Tribology International 82 (2015) 110–122

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

A fully coupled 3D thermo-elastohydrodynamics model for a bump-type compliant foil journal bearing

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ARTICLE INFO

Article history:
Received 29 June 2014
Received in revised form
7 September 2014
Accepted 7 October 2014
Available online 19 October 2014

Keywords:
Elastohydrodynamic
Aerodynamic
Finite-element method
Foil bearing

ABSTRACT

Coupled thermo-elastohydrodynamics model has been developed to predict three-dimensional thermal, structural and hydrodynamic performance of foil bearings. Bearing deformation and film pressure has been coupled in FEM considering thermal/centrifugal growths, and thermo-mechanical material properties. Energy equation has been solved for film temperature using FDM, which is also coupled to FEM via successive iterations. Augmented-Lagrangian method and thermal contact models have been applied to solve for mechanical and thermal contacts. The predicted values have been benchmarked against published measurements, which indicated reasonable correlation. A parametric study has been conducted for various shaft speeds and load conditions to visualize bearing performance.

1. Introduction

The use of foil bearings in turbomachinery has various advantages compared to the conventional rotor support technologies. The favorable features of foil bearings are improved reliability, elimination of lubrication system, operation capability at very high and low temperatures, and better tolerance to misalignment. A foil bearing consists of three main components, namely, top foil, corrugated bumps and bearing housing. The layered structure providing stiffness and damping to the system makes foil bearing unique. The inherent flexibility of the structure improves dynamic properties of the bearing. The compliant bumps can deform under hydrodynamic pressure load to form a converging wedge between the shaft and bearing surface without being significantly affected from speed and temperature variations. The working fluid in foil bearing applications is usually air. Lower viscosity of air provides superior performance at elevated temperatures compared to oil and other liquid lubricants. However, thin bearing structure with high shear speeds may result in thermal instability with increasing temperature. Foil bearings have some limitations as foil material losses strength at high temperatures, and its stiffness drops. The most important problem that is faced during experiments at high speeds or overload conditions is high local temperature gradients that cause wavy deformation of the foil surface that lead to catastrophic failure of the bearing [1]. Another important concern in terms of the thermal management is the weak conduction rate of the bearing due to thin foil structure. The contact between the top foil and supporting bumps occurs at localized small areas that prevent conductive heat removal from the system. A literature study indicates that some experimental research is available in open literature. However, extended thermo-hydrodynamic (THD) analyses are required to advance and optimize the system performance at the design level. A comprehensive compliant foil bearing (CFB) analysis that is calibrated with relevant test data will enable designers to address thermal instability issues leading to widespread usage of CFB in novel turbomachinery applications.

Conventional THD CFB analyses model the bumps as equivalent stiffness that is applied uniformly around the bearing circumference. More complex models couple elastic deformations of the top foil directly to the bump mechanism and the hydrodynamics of the gas film through considerable simplification of the structural model. The role of the top foil is to generate air film and hydrodynamic lift force when shaft rotates. Therefore, it is important that stiffness of the top foil is sufficiently high to endure the hydrodynamic pressure. However, the portions of the top foil surface that are not in contact with the bumps have practically limited bending stiffness, and deflect more when exposed to hydrodynamic pressure. Most of the previous studies in the

Abbreviations: CFD, computational fluid dynamics; CMY, Cooper–Mikic–Yovanovich correlation; FDM, finite difference method; FEA, finite element analysis; FEM, finite element method; FSI, fluid structure interaction; *Nu*, Nusselt number; PARDISO, parallel direct solver; *Pr*, Prandtl number; *Re*, Reynolds number; TC Loc., thermocouple locations; TEHD, thermo elasto hydrodynamics; THD, thermo hydrodynamics

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http://dx.doi.org/10.1016/j.triboint.2014.10.001
0301-679X/

