Application of multi-leaf foil bearings in high-speed turbo-machinery

Yu GUO*, Yu HOU*, Qi ZHAO*, Xionghao REN* and Tianwei LAI*
* State Key Laboratory of Multiphase Flow in Power Engineering
Xi’an Jiaotong University, Xi’an 710049, PR China
E-mail: laitianwei@mail.xjtu.edu.cn

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Abstract

Foil bearings, featuring various advantages (such as high-speed, good operation stability, low friction, long life, strong adaptability, wide temperature range, and oil-free), are promised to have good development prospects and application in high-speed centrifugal machinery. In order to evaluate the actual performance of the multi-leaf foil bearings and promote their further application in the high-speed turbo-machinery, the experimental research of multi-leaf foil bearings with rotor diameter of 25 mm are conducted in this paper. The static and transient tests of the multi-leaf journal bearings with different foil thickness are initiated on the journal bearing test rig to analyze their characteristics, such as the load-deflection, bearing Coulomb friction, start-up and operating friction torque, which could provide guidance for the structural parameter design and exploitation of the bearing. Afterwards, two different bearing combinations are adopted as the supporting components, namely A-type: multi-leaf foil journal bearing with aerostatic thrust bearing and B-type: multi-leaf foil journal bearing with multi-leaf foil thrust bearing. They are applied in the high-speed turbo machinery with the rotor diameter of 25 mm to assess the direct application of the multi-leaf foil bearings in high-speed turbo-machinery. The transient speed-up, high speed, and speed-down experiments of the rotor-bearing system are conducted and their transient characteristics and stability are compared. The experimental results indicate that the start-up (0.6x10^{-2}~1.8x10^{-2}N·m) and steady operation frictional torque (0.25x10^{-2}~0.5x10^{-2}N·m) of the multi-leaf foil journal bearing is relative small; the impeller-rotor system operates steadily and smoothly with good repeatability, supported by A-type and B-type, the maximum rotor speed at about 94krpm achieved in the tests is close under the support of A-type and B-type. However, there are smaller main frequency amplitude, nonsynchronous amplitude and better rotor orbit in the transient processes with the support of B-type, which indicates better stability.

Keywords: Turbo machinery, Multi-leaf Foil bearing, Aerodynamic Lubrication, Stability study, Engineering application

1. Introduction

The high-speed centrifugal machinery has many advantages (such as compact structure, high reliability and low maintenance costs), which has been widely used in mechanical engineering field. In order to achieve higher thermal efficiency, higher operation speed is prospective for the high-speed centrifugal machinery (Ghosh, 2012). Due to the special operation requirements for high-speed, high-reliability and high-cleanliness of turbo machinery, it is difficult to select and design the application bearings for high-speed turbo machinery. As the support structure of high-speed turbo machinery system, the bearing performance is of vital importance to the operation characteristics of its impeller-rotor system (Lai, 2017). Therefore, much attention has been attracted to the bearing components with the development of high-speed turbo machinery.

Many different types of aerostatic bearings and self-acting gas bearings have been developed and applied, which use ambient medium as the lubrication medium. The aerostatic bearings need the additional supplying gas mechanism, which leads to more gas consumption and complex mechanical structure system (Hou, 2015). The self-acting gas
bearings have a simpler structural system and higher reliability due to their self-lubricating and aerodynamic characteristics, which makes them have good development prospects and application significance in high-speed turbo machinery. Many different types of self-acting bearings have been developed, like grooved bearings, step bearings, tilting bearings and foil bearings. Foil bearings are considered to be a promising alternative technology due to their advantages of high speed, oil-free, low friction, strong adaptability, good stability and long life (Heshmat, 1982). The whirling energy of high-speed rotor could be dissipated due to the good damping characteristics of the structural deformation and Coulomb friction of foils, which could maintain stable operation of the high-speed rotor. Several types of underlying elastic configurations have been developed since its debut (DellaCorte, 2011), such as bump foil type (Heshmat, 1983), multi-leaf type (Oh, 1976), metal mesh type (Andrés, 2010), nested compression spring type (Song, 2006) and protuberant type (Lai, 2014), etc. There are some aspects of great concern in engineering application of foil bearings, such as ultimate capacity (Kim, 2008), bearing stability (Gu, 2019), thermal balance (Salehi, 2001), parameters optimization (Wang, 2013), start-stop performance and bearing service life.

