

## **Experimental Investigation on the Multi-Decked Protuberant Gas Foil Journal Bearing\***

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### **Abstract**

In this paper, a new kind of gas foil journal bearing using multi-decked protuberant foils as elastic support structure has been developed. Experimental test on the multi-decked protuberant foil journal bearing has been conducted on the high speed turboexpander. The maximum speed of 25 mm diameter rotor reached as high as 100 krpm. Meanwhile, at the same gas supply pressure, the rotor speed is higher in speed-down process than that in speed-up process due to the coulomb friction between the foils. In the whole test, the subsynchronous whirl can be suppressed effectively using the new structure. This all metal hydrodynamic gas foil bearing may be a promising candidate in cryogenic systems.

**Key words:** Gas Foil Bearing, Protuberant Foil, Turboexpander, Hydrodynamic Bearing

### **1. Introduction**

The concept of foil bearing was proposed by Blok and Rossum in the 1950s<sup>(1)</sup>. This kind of bearing is outperformed due to its stability, especially, in the field of high speed rotational machinery, such as turbo-expander, gas turbine, turbo-refrigerator, centrifugal compressor<sup>(2-5)</sup>. Furthermore, in extreme working states where conventional ball bearing cannot handle, foil bearing is regarded as a promising high speed rotor supporting component<sup>(4, 6-7)</sup>. Since then, quite a lot of researches have been conducted on such kind of bearing.

The stability advantage of foil compliant structure lies in the proper combination of structural and hydrodynamic stiffness<sup>(8)</sup>. Song utilized spring bumps as elastic support component with nonlinear stiffness and the equivalent damping increase with load capacity<sup>(9)</sup>. Kim analyzed multi-staged elastic bump foil bearing which can provide higher direct stiffness and damping<sup>(10)</sup>. Sim inserted metal shim between the bottom bump foil and bearing housing to improve the dynamic performance of the bearing<sup>(11)</sup>. Hou conducted experimental study on the 17 mm rotor at the speed of 151 krpm using visco-elastic rubber as supporting structure<sup>(12)</sup>, and made comparison with the elastic structure using bronze wire<sup>(13)</sup>. Lee stick visco-elastic material under the top foil, and the new structure can suppress subsynchronous vibration effectively<sup>(14)</sup>. Kumer added static supply holes in the foil bearing to reduce dry friction during start up and down process and alleviate the heat dissipation problem<sup>(15)</sup>. Andres et al. used simple and low cost bronze metal mesh as supporting structure, which promoted the damping performance of foil bearing<sup>(16-18)</sup>. Feng studied the multi-wound foil bearing in which the foil is wound triply in the housing and found that the bearing load capacity is insensitive to rotational speed and eccentricity due to foil compliance<sup>(19)</sup>. Hou analyzed gas foil journal bearing numerically with protuberant structure and found out that the sides of the top foil may warp up being beneficial for load capacity<sup>(20)</sup>. Thus, proper elastic support design is of crucial importance for the bearing

stability and reliability.

In this study, a new kind of gas foil journal bearing has been developed with multi-decked foils structure. The bearing performance has been tested in the 150 Nm<sup>3</sup>/h O<sub>2</sub> turbo-expander that is widely used in cryogenic systems in China. The relation between rotor speed and supply pressure has been recorded to monitor the operation condition. The goal of this study is to investigate the bearing performance in high speed turbo-expander, especially during the start up and slow down process.

## 2. Multi-decked protuberant foil bearing

The foil journal bearing consists of two protuberant supporting sub-foils and one flat top foil as shown in Fig. 1 (a). The arrangements of the two protuberant foils are shown in Fig. 1 (b). The assembly graph is shown in Fig. 2. The ends of the three foils are aligned and pinned in a fixing hole. The other ends of the foils are in free state. The protuberant projection can be of spherical crown shape. The foils are made of flat beryllium bronze(QBe2), and the protuberant structure array on the foil can be stamped by mould.

