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Optimization of a herringbone grooved thrust bearing

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Abstract. This paper analyses a state-of-art model of herringbone grooved thrust bearings (HGTB), i.e. dynamic gas journal bearings widely employed in various very high-speed applications where friction reduction is a key aspect. After the presentation of the state of the art about the applications of such bearings, the simplified analytical model from Whipple is briefly reviewed and the main simplifying assumptions are highlighted. This model provides a first approximation of the behavior of HGTB. Whipple model is then applied for the design of micro thrust bearings for a micro turbo-compressor applied in fuel cells systems. A sensitivity analysis is carried out to evaluate the effect of the main geometrical parameters on the maximum pressure and on the load capacity of the thrust bearing. The specific operative conditions of the given application are considered in the analysis. Finally, the geometry is optimized in order to maximize the load capacity within the bounds of the geometrical constraints of the application.

Keywords: dynamic gas bearings, narrow-groove theory, lubrication, thrust bearings.

1. Introduction

Gas bearings are employed in various high-speed applications such as turbochargers/expanders [1], gas turbines [2], air cycle machines [3] and MEMS [4, 5]. More recently, good prospects for further development are represented by the compressors for fuel cells [6] and heat pump applications [7]. Such solutions represent examples of small-scale turbomachinery, with power in ranges from some Watts to 150 kW and rotational speed from 10 to 500 krpm.

Among all types of gas bearings, the dynamic bearings with grooved surfaces have very interesting features for their compactness and simplicity of manufacture. Other advantages are related to the drastic reduction of friction compared to fluid bearings and to the absence of an external compressed air supply system. They were firstly developed in the 1950s. Automotive turbochargers [8] and turbocompressor for refrigeration [9] are two examples of successful application of such bearings.

Grooves of proper geometry can be machined on the external surface of the rotor in Herringbone Grooved gas Journal Bearings (HGJB) and on the fixed or the rotating surface of a plate in herringbone or spiral grooved thrust gas bearings (HGTB and SGTB). In [10] the so-called Narrow Groove Theory (NGT) was developed for the investigation of the herringbone pattern for HGTBs and in [11] the theory was further developed for HGJBs. According to this theory, the pressure profile along the circumferential direction is supposed to be smooth, as if the number of grooves were infinite. This is a simplification introduced by the model in order to reduce the complexity of the solution. A finite number of grooves would cause the circumferential pressure profile to be of sawtooth type, but this effect is neglect to a first approximation.

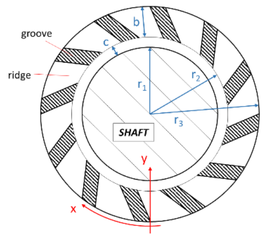
Other approaches are possible to evaluate the real pressure profile, such as the finite element method [12] and the full transient analysis [13]. Anyway, the NGT is suitable for a first design of the bearings, as in most of cases the estimation of the load capacity is quite in good accordance with experimental results [14,15].

The stability of rotors supported by HGJBs can be also analyzed with the NGT [16,17]. Another important aspect that must be considered in rotating system is the unbalance response analysis; experimental data can be used for the validation of the numerical models which estimate the linearized coefficients of the gas films [18,19].

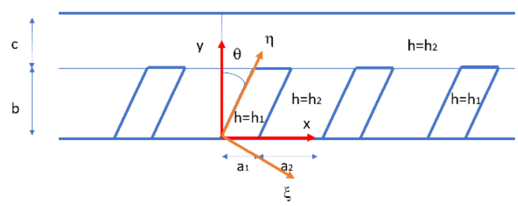
In this paper, the state-of-the-art NGT from [10] is taken into account and the geometry of a spiral grooved thrust bearing is optimized in order to maximize the load carrying capacity. A specific application is considered where micro thrust gas bearings of this type must be designed as guiding elements of a micro turbo-compressor in a fuel cells system. The specific operative conditions of the given application are considered in the analysis.