Many mathematical simulation models have been developed to provide theoretical reference and guidance for accurate performance prediction of foil bearings (Andrés, 2009). Koepsel et al. (1977) conducted a systematic theoretical analysis of the multi-leaf foil journal bearing which is applied in the engine. The elastic foil was simplified as a bending beam articulated at one end, and the foil deformation was obtained using the castigliano theorem and superposition principle. However, details about the simulation model and analysis of the numerical results were not provided in the paper. Trippett et al. (1978) developed a numerical model for multi-leaf journal bearing by considering the friction between adjacent foil, foil and rotor, the initial foil curvature and the geometric overlap. The performance characteristics of the multi-leaf foil bearing were obtained in various operation conditions through the model. Du and Zhu (2015) proposed a numerical model for multi leaf foil bearings based on the assumption of surface contact between foils. The numerical results were compared with the tested data, and the mechanism of surface contact on the bearing load-bearing performance was revealed in their research. Heshmat et al. (1995) carried out the corresponding numerical analysis on the multi-leaf foil journal bearing with elastic sup-porting components. The elastic supporting components were considered as springs in this finite element model, which simplified the numerical calculation process. Heshmat et al. (1983) proposed a numerical model of bump foil bearing. In their model, the top foil and the underlying elastic component were simplified as the independent elastic bodies with uniform stiffness, which simplified the fluid-solid coupling calculation of foil deformation and gas pressure. Dellacorte and Valco (2000) developed a correlation formula for calculating the load capacity of journal foil bearings to establish the relationship between the bearing capacity and the bearing structure with the operation condition. However, the load capacity coefficient in the correlation formula needs to be measured through experiments. Peng and Khonsari (2006) developed a thermo-elastic model to predict the thermodynamic results of foil journal bearing by taking into account the compressibility and viscous-temperature characteristics of lubrication medium.

In addition, many relevant experimental researches of the foil bearings have been carried out, which reflected their working performance directly in actual tests or application. Arakere et al. (1996) extended the experimental studies on the multi-leaf foil journal bearing with serrated elastic supporting components, which indicated that the load capacity of foil bearing could be improved effectively by adding the elastic supporting components. Feng and Zhao (2014) designed and fabricated a novel multi-cantilever foil bearing by using multi-cantilever foils as the elastic structure. The static and dynamic tests of this type of bearing proved that it possess good capacity to dissipate whirling energy, which could limit the vibration amplitude of rotor. Heshmat and Ku (1994) measured the structural stiffness and damping of foil bearings by exciting a nonrotating shaft and the dynamic force on the foil bearing. Radil and Zeszotek (2004) measured the internal temperature of the foil journal bearing in different operation conditions, which proved that both rotational speed and load have obvious effect on the bearing temperature, and the effect speed is more evident. The large number of systematic theoretical and experimental studies on foil bearings have been applied in many advanced turbo-machinery systems (Samanta, 2019), such as air circulation system (Mcauliffe, 1992), micro gas turbine (Dellacorte, 2000), micro power generator (Lee, 2011), turbocharger (Walton, 2004), centrifugal blower (Wilson, 2009), high-speed compressor and turbo alternator (Walton, 2010).

The multi-leaf foil bearing were proposed by Garrett, with the support of National Air and Space Administration funding (Emerson, 1978). The leaf foil type is easy to be manufactured owing to its simple structure, its requirements for processing technology is much lower than that of bump foil type, which results in the lower production and maintenance costs. The more complicated and higher processing and manufacturing process is needed, due to the
structural form of the bump foil type. In addition, the leaf foil type could meet the application requirements of small high-speed rotating machinery, due to its good performance. Up to now, most of the published literature about the bearing focus on mathematical model and numerical analysis due to its development and application background, and there are few related experimental studies and performance evaluation of this kind of bearing. In order to evaluate the actual performance of the multi-leaf foil bearings and promote their further application potential in high-speed turbo-machinery, the experimental research of the multi-leaf foil bearings with rotor diameter of 25 mm are conducted in this paper. The static and transient tests of the multi-leaf journal bearings with different foil thickness are initiated on the journal bearing test rig. Load-deflection, bearing Coulomb friction, start-up and operating friction torque characteristics under different foil thickness and bearing load are obtained. In order to assess the direct application of the multi-leaf foil bearings in high-speed turbo-machinery, two different bearings combinations are adopted as the supporting components, namely A-type: multi-leaf foil journal bearing with aerostatic thrust bearing, and B-type: multi-leaf foil journal bearing with multi-leaf foil thrust bearing. The aerostatic thrust bearing of the A-type needs the additional supplying gas mechanism, while the B-type have a simpler structure system. They are applied in the high-speed turbo-machinery with the rotor diameter of 25 mm. The transient speed-up, high speed, and speed-down experiments are carried out to evaluate the bearing performance. The operating characteristics of the rotor-bearing systems under the two types of bearings combinations are compared and analyzed.

