Tribology International 105 (2017) 77–84

Contents lists available at ScienceDirect

Tribology International

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

Effect of environmental pressure enhanced by a booster on the load capacity of the aerodynamic gas bearing of a turbo expander

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ARTICLE INFO

Keywords:
Aerodynamic gas bearing
Environmental pressure
Load capacity
Booster impeller

ABSTRACT

To improve the load capacity of the gas bearing by changing environmental pressure, this study analyzes the effect of environmental pressure on the load capacity of a bump foil gas aerodynamic bearing at different rotation speeds. Results show that the load capacity of the bump foil gas aerodynamic bearing increases with the increase in environmental pressure. Specifically, the load capacity of the aerodynamic gas bearing increases by 19.97%, 28.89%, and 44.28% at a speed of 60, 100 and 200 kr/min, respectively, when environmental pressure increases from 0.1 to 0.5 MPa. And the load capacity of bump foil gas bearing changes with the environment at different slenderness ratio is analyzed when rotation speed is 100 kr/min and eccentricity ratio is 0.6, the results show that under the same working load, increasing the environmental pressure of gas bearing and improve the load capacity gas bearing can reduce the slenderness ratio of the bearing; consequently, the shaft length is shortened, and the load of the aerodynamic gas bearing is reduced. Moreover, increasing the environmental pressure of gas bearing can shorten the higher the proportion of rotor length for the bigger slenderness ratio of gas bearing. This characteristic is more beneficial to the steady running of rotor at high speed. For a turbo expander that adopts a fan brake, the fan sucks and discharges air from and to the atmosphere directly; hence, the power provided by the turbo expander is wasted completely. This study recommends replacing the fan brake of a turbo expander with a booster and supplying pressurized gas to the gas bearing to enhance the environmental pressure at both ends of the bearing; this measure can improve the load capacity of the aerodynamic gas bearing and thus reduce energy dissipation. In this paper, Numeca software is utilized to analyze the internal flow of the booster impeller of a turbo expander with 69.68 kJ/s refrigerating capacity and 66.2 kW effective shaft power. On the premise of the power coupling–matching of the expander and booster, when the total pressure ratio of the booster is 3.5, 4.0, and 4.8, the slenderness ratio of the bearing can be reduced 10%, 15%, 25%, respectively, at a speed of 60, 100 and 200 kr/min for the turbo expander rotor system supported by gas bearing that the initial slenderness ratio is 1.

1. Introduction

The turbo expander is an equipment utilized to obtain low temperature and recycle energy. In the cryogenic turbo expander, the temperature of the inlet medium is low, the expansion ratio is high, and the specific enthalpy drop of the expansion medium is large. Generally, the rotor speed of the turbo expander is very high that it exceeds 10^4 r/min [1–3], e.g., the rotation speed of turbo expander reaches 196 kr/min in the literature [4]; a high speed turbo expander with gas bearing has been studied in literature [5,6], the design rotation speed is 300 kr/min and the maximum rotation speed is 342 kr/min. The operation requirements of an equipment are difficult to achieve with oil-lubricated bearing under such a high speed. Gas bearing has the advantages of light weight, high speed, frictionless, non-polluting, and it can be steady running at the high and ultra-speed [7–9]. Recent years have seen an increasing number of researchers to explore aerodynamic gas bearing. Such as active lubrication applied to gas bearings is presented [10,11]. They proved that it is possible to avoid common drawbacks associated with gas bearings, namely poor carrying capacity and low damping properties, which often lead to a reduced stability range. Xuedong Chen and Xueming He studied the effects of the recess shape of the gas-lubricated bearing on the performance analysis through numerical method [12], they found that the rectangular recess can provide larger load capacity than the spherical recess. Jianjun Du et al.

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http://dx.doi.org/10.1016/j.triboint.2016.09.027
Received 17 August 2016; Accepted 18 September 2016
Available online 20 September 2016
0301-679X/

studied the influences of structural parameters of pressure-equalizing grooves (PEGs) on the load capacity and stiffness of externally pressurized gas journal bearings and found that opening the axial PEGs is more helpful to improve the load capacity than opening the circumferential PEGs [13]. Yu Hou et al. analyzed the static and dynamic characteristics of a new type of foil journal bearings with protuberant foil structure and indicated that the bearing characteristics are significantly affected by the increase of bearing number and eccentricity ratio, particularly with respect to small bearing number [14]. Kai Feng et al. proposed a comprehensive theoretical model of flexure pivot tilting pad gas bearings (FPTPGBs) with metal mesh dampers (MMDs) in parallel by coupling the FPTPGB model and MMD model [15].

The lubrication medium of an aerodynamic gas bearing is compressible fluid. The flowing pressure of the compressible lubricating film is related to the environmental pressure. The supporting properties of aerodynamic gas bearing have been investigated extensively, but the effect of the environmental pressure of the bearing end on gas bearing properties has not been reported.

