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A NUMERICAL SIMULATION OF THRUST BEARING PERFORMANCE WITH CONSIDERATION OF THE PAD ELASTIC DEFLECTIONS

Summary. The paper presents results of numerical calculations of hydrodynamically lubricated thrust bearing with elastically deformed pads. Was taken into account two types of thrust pads : the stationary one and tilting one. The finite element method system "Algor" was used to calculate bearing sector pads deflections. The influence of bearing load, the dimensionless relative width of bearing B*= B/L and the thickness of the pad on the oil film properties have been examined.

1. The bearing and oil film geometry

In the theoretical researches the two design modifications of sector thrust bearing were considered : with stationary and tilting segments. The mentioned above types of the bearing pads are initially flat and then deflected under the oil film pressure distribution. The bearing and the oil film geometry are shown in Fig.1.

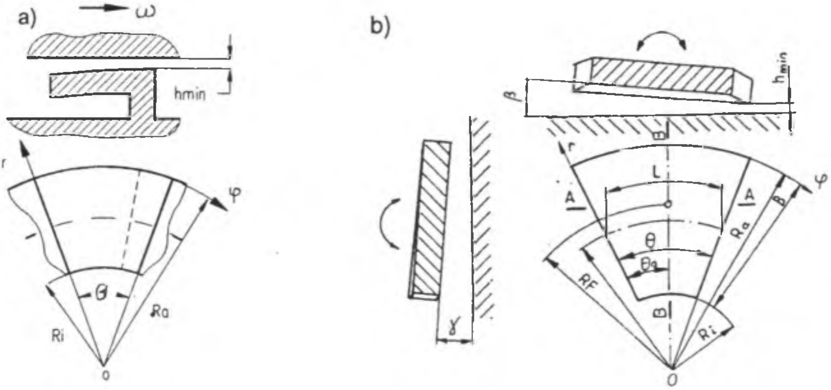


Fig.1.The bearing and oil film geometry : a) stationary pad b) tilting pad

The shape of the oil film for above mentioned modifications of the bearing pads is described by the equations (1) and (2); where disp(r,φ) is the value of deflection, h_{min} – is the minimum film thickness

$$h(r,\varphi) = h_{min} + disp(r,\varphi) \tag{1}$$

$$\begin{aligned} h(r,\varphi) = h_{min} &+ tg\beta\left[r\sin(\theta-\varphi) + R_a\sin(\theta-\theta_0)\right] + \\ &+ tg\gamma\left[r\cos(\theta_0-\varphi) + R_a\cos(\theta-\theta_0) - R_F - R_m\right] + disp(r,\varphi) \end{aligned} \tag{2}$$

The dependence (1) refers to the stationary pads and the equation (2) to the case of tilting segments.

2. Basis equations

In the paper is assumed the stationary load of the thrust bearing and laminar oil flow in the bearing clearance. The pad of a finite width is considered. It is assumed that heat transfer within the oil film takes place in the way of convection and does not exist heat flow from the oil film to surroundings. The changes of temperature across the oil film thickness were neglected. The oil viscosity depends on the temperature in agreement with the relationship :

$$\eta = \eta_0 e^{A(T-T_0)+B(T-T_0)^2} \tag{3}$$

where A, B are the constants which are determined experimentally, T is the temperature of oil film, T_0 is the fiducial temperature.

Basing of the accepted assumptions the partial differential equations :

$$\frac{\partial}{\partial r}\left(\frac{r h^3}{\eta}\frac{\partial p}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial\varphi}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial\varphi}\right) = 6\omega r\frac{\partial h}{\partial\varphi} \tag{4}$$

and

$$\rho c_v\left(q_r\frac{\partial T}{\partial r} + q_\varphi\frac{\partial T}{r\partial\varphi}\right) = \frac{h^3}{12\eta}\left[\left(\frac{\partial p}{\partial r}\right)^2 + \left(\frac{\partial p}{r\partial\varphi}\right)^2\right] + \frac{(r\omega)^2\eta}{h} \tag{5}$$

express the changes of the pressure (eq.4) and the temperature (eq.5) in the bearing clearance, where p is the oil film pressure, q_r and q_φ are the oil flows in radial and circumferential directions :

$$q_r = -\frac{h^3}{12}\eta\frac{\partial p}{\partial r} \tag{6}$$

$$q_\varphi = -\frac{h^3}{12\eta r}\frac{\partial p}{\partial\varphi} + \frac{rh}{2} \tag{7}$$

3. Numerical solution procedure

In order to calculate the values of the pressure and temperature in the oil film, the equations (3) and (4) were transformed to the dimensionless form and then to the difference equations. The bearing pad deflections caused by the hydrodynamic pressure distribution were calculated using FEA system "Algor".