2. Configurations of the multi-leaf foil bearings

The schematic diagrams and the actual image of the multi-leaf foil journal bearing is shown in Fig.1. The multi-leaf foil thrust bearing and the aerostatic thrust bearing are shown in Fig.2. The multi-leaf journal/thrust bearing consists of a bearing house and eight/six multi-leaf foils. The leading edge of the multi-leaf foils is attached to the bearing house through fixed pins or spot welding, and the trailing edge is overlapped on the adjacent foil freely. The leading edge of the thrust foils starts from the welding area. The surface of the multi-leaf foils are coated with polytetrafluoroethylene through multiple spraying, and the coating thickness is about 60~70um. The extension direction of the multi-leaf foil is consistent with the rotation direction of the rotor. The aerostatic thrust bearing has two rows of 32 holes for air supply which were distributed uniformly in the circumferential direction. The main structural parameters of the multi-leaf foil journal bearing and the multi-leaf foil thrust bearing are listed in Table 1 and Table 2 respectively. Two different combinations of the bearings were adopted as the supporting components of the high-speed turbo machinery to analyze the effect of bearing combination on the characteristics of the high-speed rotor intuitively. The two bearing combinations are named as A-type: the multi-leaf foil journal bearing with the aerostatic thrust bearing and B-type: the multi-leaf foil journal bearing with the multi-leaf foil thrust bearing, respectively.

![Fig.1 (a) Schematic diagram and (b) image of the multi-leaf foil journal bearing](image-url)
3. Tests of multi-leaf foil journal bearing

3.1 Test Rig of journal bearing

The experimental investigation and performance tests on the multi-leaf foil journal bearings were conducted through a journal foil bearing test rig. The main structure of the test rig is shown in Fig.3. The main structure of the experimental rig consists of three parts, including (A) the high-pressure gas supply, (B) the mechanical part and (C) the electronic control and data acquisition system. Air is compressed by the screw compressor (the maximal supply pressure can be up to 1.0MPa), filtered by the strainer, stabilized by the gas reservoir, and controlled by the electrical control valve controlled by the electronic control box. The turbo machinery system is driven by the adjustable high-pressure gas, and the inlet pressure of turbo is measured by the pressure sensor.
A schematic diagram of the main mechanical structure and the image of the tested bearing is shown in Fig.4 (a), (b) and (c), respectively. The main components of the mechanical structure consist of the driving turbo, loading mechanism, displacement sensor, tested bearing, rigid support bearing, moment sensors and rotor. The suspension method is applied to the experimental rig to fix the tested bearing, and the bearing house is connected and suspended on the cross beam through the load mechanism (including the hanger bolt, tension spring, turnbuckle and tension sensor). As shown in Fig.4 (b), the tested bearing is connected with two moment sensors through four groups of rubber bands. The elastic characteristics of the rubber band can reduce the fluctuation of the moment measurement. The leading edge of the multi-leaf foils is attached to the installation groove of the bearing house through fixed pins. In these experiments, the torque produced by the tension spring on the tested bearing can be neglected since the angle is too small. The sum of the torque measured by the two moment sensors is the frictional torque of the tested bearing in operation. The radial load is adjusted through the tension of the spring by twisting the turnbuckle, and the rotor is supported by two angular contact ball bearing (grease lubrication) actually. The applied radial load is measured by the tension sensor, and the weight of the relevant components has been removed in advance. Therefore, the tested compliant foil bearing has a degree of freedom in the test rig.
3.2 Analysis of experimental results

The static load-deflection, start-up and operating friction torque tests of the multi-leaf journal bearings with different foil thickness were conducted to analyze its static and transient characteristics, which could provide a reference for the structural parameter design and exploitation of the bearing.

