Numerical analysis of aerodynamic lubricated double-decked protuberant foil thrust bearing

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Abstract
Double-decked protuberant foil bearing is a type of foil bearing that uses two stacking-up layers of protuberant foils as supporting structure. In preliminary experiments, wear failure of foil thrust bearing occurs occasionally due to excessive axial load in the bearing. Further improvement of the bearing performance is necessary for its better engineering application. In this paper, a numerical model of this kind of foil bearing is applied to predict its static characteristics. Reynolds equation and Kirchhoff equation are coupled via successive iterations of film pressure and foil deformation. A parametric study is conducted to improve the static characteristics of the bearing, which is conducive to load capacity improvement. The load capacity of the bearing is represented by the bearing load with the given nominal film clearance. Due to point and surface contact between the foils, distribution of protuberances plays an important role in the stiffness distribution of the protuberant foil thrust bearing. Effect of radial location of protuberances on bearing load is analyzed. Furthermore, parametric studies with different foil thickness and diameter are conducted for further optimization. The results indicate that large unevenness of stiffness distribution will deteriorate its static characteristics. Through adjustment of the radial locations of protuberances, static performance of the double-decked protuberant foil thrust bearing can be improved prominently under different operating conditions.

Keywords: Gas lubrication, Foil bearing, Thrust bearing, Static characteristics, Parametric study

1. Introduction

Foil bearings are thought to be a promising alternative technology due to its advantages of high speed, oil-free, low friction, and long life (Heshmat, 1982). It has been widely applied in turbomachinery in the past few decades (DellaCorte, 2011; André, 2014). Several types of foil bearings have been presented (Somaya, 2009), such as multi leaf type (Oh, 1976), metal mesh type (André, 2010), viscoelastic rubber type (Zhou, 2009), bump type and protuberant type (Lai, 2014).

There are some aspects requiring great attention in design stage of foil bearings, such as stability (Gu, 2017), thermal balance (Ryu, 2012), ultimate capacity and structural parameters study (Li, 2018). Reliable mathematical models are crucial to predict the performance of foil bearings accurately (André, 2009). Up to now, many numerical models have been developed to provide theoretical reference and guidance. Feng and Kaneko (2009) presented a mathematical model for multi wound foil bearings, which considered the effect of temperature on gas viscosity. Peng and Carpino (2009) proposed a mathematical model for bump foil journal bearings, which predicted the stiffness and damping coefficients of bump foil bearings and discussed the effect of the bearing compliance on the dynamic coefficients. Du and Zhu (2015) presented a mathematical model for multi leaf foil bearings, which considered the effect of surface contact between foils on bearing characteristics. DellaCorte and Valco (2000) presented a simple rule of thumb method calculating the load capacity of foil bearings. Kim and André (2007) utilized a flow advection model to solve the partial differential equations for the zeroth and first-order pressure fields. Lee et al. (2008) proposed a
mathematical model considering the three-dimensional structure of bump foils. Peng and Khonsari (2006) developed a thermal-elastic model for predicting the three-dimensional temperature field in bump foil journal bearings, which considered compressibility and viscosity-temperature characteristic of air. Iordanoff (1999) proposed a design method for bump foil thrust bearing based on a three-dimensional model. God and Kaneko (2015) tailored the thrust foil bearing stiffness in the radial and circumferential directions to enhance its load capacity. New configuration for the bump foil are introduced to accommodate the bearing deformation, which could achieve improved load capacity. Hou et al. (2011) presented a model for the single-decked protuberant foil journal bearing. However, the model was not applicable to the supporting sub-foils with multi-layer protuberant foils. Zheng et al. (2016) presented a mathematical model for protuberant foil journal bearings. In their model, the foils in the bearing were treated as thin plates. The deformation of top foil and middle protuberant foil was calculated by considering the support force between different foils. However, parameter optimization has not been conducted for the bearing yet.

