Study on double-layer protuberant gas foil journal bearings with different foil layers arrangement

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Abstract
With the excellent damping performance, gas foil bearing (GFB) is becoming the promising hydrodynamic lubrication structure for high speed turbomachinery. To improve the performance of the conventional GFBs, a new double-layer gas foil bearing (PGFB) with double-layer protuberant strip as elastic support is developed. A 2D numerical model based on Reynolds equation and Kirchhoff equation is established to determine the gas film thickness, gas film pressure and foil deflection. Static performance and dynamic performance of three PGFBs with different elastic foil layers arrangement are presented and compared qualitative dues to the difficulties in measuring the practice eccentricity and load capacity with high speed rotor-bearing system. By applying in a turboexpander with \( \phi \) 25mm rotor diameter, the three PGFBs are tested. Because the three PGFBs share the same rotor structure and the same air supply pressure, the rotor drag torque and the bearing static load are almost the same in the tests. The experimental results show that all the PGFBs run well in turboexpander. Rotating speed, temperature and enthalpy differences and rotor synchronous vibration under the same turboexpander air supply pressure are compared. The numerical model predicted well the qualitative difference of the loading performance of the three PGFBs.

Key words: Hydrodynamic lubrication, Gas foil journal bearing, Double-layer protuberant foil bearing, Static and dynamic performance, Turboexpander

1. Introduction
The application of GFBs such as multi-pad foil bearings, Hydresil gas foil bearings, and multilayer foil bearings, in high speed turbomachineries (Agrawal, 1997) showed much better operation stability than hydrodynamic gas bearing with rigid lubrication surface. In fact, the most significant advantages of GFBs is the excellent damping performance in restraining and attenuating the vibration energy of the high speed rotor. For better stability at super high speed and extreme working condition, development and research on GFBs have attracted much attention. New structures of GFBs have been proposed and investigated in recent years by many researchers.

Among those novel foil bearings, the metal mesh gas foil bearing which was first proposed by Lee and Kim et al (2008) is representative. The metal mesh air foil bearing comprises a bearing cartridge, a top plate foil and a porous metal mesh foil. The structure is similar with the conventional Hydresil foil bearing. And the only difference is that the bump foil in Hydresil foil bearing was replaced by a metal mesh strip. Luis and Thomas et al. (2010) measured the stiffness and damping coefficients in a metal mesh foil bearing. The diameter and the length of the tested metal mesh foil bearing are 28.0 mm and 28.05 mm, respectively. 0.3 mm copper wire was used in the metal mesh strip, and the compactness was 20%. The large mechanical energy dissipation was indicated by the nonlinear load-deflection relationship with a sizable hysteresis loop. The motion amplitude was about 12.7 \( \mu \)m to 38.1 \( \mu \)m (frequency 25 Hz to 400 Hz) for the tested rigid shaft of diameter about 28.0 mm. The drag torque, lift-off journal speed and temperature of the same metal mesh foil bearing was also measured by Luis and Thomas et al. (2010). The obvious hydrodynamic film
was formed in their experiment. The friction torque decreased with the increasing static load, while the lift-off speed and bearing drag torque increased with the increasing static load. The influence of metal mesh density (compactness) on bearing static load performance was investigated by Lee and Kim et al. (2012). The tested 0.15 mm stainless steel wire metal mesh strip had the compactness of 13.1%, 23.2% and 31.6%. The tested rigid shaft diameter was 60 mm. With the increasing static load, the bearing deflection, stiffness and energy dissipation was analyzed. Large hysteresis loops was recorded during the loading process, which meant that the tested metal mesh foil bearing has excellent energy dissipation ability. The metal mesh foil bearing with 31.6% mesh density showed the best damping performance. Except the advantages of simple structure and inexpensive cost, the bearing performance was also disclosed by Luis and Thomas et al. (2012). Through the comparison of a metal mesh foil bearing and a bump-type foil bearing (Hydresil foil bearing), the improvements of metal mesh foil bearing in bearing performance were analyzed thoroughly. The tested new GFB showed the larger mechanical hysteresis loop, and its loss factor was 2 to 3 times larger than that of bump-type foil bearing. Meanwhile, metal mesh foil bearing showed the lesser dynamic stiffness and viscous damping and the much more structural damping.