The static load-deflection of the multi-leaf foil journal bearing with different foil thicknesses (0.1mm, 0.12mm and 0.14mm) is shown in Fig.5. As shown in the trend of the static load-deflection, the multi-leaf foil journal bearing with different foil thicknesses all exhibit nonlinear stiffness characteristics. According to the linear fitting of the static load-displacement curves, the structural stiffness of these bearings is about 1.70×10^6 N·m^-1 (0.1 mm), 2.38×10^6 N·m^-1 (0.12 mm), and 3.04×10^6 N·m^-1 (0.14 mm), respectively. The structural stiffness of the multi-leaf foil journal bearing increased evidently with the foil thickness. There is an obvious hysteresis phenomenon appears in the measured curve of the static load-deflection tests, which is due to the Coulomb friction inside the bearing which hinders the recovery of the deformed foil to its initial state. The hysteresis makes the bearing have certain damping characteristics in operation, which is beneficial for the vibration and whirling energy dissipation in the rotor-bearing system. Due to the effect of the Coulomb friction, the absolute deformation of the unloading curve is always higher than that of the loading process under the same load, in the deformable range of the foil bearing. A cyclic process of loading and unloading is actually a process of mutual conversion between energy working and elastic potential energy. The area enclosed by the load-deflection curve is the energy converted by the applied load in a load-unload cycle. Fifth-order polynomial was applied to fit the loading and unloading curves separately, based on the weighted least-square method. The polynomial fitting expressions of the loading and unloading curves for multi-leaf foil journal bearing with 0.1 mm, 0.12 mm and 0.14 mm foil thickness are shown in Eq. (1), Eq. (2) and Eq. (3), respectively. Then, the area enclosed by the static load-deformation curve is calculated by numerical integration—Eq. (4). The power consumption of the multi-leaf foil journal bearing with different foil thicknesses in one cycle is 4.26×10^{-4} J, 3.92×10^{-4} J and 3.44×10^{-4} J, respectively.

![Fig.5 Static load-deflection of the multi-leaf foil journal bearing with different foil thicknesses](image)