This study focuses on the bump foil gas aerodynamic bearing to analyze the effect of the environmental pressure of the bearing end on the gas bearing properties. The finite difference method and Newton-Raphson iteration method are adopted to solve Reynolds equation and gas film thickness equation.

In addition, for several turbo expanders that employ a fan brake (e.g., small cryogenic turbo expander), the fan sucks and discharges air from and to the atmosphere directly; hence, the power provided by the turbo expander is wasted completely. To recycle this part of energy, we recommend replacing the fan impeller with a booster and supplying pressurized gas to the gas bearing to enhance the environmental pressure at both ends of the bearing. As a result, the load capacity of the aerodynamic bearing can be improved. The design of the fan brake often focuses on simplified structure and improved mechanical strength; efficiency is often a secondary consideration. This study investigates the fan impeller as a booster, which not only consume the mechanical work of the turbo expander but also compresses the gas and does work on it to increase the gas pressure from the outlet nozzle. The flow in the internal mechanical part of the impeller is a highly complicated viscous and compressible unsteady 3D flow. The FINE/Turbo solver of the NUMECA software employs the time marching method to solve Reynolds averaged N–S equations and is applied to compressible and incompressible flows across a range of low, transonic, and hypersonic speeds. In this paper, the internal flow of the booster impeller is numerically analyzed with the NUMECA software to provide a theoretical foundation for the feasibility of improving the environmental pressure of the aerodynamic gas bearing through a booster.

2. Effect of environmental pressure on the load capacity of the bump foil gas aerodynamic bearing

2.1. Mathematical model

Fig. 1 shows a schematic of a bump foil gas aerodynamic bearing. φ is attitude angle of the bearing.

(1) Reynolds equation

In this paper, the N-S equation for three-dimensional compressible flow is solved for numerical simulations. The dimensionless form of compressible gas static Reynolds equation at the isothermal condition is:

$$\frac{\partial}{\partial\theta}\left(PH^3\frac{\partial P}{\partial\theta}\right)+\frac{\partial}{\partial\xi}\left(PH^3\frac{\partial P}{\partial\xi}\right)=\Lambda\frac{\partial(PH)}{\partial\theta} \quad (1)$$

The dimensionless parameters are:

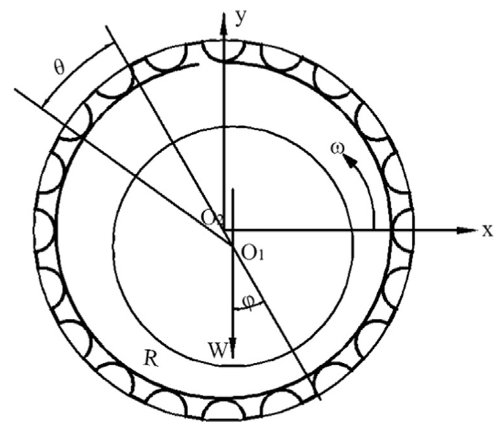

Fig. 1. Schematic of the bump foil gas aerodynamic bearing.

$$x=R\theta;\ y=R\xi;\ p=P_aP;\ h=cH;\ \Lambda=6\frac{\omega\mu}{p_a}\frac{R^2}{c^2} \quad (2)$$

Where: P—dimensionless gas film pressure;
P_α— environmental pressure (Pa);
R—bearing radius (m);
H—dimensionless gas film thickness;
c—bearing gap (m);
ω—angular speed of bearing collar ($rad{\cdot}s^{-1}$);
ξ—dimensionless axial coordinate;
θ—dimensionless circumferential coordinate;
Λ—bearing number.

(2) Dimensionless gas film thickness equation

Considering the specific conditions of bump foil and top foil, it has to make the following assumptions: (1) using single bump foil, the stiffness of bearing surface is well distributed and a fixed value; (2) top foil with the whole displacement of bump foil is only considered between the two adjacent wave crest; (3) deformation caused by load only depends on the load of the point.

The lubricating gas film thickness is formed by the film thickness that take no account of the foil deformation and bump foil deflection, the corresponding expression of dimensionless gas film thickness is:

$$H=1+\varepsilon\cos\theta+\alpha(P-1) \quad (3)$$

Where the dimensionless parameter is:

$$\alpha=\frac{P_a s}{k_{b0}c} \quad (4)$$

Where: ε—eccentricity ratio;
s—element length of bump foil (m);
k_{b0}—bump foil stiffness of horizontal element length (N/m^2).

(3) Boundary conditions

The dimensionless boundary conditions of the computational domain are as follows:

$\xi=\pm1,\ P=1$
$\theta=0,\ P=1$
$\theta=2\pi,\ P=1$

2.2. Results and discussion

Bearing structure and the relevant parameters of lubricating gas are shown as Table 1.

Fig. 2 shows the comparison between numerical analysis results of effect of rotation speed on load capacity and document data at different eccentricity ratio of foil bearing when the slenderness ratio L/D is 0.75.