2. The analytical model

The model from Whipple [10] analyzes the geometry of a thrust bearing which presents some grooves machined on the thrust surface starting from the external radius r_3 to intermediate radius r_2, see figure 1. The internal part of the thrust bearing ($r_1<r<r_2$) is ungrooved. The film thickness above the ridge is h_2, while in the grooves it is h_1, so that the groove depth is h_1-h_2. The width of grooves measured along the circumferential direction is a_1, while the width of the ridges is a_2. One assumption made in [10] is that the curvature can be neglected, so that the geometry depicted in figure 1a can be unrolled along the circumferential direction x, so that it is equivalent to an unbounded set of rectilinear parallel grooves with periodicity, as depicted in figure 1b. The reference system Oηξ is defined along the grooves, and in the equivalent geometry the grooves are inclined of angle θ with respect to the radial direction (y-direction in the figure 1b).



1a



1b

Fig. 1. Geometry of a SGTB considered by Whipple's model.

The simplifying assumptions of the model are listed below.

- Curvature of grooves is neglected; spiral grooves are approximated as straight grooves.
- An equivalent straight thrust bearing problem with indefinite length (periodic geometry condition) is solved for.
- Newtonian fluids are considered.
- Grooves and ridges are considered sufficiently narrow so that a linearized pressure distribution inside the thrust bearing is hypothesized ($\partial p/\partial \xi \approx cost$ and $\partial p/\partial \eta \approx cost$).
- Inertial effects of the fluid are neglected.
- Pressure variation along the z-direction is neglected ($\partial p/\partial z \approx 0$).
- Velocity component w of the fluid along the z-direction is neglected ($w \approx 0$).
- Velocity components u and v in a direction perpendicular to the groove are neglected ($\partial u/\partial x \approx \partial v/\partial x \approx 0$ and $\partial u/\partial y \approx \partial v/\partial y \approx 0$).
- Possible ends effects at the ends of the grooves are neglected to set the boundary conditions of the problem.
- Pressure at both ends of the grooves, i.e. $y = b$ ($r = r_2$) and $y = 0$ ($r = r_3$), is constant.
- Continuity of pressure and flux exists at $\xi = 0$ at the boundary between a grooved and ungrooved region.
- The mean mass flow V across the bearing (average of the flow on a ridge and in a groove) occurs in the y-direction only.
- V is independent of y and its value is constant both in the grooved and flat region.
- The pressure trend along the x-direction is periodic with period $a_1 + a_2$.
- Isothermal and incompressible fluid

The mean pressure at the end of the grooves ($y = b$ or $r = r_2$) is named P. Therefore, the boundary conditions of the fluid static problem are set according to equation (1).

$$\begin{cases} p(y=0)=p_0 \\ p(y=b)=P \end{cases} \tag{1}$$

where p_0 is the ambient pressure. The pressure in the groove and in the ridge is named p_1 and p_2 respectively.

Naming β the gradient of pressure along η direction and α_1, α_2 the gradients of pressure p_1, p_2 along ξ direction, the linearized pressure variations are assumed according to equation (2).

$$\begin{aligned} \Delta p_1 &= \alpha_1 \Delta\xi + \beta\Delta\eta \\ \Delta p_2 &= \alpha_2 \Delta\xi + \beta\Delta\eta \end{aligned} \tag{2}$$

By imposing the simplifying assumptions to the thin film problem, it is possible to obtain the analytical expressions for coefficients α_1, α_2 and β. If radial flow V that crosses the grooved region is imposed to be coincident with the flow across the ungrooved region, it is possible to obtain the expression of pressure P as a function of the geometrical parameters:

$$P = p_0 + \frac{Ecb}{Fc + F'b} \tag{3}$$

where

$$E = \frac{a_2 a_1}{a_1 + a_2} \frac{\rho}{24\mu} \frac{6\mu u_0 (h_1 - h_2)}{(h_2^3 a_1 + h_1^3 a_2)} (h_1^3 - h_2^3) \sin 2\theta$$
$$F = \frac{1}{a_1 + a_2} \frac{\rho}{24\mu} \frac{2(a_1^2 + a_2^2) h_2^3 h_1^3 + (h_1^3 + h_2^3)^2 a_1 a_2 + \cos 2\theta \, a_1 a_2 (h_1^3 - h_2^3)^2}{h_2^3 a_1 + h_1^3 a_2} \tag{4}$$
$$F' = \frac{\rho h_2^3}{12\mu}$$

The load capacity of the thrust bearing per unit of length x results from the integral of the pressure distribution:

$$T = \frac{b + c}{2} (P - p_0) = \frac{b + c}{2} \frac{Ecb}{Fc + F'b} \tag{5}$$

The mean flow along y per unit of circumferential length is

$$V = E - \frac{P - p_0}{b} F \tag{6}$$

Figure 2 is the flow chart of the model analysis where it is presented how the application of the main assumptions of Whipple's model allow to obtain the above equations.