By replacing the bump foil layer with an elastic rubber strip in the bump-type foil bearing, a new rubber pad foil bearing was proposed by Hou and Xiong et al. (2004). The uniform stiffness and viscous damping are the most significant advantages. The new structure displayed excellent stability when it was applied in a high speed turboexpander. The maximum speed reached 147,000 rpm (linear speed of the outer edge of the expansion wheel was about 280.8 m/s) which was 37% above the designed speed. The similar rubber strip was also applied by Lee and Kim et al. (2004) in their new structure. An elastic rubber strip was placed between the bump foil layer and the bearing cartridge. Although the viscous damping was improved, the structure with a rubber strip was limited in applications in extremely high or low temperature.

A new structure of multi-wound foil bearing was proposed by Feng and Kaneko (2007) and Feng and Kaneko (2009). The new foil bearing comprises a foil strip and the bearing cartridge. The foil strip was wounded triply in the inner surface of the bearing cartridge. The beginning 1/3 plate foil was placed on the top, with hemispherical projections the rest 2/3 foil was placed twice around the 1/3 top foil layer as the middle and bottom layers. The new structure had two layers of elastic foils which is referenced to the double-layer structure in this paper. The difference is that the foil bearing in this paper has two individual elastic layers. A 2D numerical model for multi-wound foil bearing was firstly developed by Feng and Kaneko. And the numerical results was validated by the experimental data. The deflections of the top foil and the mid-foil were obtained and displayed clearly in their literature (2007). Furthermore, their original research works gave the primary suggestions of the research works in the present article. Recently, Feng reported a novel multi-cantilever foil bearing, and tested the structural stiffness and equivalent viscous damping (Feng and Zhao, et al. 2014).

Hou and Chen reported a numerical simulation on the single layer protuberant foil journal gas bearing (Hou and Chen, et al. 2011). The static and dynamic performance were investigated. Due to the near rigid support stiffness of the elastic foil, the damage happened to the top plate foil in the experiment. Therefore, Hou and Lai designed a double-layer protuberant gas foil journal bearing (Hou and Lai, et al. 2013). The new structure run well in a φ 25 mm shaft turboexpander, and the maximum speed reached 100,000 rpm (linear speed of the outer edge of the expansion wheel was about 191 m/s). The rotor vibration was well suppressed due to the double-layer elastic foil structure. However, theoretical or experimental study on the effect of the arrangement of the two elastic foil layers on the bearing running performance have not been reported.

In this paper, a 2D numerical model is developed for describing the film thickness, pressure and foil layer deflections of the double-layer protuberant foil journal bearing. In this model, the attitude angle is obtained through iteration and the deformation of the middle foil layer is taken as the boundary condition of the top plate foil. FEM method is used to solve the coupled Reynolds and Kirchhoff equations. Three foil bearing with different middle foil and bottom foil arrangements are compared qualitative in respects of attitude angle, gas film thickness, gas film
pressure distribution, load capacity, gas film friction torque, stiffness and damping coefficients. Then the three double-layer protuberant foil bearings are applied in a turboexpander with a shaft diameter equal to 25 mm. with the uniform changing air supply pressure, the turboexpander flowrate, the rotating speed, the temperature and enthalpy differences and the synchronous vibration amplitude are compared experimentally. The AXI-PGFB shows the largest loading capacity from both the numerical results and the experimental results.

Nomenclature

| Symbol | Description                   | Unit   |
|--------|-------------------------------|--------|
| c      | bearing radial clearance (m)  |        |
| D      | rotor diameter                |        |
| e      | eccentricity (m)              |        |
| h      | film thickness (m)            |        |
| L      | half length of the bearing (m)|        |
| p      | pressure (m)                  |        |
| p_a    | ambient pressure              |        |
| q_0    | loading force of the foil layer (N) |    |
| r      | rotor radius (m)              |        |
| t      | time (s)                      |        |

2. Theoretical analysis

Figures 1(a) and (b) show the model of double-layer protuberant gas journal foil bearing (PGFB) and geometric parameters of the bottom and mid foil layers which have uniform hemispherical shell supporting points and placed between the top plate foil and the bearing cartridge. Three typical positional relations are showed in Figs.1 (d) to (f). The positional relation in Fig. 1(d) is marked with AXI which the supporting points of bottom foil layer are located between the supporting points of mid foil layer in axial direction. The CIR arrangement is showed in Fig. 1(e), the bottom foil supporting points are located between the supporting points of mid foil circumferentially. The arrangement that the supporting point of the bottom foil is located on the middle position of the adjacent four supporting points of the mid foil is named as MIX in Fig. 1(f). All of the three foil layers are fixed by the same ends opposite to the journal rotation direction, and free on the other end.