Nomenclature

c	nominal clearance [m]
c_p	specific heat capacity of fluid [J/(kg K)]
D	diameter [m]
D_{s-i}	shaft inner diameter [m]
D_{sl-i}	sleeve inner diameter [m]
D_{sl-o}	sleeve outer diameter [m]
e	eccentricity [m]
$h(x,y)$	fluid film thickness [m]
H_{bf}	bump height [m]
h_{conv}	convective heat flux coefficient [W/(m^2 K)]
h_s	convective heat flux coefficient of shaft surface [W/(m^2 K)]
h_{max}	maximum film thickness [μm]
h_{min}	minimum film thickness [μm]
h_{tf}	convective heat flux coefficient of top foil surface [W/(m^2 K)]
k	thermal conductivity [W/(m K)] or changed node index in FDM
L	bearing length [m]
L_{bf}	bump length [m]
m	node number in circumferential direction
$\mathbf{n}$	surface normal
n	node number in axial direction
P	pressure [Pa]
q''_s	Shaft surface heat flux [W/(m^2 K)]
q''_{tf}	top foil surface heat flux [W/(m^2 K)]
R	radius [m]
$v(x,y,z,)$	film velocity in y-direction [m/s]
R_{bf-sl}	thermal resistance between bump foil and sleeve [(m^2 K)/W]
R_{tf-bf}	thermal resistance between top foil and bump foil [(m^2 K)/W]
S_{bf}	bump pitch [m]
T	temperature [K or °C]
T_a	ambient temperature (293.15 [K])
t_{bf}	bump foil thickness [m]
T_{cool}	cooling flow temperature [K]
$u(x,y,z,)$	film velocity in x-direction [m/s]
$z(x)$	finite difference function
Z_{bf}	bump foil plain segment

Greek letters

δ	some small value
ε	eccentricity ratio
ε_{err}	error rate for two consecutive iteration
θ	circumferential angle
μ	dynamic viscosity [Pa s]
ρ	density [kg/m^3]
φ	attitude angle
ω	angular speed [rad/s]

Subscripts

a	ambient
bf	bump foil
cool	coolant
f	film gap
g	gas
i	inner or tangential direction index in FDM
j	journal or axial direction index in FDM
leading	leading edge
L	length
nc	natural convection
o	outer
s	shaft
sf	surface
sl	sleeve
tf	top foil
trailing	trailing edge
x	*x* direction
y	*y* direction
z	*z* direction

Superscripts

k	iteration number
t	traction

literature neglect this deflection of the top foil. Therefore, studying the damping characteristics due to the top foil deflection is impossible using these models. However, the above mentioned top foil deflection phenomenon, which is called as sagging, radically affects the overall behavior of the bearing.

Salehi et al. [2] performed the first study to characterize thermal properties of gas foil bearings by utilizing a simple elastic foundation model [3] to resolve bump deformation. The Couette flow approximation is applied to simplify the energy equation by neglecting the work done by pressure such that energy and Reynolds equations are uncoupled. Only the circumferential temperature distribution at the bearing mid-plane is calculated, and the axial temperature distribution is assumed to decrease linearly towards the bearing edges. Peng and Khonsari [4] proposed a more advanced THD model to predict the performance of the CFB at steady state conditions. The foil structure is represented by simple elastic foundation whereas coupled Reynolds and thermal energy transport equations are solved simultaneously to predict gas film pressure and temperature. However, the model allows sub-ambient pressure, which is not realistic for foil bearings, and ignores heat flux towards shaft and sleeve. According to predictions of this model, temperature distribution in axial direction is almost uniform. Sim and Kim [5] developed a 3D THD model for compliant flexure pivot tilting pad gas bearings. The model predicts the rotor and pad temperatures, as well as the gas film temperature. The model employs global thermal balance with thermal boundary conditions for bearing housing and rotor. In a succeeding study, Sim and Kim [6] presented an enhanced version of the THD analysis by adding analytic thermal contact formulation and numerical inlet flow mixing models. The proposed model demonstrates the characteristics of the inlet flow mixing and determines the thermal mixing parameter. The model predictions are also compared to experimental data. Kim and San Andrés [7] measured rotor response with foil bearings cooled by pressurized side flow. They benchmarked the measurements for onset speeds of instability and whirl sub-synchronous frequency to the predictions with a computational model. In a following study, San Andrés and Kim [8] compared 1D and 2D finite element models to estimate the static and dynamic load capacity of foil bearing. The deformation of the top foil structure is coupled to Reynolds equation through a global stiffness matrix. It is found that predictions in 1D model are closer to the experiments, and 2D model