Preliminary experimental study on static loading characteristics of multi-decked protuberant foil thrust bearing

Tianwei LAI*, Shuangtao CHEN*, Bin MA*, Liqiang LIU** and Yu HOU*

* State Key Laboratory of Multiphase Flow in Power Engineering, Xi’an Jiaotong University, Xi’an 710049, PR China
** Key Laboratory of Cryogenics, Technical Institute of Physics and Chemistry, Chinese Academy of Sciences, Beijing 100190, PR China
E-mail: yuhou@mail.xjtu.edu.cn

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Abstract
In the fluid film lubrication of high speed turbomachinery, bearing performance plays a significant role in the dynamics of the rotor-bearing system. In this paper, a novel foil thrust bearing using multi-decked protuberant foils as supporting structure is developed and tested. Experimental bearing test rig using gas-driven high speed turbo-expander is set up to evaluate static loading characteristics of the bearing. Static stiffness, friction torque, nominal bearing clearance and axial vibration amplitude are discussed in the transient processes. Static loading characteristic of the bearing is evaluated based on the relationship between displacement and loading force. Test results indicate that this simply configured thrust bearing has a good prospect in high speed hydrodynamic lubrication.

Key words : Thrust bearing, Foil bearing, Multi-decked, Protuberant, Loading character

1. Introduction

Gas foil bearing (GFB) is a type of compliant surface hydrodynamic bearing that has been applied in rotating machinery with higher speed and heavier load, such as turbocharger (Lee, et al., 2013), cryogenic turbo-expander (Andres, et al., 2014; Hou, et al., 2004), compressor (Conboy, 2013), generator (Kim, et al., 2013), micro turbomachinery (Dykas, et al., 2013). Comparing to conventional ball bearing, foil bearing is outperformed with high speed, reliability, durability and wide temperature capabilities (Agrawal, 1997; Ryu K and Andres, 2013; Howard, et al., 2001a, 2001b; Heshmat, et al., 2005). It is considered to be a competitive alternative to conventional bearings. Nevertheless, gas lubrication has its own limitation of low viscosity that leads to smaller load capacity. Besides, in high speed range, stability is another concern.

Since its first application in hydrodynamic lubrication (Blok and Van Rossum, 1953), extensive effort has been devoted in developing the elastic supporting structure to improve bearing performance. One of the earliest and traditional supporting structures is the corrugated bump type (Heshmat, et al., 1981; Heshmat, et al., 1983a, 1983b.). In order to improve the bearing performance, many other configurations have also been developed. Luis Andres et al. adopted metal mesh as supporting structure in journal bearing (Chirathadam and Andres, 2013; Andres and Chirathadam, 2012). Ju-ho Song used compression springs and presumed that the stick-slip at the interface between spring and bearing sleeve induces rather high stiffness (Song and Kim, 2007). Tae Ho Kim presented multi-stage elastic supports that provide larger stiffness and damping coefficients for enhanced material hysteresis (Kim and Andres, 2007). Quan Zhou tested dynamic characteristics of viscoelastic supported air foil thrust bearing (Zhou, et al., 2009). However, due to its physical property, fluorine rubber can not be used in extreme temperatures. Daejong Kim et al. combined hydrostatic injection with conventional foil bearing to develop the hybrid foil bearing (Kumar and Kim, 2010). This hydrostatic supply gas ensures smooth running in avoidance of the dry rubbing and wear issues during the start-up and coast-down processes (Kim and Park, 2010). Yu Hou et al. performed numerical studies on the performance of protuberant foil journal bearing (Hou, et al., 2011). Then, the same research group experimentally studied the effects of bearing clearance and supporting stiffness of the foil journal bearing with multi-decked protuberant configuration. In application tests on a 25mm-diameter turboexpander around 100 krpm, subsynchronous
vibration was suppressed quite well (Hou, et al., 2013; Lai, et al., 2014).