The scheme of the finite element mesh for the stationary and tilting pads shows Fig.2.

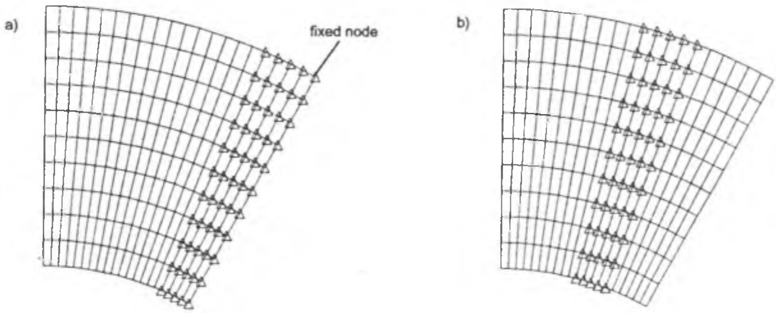
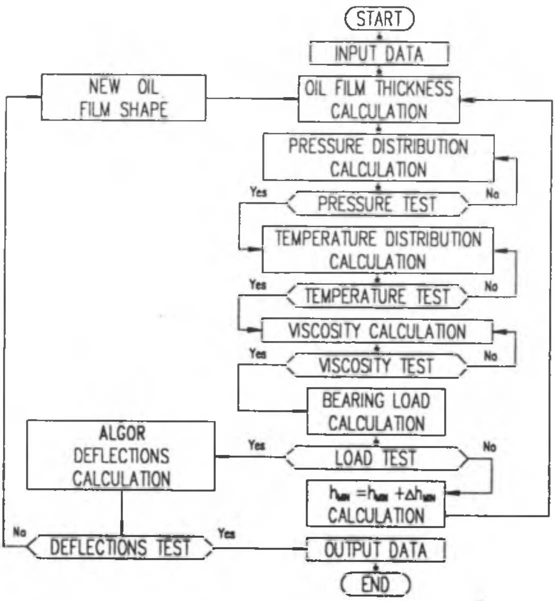


Fig.2.Scheme of finite element mesh for :

a) stationary pad

b) tilting pad



The scheme of the iterative procedure making possible the calculation of the pressure, the temperature, the oil film thickness and the elastic deflection values in the nodes of the finite difference mesh shows Fig.3 (in the case of the stationary thrust pads).

Fig.3. Scheme of numerical procedure.

The input data to the program are the following:

- bearing dimensions R_a , R_i , θ , θ_o
- angle γ
- oil viscosity η_o , oil density ρ_o , oil specific heat – c
- coefficients A, B (eq.1)
- fiducial temperature T_o , leading edge temperature T_p
- rotational speed ω
- number of pads N
- bearing load
- coordinates of the load position R_F , θ_F
- iteration accuraces

4. Numerical calculations

The numerical calculations were realised to examine the influence of :

- type of the pad
- segment stiffness
- bearing load
- relative width $B^* = \dfrac{B}{L}$

on the oil film properties. The computations were carried out for :

- 3 variable values of the pad thickness : H1=16 , H2=18 , H3=20 [mm]
- 2 variable values of the bearing load : F1=150 , F2=200 [kN]
- 5 variable values of B^* parameter : 1.2–2.0 with step 0.2

In Table 1 was shown the values of inner and outer radii (R_i and R_a) of the pads in dependence upon the B^* parameter (for number of segments N=10)

Table 1 Values of R_i , R_a radius of the pad

B*	1.2	1.4	1.6	1.8	2.0
R_i [m]	0.128	0.110	0.095	0.082	0.071
R_a [m]	0.238	0.228	0.221	0.216	0.212

For all computation variants was accepted : the oil viscosity η_o=0.109[Pa·s], coefficients A=–0.0572 , B=0.00025 [5], oil density ρ_o= 883.5[kg/m^3] , the leading edge temperature T_p=30[°C] and angular velocity of the rotor ω = 50[rad/s] . In Fig. 4 was shown the thrust pads deflections (in a contour map form) caused by the oil film pressure for the stationary and the tilting pads.