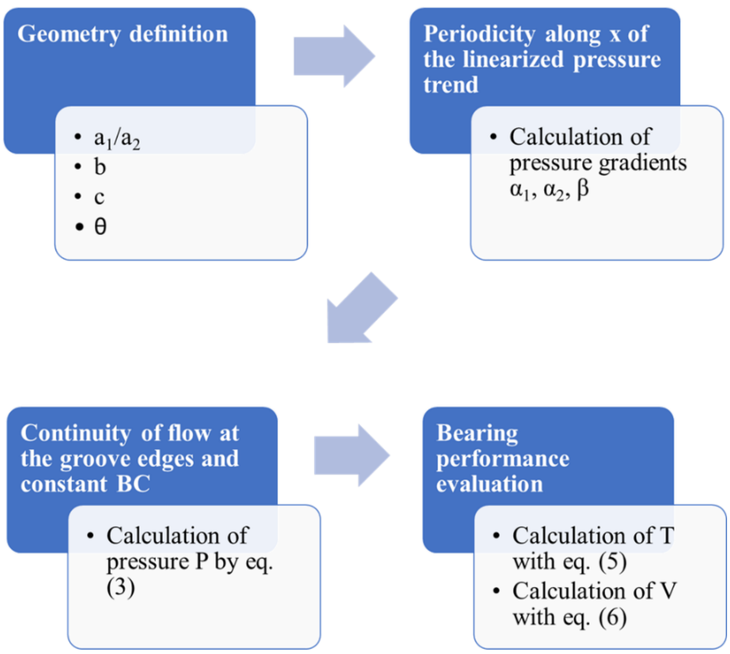


Fig. 2. Flow chart of the Whipple model

3. Sensitivity analysis

The number of grooves in this model is not considered, as the formulas are valid for an infinite number of grooves. Moreover, the fluid (air) is considered uncompressible with density $\rho = 1.2$ kg/m^3 and viscosity $\mu = 1.81 \cdot 10^{-5}$ Pas at 20°C. The pressure distribution is affected by the angle θ and by the following dimensionless quantities that are introduced to represent the bearing geometry: $X = \frac{a_1}{a_2}$, $Y = \frac{r_3 - r_2}{r_3 - r_1} = \frac{b}{b+c}$.

A sensitivity study is carried out in this paper in order to evaluate the effect of these three parameters on the maximum pressure *P* and on the load capacity *T*. The following reference geometry is considered from which variables are changed one at a time: r_3=20 mm, r_1/r_3=0.5, θ=45°, *X*=1, *Y*=0.5. r_3 is a fixed parameter from the target size of the compressor unit for fuel cells. The curves are plotted for three different groove depths (h_1-h_2=3, 5 and 10 µm) with h_2=5 µm and rotational speed ω=200 krpm, which is the target rotational speed for the turbocompressor to be designed.

From the sensitivity analysis it is evident that the load capacity increases with the groove depth; the angle θ at which the maximum is located increases with the groove depth, see figure 3. Notice that in case the angle is null or 90° the load capacity is null. In the first case, the groove would be radial and the speed component of air along the groove is null. In the second case, the groove would be circumferential, with no possibility of pumping the air inside the bearing.