In bearing design, load capacity is an important parameter. It is affected by many factors such as the compliant foil deformation, the lubricant thermal properties and the flow regime in bearing. Bearing stiffness measurement is the basis in the selection of bearing clearance, preload, etc. Franck Balducchi experimentally investigated the static stiffness and the start-up torque of a foil thrust bearing with a mild static load ranging from 5 N to 60 N and rotational speed from 20 krpm to 35 krpm (Balducchi, et al., 2013). Hooshang Heshmat et al. analyzed bump geometry parameters and load distribution profile effect on bearing stiffness; they also found that Coulomb friction and higher stiffness could be realized by increasing friction coefficient between top foil and bumps (Ku and Heshmat, 1992; Ku and Heshmat, 1993). Yong-Bok Lee attributed the stiffness increase to the Coulomb friction between the foils (Lee, et al., 2008). In addition, the tilting of thrust bearing could increase the bearing load and friction torque (Park, et al., 2008). Le Lez studied the static characterization of a bump type foil bearing and figured out that loading profile affected area of hysteresis plots by increasing or decreasing the stick or slip phases (Le Lez, et al., 2007). Besides, the nonlinearity in stiffness is imperiously necessary in the development of a clear design procedure (Rudloff, et al., 2011).

Generally, thrust bearing is designed to constraint the axial vibration. Actually, it strengthens the overall supporting stiffness to some extent. Experimental study on the load capacity of gas foil thrust bearing is fundamental and basic research for the precise design of rotor bearing system. After decades of years of improvements, bump foil bearing is substantially developed. However, it is enslaved to its high investment and complex crafting processes in manufacturing. In this paper, a new type of thrust bearing with multilayer of protuberant foils as supporting structure is presented. Multi-decked protuberant foil thrust bearing is characterized for its simple manufacturing and maintenance. A gas-driven thrust bearing test rig has been built for the tests. The stiffness and transient load capacity of the thrust bearing are assessed, which provides good insight for further studies.

2. Experimental setup
2.1. Multi-decked protuberant foil thrust bearing

The schematic diagram of multi-decked protuberant foil thrust bearing is shown in Fig. 1. The thrust bearing is consisted of stainless steel bearing housing, supporting metal foil pads and fixing pins. Four thrust loading pads are used in the bearing and each thrust pad spans 90˚ in circumferential direction. Twelve fixing holes are bored on the outer diameter range of bearing housing and loading pads. The four loading pads are fixed on the bearing housing by pin. One advantage of pin fixing is the convenience in assembly and disassembly during test processes. The supporting structure in each loading pad consists of two parts: top flat foil and supporting structure. The top foil provides compliant loading surface for the bearing. The elastic supporting structure has two layers of protuberant foils that stack up one over another. The protuberant foil is placed with convexes facing upward to support the upper foil.

![Figure 1 Three dimensional diagram of the protuberant foil thrust bearing](image)

The layout of the two multi-decked protuberant foils is shown in Fig. 2. Convexes on bottom protuberant foil are denoted by the crossing symbols, while the convexes on upper protuberant foil are denoted by the circles. In order to provide support for upper foil, the convex distribution of bottom protuberant foil is different from that on upper foil.
2.2. Gas-driven thrust bearing test rig

High speed gas-driven bearing test rig is built based on the test rig (Zhou, et al., 2009). The process diagram of the test rig is shown in Fig. 4. The test rig is made up of three systems: the rotor-bearing system, the high pressure gas supply system and the data acquisition system. Pressurized gas is provided by a double screw compressor with supply pressure up to 1.1 MPa. After air filter, oil-water separation, and pressure reservoir, gas supply pipeline is connected to indoor test rig. It is distributed into three routes: the hydrostatic supply gas, the expander gas and the loading gas. Hydrostatic supply gas is for the journal bearings that support the weights of rotor and loading cell. Expander gas is used for high speed turbo-expander that converts high pressure potential energy into high rotational speed. The load on the thrust bearing is controlled by loading unit pressure.
The detailed thrust bearing testing part is shown in Fig. 5. The bearing test part is composed of hydrostatic rotor-bearing system and loading unit. The 17 mm-diameter rotor is driven by a turbo-expander, which provides high velocity to generate hydrodynamic pressure between the thrust disc on the shaft and top foil of thrust bearing. The double-row hydrostatic journal bearings are utilized to provide radial support. Axial load of the shaft with turbo expander is balanced by a hydrostatic bearing with gas supply holes. Expansion wheel is reaction type and its rotational speed depends on the expansion gas supply pressure. The loading unit consists of a connecting disc, a piston rod, a cylinder and a hydrostatic journal bearing. The thrust bearing is connected with piston rod by the connecting disc, and its weight is supported by hydrostatic journal bearing. The tested hydrodynamic foil thrust bearing is bolted at the left end of loading unit. Friction torque transducers are linked with the connecting disc with elastic strings. For alignment of the connecting disc, two symmetrically placed torque transducers are applied to measure the friction torque of the bearing. The other main parameters of the rotor are shown in Table 1.
The thrust bearing test part and tested foils are shown in Fig. 6. The high speed rotor-bearing system and the loading unit are fixed on a stainless supporting frame individually, positioned by high-precise guiding rail. For convenience of assembly and instrumentation, separated supporting structure is adopted. Stainless supporting frames are subdivided from one pre-machined component to ensure coaxiality.