Along with these theoretical studies, many related experimental researches have been carried out to study the performance of foil bearings. Heshmat and Ku (1994) measured structural stiffness and damping of bump foil bearings by exciting a nonrotating shaft and measuring dynamic force on the foil bearing. Song and Kim (2006) extended experimental studies on structural stiffness and damping characteristics of a foil bearing with compression springs. Hou et al. (2004) developed two kinds of compliant foil bearings which use copper wires and elastic material as the supporting component. The performance of these two compliant bearings were compared. Kim and San Andrés (2008) suggested that load capacity of isothermal foil bearings was affected mainly by the allowable deflection of the supporting component. Because of the different forms of supporting structure, characteristics of protuberant foil thrust bearings were different from bump foil bearings. Radil and Zeszotek (2004) measured the internal temperature foil of bump foil radial bearing with different speeds and loads. Their experimental results indicated that the bearing temperature is affected by rotational speed and load, and speed is more influential. Hou et al. (2015) conducted experimental studies on protuberant foil journal bearings and the conventional hydresil foil journal bearing. Lai et al. (2016) carried out a series of experimental investigations on the multi-layer protuberant foil thrust bearing, which proved that the protuberant foil thrust bearing had good static characteristics and stability. The load capacities of the protuberant foil bearings were greatly affected by its configuration. However, the effect of protuberances distribution on bearing performance was not analyzed in their study.

Wear failure of the protuberant foil thrust bearings occurs occasionally in preliminary experiments due to excessive axial load in the bearing. It is necessary to improve the bearing performance for its better engineering application. In this paper, double-decked protuberant foil thrust bearing is sampled to conduct the parametric study of the bearing. A numerical model of this kind of foil bearing is applied to predict its static characteristics. Reynolds equation and Kirchhoff equation are coupled via successive iterations of film pressure and foil deformation. Radial location of protuberances on bearing performance is analyzed for parametric study. Additionally, foil thickness and foil diameters are evaluated for further study.

2. Configuration of double-decked protuberant foil thrust bearing

A schematic diagram of double-decked protuberant foil thrust bearing, protuberant foils and gas film thickness distribution are shown in Fig.1(a), Fig.1(b) and Fig.1(c), respectively. The bearing consists of six thrust pads. Each pad contains three layers of foils, which are labeled as top foil, middle foil and bottom foil according to their axial positions. Top foil and protuberant foils are all taped to form a convergent clearance from the leading edge. There are protuberances with different radial locations on the middle foil and bottom foil, which stack up as supporting structure. The distribution of protuberances is determined by radial locations of protuberances. On each protuberant foil, the two protuberances located at the inner radial location are named as inner protuberances, the four protuberances located at the outer radial location are named as outer protuberances. Radial locations of the inner protuberances and outer protuberances on middle foil and bottom foil are named as \( R_{m1}, R_{m2}, R_{b1}, \) and \( R_{b2}, \) respectively. Structural parameters of the protuberant foil thrust bearing are listed in Tab.1. \( R_{m1}, R_{m2}, R_{b1} \) and \( R_{b2} \) are different to avoid the overlap of protuberances and form the elastic supporting structure. Even distribution of protuberances in circumferential direction is adopted for convenience of fabrication. Movement of foils is restricted by pins, which ensure that the foils will not move in the circumferential or radial direction, and four edges of these foils are all free.
Fig. 1 Schematic diagram of (a) multi-layer protuberant foil bearing and (b) protuberant foils and (c) gas film thickness distribution

Table 1 Structural parameters of the double-decked foil thrust bearing

| Variables                                | Values |
|------------------------------------------|--------|
| Thickness of top foil $l_t$/mm           | 0.07   |
| Thickness of protuberant foil $l_p$/mm   | 0.07   |
| Height of protuberance / mm              | 1.2    |
| Modulus of elasticity /GPa               | 130    |
| Nominal clearance $h_0$/μm               | 10     |
| Height of taper /μm                      | 20     |
| Material of protuberant foil Qbe        | 1.7    |
| Material of top foil Qbe                 | 1.7    |

