INSTITUTE OF FLUID-FLOW MACHINERY
POLISH ACADEMY OF SCIENCES

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

114

GDAŃSK 2003

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

Appears since 1960

Aims and Scope

Transactions of the Institute of Fluid-Flow Machinery have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

EDITORIAL AND PUBLISHING OFFICE

IFFM Publishers (Wydawnictwo IMP), Institute of Fluid Flow Machinery, Fiszera 14, 80-952 Gdańsk, Poland, Tel.: +48(58)3411271 ext. 141, Fax: +48(58)3416144, E-mail: esli@imp.gda.pl http://www.imp.gda.pl/

Terms of subscription

Subscription order and payment should be directly sent to the Publishing Office

Warunki prenumeraty w Polsce

Wydawnictwo ukazuje się przeciętnie dwa lub trzy razy w roku. Cena numeru wynosi 20,- zł + 5,- zł koszty wysyłki. Zamówienia z określeniem okresu prenumeraty, nazwiskiem i adresem odbiorcy należy kierować bezpośrednio do Wydawcy (Wydawnictwo IMP, Instytut Maszyn Przepływowych PAN, ul. Gen. Fiszera 14, 80-952 Gdańsk). Osiągalne są również wydania poprzednie. Prenumerata jest również realizowana przez jednostki kolportażowe RUCH S.A. właściwe dla miejsca zamieszkania lub siedziby prenumeratora. W takim przypadku dostawa następuje w uzgodniony sposób.

ISSN 0079-3205

No. 114, 2003, 3-4

Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100$^{\text{th}}$ Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery.* At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

Prof. Jarosław Mikielewicz
Editor-in-Chief

Dr Edward Śliwicki
Managing Editor

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 114, 2003, 201-208

STANISŁAW STRZELECKI[a*] and SOBHY. M. GHONEAM[b]

Application of tilting-pad bearings in the design of large output turbines

[a] *Technical University of Łódz, Institute of Machine Design, Stefanowskiego 1/15, 90-924 Łódz, Poland*

[b] *Mechanical Design & Production Engineering Department, Faculty of Engineering, Menoufya University, Shebin-el-kom, Egypt*

Abstract

Modern steam turbines for electric power utilities have reached the output of 1000 MW per unit. It causes the need for an increase in shaft diameters and application of special tilting-pad journal bearings. The tilting-pad journal bearings are one of the options as they have very good hydrodynamic stability at high speed and are less sensitive to load direction and shaft misalignment. They allow for minimizing the oil flow too. In case of large output turbines, large diameter of heavy rotors the number of pads is restricted to three. The reliable design of journal bearings for responsible turbines is assured by calculation and analysis of static and dynamic characteristics of bearing. These characteristics can be determined from the oil film pressure and temperature distributions. The paper describes the possibilities of the application in large output turbines of some tilting-pad bearings. Static characteristics of these bearings can be obtained from the solution of Reynolds, energy, geometry and viscosity equations. Numerical method, assumption of incompressible lubricant, the laminar and adiabatic flow of oil in the bearing gap of finite length bearing and static equilibrium position of journal have been assumed.

Keywords: Hydrodynamic lubrication; Tilting-pad bearings; Hybrid bearings

1 Introduction

The tilting 3-pad journal bearings are used in the turbines and compressors operating at the high rotational speeds and medium or low external loads [1-4].

*Corresponding author. E-mail address: strzelec@pkm1.p.lodz.pl

One of the advantages of these bearings consist in a progressive increase of the assembly stiffness with geometric preload. If this is correct, the fractional frequency whirl can be completely suppressed. Some disadvantages of these type of bearings include a higher power loss and lower load capacity than other types of bearings, such as the multilobe journal ones [1-4].

Tilting pad radial bearings are characterised by very good hydrodynamic stability at high speed. They are less sensitive to load direction and shaft misalignment, oil flow can be minimised, able to use standard components, and spares consist of pads only.