The stiffness of the bearing can be regulated in axial or circumferential direction through combination of the two protuberant foils according to the relative position of protuberant projections on the foils. The bearing can provide well distributed stiffness in axial direction that makes corresponding adjustment with the load which suggests high adaptability. The coulomb friction between the foils can consume the wobble energy and suppress the subsynchronous vibration which implies high stability. Besides, this kind of bearing is easy to be fabricated and equipped in the turbo-expander.

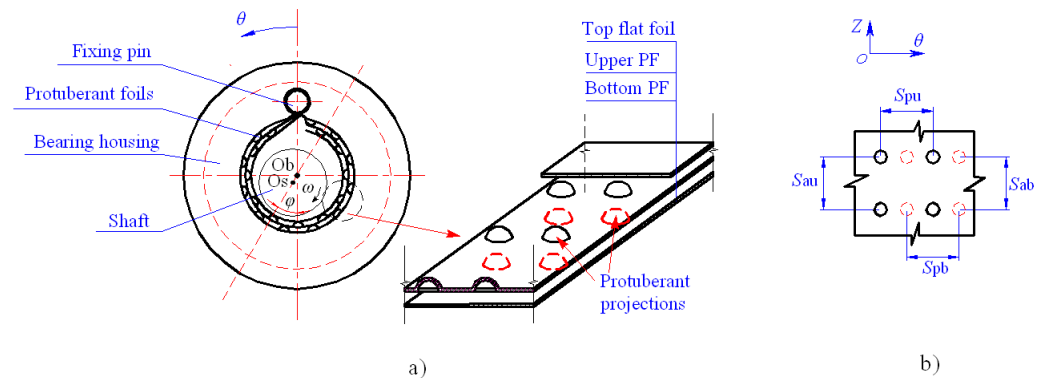


Fig. 1 Multi-decked protuberant gas foil journal bearing (PF-Protuberant Foil)  
a) Schematic view; b) Arrangement of supporting foils

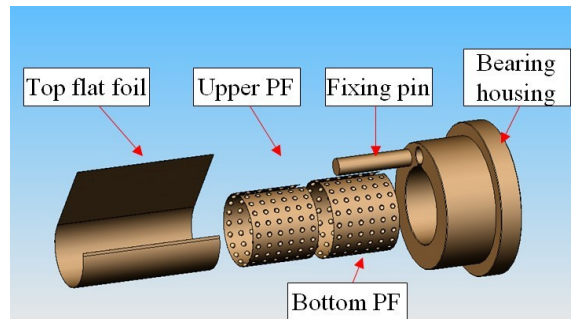


Fig. 2 Assembly of multi-decked protuberant foil journal bearing

## 3. Experimental test rig

The newly developed gas foil journal bearing was assembled in a rotor-bearing system as

shown in Fig. 3. This rotor-bearing system is composed of the expansion wheel, brake wheel, shaft, two static thrust bearings and two foil journal bearings. Orthogonally positioned displacement sensors were used to monitor the vibration at the middle section of the rotor. The test rig used to examine the bearing performance in high speed turbo-expander is presented in Fig. 4. The system composed of compressor, air storage reservoir and turbo-expander. The main parameters of the turbo-expander are listed in Table 1. The compressed high pressure air is supplied by the 75 kW screw air compressor with supply pressure up to 1.3 MPa, the maximal flow rate is 600 Nm<sup>3</sup>/h. The single row static thrust bearing is used to balance the thrust force with clearance being around 30 μm. The inlet pressure of the expander is controlled through a manual regulating valve. Industrial personal computer is chosen as the connecting, logic, controlling and computing core in the data acquisition system. The rotational speed, vibration signals are monitored by the 4 mm eddy current sensor with linearity of <±2% and static resolution 0.1 μm) and data acquisition card processed and stored in the IPC.