3. Mathematical Model

3.1 Equation for gas lubrication

The gas film is described by Reynolds equation. Its dimensionless form can be written as (Heshmat, 2007):

$$\frac{1}{R^2} \frac{\partial}{\partial R} \left( RP \frac{\partial P}{\partial R} \right) + \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( RP \frac{\partial P}{\partial \theta} \right) = \frac{\partial (PH)}{\partial \theta} + 2 \frac{\partial (PH)}{\partial R} (1)$$

Where $P = \frac{p}{p_a}$, $R = \frac{r}{R_2}$, $\gamma = \frac{v}{\omega}$, $\tau = \frac{v t}{R_2}$, $\Lambda = 6 \mu \omega r^2 / p_a c^2$

The boundary condition for Eq.(1) are:

$p = p_a$ at $r = R_1$ and $R_2$ and at $\theta = 0$ and $\beta$

Where $R_1$ is the inner radius of foil, $R_2$ is the outer radius of foil, $\beta$ is angular extent of bearing pad.

The film thickness is described as:
\[
\begin{align*}
    h &= h_n + g(r, \theta) + w \\
    g(r, \theta) &= \begin{cases} 
        h_{super} \left(1 - \frac{\theta}{b\beta}\right) & 0 < \theta < b\beta \\
        0 & b\beta < \theta < \beta
    \end{cases}
\end{align*}
\]  

(2)

Where \( b \) is the taper land ratio, \( h_n \) is the normal clearance, \( h_{super} \) is the height of taper, \( w \) is the foil deflection of top foil.

Finite element method is applied to solve Reynolds Equation.

### 3.2 Finite element model for compliant foil

Since thicknesses of the foils are small compared to their length and width, Kirchhoff equation is used to solve the stress and deformation of the top foil and protuberant foils, which can be written as follow (Timoshenko, 1959):

\[
D_0 \left( \frac{\partial^2}{\partial r^2} + 1 \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2}{\partial \theta^2} \right) \left( \frac{\partial^2 w}{\partial r^2} + 1 \frac{\partial}{\partial r} + \frac{1}{r^2} \frac{\partial^2 w}{\partial \theta^2} \right) = p(x, y) 
\]

(3)

Where \( D_0 = \frac{Eh^3}{12(1-v^2)} \)

Finite element method is used to solve the equation of the corresponding plate theory. Standard rectangular element with four nodes are used to discretize the foils (Zienkiewicz, 2005).

Foil structural stiffness matrix is obtained by coupling the element stiffness matrix. Stiffness equation of foil is defined as:

\[
[K][q] = [f]
\]

(4)

Where \([K]\) is foil structural stiffness matrix, \([q]\) is generalized displacements vector, \([f]\) is generalized force vector.

The mesh of the numerical model is divided according to the location of protuberances. The stiffness of protuberance is assumed to be infinity in this model, due to its structural characteristics.

### 4. Parametric study

Based on the mathematical model, a parametric study is conducted to improve the static characteristics of the protuberant foil thrust bearing. The load capacity of the bearing is represented by the bearing load with the given nominal film clearance. As all pads are periodic symmetrical in the circumferential direction, the model of one pad is used in this numerical calculation when there is no tilting. The same mesh is applied in gas film and foils for convenience of data transmission, which contains 24 elements in radial direction and 64 elements in circumferential direction. Mesh-independence is checked by doubling the mesh density. The difference between two calculated bearing loads with different mesh density is less than 1% at the same operating condition.