Figure 1 (a) also shows the coordinate system used in the following model. Reynolds equation is used to describe the hydrodynamic gas lubrication in conditions of isothermal and isoviscous. At the side edges of the foil layers i.e. \( z = \pm L, \ \theta = 0 \) or \( \theta = 2\pi \), the pressure is set as the ambient pressure \( (P_a = 0.1 \text{ MPa}) \).

\[
\begin{align*}
\frac{1}{R^2} \frac{\partial}{\partial \theta} \left( p h \frac{\partial}{\partial \theta} \right) + \frac{\partial}{\partial z} \left( p h \frac{\partial}{\partial z} \right) &= 6 \mu_0 \frac{\partial}{\partial \theta} (p h) + 12 \mu \frac{\partial (p h)}{\partial t} \\
\frac{\partial}{\partial z} & = \pm L, \quad p = p_a \\
\theta = 0 \quad \text{or} \quad \theta = 2\pi, \quad p = p_a
\end{align*}
\]

(1)

According to the geometric relationship in Fig.1 (a), the film thickness \( h \) is

\[
h = c + e \cos (\theta - \phi) + \frac{z}{L} \xi + w
\]

(2)

where \( c \) (Hou and Lai, et al. 2013) is the bearing clearance, and \( e \) is the eccentricity. \( w \) is the transverse deflection of the top foil structure due to the hydrodynamic pressure of the fluid film. \( \phi \) is attitude angle.

The top foil and mid foil is modeled as two-dimensional rectangular thin plate as shown in Fig. 2(a). The supporting points of mid foil and bottom are modeled as springs with huge stiffness (Hou and Chen, et al. 2011). Therefore, Kirchhoff equation is used to determine the transverse deflection \( w \) (perpendicular to the foil layer surface).

\[
\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^2 w}{\partial x^2 \partial y^2} + \frac{\partial^2 w}{\partial y^2} = \frac{q_0}{D}
\]

(3)
In the model, the circumferential slip of foil layers and elastic foil layers structure damping are ignored. As shown in Fig. 2(b), the reacting force of the top foil from the rigid supporting point acts on the corresponding points of the mid foil. The new supporting position of the top foil is lower than its original position. When the nodes displacement of the mid foil reach the maximum value, deflection occurs to the top foil. Due to the connection of the supporting points, the deformation of supporting nodes on mid foil can be transferred to the corresponding nodes on the top plate foil directly. However, the non-supporting nodes of mid foil and top plate foil has no direct relationship. The displacements of the mid foil supporting nodes can be used as the displacement boundary conditions in the calculation of top foil deformation. And the foil structure are fixed by the same end. Therefore, the boundary condition for Eq. (3) is

\[
\begin{align*}
\chi = 0, \theta = 0, & \text{ then } w = 0 \\
at \text{ contact points}, & \quad w_{\text{top}} = w_{\text{mid}}
\end{align*}
\]

(4)

Figure 1 Double-layers protuberant gas foil journal bearing and foil layer geometric parameters

Figure 2 Model and deformation of PGFB foil structure

In order to generalize the solution, Reynolds equation and the Kirchhoff equation are nondimensionalized, and the details were described by Hou and Chen et al. (2011). The static performance of PGFB can be obtained by solving the combined equation of Eqs. (1) to (4). Besides, Linear Perturbation Method (Hou and Chen et al. 2011, Peng and Carpino, 1993, Lund, 1987) is used to determine the dynamic parameters such as the stiffness and damping coefficients of PGFB. Finite Element Method is used to discretize the governing equations.