Design variation of tilting-pad journal bearings allow operation in modern, high speed, high output power turbounits. Taniguchi [1] gives an example of special design journal bearing called "two-pad bearing" (Fig. 1) for large turbines. This bearing is a hybrid one with fixed pad (lobe) in the upper part of sleeve and two tilting-pads in the bottom part. The load is directed between the pads. The lower half of bearing with two tilting-pads allows to maintain high reliability against rotor vibration and upper half is a semicircular sleeve widely scooped to reduce bearing friction.

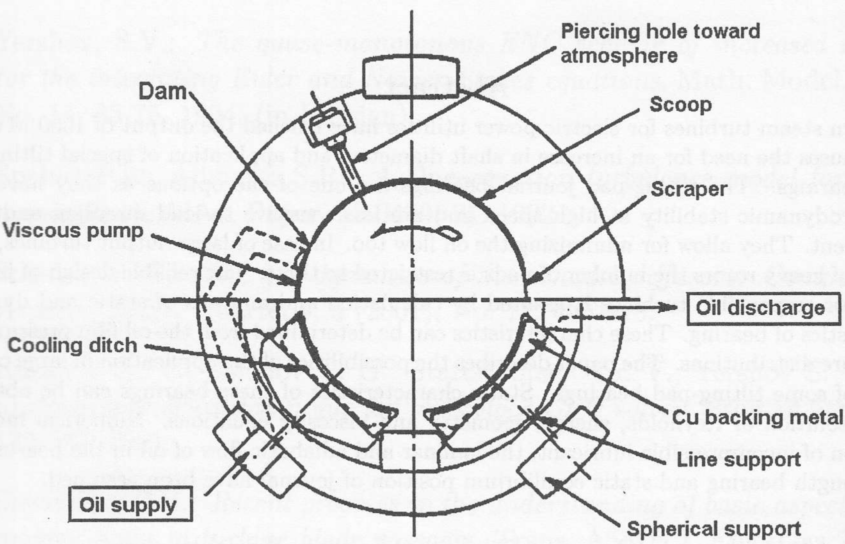

Figure 1. Hybrid journal bearing with upper fixed pad (lobe) and tilting 2-pads in the bottom part of bearing [1] applied in the bearing system of turbogenerator.

In experimental investigation Taniguchi checked the bearing of diameter 535 mm, loaded by 290 kN, operating at 3600 rpm which would be applied to

1000 MW steam turbine. The large two-pad journal bearing had the cooling ditches and viscous pump.

The paper describes the design and some static characteristics of tilting 3-pad and hybrid one with fixed pad and 2-tilting pads. The Reynolds, energy, geometry and viscosity equations were solved numerically on the assumption of incompressible lubricant, the laminar and adiabatic flow of oil in the bearing gap of finite length bearing. Aligned orientation of bush and journal axis without deflections of bearing and journal was assumed.

Solution of the basic equations of the thermo-hydrodynamic theory of lubrication allows to receive the necessary data on pressure and temperature distributions, the maximum value of pressure and temperature of oil film, minimum oil film thickness, oil flow and friction forces which are the static characteristics determining the input variables for the design of bearing tribosystem. Static characteristics are also required for determination of dynamic characteristics expressed by four stiffness and four damping coefficients applied in the vibration analysis and calculation of the stability of the rotor-bearing system.

2 Geometry of lubricating gap

The tilting-pad bearing consists the set of pads supported individually in a way shown in Fig. 2. The geometric parameters defining the bearing are: bearing diameter (D), length (L), number of pads (Z), angular length of pad (Ω), relative length of bearing (L/D), pad clearance (ψ_s), orientation of support point (Ω_F/Ω), pad mass and inertia. During the operation of bearing each of pads displaces in such a way that the oil film resultant force goes through the support point. The position of support point effects the pad displacement and the magnitude of generated oil film.

The geometric parameters of tilting-pad journal bearing are introduced in Fig. 2. Each of pads of tilting-pad bearing aligns in static equilibrium position in such a position, that the resultant force of oil film goes through the support point of pad. The geometric sum of resultant forces of single pad gives the reaction corresponding the external load [1-3].