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\begin{align*}
\text{Y}_{\text{loading}} &= 9 \times 10^{-5} x^5 - 0.0015 x^4 - 0.005 x^3 + 0.12 x^2 + 1.299 x + 9.667 \\
\text{Y}_{\text{unloading}} &= 7 \times 10^{-5} x^5 - 1 \times 10^{-4} x^4 - 0.0143 x^3 + 0.0077 x^2 + 2.5051 x - 10.537 \\
\text{Y}_{\text{loading}} &= 8 \times 10^{-4} x^5 - 0.0107 x^4 + 0.0181 x^3 + 0.1379 x^2 + 1.4123 x + 16.27 \\
\text{Y}_{\text{unloading}} &= 1.4 \times 10^{-3} x^5 + 0.0052 x^4 - 0.0892 x^3 - 0.232 x^2 + 3.6134 x - 11.169 \\
\text{Y}_{\text{loading}} &= 5 \times 10^{-4} x^5 - 0.0251 x^4 + 0.2008 x^3 - 0.2818 x^2 + 0.9833 x + 20.586 \\
\text{Y}_{\text{unloading}} &= 7.1 \times 10^{-3} x^5 + 0.0262 x^4 - 0.2021 x^3 - 0.5636 x^2 + 4.181 x - 10.186 \\
W &= \int_{a}^{b} y(x)dx
\end{align*}
\]
The frictional torque at start-up and steady operation of the bearing is an important factor for evaluating the bearing lubricating performance. The frictional torque at start-up and steady operation (40 krpm) of the bearing with bearing load is shown in Fig. 6. It can be seen that the frictional torque at start-up and steady operation of the bearing increases with the bearing load. However, it should be noted that the frictional torque at start-up and steady operation are all not 0 under the 0 N bearing load, which is caused by the bearing pre-load. A schematic diagram of the state of multi-leaf foils before and after assembly of the rotor is shown in Fig. 7 (a) and Fig. 7 (b). As shown in Fig. 7 (a), the curvature radius of multi-leaf journal foils is larger than that of the rotor, an aperture hole which is smaller than the diameter of the rotor will be formed by the overlapping multi-leaf journal foils when all the foils are assembled into the bearing house. The size of the aperture hole changes with the deformation of the multi-leaf journal foils. When the rotor is assembled with the multi-leaf foil journal bearing, the aperture hole with variable size is enlarged due to a certain pre-deformation of the multi-leaf journal foils under the action of the rotor, which is shown in Fig. 7 (b). Therefore, there is a certain initial contact force (i.e. pre-load) between the bearing surface and the rotor surface due to the flexible characteristics and pre-deformation of the multi-leaf foils. The initial contact force between the bearing and the rotor increased with the foils thickness under the same other structural parameters, including rotor diameter, foil radius, foil length, foil angle and bearing house inner diameter, etc. and the operation conditions including bearing load, lubricating medium and ambient temperature, etc. Therefore, a thicker multi-leaf foil lead to a greater pre-load of the bearing, which causes a larger frictional torque at start-up and steady operation. Similarly, a greater bearing load brings a larger contact force for the multi-leaf foil journal bearing, which caused a bigger start-up frictional torque. The low start-up frictional torque of the bearing could be obtained under different bearing loads, which ensures that the turbo machinery with the multi-leaf foil journal bearing could start up easily. The larger start-up frictional torque of the bearing will cause a higher turbo start-up pressure and larger axial forces (which are decided by the pressure difference between the impeller passage and the impeller sidewall gap), possibly leading to excessive speed-up, unsteady rotor speed, wear and tear of bearings and rotor, etc. Furthermore, the excessive start-up frictional torque of the bearing may result in unsuccessful start-up of the turbo machinery, in the given range of the supply pressure. Under the same bearing load, frictional torque at steady operation is much less than the start-up frictional torque, due to the effective hydrodynamic lubrication film. Due to the low viscosity of gas, the frictional torque at steady operation is in the order of 10^{-3} N m. Besides, the frictional torque of the multi-leaf foil journal bearing at steady operation only increased by 48% (0.1 mm), 47% (0.12 mm) and 51% (0.14 mm) when the bearing load increased from 5 N to 20 N, and the frictional torque at steady operation is not proportional to the bearing load. Compared with the start-up frictional torque, the effect of the bearing load and foil thickness on the frictional torque at steady operation can be ignored due to the gas lubrication effect. The tested bearings have a low frictional torque at steady operation under different bearing loads, which ensure the low friction loss at the high-speed.

![Fig. 6 Frictional torque versus bearing load at the start-up and steady operation (40 krpm) of the bearing](image-url)
4. Application of multi-leaf foil bearing in high-speed turbo machinery

4.1 High-speed turbo machinery test rig

In this study, the turbo expander with the rotor diameter of 25mm is employed as the high-speed turbo machinery in the foil bearing application test rig. The configuration of the assembly of the high-speed turbo machinery, an industrial product, is shown in Fig.8. The mechanical part of the high-speed turbo machinery is mainly composed of the nozzle, expansion wheel, expansion volute, rotor, journal bearing, thrust bearing, shell, braking wheel and the braking volute, etc. The driving power of the wheel-rotor system comes from the high-pressure gas expansion in the expansion wheel, while the braking force is provided by the braking wheel located at the other end of the rotor to dissipate coaxial power. The rotor is arranged horizontally and supported by the two journal bearings, while the net axial load generated by the pressure difference between the impeller passage and impeller sidewall gap of the expansion wheel and braking wheel is balanced by the two thrust bearings. Two displacement sensor (with the linearity ±2% and static resolution of 0.1 μm) are arranged vertically to acquire the vibration signal of the high-speed rotor. An image of the test rig is shown in Fig.9. The high-speed turbo machinery test rig consists of the high-pressure gas supply, the mechanical part and the electronic control and data processing system (Lai, 2018). The multi-leaf foil journal (0.12mm) and thrust bearing (0.1mm) are applied and tested in the high-speed turbo machinery. The actual image of the rotor-bearing system in the high-speed turbo machinery test rig is shown in Fig.10. Main structural parameters of the wheel-rotor system are listed in Table 3. The unbalance of the impeller-rotor system is located on the outer diameter surface of the two thrust disks, respectively.