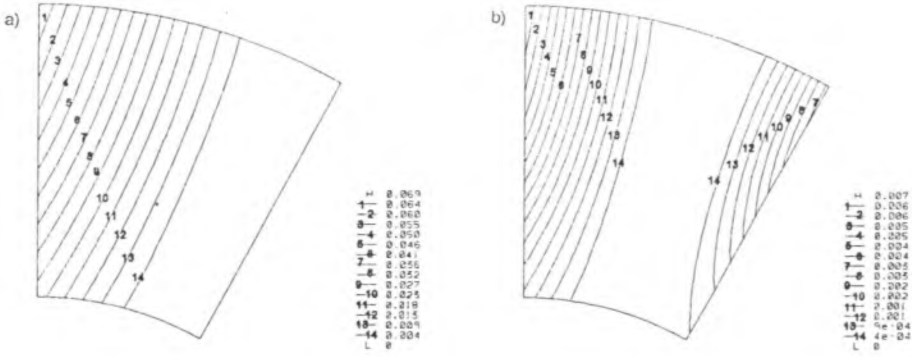


Fig.4.Thrust pad deflections distribution (for B^*= 1.2 , H= 16 [mm] , F= 200 [kN]) [mm].
a) stationary pad
b) tilting pad

Tables 2 and 3 give the results of computations : the minimum film thickness h_{min} , the coefficient of friction μ , the oil film temperature rise ΔT , the inlet oil flow Q_{in} and the oil film maximum pressure p_{max} .

Table 2 Results of computations in the case of stationary bearing pad

		F = 150 [kN]					F = 200 [kN]				
B^*		1.2	1.4	1.6	1.8	2.0	1.2	1.4	1.6	1.8	2.0
H=16 [mm]	h_{min} [μm]	36.8	36.3	35.8	36.1	36.4	31.7	31.2	31.2	31.1	31.4
	μ [-]	0.0074	0.0074	0.0074	0.0074	0.0078	0.0059	0.0059	0.0059	0.0060	0.0063
	ΔT [K]	13.85	13.4	13.0	12.7	13.0	15.8	15.5	15.0	14.7	15.0
	Q_{in} [dm^3/s]	0.074	0.065	0.060	0.056	0.047	0.078	0.067	0.060	0.056	0.045
	p_{max} [MPa]	3.32	3.28	3.15	3.20	3.16	4.68	4.53	4.50	4.40	4.35
H=18 [mm]	h_{min} [μm]	36.9	36.3	36.0	36.2	36.0	31.8	31.3	31.4	31.2	31.0
	μ [-]	0.0078	0.0077	0.0077	0.0076	0.0077	0.0061	0.0061	0.0061	0.0062	0.0062
	ΔT [K]	14.1	13.5	13.1	12.9	12.6	16.3	15.7	15.4	15.0	14.6
	Q_{in} [dm^3/s]	0.064	0.058	0.053	0.051	0.049	0.066	0.059	0.052	0.050	0.047
	p_{max} [MPa]	3.24	3.19	3.13	3.18	3.15	4.48	4.45	4.46	4.32	4.30
H=20 [mm]	h_{min} [μm]	37.0	36.5	36.6	36.4	36.1	32.0	31.4	31.5	31.4	31.1
	μ [-]	0.0080	0.0079	0.0078	0.0078	0.0079	0.0063	0.0064	0.0063	0.0063	0.0064
	ΔT [K]	14.3	13.8	13.4	13.0	12.7	16.4	16.0	15.6	15.0	14.9
	Q_{in} [dm^3/s]	0.058	0.052	0.049	0.047	0.045	0.059	0.051	0.047	0.045	0.043
	p_{max} [MPa]	3.21	3.17	3.23	3.16	3.14	4.45	4.35	4.43	4.35	4.28