During the variation of journal position, the hydrodynamic force of pad acting on the journal changes only with respect to the magnitude not the direction [2]. Oscillations of pad around the equilibrium position causes the displacements of pad centre on the radius R_k around the support point on the assumption that the pads and supporting surface are stiff. The geometry of lubricating gap determines Eq. (1).

$$\bar{H}(\varphi) = \psi_s + \frac{\psi_s - 1}{\cos(\tau_1 - \tau_o)} \cdot cos(\varphi - \tau_1), \tag{1}$$

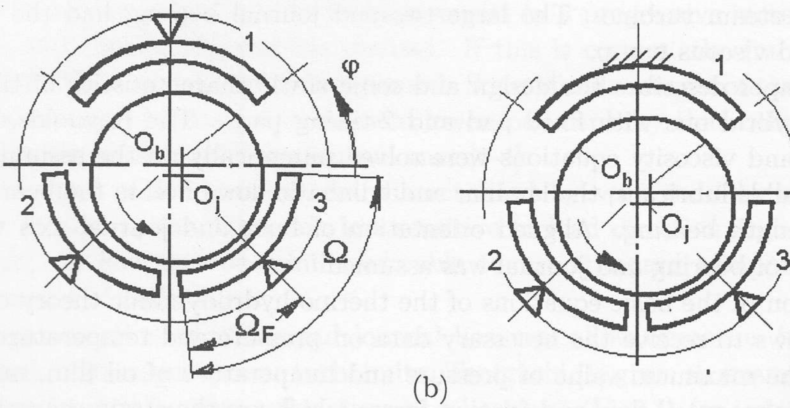

Figure 2. Tilting 3-pad (a) and hybrid journal bearing (b) with fixed upper pad and tilting 2-pads in the bottom part of bearing; O_b , O_j – centre of bearing and journal, φ – peripheral co-ordinate, 1, 2, 3 – lobe and tilting pads.

Table 1. Ω_F/Ω – relative length of pad support ($\Omega_F/\Omega = 0.5$).

	Angle of pad			
Lobe No	Begin $\varphi_b[°]$	End $\varphi_e[°]$	Support $\varphi_s[°]$	Centre $\varphi_c[°]$
1	28.0	80.0	234.0	54.0
2	100.0	152.0	306.0	126.0
3	172.0	224.0	378.0	198.0

where: φ – circumferential co-ordinate (°), ψ_s – lobe relative clearance, τ_o – angle of the lobe centre line (°), τ_1 – angular orientation of centre point of pad (°). Relative clearance of tilting pad (Fig. 2) describes Eq. (2).

$$\psi_s = \frac{E_o}{\Delta R_o} + 1. \tag{2}$$

It results from Eq. (1), that the geometry of lubrication gap, besides the pad relative clearance ψ_s and angle τ_o, is decided by angular orientation τ_1 of the pad centroid. During determination of static equilibrium position of pad, the angle τ_1 should be varied as long as the magnitude of lubricating gap generates oil film pressure distribution which gives the resultant force to go through the support point of pad.

3 Oil film pressure and temperature distribution

Pressure, temperature and viscosity distributions in the oil film were determined on the basis of geometry, Reynolds, energy and viscosity equations [2, 3]. The assumptions about fluid film lubrication, incompressible flow, Newtonian lubricant, absence of deformation of the bearing or the journal and parallel axis of bearing and journal were made. The equation of the generated pressure field is described by the following non dimensional form of the Reynolds equation.

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

where: D – bush diameter, m; L – bearing length; H – dimensionless oil film thickness; p – dimensionless oil film pressure; η – dynamic viscosity of oil, Ns/m^2; z – dimensionless axial co-ordinate; ϕ – dimensionless time, $\phi=\omega\cdot t$; t – time, s; ω – angular velocity, 1/s.