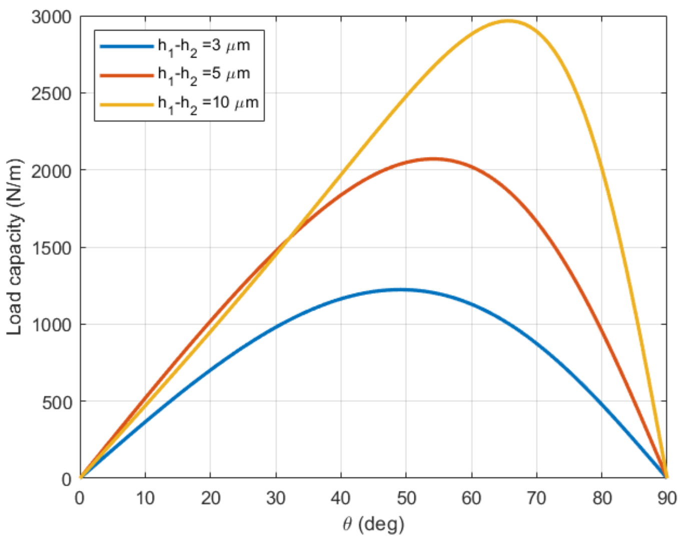


Fig. 3. Evolution of load capacity with respect to the angle θ at 200 krpm.

Ratio *X* influences the load capacity for values lower than unity; for higher values, the effect is less evident as visible in figure 4. Notice that in case *X* = 0 there is no groove, and the load capacity is null.

The effect of Y is evident in figure 5: the optimal value of Y depends on the grooved depth and increases with it. In case $Y = 0$ there is no groove, while in case $Y = 1$ the groove is complete. In the latter case the pressure difference across the groove is null, so the air is pumped inside the bearing without a compression ratio. In both cases, the load capacity is null.

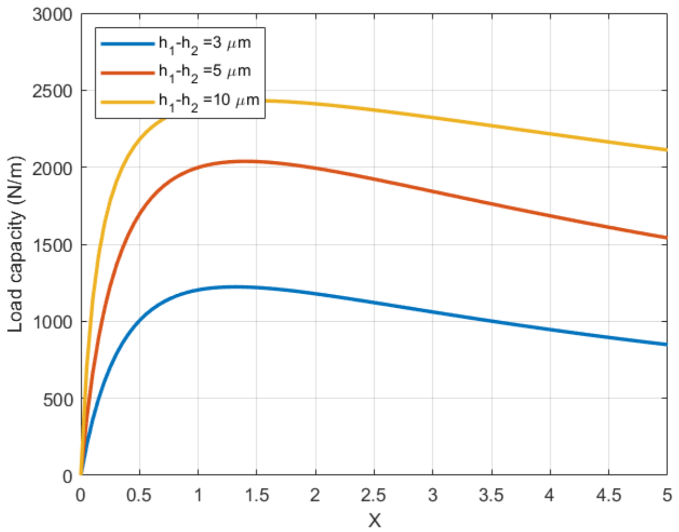


Fig. 4. Evolution of load capacity with respect to X at 200 krpm

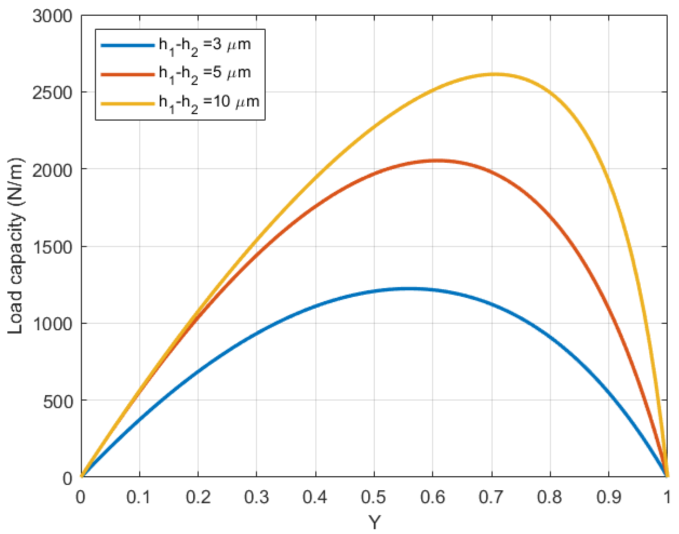


Fig. 5. Evolution of load capacity with respect to Y at 200 krpm

4. Geometry optimization

Defined the groove depth equal to 5 μm and the film thickness above the ridge h_2=5 μm, the optimal geometry is searched for maximizing T at ω=200 krpm in the following ranges:

$$0.5 < X < 1.5$$
$$0.2 < Y < 0.8$$
$$30° < \theta < 70°$$

Maximization of the load capacity was carried out in Matlab by means of a goal-seeking algorithm that looks for the values of X, Y and θ such that the function T has the highest value within the above ranges. The outcome of the geometry optimization process is shown in figure 6 and the corresponding bearing geometry is schematically represented in figure 7. The maximum specific load capacity of a SGTB operated at 200 krpm is about 2090 N/m.

