRODUCTION
Performances of turbomachines and compressors tilting-pad journal bearings are affected by thermal and elastic deformation of pads[1-4]. 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 [5] 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.

NOMENCLATURE
D bush diameter (m) 
$e$ journal eccentricity (m) 
h oil film thickness (μm) 
$\tilde{H}$ dimensionless oil film thickness 
$K_T$ heat number, $K_T=\frac{w \times h}{c \times r \times T_0 \times y^2}$ 
$L$ bearing length 
$\tilde{p}$ dimensionless oil film pressure, $\tilde{p} = p \cdot \psi^2 / \eta \cdot \omega$ 
r journal radius (m) 
$R_b$ bush radius (m) 
s number of pads 
$S_0$ Sommerfeld number, $S_0 = F \cdot \psi^2 / (L \cdot D \cdot \eta \cdot \omega)$ 
t time 
$\tilde{\tau}$ dimensionless axial co-ordinate 
$\tau_c$ angle of the lobe centre line (°) 
$\tau_1$ angular orientation of centre point of pad (°) 
$\eta$ dynamic viscosity of oil, (Ns/m²) 
$\eta_o$ dynamic viscosity at ambient temperature (Ns/m²) 
$\tilde{\eta}$ dimensionless viscosity of oil, $\tilde{\eta} = \eta / \eta_o$ 
$\varphi$ circumferential co-ordinate (°) 
$\varepsilon$ relative eccentricity, $\varepsilon = e / (R_b - r)$ 
$\psi$ bearing clearance, %e 
$\psi_c$ pad relative clearance, 
$\omega$ angular velocity (1/s) 
$\phi$ dimensionless time, $\phi = \omega t$
PAD THERMAL DEFORMATION

Geometry of considered tilting 5-pad journal bearing with the load on the pad (LOP), is shown in Fig. 1.

![Figure 1 Lay-out of tilting 5-pad journal bearing](image)

The geometry of lubricating gap determines Eqn. (1).

\[
\bar{H}(\varphi) = \psi_1 + \frac{\psi_2 - 1}{\cos(\tau_1 - \tau_2)} \cdot \cos(\varphi - \tau_1) - \varepsilon \cdot \cos(\varphi - \alpha)
\]

Pressure, temperature and viscosity distributions of the oil film were determined on the basis of Reynolds, energy and viscosity equations [2]. The generated oil film pressure field is described by the following non dimensional form of Reynolds equation:

\[
\frac{\partial}{\partial \varphi} \left( \frac{H^3}{\bar{F}} \frac{\partial \bar{p}}{\partial \varphi} \right) + \left( \frac{D}{L} \right)^2 \frac{\partial}{\partial \varepsilon} \left( \frac{H^3}{\bar{F}} \frac{\partial \bar{p}}{\partial \varepsilon} \right) = 6 \cdot \frac{\partial \bar{H}}{\partial \varphi} + 12 \frac{\partial \bar{H}}{\partial \varepsilon}
\]

(2)

The oil film pressure distribution computed from Eqn. (2) was put into energy equation to obtain the oil film temperature and viscosity fields [3-5]. An example of oil film temperature calculation results is given in Fig. 2. Turbine oil of viscosity \(\eta_0=0.0487 \text{ Ns/m}^2\) at 40\(^\circ\)C 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 [3].

![Figure 2 Oil film temperature distribution of 5-pad bearing](image)

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) [4]. The system CAD/CAM I-DEAS Master Series v.7m2 [6] 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 modelled based on the results obtained from the numerical calculation [2, 3]. 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 are shown in Fig. 3 and Fig. 4.

![Figure 3 Deformation of pad No. 4 caused by the pressure](image)

![Figure 4 Deformation of pad No.4 caused by pressure and temperature](image)

CONCLUSIONS

1. 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.
2. The procedure of calculation consisting in application of own developed thermo-hydrodynamic lubrication program and finite-element one have been proved successfully.
3. Developed procedure of static characteristics and elastic deformation calculation can be used for determination of deformation of any realistic bearing form.

REFERENCES