![Fig.8 Assembly of the high-speed turbo machinery with rotor diameter of 25 mm](image-url)
Fig. 9 The high-speed turbo machinery test rig

Fig. 10 The rotor-bearing system in high-speed turbo machinery test rig

Table 3 Structural parameters of the wheel-rotor system

| Variables                        | Values | Variables                        | Values |
|----------------------------------|--------|----------------------------------|--------|
| Rotor diameter /mm               | 25     | Rotor length /mm                 | 250.5  |
| Expansion wheel diameter /mm     | 36.5   | Braking wheel diameter /mm        | 60     |
| Outer radius of thrust disk /mm  | 44     | Distance between journal bearings /mm | 127  |
| Weight of rotor /kg              | 0.891  | Material of rotor                | Stainless steel |
| Density of shaft /kg · m⁻³        | 7830   | Density of wheel /kg · m⁻³        | 2730   |
| Unbalance(braking side) / g · mm | 0.637  | Unbalance(expansion side) / g · mm | 0.249  |

4.2 Application test in high-speed turbo machinery

The transient speed-up, high speed, and speed-down tests were carried out to assess the performance of the two types of bearing combinations in the high-speed turbo machinery. The rotation speed-supply pressure of the expansion wheel in transient processes is shown in Fig. 11. The transient processes could be divided into six stages according to the variation of the rotor speed, which are indicated as: (I) Non-rotating stage, (II) Start-up stage, (III) Speed-up stage, (IV) High Speed stage, (V) Speed-down stage, (VI) Stop stage. In stage I, the supply pressure increases gradually from 0, and the rotor remains stationary for the driving torque of the expansion wheel is less than the static frictional torque between the rotor and the bearing surfaces. In stage II, the driving torque of the expansion wheel increases with the supply pressure. When the supply pressure reaches a certain value, the rotor begins to rotate for the driving torque is greater than the static frictional torque. The supply pressure is maintained at this constant value, indicated as the start-up pressure, until the rotation speed reaches a constant value. The aerodynamic lubrication takes effect due to the
generation of a stable lubrication film in bearing clearance. In stage III, the rotation speed increases gradually with the supply pressure, until it reaches the maximum value. In stage IV, the supply pressure is maintained at 0.96 MPa, and the rotor operate steadily at high speed for a period of time. In stage V, a continuous decrease of the supply pressure leads to a gradual drop of the rotation speed. In stage VI, the driving torque of the turbo is less than the frictional torque of the impeller-rotor system, when the supply pressure is reduced to a certain value which is denoted as the stop pressure. The rotation speed decreases continuously until it stops. It can be seen that the application of the multi-leaf foil bearings in the high-speed turbo machinery is completely feasible. The impeller-rotor system can complete the experiment of speed-up, speed-down and high-speed operation successfully, under the support of both A-type and B-type. The key operating parameters of the transient processes under A-type and B-type is shown in Table 4. It can be seen that the start-up pressure of A-type is much lower than that of B-type, due to the lower frictional torque between the thrust disc and the lubricating film of the aerostatic thrust bearing. Besides, the effect of the increased rotor axial force on the frictional torque of aerostatic thrust bearing could be ignored almost, compared to the dry friction between the rotor and the thrust bearing surface. The direct contact between the thrust disc and the thrust foil bearing surface leads to a larger frictional torque with the further application of the multi-leaf foil thrust bearing. The dry frictional torque between the foil thrust bearing and the thrust disc increases gradually with the increase of axial force, when the supply pressure of the expansion wheel increases continuously. All these factors lead to the greater start-up frictional torque of the rotor-bearing system under B-type, which results in the higher start-up pressure. Under the same maximum supply pressure, the highest achievable rotation speed with A-type is close to that of B-type.