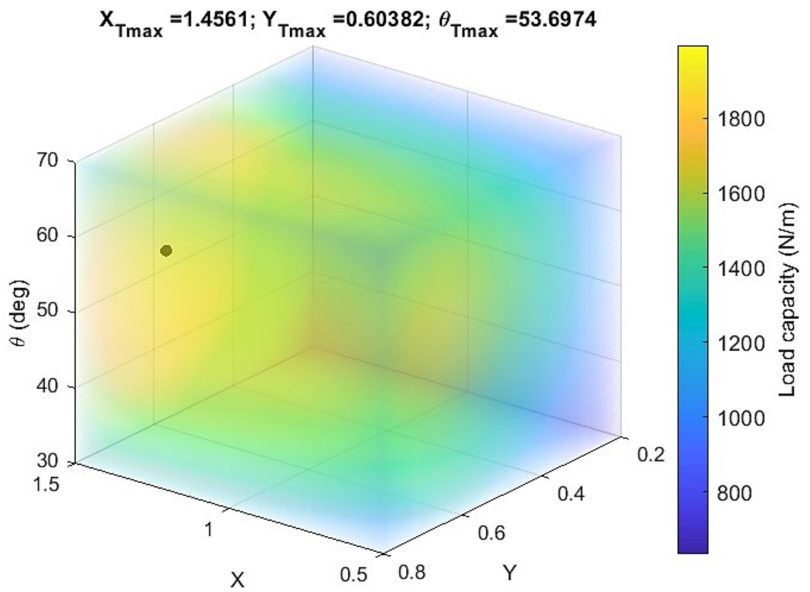


Fig. 6. Maximization of function T with respect to X, Y and θ at a time

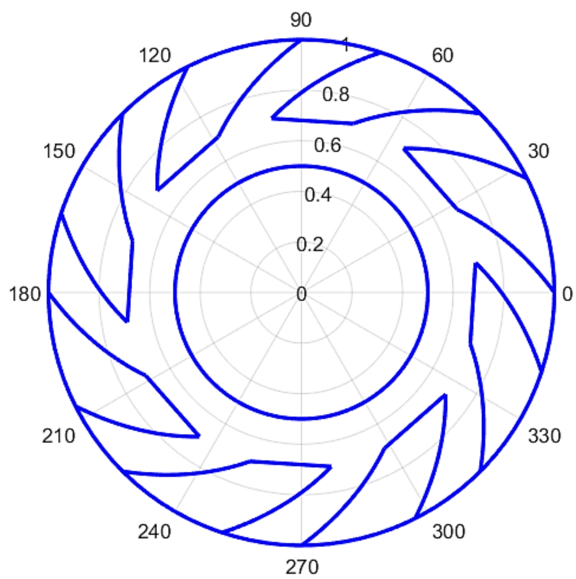


Fig. 7. Optimal geometry of a SGTB to maximize the load capacity at 200 krpm according to the NGT, considering 8 grooves.

5. Conclusions

This paper recaps the so-called Narrow Groove Theory (NGT) developed for the investigation of gas bearings with spiral and the herringbone groove pattern and proposes an optimization analysis to maximize the load carrying capacity of a SGTB operated at 200 krpm. Maximization of the performance is crucial for small-scale applications where thrust bearings as small as possible are needed to limit side-effects due to dynamics at extremely high rotational speed.

The optimization was carried out considering a specific mobility application where SGTBs are to be designed as guiding elements of a turbocompressor for fuel cells.

Dimensionless parameters describing the key features of the bearing geometry are taken into account to maximize the load carrying capacity so that the results thus obtained can be applied to a whole family of SGTBs geometrically similar with each other but with different sizes.

The sensitivity analysis allows to evaluate the effect of each parameter and find the local maxima of the load capacity function around the reference geometry. The result of the optimization suggests that the maximum load carrying capacity of about 2090 N/m is reached for SGTBs as small as few tens of millimeters operated with air at 200 krpm.

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