![Figure 6 Gas-driven thrust bearing test rig and tested foils](image)

| Parameter                      | Value    |
|--------------------------------|----------|
| Rotor diameter/mm             | 17       |
| Rotor length/mm               | 170      |
| Outside diameter of rotor disc/mm | 34.5   |
| Outside diameter of expander wheel/mm | 12    |

2.3. Experimental instrumentation

Data acquisition is developed with sampling interval of 1 second. Logic, control, computing and operation platform are connected to an industrial computer (2.8GHz). Rotor speed and vibration signals are measured by eddy current displacement transducer (linearity <±2% and static resolution of 0.1 μm). Friction torque is measured by a key type torque transducer with sensitivity 1.5 ±0.01MV/V and accuracy ±0.05%F.S. Multi-functional board is used to ensure high accuracy under high sampling rate.

2.4. Experimental procedures

In the transient processes, the tested parameters include nominal bearing clearance and rotor speed. On account of the high rotor speed of 100 krpm, dynamic responses to the loading gas pressure are around 100 ms. Static and transient tests are both conducted to monitor the bearing working condition. The testing procedure is arranged as follows:

1) Set the output pressure of gas compressor around 1.0 MPa to meet the pressure requirement of turboexpander; stabilize the hydrostatic pressure around 0.65 MPa; adjust the supply pressure of loading cell until bearing preload is around 4.3 N;

2) Open the valve to increase the supply gas pressure of turboexpander until rotor begins to rotate, and then raise supply gas pressure further to attain smooth operation of turboexpander with the rotor speed stabilized in the range from 40000 rpm to 115000 rpm for at least 60 seconds;

3) Increase load unit pressure until turboexpander speed drops, repeat step (2);

4) Repeat step (3) until instability occurs. In order to prevent the rotor damage, it is necessary to reduce the expander supply pressure until the rotor comes into a complete stop;

5) Reduce the supply pressure of turboexpander, loading unit and hydrostatic journal bearing. Inspect the experimental test parts and prepare for the next set of tests.
In actual manufacturing, from the molding of protuberant foil to the assembly processes, the error accumulation is inevitable. Therefore, several sets of foil bearing have been tested to validate the feasibility of this new configuration.

3. Test results and discussion

3.1 Static stiffness of the thrust bearing

Static structural stiffness of thrust bearing is crucial for the transient loading process. Bearing deflection and stiffness with thrust load are shown in Fig. 7 a) and b), respectively.

As shown in Fig. 7 a), the deflection of compliant foil increases with load magnitude. In the loading process, the deflection decrement reduces with thrust load. Relation between load and deflection can be described by a quadratic curve. In the load releasing process, the deflected foils have a tendency to restore to its initial position, but the friction force prevents the relative movement between the foils. Thus, under the same thrust force, the axial deflection of the bearing is higher than that in loading process. As a consequence, the hysteresis loop appears during the loading and unloading process. To a certain extent, the energy can be dissipated under this loading and unloading process. Furthermore, it provides more coulomb damping for the foil bearing in operation.