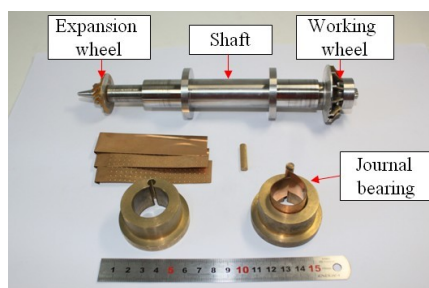


Fig. 3 Tested gas journal bearing

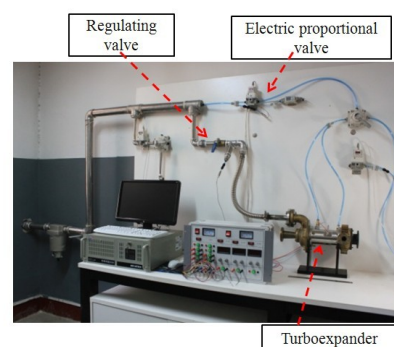


Fig. 4 Experimental test rig

#### 4. Test procedure

1. Set the supply pressure of static thrust bearing at 0.55 MPa at the start up process;
2. Turn up the regulation valve at the entrance of the turbo-expander rapidly before the rotor lifted up from the bearing, and increase the supply pressure up to 1.15 MPa gradually after the rotor lift up;
3. The initial pressure of the static thrust bearing at the start up process is 0.55 MPa and raised gradually to 0.7 MPa with the rotor speed;
4. Close down the regulating valve at the inlet of turbo-expander gradually until the rotor touch the bearing.

#### 5. Results and analysis

The spherical crown surface is regarded as stiff without deformation during the entire test. Therefore, the nominal radial clearance of the compliant foil journal bearing is defined as the following:

$$c_n = (D_n - D_s - 4H_p - 2T_f) / 2 \quad (1)$$

Once the hydrodynamic pressure forms in the gap between the rotor and top foil, the hydrodynamic pressure will push the compliant top foil outwards and deformation will be generated on the upper protuberant foil. There will be larger clearance between the top foil and rotor. Therefore, the compliant surface bearing can work with moderate preload on the journal, and over compression or over relaxation will cause unstable performance<sup>(21)</sup>. In this experiment, a set of clearances shown in table 2 have been tested using different

bearing housing pairs. The bearing suffers from thermal runaway under over preload with nominal radial clearance of  $-20\text{ }\mu\text{m}$  and from unstable subsynchronous wobble under over relaxation. The performance of the bearing with  $0\text{ }\mu\text{m}$  clearance is chosen as a representative sample in this report.

Table 1 Main parameters of the high speed turbo-expander

parameter	magnitude	parameter	magnitude
Outside diameter of fan wheel $D_1/\text{mm}$	60	Outside diameter of working wheel $D_2/\text{mm}$	36.5
Diameter of journal $D_s/\text{mm}$	25	Inside diameter of thrust bearing $D_{in}/\text{mm}$	28
Outside diameter of thrust bearing $D_{out}/\text{mm}$	44	Length of rotor $L_r/\text{mm}$	250.5
Mass of rotor $G_m/\text{g}$	830	Protuberant foil thickness $T_p/\text{mm}$	0.05
Height of protuberant foil $H_p/\text{mm}$	0.2	Top foil thickness $T_f/\text{mm}$	0.07
Crown pitch in peripheral direction of upper layer $S_{pu}/\text{mm}$	4	Crown pitch in peripheral direction of upper layer $S_{pb}/\text{mm}$	4
Crown pitch in peripheral direction of lower layer $S_{au}/\text{mm}$	4	Crown pitch in peripheral direction of lower layer $S_{ab}/\text{mm}$	4

Table 2 Nominal clearances in the tests

No.	Diameter of bearing housing $D_h/\text{mm}$	Nominal radial clearance $c_n/\mu\text{m}$
1	25.90	-20
2	25.94	0
3	25.98	20
4	26.02	40
5	26.06	60

In the speed-up and down test, the rotor speed changes with the supply pressure, as is shown in Fig. 5. It is obvious that the speed-up and speed-down processes do not coincide at the same pressures. On the whole, the rotating speed at speed-up processes is lower than that in the speed-down process at the same pressure. In startup process, the coulomb friction between the foils impedes the relative movement of foils which leads to the increase of friction torque on the rotor. However, in the speed-down process, the coulomb friction and the gas film friction are in opposite directions. At high supply pressure range, namely high rotational speed range, the rotational speed increment with the pressure decrease in the two

processes. Besides, the rotor lift-up speed is higher than the landing speed in the experiment.