The pressure distribution computed from Eq. (3) has been introduced in energy Eq. (4) giving the temperature and viscosity fields for adiabatic model of oil film.

$$\frac{12\cdot H}{\mathrm{Pe}}\left(\frac{\partial^2 T}{\partial\varphi^2}+\left(\frac{D}{L}\right)^2\frac{\partial^2 T}{\partial z^2}\right)+\left(\frac{H^3}{L}\frac{\partial p}{\partial\varphi}-6\cdot H\right)\cdot\frac{\partial T}{\partial\varphi}+\left(\frac{D}{L}\right)^2\frac{H^3}{\eta}\frac{\partial p}{\partial z}\cdot\frac{\partial T}{\partial z}=$$

$$-\frac{H^3}{\eta}\left(\left(\frac{\partial p}{\partial\varphi}\right)^2+\left(\frac{D}{L}\right)^2\left(\frac{\partial p}{\partial z}\right)^2\right)-\frac{12\cdot\eta}{H}, \tag{4}$$

where: Pe – Peclet's number, $\mathrm{Pe}=\rho\cdot c\cdot\omega\cdot R_j^2/h$, T – dimensionless temperature, η_o – dynamic viscosity at ambient temperature, Ns/m^2; c – specific heat of lubricant, J/(kgK); ρ – density of lubricant, kg/m^3; R_j – journal radius, m; h – oil film thickness, μm.

The boundary conditions for the pressure and temperature include their inlet values. The pressure boundary condition assumes only the positive values of oil film pressure distribution and the ambient pressure on the sides of the bearing. Oil film pressure, temperature and viscosity fields can be obtained by numerical solution to Eqs. (1) to (4) including the equation describing viscosity [3].

4 Results of calculation

Tilting 3-pad journal bearing is shown in Fig. 2a and the hybrid one which consists of one lobe and tilting 2-pads is introduced in Fig. 2b. Some results of calculation of static equilibrium position angle α_{eq}, Sommerfeld number So, minimum oil film thickness H_{min}, power loss P_{loss}, oil flow q_{fl} for 3-lobe (3L) and tilting 3-pad (3PT) journal bearings are given in Tab. 2. There are differences in the values of bearing characteristics obtained in case of 3-lobe or tilting 3-pad bearings (Tab. 2). The values of static equilibrium position angle α_{eq}, minimum oil film thickness H_{min}, and power loss P_{loss}, are smaller in case of tilting 3-pad bearing than in case of 3-lobe one but Sommerfeld number So and oil flow q_{fl} show an increase.

Oil film pressure and temperature distributions obtained for tilting 3-pad

Table 2. $L/D = 0.8$ $\varepsilon = 0.4$, $\psi_s = 2$.

Bearing type	α_{eq}	So	H_{min}	P_{loss}	q_{fl}
3-lobe	317.54	0.04497	0.60028	24.864	3.13
Tilting 3-pad	270.35	0.10504	0.45383	13.345	3.75

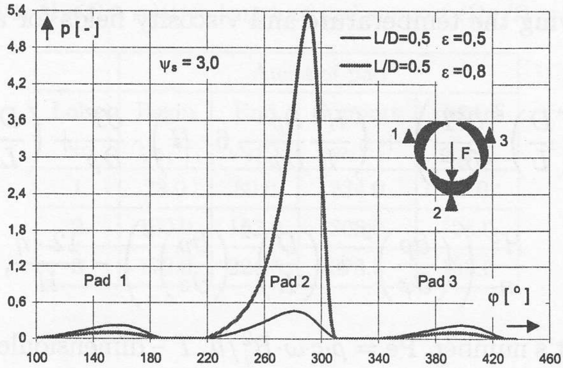

Figure 3. Oil film pressure distribution in tilting 3-pad bearing at different values of relative eccentricity.

journal bearing at two values of relative eccentricity of journal ε are shown in Figs. 3 and 4. It can be observed in Figs. 3 and 4 that the largest value of pressure or temperature of oil film is in 2-nd pad in both cases of considered relative eccentricities of the journal. The value of thermal coefficient was $K_T =$ 0.[illegible] ($K_T = \omega \cdot \eta_0 / (c_t \cdot \rho \cdot g \cdot T_0 \cdot \psi^2)$ where: c_t – specific heat of oil, ρ – oil density, g – acceleration of gravity, T_0 – temperature of supplied oil).