![Figure 7 Static performances of the thrust bearing: a) load b) stiffness](image)

Based on the relationship between the thrust load and deflection, stiffness of the thrust bearing can be derived. Bearing stiffness with load is shown in Fig. 7 b). In small loaded regime 0 to 10 N, the bearing stiffness is almost constant around 0.1 MN/m. Thereafter, the stiffness increases quadratically with the thrust force. During the process of the transition from loading to unloading, bearing stiffness jumped from 1.25 MN/m to 2.0 MN/m. Coulomb friction forces between foils change acting directions to prevent the relative displacement between the foils. Some of protuberant structures may be still kept in squeeze contact with upper foil. Therefore, there is a very stage when axial deflection with load being too small to cause a sudden increase of stiffness. In the unloading process, the stiffness and thrust force are also in a quadratic relation. At the beginning of unloading, bearing stiffness is greater than that in the loading process. When bearing load is reduced to a value around 25 N, bearing stiffness is almost the same as that in loading process. It is observed that, for this type of bearing, stiffening effect from Coulomb friction is more prominent in heavier loaded regime.

Unlike bump type structure, there is no clear evidence of the small hysteresis loop contained in a large hysteresis loop (Ku and Heshmat, 1993). Another important distinction exists in respect of the loading-stiffness characteristics. For the bump foil structure, stiffness versus load relation is nonlinear in lightly loaded region and trends to be asymptotically stable in heavier loaded region (Ku and Heshmat, 1993; Rudloff, et al., 2011). For this multi-decked structure with 0.05 mm thick protuberant foils, bearing stiffness is almost linear in lightly loaded region and highly nonlinear in heavier loaded range.

3.2 Bearing performance in transient loading and unloading

Based on the static structural stiffness test, the basic mechanical feature of the bearing is obtained. In practical application of hydrodynamic thrust bearing, however, actual stiffness of thrust bearing is generated by both the elastic structure and the gas film. Therefore, it is necessary to evaluate the transient loading capacity of the bearing under different loads.

3.2.1 Preliminary mild load test

Following the test procedures in previous section, preliminary mild loading and unloading tests are carried out. A complete transient loading test of protuberant foil thrust bearing is shown in Fig. 8.
The nominal operating speed of turboexpander is 90 krpm with a supply pressure of 0.075 MPa. The preload force on the thrust bearing is 4.3 N. After smooth operation for 60 seconds, the loading force is increased by raising the loading pressure. The load is increased from 4.3 N to 5.7 N. Within this range of load, the input pressure at expander keeps constant at 0.075 MPa. Rotor speed decreases due to the higher friction torque between the thrust bearing and the rotor disc under the same driving pressure. When the friction loss at the thrust bearing increases, the rotor speed drops from 90 krpm to 80 krpm.

In the coast-down process, expander supply pressure is reduced from 0.075 MPa to 0.042 MPa with a constant bearing load of 4.7 N. Afterwards, the rotor speed is reduced from 102 krpm to 65 krpm in a stepwise shape. Then, rotor speed decreases gradually in that the friction loss at bearing side is larger than the driving power at expander.

3.2.2 Higher load variation tests

After the transient mild loading tests, higher thrust load is imposed on the bearing in high speed operation. The expander supply pressure, axial vibration, nominal bearing clearance, and friction torque are compared under four different loads: 10 N, 15 N, 20 N and 25 N.

For gas-driven high-speed turboexpander, vibrations in radial and axial directions are inevitable in transient process. Axial vibration of the turboexpander is shown in Fig. 9. The vibration amplitude in axial direction is almost linear with the rotational speed in the tests. When the load is 10 N, axial vibration is around 1.5 μm at 46 krpm and increases to 2.5 μm at 107 krpm. With increase of bearing load, vibration amplitude decreases at the same rotational speed. For instance, at 100 krpm, vibration amplitude is around 2.0 μm when the load is 10 N. When the load reaches 25 N, vibration amplitude is reduced to about 1.4 μm at the same rotor speed.
After taking off of the thrust disc from the bearing, the deformation of compliant supporting admits lubricating gas into the clearance. In actual operation, the gas film thickness is not uniform for this foil bearing. Therefore, in order to calculate the stiffness of bearing in operation, nominal bearing clearance is conceptually proposed by measuring the relative displacement between thrust disc and bearing surface. In transient processes, nominal bearing clearance maintains almost a linear relationship with rotational speed in Fig. 10. Under the same load, e.g. 10 N, nominal bearing clearance increases from around 15 μm at 50 krpm to 33 μm at 105 krpm. This can be attributed to the hydrodynamic strengthening effect between rotor disc and bearing. With increase of bearing load, thinner nominal bearing clearance is required to generate higher hydrodynamic pressure at the same rotor speed.

