Tribology International 102 (2016) 305–318

Contents lists available at ScienceDirect

Tribology International

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

Influence of surface texturing on the hydrodynamic performance of a thrust bearing operating in steady-state and transient lubrication regime

A. Gherca*, A. Fatu, M. Hajjam, P. Maspeyrot

Institut Pprime, UPR 3346, Département Génie Mécanique et Systèmes Complexes, CNRS – Université de Poitiers – ENSMA, 4, Avenue de Varsovie, 16021 Angoulême, France

ARTICLE INFO

Article history:
Received 7 March 2016
Received in revised form
27 May 2016
Accepted 28 May 2016
Available online 30 May 2016

Keywords:
Thrust bearing
Textures
Hydrodynamic lubrication
Cavitation

ABSTRACT

This paper presents a finite element modeling of hydrodynamic thrust bearings operating both in steady-state and transient lubrication regime. The presented numerical algorithm is conservative and makes it possible to determine, with a high degree of accuracy, the operating characteristics such as the friction torque, the film pressure, the film thickness and the oil flow. For different operating conditions, it was shown that the behavior of the thrust bearing is not the same when textures are applied on the rotor or on the stator. Thus, the difference in behavior between the stationary and non-stationary case was revealed. Lastly, it was predicted that placing the textures on the rotor could, under certain conditions, improve the hydrodynamic performance of the thrust bearing.

1. Introduction

Thrust bearings are load-supporting mechanical parts, which are essential in the operation of various devices such as turbomachinery. Their main purpose is to balance an axial force, which is usually exerted by the weight of various machine parts. The optimization of thrust bearings geometry has been the subject of research for more than a century, and many improvements have been made over the years. The methods used to optimize the hydrodynamic behavior of thrust bearings (the objective being an increase of the load-carrying capacity) are very often related to the geometry of the stator and rotor.

Today, the development of new machining techniques allows the control of surface geometry at micro and even nano level [1]. In this context, surface texturing becomes a particularly interesting optimization method, since it can be applied to all types of thrust bearings. The increase of the load-carrying capacity of thrust bearings would allow a reduction of their dimensions, which would be consistent with the current technological trends that require a downsizing of mechanical components. Therefore, the optimization of thrust bearings through surface texturing has been the subject of an increasing number of both theoretical and experimental works [2–10].

Textures may be applied onto a limited area or to the entire surface of thrust bearings. At the same time, textures may be applied on the stationary surface (the stator), but also on the moving surface (the rotor). In a perspective of simplification, the current study focuses exclusively on fixed geometry configurations. Thus, the tilting pad thrust bearings, which require additional conditions, were not included in this study.

It is well known that the physical phenomena that can occur during operation of the thrust bearings are very diverse. Thermal and mechanical deformations, misalignments or lubricant starvation are a few examples. All these effects are very complex and undoubtedly require a very thorough theoretical treatment. These effects are also not included in this study, which allows the authors to better highlight the effects introduced by surface textures.

The first part of the paper focuses on a description of the lubrication mechanisms governing the operation of thrust bearings. The numerical model used in this paper, which is based on a finite element formulation, is briefly presented next. To prove the applicability of this algorithm in the case of thrust bearings, the paper also includes a validation procedure.

The main part of the paper is dedicated to a thorough analysis of the effects induced by different geometrical characteristics of textures on the hydrodynamic performance of a thrust bearing operating in both steady-state and transient lubrication regimes.

* Corresponding author.
E-mail address: andrei.gherca@univ-poitiers.fr (A. Gherca).

http://dx.doi.org/10.1016/j.triboint.2016.05.041
0301-679X/

Nomenclature

C	Friction torque [N m]
D	Universal function
F	Switch function
h	Oil film thickness [μm]
h_0	Minimum oil film thickness [μm]
h_d	Pocket/Texture depth [μm]
L_r	Width of textured area [mm]
l_c	Cell length [mm]
l_d	Pocket length [mm]
N	Cell number
N_p	Pad number
p	Lubricant pressure [MPa]
p_0	Ambient pressure [MPa]
p_{cav}	Cavitation pressure [MPa]
Q	Lubricant flow rate [m^3/s]
Q_T	Lubricant outflow rate [m^3/s]
R_1	Inner radius [mm]
R_2	Outer radius [mm]
r, θ, y	Cylindrical coordinates [m, rad, m]
T_t	Period [s]
t	Time [s]
U	Relative speed of surface [m/s]
W	Load carrying capacity [N]
W_a	Axial applied load [N]
α	Texturing ratio [%]
β	Pad angle [deg]
β_t	Angular amplitude of textured area [deg]
γ	Filling ratio
Δt	Time step [s]
μ	Dynamic viscosity [Pa s]
ρ	Density of lubricant [kg/m^3]
ρ_0	Density of mixture of gas and lubricant [kg/m^3]
ρ_t	Texture density [%]
Ω	Active region of lubricant film
Ω_0	Non-active region of lubricant film
ω	Rotation speed [rpm]

Indices and exponents

$(\ )_1$	Parameter corresponding to the rotor
$(\ )_2$	Parameter corresponding to the stator
$(\ )_x$	Component in the x direction
$(\ )_z$	Component in the z direction
$(\ )_r$	Component in the radial direction
$(\ )_\theta$	Component in the circumferential direction
$(\ )_{avg}$	Average value in time
$(\ )_{min}$	Minimum value
$(\ )_{max}$	Maximum value
$(\ )^{\Omega}$	Component in the active region
$(\ )^{\Omega_0}$	Component in the non-active region