![Fig.11 Rotation speed-supply pressure of expansion wheel in transient processes: (a) A-type; (b) B-type](image)

Table 4 Key operating parameters of the transient processes

| Operating parameters                        | A-type      | B-type      |
|---------------------------------------------|-------------|-------------|
| Start-up pressure /MPa                      | 0.22        | 0.29        |
| Maximum supply pressure /MPa                | 0.96        |             |
| Maximum rotation speed /rpm                 | 96414       | 93993       |
| Stop pressure /MPa                          | 0.14        | 0.16        |
| Maximum axial load /N                       | 48.8        | 49.2        |
| Clearance of the thrust bearing at maximum axial load /μm | 32.2 | 12.3 |

In order to analyze the operating characteristics of the high-speed rotor supported by A-type and B-type, the vibration characteristics of the high-speed rotor were compared and analyzed. The rotor vibrations in X and Y directions are transformed into frequency domain by FFT. The waterfall plot of the rotor supported by A-type and B-type in the transient process is shown in Fig.12 and Fig.13, respectively. The main frequencies of the high-speed rotor are all dominant among all the vibration signals. The amplitude at the main frequencies has a slight fluctuation trend with the rotational speed. There are sub-synchronous and 2X frequency vibrations which are speed-dependent. In X direction, the main frequency amplitude and the sub-synchronous amplitude under B-type are much smaller than that under A-type. However, the difference of the main frequency amplitude and sub-synchronous amplitude between A-type and B-type is not obvious in Y direction, compared with that in X direction. The amplitude of the synchronous vibrations has a slight upward trend with the increase of the rotation speed, while the amplitude of 2X frequency
decreases gradually until almost disappears. Therefore, the ratio of 2X frequency amplitude to the main frequency amplitude decreases gradually with the rotational speed. The 2X frequency vibrations reappear when the rotational speed decreases to a certain value, and its amplitude increases gradually with the decrease of the rotational speed. Many factors could lead to the generation of 2X frequency vibration, such as misalignment of the rotor, asymmetric stiffness of the rotor, bearing nonlinearity, reciprocating force, resonance, supporting loosening and aerodynamic forces. The speed frequency moves away from the natural frequency of other components in the high-speed turbo machinery with the increase of the speed frequency, which lead to the decrease of the 2X frequency amplitude. Furthermore, the misalignment, supporting loosening and bearing nonlinearity may be the reasons for the change of 2X frequency in this study, as well.

Fig.12 Waterfall plot (X direction) of the rotor in the transient process: (a) A-type; (b) B-type

Fig.13 Waterfall plot (Y direction) of the rotor in the transient process: (a) A-type; (b) B-type

In order to analyze the transient high-speed rotor equilibrium positions supported by A-type and B-type, the rotor centerline during the transient speed-up and speed-down processes is shown in Fig.14. The rotor rotates counterclockwise with the support of the bearings, and its eccentricity and attitude angle are reflected through its polar coordinates. In the polar coordinates, the initial coordinates of the rotor centerline is set as (100μm, 270°), according to its actual initial position. There is an obvious hysteresis phenomenon in the rotor centerline trajectory of B-type, due to the deformation of multi-leaf foils and Coulomb friction between foils. The Coulomb damping characteristics of the bearing are beneficial for the vibration and whirling energy dissipation in the rotor-bearing system, which could improve the stability of the rotor-bearing system.
To analyze the stability of the high-speed rotor under different bearing combinations, the rotor orbit at different rotational speeds are compared and analyzed. The rotor orbit at different rotational speeds under the support of A-type and B-type is shown in Fig. 15. It can be seen that the overall trend of the rotor orbit with the increase of the rotational speed is similar, supported by A-type and B-type. With the increase of the rotational speed, the rotor orbit changes from irregular dumbbell shape to semicircle firstly, and then become a regular oval shape. There is obvious difference between the main frequency amplitude of A-type and that of B-type, while the 2X frequency amplitude of these two types is close, which lead to the difference in the shape of the rotor orbit. For example, the irregular dumbbell shape is more obvious for the rotor orbit in the low rotational speed, supported by B-type. Besides, it is worth noting that the rotor orbits of B-type is more clear and repeatable. The damping characteristics of the aerostatic thrust bearing only comes from the gas film, while that of the multi-leaf foil thrust bearing is derived from the gas film and multi-leaf foils. The gas film-whip energy could be dissipated effectively due to the slip and local elastic deformation of the multi-leaf foils, which could substantially improve its operation stability. The effect of the thrust bearing on the rotordynamic characteristics of the high-speed rotor is complicated and uncertain. There is angular misalignment exists when the rotor operates at high speed, so the angle and displacement of the rotordynamic characteristics of the high-speed rotor are all affected by both the journal bearing and thrust bearing. Therefore, the rotor orbit and displacement in the radial direction are influenced by the thrust bearing.