Thrust bearing friction torque is proportional to relative velocity between rotor and disc, viscosity and fluid film shear rate. Friction torque of the bearing is shown in Fig. 11. Bearing friction torque increases almost linearly with the bearing load, but the magnitude of increment is generally small. Taking the friction torque under 20 N for instance, it increases by 6.9% from $4.3 \times 10^{-3}$ N·m to $4.6 \times 10^{-3}$ N·m when the rotor speed increases from 57 krpm to 118 krpm. For the bearing with a load of 15 N, friction torque remains almost constant around $3.7 \times 10^{-3}$ N·m. Under the same operating speed around 80 krpm, frictional torque is $4.7 \times 10^{-3}$ N·m with a load of 25 N. For the bearing with a load of 10 N, the friction torque is about $3.5 \times 10^{-3}$ N·m.
In the test, the main loss of turboexpander comes from three parts: multi-decked protuberant foil thrust bearing, the reaction type expansion wheel and the hydrostatic journal bearing. When the friction torque at the thrust bearing varies, supply pressure has to be adjusted correspondingly to maintain the rotational speed. The supply pressure at expander is shown in Fig. 12. Under the same load, supply pressure increases linearly with rotational speed to compensate for the friction loss at the bearing side. Under heavier load, higher gas supply pressure is required to maintain the same rotational speed.

The load capacity and load capacity coefficient of the thrust bearing are shown in Fig. 13. After stable start-up, bearing load increases from 16.6 N to 26.3 N linearly with the rotational speed until 70 krpm. From the perspective of load capacity coefficient, it increases slightly from 0.81 N/(mm³·krpm) to 0.82 N/(mm³·krpm) with rotational speed from 45 krpm to 70 krpm. Since then, further increase of the rotational speed does not make any contribution to the bearing load. Bearing load reaches a limit and remains stable regardless of rotational speed in the range from 70 krpm to 105 krpm. Bearing load capacity coefficient drops with further increase of rotational speed. It is interesting to note that bearing load declines as rotational speed exceeds 105 krpm. It might be explained by the side leakage of hydrodynamic film from the compliant top foil. However, this cannot be avoided although there are some sealing effects (Dellacorte and Valco, 2000). Especially in the higher rotational speed regime, the hydrodynamic pressure at the outer circumferential region increases, which weakens the sealing effectiveness of the top foil. Besides, the thermal effects caused by high shear-rate in fluid film might be another possible incentive.
4. Conclusion

In this paper, a new type of thrust bearing with multi-decked protuberant foils as supporting structure is developed. Static structural stiffness and transient load capacity of the thrust bearing is experimentally investigated. Unlike the bump foil bearing, structural stiffness of multi-decked protuberant structure is almost linear with the deflection in lighter loaded range, and the nonlinearity emerges when heavier load is imposed. At the instant of transition from loading to unloading state, smaller deflection and higher stiffness is observed in heavier loaded region. In the unloading process with lighter load, bearing structural stiffness is larger than that in the loading process. Stiffening effect from Coulomb friction is more prominent in heavier loaded regime.

In transient loading and unloading processes, experimental parameters in a complete test are presented. Under the same supply pressure, rotor speed decreases with smaller lubricant film and higher friction loss. In the static loading test, friction loss of bearing keeps almost a linear relation with the rotational speed but the increased percentage is quite small. Loading limit is observed when rotational speed exceeds a certain magnitude. In this speed range, hydrodynamic bearing load is independent of rotational speed. Further increase of speed causes the unexpected decline of bearing load.

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