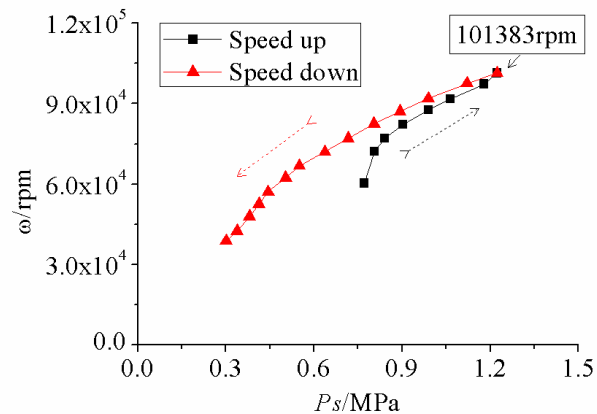


Fig. 5 The rotor speed( $\omega$ ) with the supply air pressure( $P_s$ )

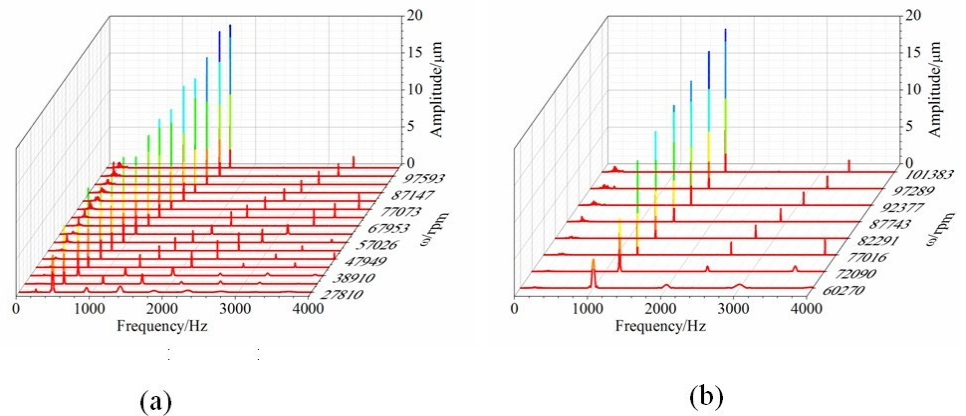


Fig. 6 Waterfall of the turbo-expander (a) Speed-up and (b) Speed-down

The frequency content of the bearing response in the speed-up process is given in the waterfall plot depicted in Fig. 6 (a). Due to the unstable transient start up and the available data acquisition system, the minimal obvious frequency of the rotor is around 60000rpm. The maximal speed of the rotor reached 101383rpm and the subsynchronous vibration is suppressed quite well with this new structure. The maximal vibration amplitude in the speed up process is around 20  $\mu\text{m}$  and there is no wear down under nominal clearance which means that there are gas film between the rotor and the top foil. After a period of smooth running around the maximal speed, the regulating valve is turned down. The speed down waterfall is shown in Fig. 6 (b). The subsynchronous vibration is relative small. Thus, the bearing using the new structure runs smoothly in the speed-up and down processes. The minimal speed in the speed down process is around 30000 rpm which means that the rotor landing speed is around this value. It can be concluded that the rotor lift-up speed is higher than the touch-down speed where the clearance plays an important role. Andres et al. have reported subsynchronous vibrations with the exception that the subsynchronous frequency to track the rotor speed at a ratio of 50% in bump foil bearing <sup>(22)</sup>, however, the subsynchronous frequencies of the protuberant bearing seem to be somewhat constant around 200 Hz being independent of the rotor speed. This may be attributed to the hardening effects of supporting foils and the rotor mass imbalance, particularly for operation at super

critical speeds <sup>(23)</sup>.