2. Numerical model

2.1. Principle

Fig. 1 shows the diagram of a classic inclined pad thrust bearing, as proposed by Frêne et al. [11]. This type of thrust bearing can include from 6 up to 20 pads. The pads are formed of inclined planes, but in some cases can also have a flat region, parallel to the rotor. Furthermore, the oil-supply grooves that separate the pads can have an inclination of 10–20°. The load and the friction torque calculated for a pad are multiplied by the number of pads in order to obtain the overall characteristics of the thrust bearing. Four different flow rates are used to describe the total flow rate, as shown in Fig. 1: the radial flow to the inner radius Q_{r1}, the radial flow to the outer radius Q_{r2}, the incoming flow Q_e and the outgoing flow Q_s. The movement of the rotor is characterized by the rotation speed ω (rpm), W_T is the total load applied to the thrust bearing, while the β parameter defines the angular amplitude of the pads.

2.2. Reynolds equation in cylindrical coordinates

First, it should be mentioned that the particular geometrical configuration of the thrust bearing requires an adapted reference system. Regarding the modeling of the flow of lubricant, this requires an adaptation of the Reynolds equation to a cylindrical coordinate system.

Let us consider a cylindrical reference system, as shown in Fig. 2. A point M_1 (conveniently associated with the lower surface of the thrust bearing) has a H_1 coordinate with regard to the y direction and the following velocity components: U_1^r (radial velocity), U_1^θ (tangential velocity) and U_1^y (axial velocity) following the directions r, θ and y. Similarly, the point labeled M_2 (conveniently associated with the upper surface of the thrust bearing) has a H_2 coordinate following y and has similar velocity components: U_2^r, U_2^θ and U_2^y.

In these circumstances, the equation of thin viscous fluid films in cylindrical coordinates is written as [11]:

$$\begin{aligned} &\tfrac{\partial}{\partial r}\left(\tfrac{\rho r}{\mu}(H_2-H_1)^3\tfrac{\partial p}{\partial r}\right)+\tfrac{\partial}{\partial \theta}\left(\tfrac{\rho}{r\mu}(H_2-H_1)^3\tfrac{\partial p}{\partial \theta}\right)= \\ &=6\tfrac{\partial}{\partial r}\left[\rho r\left(U_1^r+U_2^r\right)(H_2-H_1)\right]-12\rho r U_2^r\tfrac{\partial H_2}{\partial r}+12\rho r U_1^r\tfrac{\partial H_1}{\partial r}+ \\ &+6\tfrac{\partial}{\partial \theta}\left[\rho\left(U_1^\theta+U_2^\theta\right)(H_2-H_1)\right]-12\rho U_2^\theta\tfrac{\partial H_2}{\partial \theta}+12\rho U_1^\theta\tfrac{\partial H_1}{\partial \theta}+ \\ &+12\rho r\left(U_2^y-U_1^y\right)+12r(H_2-H_1)\tfrac{\partial \rho}{\partial t} \end{aligned} \quad (1)$$

Supposing the velocity components corresponding to points M_1 et M_2 satisfy the following conditions:

$$\vec{U}(M_1)\begin{cases}U_1^r=0\\U_1^\theta\neq 0\\U_1^y=0\end{cases} \text{ and } \vec{U}(M_2)\begin{cases}U_2^r=0\\U_2^\theta=0\\U_2^y\neq 0\end{cases} \quad (2)$$

Therefore, in the case of an incompressible fluid, Eq. (1) can be written as:

$$\frac{\partial}{\partial r}\left(\frac{r}{\mu}h^3\frac{\partial p}{\partial r}\right)+\frac{\partial}{\partial \theta}\left(\frac{1}{\mu r}h^3\frac{\partial p}{\partial \theta}\right)=6U_1^\theta\frac{\partial(H_1+H_2)}{\partial \theta}+12rU_2^y \quad (3)$$

where $h=H_2-H_1$.

In order to comply with the laws of mass-conservation, Eq. (3) is further adapted. Therefore, the final form of the Reynolds equation applied in the study is as follows [12,13]:

$$\begin{aligned} &F\left\{\frac{\partial}{\partial x}\left[\frac{h^3}{\mu}\left(\frac{\partial D}{\partial x}x+\frac{\partial D}{\partial z}z\right)\right]\frac{x}{r}+\frac{\partial}{\partial z}\left[\frac{h^3}{\mu}\left(\frac{\partial D}{\partial x}x+\frac{\partial D}{\partial z}z\right)\right]\frac{z}{r}-\right. \\ &\left.-\frac{\partial}{\partial x}\left[\frac{h^3}{\mu r}\left(\frac{\partial D}{\partial z}x-\frac{\partial D}{\partial x}z\right)\right]z+\frac{\partial}{\partial z}\left[\frac{h^3}{\mu r}\left(\frac{\partial D}{\partial z}x-\frac{\partial D}{\partial x}z\right)\right]x\right\}= \\ &=6U_1^\theta\left(\frac{\partial h}{\partial x}x-\frac{\partial h}{\partial z}z\right)+12U_1^\theta\left(\frac{\partial H_1}{\partial z}x-\frac{\partial H_1}{\partial x}z\right)+ \end{aligned}$$