Table 2 Results of computations in the case of tilting bearing pad

		F = 150 [kN]					F = 200 [kN]				
B^*		1.2	1.4	1.6	1.8	2.0	1.2	1.4	1.6	1.8	2.0
H=16 [mm]	h_{min} [μm]	46.2	44.2	42.6	41.6	40.5	37.2	35.7	34.7	34.4	33.2
	μ [−]	0.0034	0.0034	0.0035	0.0035	0.0036	0.0028	0.0028	0.0028	0.0028	0.0029
	ΔT [K]	47.5	44.9	42.6	41.4	41.3	56.2	54.7	52.3	53.7	51.5
	Q_{in} [dm^3/s]	0.109	0.102	0.098	0.096	0.095	0.096	0.091	0.087	0.087	0.084
	p_{max}[MPa]	3.65	3.65	3.71	3.79	3.89	4.79	4.82	4.92	5.13	5.19
H=18 [mm]	h_{min} [μm]	45.6	43.7	42.5	41.1	40.2	36.4	35.0	34.4	33.9	32.9
	μ [−]	0.0034	0.0034	0.0035	0.0036	0.0036	0.0028	0.0028	0.0028	0.0028	0.0029
	ΔT [K]	47.8	44.7	44.0	41.0	41.7	56.0	53.4	53.9	54.0	53.1
	Q_{in} [dm^3/s]	0.107	0.100	0.098	0.094	0.094	0.093	0.087	0.086	0.085	0.083
	p_{max}[MPa]	3.65	3.65	3.77	3.77	3.90	4.77	4.77	4.98	5.13	5.29
H=20 [mm]	h_{min} [μm]	45.3	43.6	42.4	41.4	40.0	36.0	34.9	33.9	33.3	32.6
	μ [−]	0.0034	0.0034	0.0034	0.0035	0.0036	0.0028	0.0028	0.0028	0.0029	0.0029
	ΔT [K]	48.4	46.0	45.0	43.7	41.6	55.8	55.6	53.0	51.7	53.6
	Q_{in} [dm^3/s]	0.106	0.100	0.098	0.096	0.093	0.091	0.087	0.084	0.083	0.083
	p_{max}[MPa]	3.66	3.70	3.80	3.87	3.90	4.75	4.87	4.94	5.05	5.33

Fig. 5 to 8 show the changes of the above presented results of computations.

a) b)

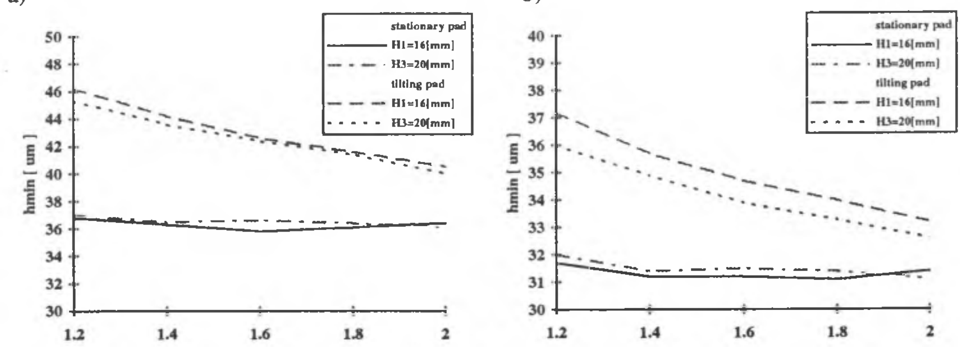


Fig.5.The minimum film thickness variation with B* parameter
a) F = 150 [kN] b) F = 200 [kN]

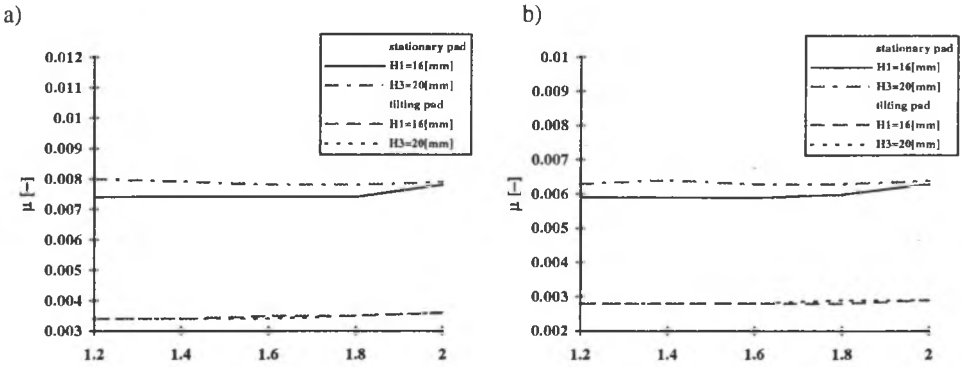


Fig.6.The coefficient of friction variation with B* parameter
a) F = 150 [kN] b) F = 200 [kN]