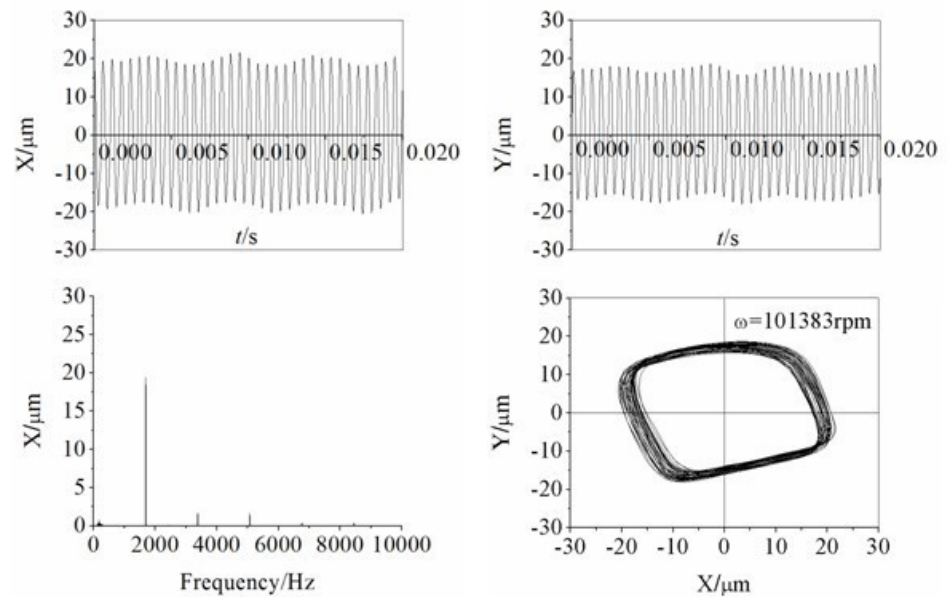


Fig. 7 Time and frequency domain analysis at maximum speed

The time domain and frequency domain analysis of the vibration signals at maximal rotational speed is shown in Fig. 7. The maximum vibration amplitude is around 20  $\mu\text{m}$ . The locus at maximum speed is quite clear and regular. Comparing to main frequency, the subsynchronous vibration is quite small implying that there is a further speed-up margin.

The rotor locus of the bearing in the speed-up and down process is shown in Fig. 8. The vibration amplitude is increasing with the rotational speed. In the processes, the rotor locus presents as a diamond shape with keen edges. When the rotor runs around 60000 rpm or at lower speed, there will be overlapped locus which can be attributed to the relative small stiffness of the gas film at lower speed. As the rotational speed increases, the rotor locus gets smoother and the vibration amplitude increases. The rotor locus is clear and regular in the entire speed-up and down process. Generally, this bearing with protuberant foils exhibits smooth operation in the tests.

## 6. Conclusion

The present experimental results indicate that multi-decked protuberant foil bearing is a promising foil bearing which can be adopted in the turbo-expander. Due to the multi-layer structure, the vibration of the rotor is confined in small magnitude. In high speed range, the subsynchronous vibrations in the system are suppressed quite well. The rotor speed in speed-up process is smaller than that in speed-down process because the coulomb friction between the foils acts in different directions during the two processes.

This kind of bearing is highlighted for its easy fabrication and assembly process, as well as its high stability in the whole test. It can be considered a good candidate for rotor supporting component in high speed cryogenic turbo-expander.