The flow chart of the program is concluded in Fig. 3. Through the solution of Reynolds equation the node force of the top plate foil is determined by the gas film pressure via Kirchhoff equation. The supporting node force of the mid foil is generated from the corresponding node force of the top plate foil. Therefore, the mid foil node displacement can be determined. Because the top plate foil and the mid foil are connected through the supporting nodes, the displacement of the mid foil supporting nodes can be used as parts of the boundary conditions of top plate foil deformation. With the fixed end boundary condition, the top foil deformation can be obtained. Then, the bearing static parameters and dynamic stiffness and damping coefficients can be obtained. There are 2132 elements in the calculation area, 27 nodes in \( z \) direction and 83 nodes in \( \theta \) direction. The supporting points are included in the FEM nodes.
3. Experimental study

Figure 4 shows the test rig and the tested turboexpander which was driven by a 75kW ATLAS screw air compressor. The buffer after the compressor was used to stabilize the pressurized air flow. The turboexpander inlet pressure and the supply pressure of the pressurized thrust air bearing were regulated by electric proportional valve which has the precision of 0.005MPa. Two 4mm diameter eddy current sensors (8.33 V/mm, linearity ±2%, static resolution 0.05 μm) were mounted with an angle of 90 degree in the middle section perpendicular to the turboexpander rotor.

The three tested PGFBs are made of two layers of stainless steel bump foil and a piece of beryllium bronze top plate foil. The arrangement of the three PGFBs are listed in Fig.1 (d) AXI-PGFB (e) CIR-PGFB and (f) MIX-PGFB. The parameters of the thrust bearings and the test PGFBs are listed in Table 1. The thickness of the foil stainless steel bump foil is 0.05 mm, and the thickness of the beryllium bronze top plate foil is 0.07 mm. The total height of the foil structure is 0.47 mm. The radius of the bearing housing is 12.99 mm. The main parameters of the turboexpander are listed in Table 2.

4. Results and discussions
4.1 Numerical results

In the numerical simulation, the nominal radial clearance is set as 20 μm, which is also the configuration radial clearance. The eccentricity ratio is set as 0.8. As shown in Fig.5, the numerical model in this article predicted successfully the dimensionless distributions of lubrication gas film thickness, pressure and foil deflection. Fig.5 (a) shows the distribution of dimensionless gas film thickness. The original average film thickness is the radial clearance 20 μm. Due to the eccentricity of the rotor, the film thickness shows the nonuniform distribution. In the unloaded area (≈ 0° to 60° and 270°-360°), the film thickness is larger than 20 μm. In the loading area (≈ 60° to 260°) the gas film thickness is smaller than 20 μm (the dimensionless value is less than 1). As shown in Fig.5 (b), the high pressure area corresponding to the area in which the gas film thickness is less than 20μm. MIX-PGFB has the largest high pressure (P > 1) area but the smallest maximum pressure value. AXI-PGFB and CIR-PGFB have the similar pressure distribution. It can be seen from Fig.5(c) that the deflection of the supporting node is prominent. AXI-PGFB has the smallest maximum top foil transverse deflection and MIX-PGFB has the largest maximum top foil transverse deflection.

The attitude angle decreases with the increasing rotating speed (see Fig. 6 (a)). Small attitude angles are desirable features for GFBs since they denote a journal displacement parallel to the load direction and a reduction in cross-coupled effects which are destabilizing (Kim and Luis, 2007). Under the same rotating speed, CIR-PGFB has the smallest attitude angle. AXI-PGFB and MIX-PGFB shows the similar attitude angle. The minimum film thickness and the dimensionless load capacity show the reverse turn for the three PGFBs, i.e. MIX-PGFB shows the largest minimum film thickness and the smallest dimensionless load capacity (Fig. 6 (c)). The three PGFBs share the much similar gas
film friction torque under the same rotating speed.

Figure 5 Dimensionless distributions under 60,000rpm (a) gas film thickness (b) gas film pressure (c) top plate foil transverse deformation

Figure 6 Bearing static parameters (a) attitude angle, (b) dimensionless minimum gas film thickness, (c) dimensionless load capacity, (d) dimensionless friction torque
The damping coefficient of the three PGFBs under the rotating speed of 10,000 rpm to 80,000 rpm. The damping coefficients $B_{xx}$, $B_{yy}$, and $B_{xy}$ for the three PGFBs are positive, which is good for dissipating the system energy. MIX-PGFB has the weakest energy dissipation ability among the three PGFBs due to the smallest values of $B_{xx}$, $B_{yy}$, and $B_{xy}$. The negative $B_{xx}$ suggests that the journal rotation in $y$ direction is unstable due to the energy supply from $y$ direction to the motion in $x$ direction (loading direction), and the same situation was introduced by Peng (Peng and Carpiino, 1997). The absolute values of $B_{xx}$ of the three PGFBs tend to smaller with the increasing rotating speed which means that the disturbing energy supply from $y$ direction to the motion in $x$ direction becomes weaker at higher speed.