In order to analyze the motion state of the high-speed rotor directly, the time domain analysis of high speed rotor in the highest rotational speed (A-type: 96414; B-type: 93993rpm) under different bearing combinations is shown in Fig. 16 and Fig. 17, respectively. It can be seen that periodic vibration appears in both X and Y directions, resulting from the nonsynchronous whirl of high-speed rotor. The oscillation frequency of the rotor amplitude is (A) 195.3 Hz and (B) 198.41Hz respectively, under different bearing combinations. Comparing the nonsynchronous whirl frequency (in
Fig.12 and Fig.13) and oscillation frequency of the rotor amplitude in the time domain analysis (in Fig.16 and Fig.17), it can be found that they are all close to 200Hz, which is the response of the natural frequency of the rotor-bearing system (Lai, 2017). The nonsynchronous whirl of high-speed rotor leads to the oscillation of rotor vibration amplitude at a certain frequency, which is the main reason for the widened band of the rotor orbit. The nonsynchronous whirl is not conducive to the stable operation of the high-speed rotor.

![Fig.16 Time domain analysis of high speed rotor under A-type](image1)

![Fig.17 Time domain analysis of high speed rotor under B-type](image2)

In order to observe the wear condition of the multi-leaf foil bearings after repeating the start-stop experiments for dozens of times, the actual image of the worn multi-leaf journal and thrust foils after multiple tests are shown in Fig.18 (a) and Fig.18 (b), respectively. There are two kinds of wear marks (which are marked as Mark-1 and Mark-2) on each multi-leaf journal foil, which are located at the trailing edge of the journal foil and the contact area between adjacent journal foils, respectively. The wear mark of the trailing edge is caused by the direct contact and dry friction between the foil surface and rotor in the transient start-stop process. The wear mark of the contact area between adjacent journal foils is due to the flexible deformation and slippage of the multi-leaf foils. The flexible deformation of the multi-leaf journal foil results in the slippage of its trailing edge, which causes the sliding friction and wear marks in the contact area between adjacent journal foils. As shown in Fig.18 (b), there are similar wear marks (marked as Mark-3) on the trailing edge of the thrust foils, due to their direct contact with the thrust disc surface in the transient start-stop process. However, the wear marks of coating and foils in the multi-leaf journal and thrust bearings have no serious adverse effects on the start-up, steady operation and speed-down of the turbo machinery. The good state of the multi-leaf foils after repeated start-stop tests indicates that the bearings have good applicability and adaptability.
5. Results and discussion

Both the multi-leaf foil journal and thrust bearing were designed and applied in the high-speed turbo machinery to evaluate their application performance. Static load-deflection, start-up and steady operating friction torque tests of the multi-leaf journal bearings with different foil thickness were conducted to analyze their static and transient characteristics. Afterwards, A-type and B-type bearing combinations were applied and tested in the high-speed turbo machinery, and the transient speed-up, high speed, and speed-down processes of the rotor-bearing system were compared and analyzed. The results can be summarized as follows:

1. The multi-leaf foil journal bearing with different foil thicknesses all have nonlinear stiffness characteristics due to the Coulomb friction effect between foils. The structural stiffness of the multi-leaf foil journal bearing increased with the foil thickness. There is an obvious hysteresis phenomenon, which indicates that the bearings have certain damping characteristics in operation.

2. Frictional torque at start-up and steady operation of the multi-leaf journal bearing increases with the bearing load and foil thickness. In the given bearing load range, the start-up and steady operation frictional torque of the bearing could be maintained at a low value, which ensures that the turbo machinery could start up and operate easily.

3. The turbo machinery with integrated aerodynamic multi-leaf foil journal and thrust bearing shows good stability and reliability. The main frequencies and sub-synchronous amplitude of B-type are much smaller than those of A-type. The application of the integrated aerodynamic multi-leaf foil bearings in the high-speed turbo machinery could simplify the system and improves the operation stability effectively.

4. After dozens of start-stop tests, the wear marks of the journal foils and thrust foils are concentrated in their trailing edge area, due to their direct contact and dry friction with the rotor surface in the transient start-stop process. Besides, the elastic deformation of the journal foils and the corresponding slippage of their trailing edge result in the wear marks in the contact area between adjacent journal foils. The worn multi-leaf foils have no obvious adverse impact on the bearings performance, which indicates their good applicability and adaptability.

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