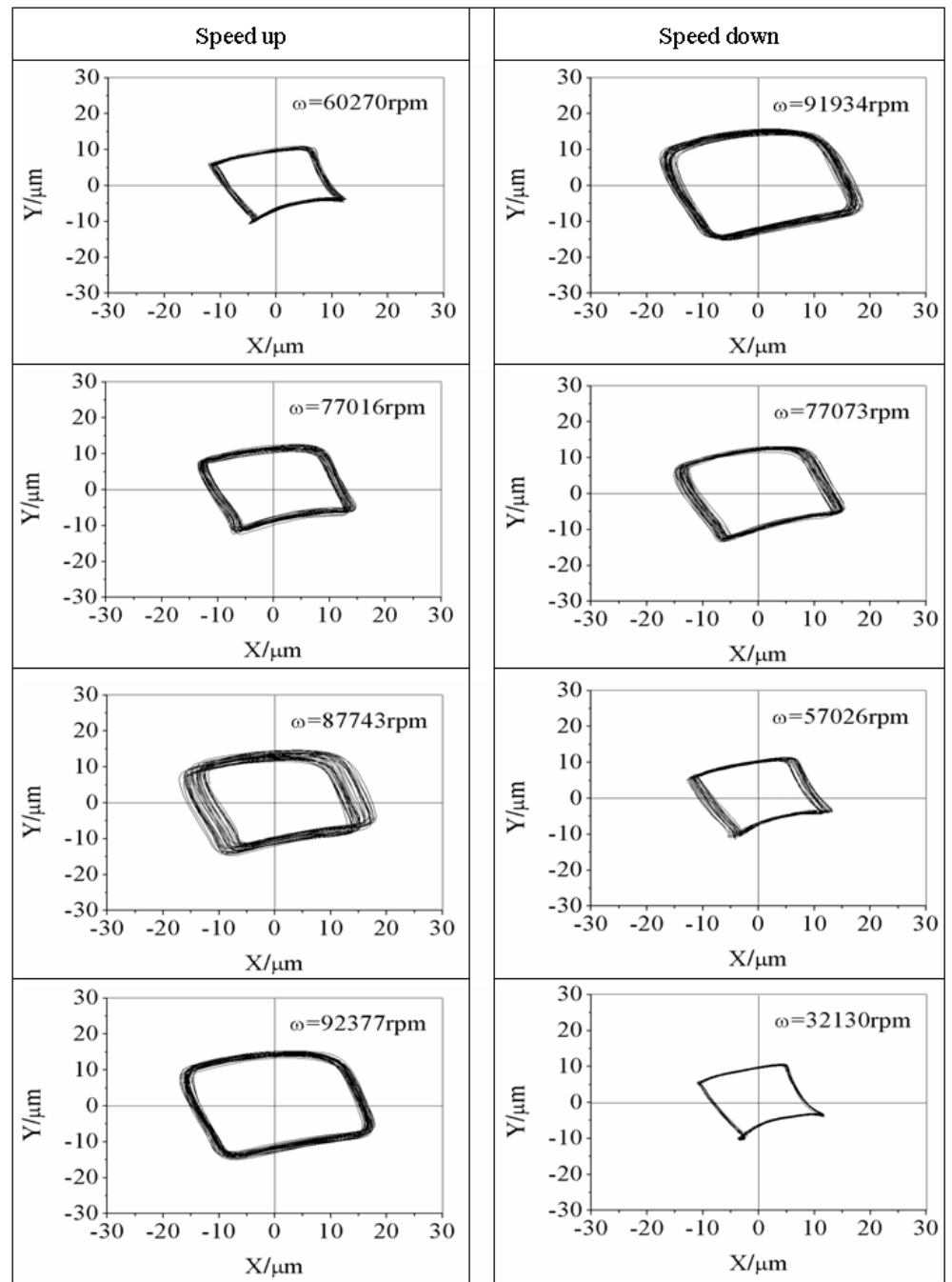


Fig.8 Rotor locus in the speed-up and speed-down processes

### Acknowledgement

This project was supported by the National Basic Research Program of China(2011CB706505), Fundamental Research Funds for the Central Universities, NSAF (Grant No. 11176023) and partially supported by the Open Research Project of Key Laboratory of Cryogenics, TIPC, CAS (CRYO201226).

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