4.2 Experimental results

Air supply pressure and volume flowrate are shown in Fig. 9. Air flowrates in the tests are similar due to the same air supply pressure. In speed up process, large air supply pressure is needed to overcome the dry friction on the surface of the top foil. Therefore, the speed up process of the rotor-bearing system is too fast to be recorded and displayed clearly. The same situation has been reported by Lee and Kim et al. (2012). Therefore, the experimental data
in speed down process is displayed. In the tests, the maximum air supply pressure is about 0.65 MPa, and the maximum air volume flowrate is about 2800 L/min.

The turboexpander rotor supported by different PGFBs share the same drag torque due to the same air supply. In fact, the three PGFBs also share the same bearing static load which is the weight of the rotor (about 8.73 N). With the same rotor drag torque and the same bearing static load, the better hydrodynamic effect could favor bearing performance, such as the rotating speed, loading capacity and the enthalpy difference of the turboexpander. Fig. 10
shows the rotating speed of the three PGFBs under the decreasing air pressure. The three PGFBs show different rotating speeds with the same air pressure (drag torque), which indicates that the bearing torques are different. AXI-PGFB has the smallest bearing torque, and MIX has the largest bearing torque. AXI-PGFB achieves the largest speed among the three PGFBs under the same air supply pressure. The maximum rotating speed under 0.65MPa air supply for AXI-PGFB, CIR-PGFB and MIX-PGFB are 67,369 rpm, 65,849 rpm and 65,242 rpm, respectively. The changes of the speed for the three PGFBs are almost the same for every 0.05 MPa pressure decrease.

A lift-off speed is usually used to define the effective hydrodynamic lubrication of GFBs. In the tests, the lift-off speed is hard to be obtained from the short speed up process. Therefore, the shutoff speed which is the minimum rotating speed for keeping an effective hydrodynamic gas film lubrication of an airborne rotor is defined. The lower shutoff speed allows the softer landing of the high speed rotor, which is good for relieving the damage caused by the dry friction between the rotor surface and the surface of the top plate foil. The maximum shutoff speed for MIX-PGFB is about 37,500 rpm at 0.3 MPa. And the shutoff speed for AXI-PGFB and CIR-PGFB are 30,000 rpm at 0.2 MPa and 36,000 rpm at 0.28 MPa. Obviously, AXI-PGFB could perform the better hydrodynamic lubrication effect than the other two PGFBs. AXI-PGFB can bear the load of the rotor at the lower speed range, which means that the AXI-PGFB could supply more loading capacity with the same air supply. The experimental data in Fig. 10 agrees well with the predicted results in Fig. 6 (c).

The better expansion performance can be indicated by the enthalpy difference through the expansion process. With the same expansion ratio 7.5 (turboexpander inlet absolute pressure is 0.75 MPa, turboexpander outlet absolute pressure is 0.1 MPa), the temperature difference and enthalpy difference are listed in Table 3. As shown in Fig.11, the expansion process using AXI-PGFB as the supporting bearings shows the largest temperature difference and enthalpy difference with the same air supply pressure. Although the temperature difference 44.2 K for expansion process using MIX-PGFB is larger than the temperature difference 43.6 K for expansion process using CIR-PGFB, the lower flowrate leads to the smaller expansion enthalpy difference for turboexpander using MIX-PGFB. Furthermore, the smaller shutoff speed of AXI-PGFB achieves its stable and effective running in lower speed which is benefit for the cooling capacity adjustment of turboexpander.

| Expansion with PGFBs          | Temperature difference (K) | Enthalpy difference (kJ/kg) |
|------------------------------|----------------------------|-----------------------------|
| Expansion with AXI-PGFB      | 44.7                       | 7237.2                      |
| Expansion with CIR-PGFB      | 43.6                       | 7112.6                      |
| Expansion with MIX-PGFB      | 44.2                       | 7082.0                      |