Sommerfeld number and power loss P_{loss} for tilting-pad (3PT) and 3-lobe (3L) journal bearings are shown in Figs. 5 and 6. It results from Figs. 5 and 6 that the tilting 3-pad bearing has a larger load capacity and smaller power loss (particularly at small relative eccentricities).

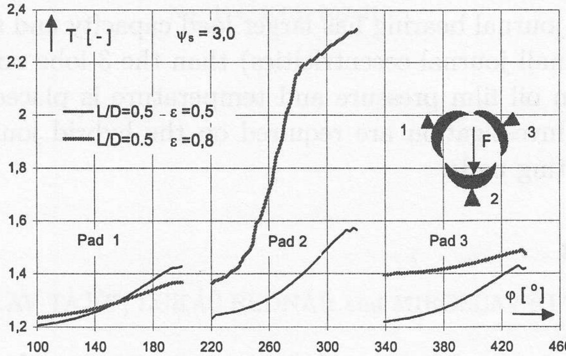

Figure 4. Oil film temperature distribution for different values of relative eccentricity.

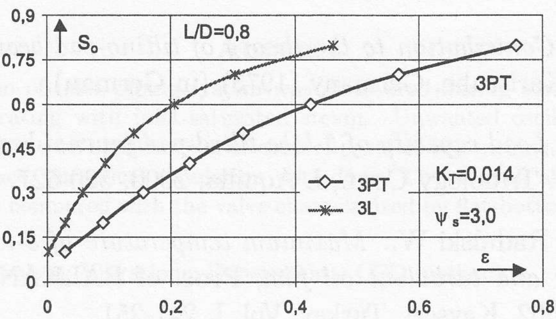

Figure 5. Sommerfeld number for tilting-pad (3PT) and 3-lobe(3L) journal bearings.

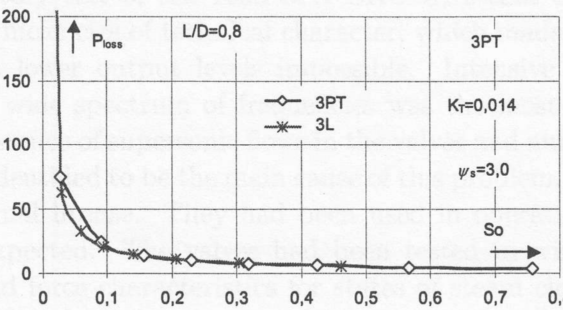

Figure 6. Power loss P_{loss} for tilting-pad (3PT) and 3-lobe(3L) journal bearings; (for 3L bearing the power loss at So=0.00204 is $P_{loss} = 865.4$).

5 Conclusions

Tilting 3-pad journal bearing has larger load capacity and smaller power loss (particularly at small journal eccentricities) than the 3-lobe one. At the load on pad the maximum oil film pressure and temperature is placed on the 2^{nd} pad. More theoretical investigation are required on the hybrid journal bearing with stationary and tilting pads.

Received 5 May 2003

References

[1] Taniguchi S., at al.: *A thermohydrodynamic lubrication analysis to design large two-pad journal bearing with cooling ditches*, Trans. of ASME, Vol. 120(1998), 214-220.

[2] Klumpp R.: *Contribution to the theory of tilting-pad bearings*, PhD Thesis, Tech. Univ. Karlsruhe, Germany (1975) (in German).

[3] Strzelecki S.: *Load capacity of 5-lobe tilted-pad journal bearing.* Proc. of 2000 AIMETA Int. Tribology Conf., L'Aquilla, 2000, 520-527.

[4] Strzelecki S., Radulski W.: *Maximum temperature of 3-lobe journal bearing with laminar and turbulent oil film*, Proc. of BALKANTRIB'2002 Conf., 12-14 June 2002, Kayseri, Turkey, Vol. I, 241-251.