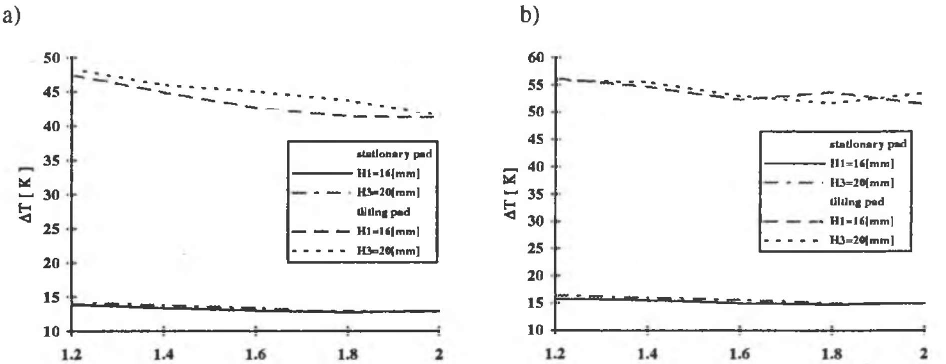


Fig.7. The temperature rise in oil film variation with B* parameter
a) F = 150 [kN] b) F = 200 [kN]

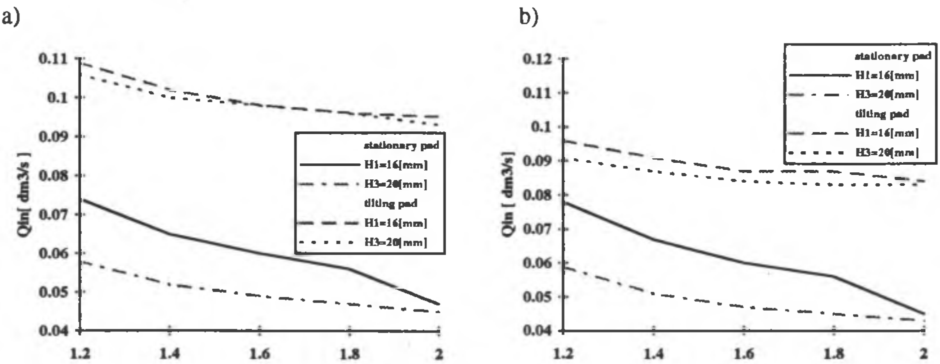


Fig.8.The inlet oil flow variation with B* parameter
a) F = 150 [kN] b) F = 200 [kN]

5. Conclusions

1. The results of computations show the evident influence of the type of thrust bearing pad on the oil film properties.
2. The changes of the pad stiffness have the influence on the bearing characteristics.

REFERENCES

[1] Ettles C. : The Development of a Generalized Computer Analysis for Sector Shaped Tilting Pad Thrust Bearings. ASLE Transactions,vol.19,1975.

[2] Tieu A. K. : An Investigation of " Diaphragm " Type Thrust Bearings. Part I : Experiment. Transactions of the ASME, vol. 97 ,October 1975.

[3] Tieu A. K. : An Investigation of " Diaphragm " Type Thrust Bearings. Part II : Theory. Transactions of the ASME, vol. 97 ,October 1975.

[3] Kaniewski W. , Murdzia E. : Algorytm metody obliczeniowej wzdłużnego łożyska ślizgowego ze stałymi segmentami. VI Konferencja " Metody i środki projektowania wspomaganego komputerowo". Warszawa 1987.

[4] Świderski W. , Latuśkiewicz J. : Program do wspomagania projektowania poprzecznego ł ożyska ślizgowego. VII Konferencja " Metody i środki projektowania wspomaganego komputerowo". Warszawa 1989.

[5] Kisiel J. , Kuśmierz L. , Olszewski M. : Wspomagana komputerowo metoda wyznaczania charakterystyk pracy wzdłużnego łożyska ślizgowego ze stałymi segmentami. VIII Konferencja " Metody i środki projektowania wspomaganego komputerowo". Warszawa 1991.

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