The whole rotating processes of the three PGFBs are recorded by waterfall maps via FFT analysis as shown in Fig. 12. There is no valid data recording for the speed up process of AXI-PGFB due to the flash growth of the rotating speed. Meanwhile, the speed up process of CIR-PGFB and MIX-PGFB is short too. The subsynchronous vibration of the three PGFBs in the test speed range is insignificant, which means that the running performance of the
A turboexpander supported by PGFBs is reliable and stable in the tested speed range of about 30,000 rpm to 65,000 rpm.

![Figure 11](image1.png) Expansion performance in the turboexpander (a) temperature difference and (b) enthalpy difference

![Figure 12](image2.png) Waterfall maps (a) AXI-PGFB, (b) CIR-PGFB and (c) MIX-PGFB

![Figure 13](image3.png) Maximum synchronous vibration amplitude under different rotating speed (a) AXI-PGFB, (b) CIR-PGFB and (c) MIX-PGFB
Figure 13 shows the maximum synchronous vibration amplitudes in \( x \) and \( y \) directions of the turboexpander supported by the three PGFBs under different rotating speeds. Most of the synchronous vibration amplitudes of the three PGFBs in both \( x \) direction and \( y \) direction are smaller than 3 \( \mu \)m. AXI-PGFB has the minimum synchronous vibration amplitude compare to that of CIR-PGFB and MIX-PGFB. It is believed that the better damping performance is crucial in reducing the vibration amplitude of the high speed rotor. The larger damping coefficients in Fig. 8 could be responsible for the smaller synchronous vibration amplitude of AXI-PGFB. And all of the synchronous vibration amplitudes decrease with the deceasing rotating speed except that when the speed is lower than 35,000 rpm. The increase of the amplitudes at low speed are probably caused by the natural frequency of the rotor bearing system, especially, the sharp increase of the amplitude for AXI-PGFB. The situation was predicted by Kim and Luis (2009), and the shutoff speeds of CIR-PGFB and MIX-PGFB are too large to allow the appearance of the same sharp increase. The small cusps of the three amplitude curves are the disturbances in the temporary decreasing pressure during the pressure adjustment.

Figure 14 shows the rotor rotation trajectory at 50,000 rpm and 60,000 rpm. Although the radius difference among those circles are small after going through the virtual filter, the radius of the circles under 50,000 rpm are smaller than that the corresponding circles under 60,000 rpm. The increasing centrifugal force caused by the increasing linear speed of the rotor might be responsible for the increasing radius of the rotor rotation trajectory.

![Figure 14 Rotor rotation trajectory (a) AXI-PGFB, (b) CIR-PGFB and (c) MIX-PGFB](image)

5. Conclusion

Numerical and experimental studies are performed for a new double-layer protuberant gas foil journal bearing with three different elastic foil layers arrangement. The major findings can be summarized as follows.

1 The key static performance and the dynamic performance of PGFBs such as the gas film pressure distribution, the gas film thickness distribution, the foil deflection, the stiffness and damping coefficients are predicted successfully.

2 The qualitative comparison of the cross couple stiffness coefficients shows that AXI-PGFB and CIR-PGFB have the wider stable running speed area than that of MIX-PGFB.

3 PGFBs with the tested three different elastic foil layers arrangements run well in the turboexpander, and the maximum speed (67369 rpm at 0.65 MPa gauge pressure) and expansion performance (enthalpy difference 7237.2 kJ/kg at temperature difference 44.7 K) are achieved by AXI-PGFB.

4 AXI-PGFB has the lowest shutoff speed (30,000 rpm at 0.2 MPa gauge pressure), and MIX shows the largest shutoff speed (37500 rpm at 0.3 MPa gauge pressure). The shutoff speed for CIR-PGFB is 36,000 rpm at 0.28 MPa gauge pressure.

5 AXI-PGFB shows the best loading performance which is also predicted qualitatively by the numerical model.

6 The maximum synchronous vibration of the tested three PGFBs is smaller than 3 \( \mu \)m. And the larger direct
damping coefficient of AXI-PGFB is helpful in reducing the vibration amplitude.

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