The flow chart for the bearing load solutions is shown in Fig.2. In the process of numerical calculation, the gas film pressure is calculated by solving the Reynolds equation. With the gas film pressure acting on the top foil, the reaction force on the middle foil at the contact points is calculated. Afterwards, the action force at contact points for the middle foil is calculated by opposing the reaction force acting on middle foil, so the deformation of the middle foil is obtained. Then, the deformation of the top foil is calculated using the deformation of the middle foil as the displacement boundary condition, which means \( W = W_2 \) at contact points between the top foil and middle foil. The Newton-Raphson iteration of total foils deformation is iterated until the deformation of the top foils is converged. Finally, the film pressure is obtained when the minimal value of the film thickness converges to the nominal clearance.
In order to improve its static characteristics, the bearing load with different radial location of the protuberances is calculated. Due to the stacking of two layers of protuberant foils, the distribution of protuberances on both the middle foil and bottom foil should be considered. Namely, both the distribution of support points for the top foil ($R_{m1}$ and $R_{m2}$) and the stiffness characteristics of the support points should be taken into account. The stiffness characteristics of the support points are decided by the relative position between protuberances of the middle foil and that of the bottom foil ($R_{m1}, R_{m2}, R_{b1},$ and $R_{b2}$). The flow chart for a parametric study of the double-decked protuberant foil thrust bearing is shown in Fig.3. Firstly, bearing load is calculated with varied $R_{b2}$ when $R_{b1}, R_{m1}$ and $R_{m2}$ are all constant. The value of $R_{b2}$ is limited by $R_1$ and $R_2$. Then, the value of $R_{b1}$ is revised after calculation of $R_{b2}$, and the above calculation is repeated. Effect of $R_{b1}$ and $R_{b2}$ on bearing load is obtained with a constant value of $R_{m1}$ and $R_{m2}$. The maximal bearing load obtained in this calculation process is considered as the bearing load with $R_{m1}$ and $R_{m2}$. The effect of $R_{m1}$ and $R_{m2}$ on the bearing load is obtained through a similar process.
Influence of middle foil on the bearing characteristics is investigated for its contact with the top foil directly. Radial location of protuberances on the middle foil is shown in Fig.4(a). The top foil consists of three parts based on the inner and outer protuberances on the middle foil. The area between the inner edge and inner protuberances is defined as region I, the area between the inner and outer protuberances is defined as region II and the rest of the top foil is defined as region III. The effect of $R_{m1}$ and $R_{m2}$ on the bearing load is shown in Fig.4(b). Every point on these curves represents the maximal bearing load with different value of $R_{b1}$ and $R_{b2}$ in the coordinates of $R_{m1}$ and $R_{m2}$. The calculation results show that both the values of $R_{m1}$ and $R_{m2}$ have obvious influence on bearing load. There are local optimization values for the bearing load with $R_{m2}$ in different value of $R_{m1}$. Furthermore, the value of $R_{m2}$ is almost invariable when the bearing load reaches the local optimization value in different value of $R_{m1}$. Maximal bearing load is obtained when $R_{m1}$ is 9.7mm and $R_{m2}$ is 14.3mm. The division of the top foil based on the inner and outer protuberances leads to the effects of $R_{m1}$ and $R_{m2}$ on the bearing load. With a certain value of $R_{m1}$, area of region II enlarges with an increase of $R_{m2}$, while it is opposite for region III. When the value of $R_{m2}$ is relatively small, the deformation of region III is too large compared with other regions, high gas pressure could not persist in region III. Similarly, when the value of $R_{m2}$ is larger, high gas pressure could not persist in region II due to inordinate local
deformation. Therefore, bearing load declines when the value of $R_{m2}$ increases or decreases near the local optimization value. Also, the trend of bearing load is applicable for region I. Area of region I and II is decided by the value of $R_{m1}$. In other words, local deformation of region I and II is affected by $R_{m1}$. The optimal value of $R_{m1}$ for the bearing load is obtained when inordinate local deformation is avoided in both region I and II. It can be seen that areas of the three parts should be controlled appropriately by adjusting the radial location of protuberances to avoid inordinate local deformation in the top foil.

![Fig.4](a) Area division of top foil and (b) effect of $R_{m1}$ and $R_{m2}$ on bearing load

![Fig.5](a) A schematic diagram of protuberance distribution on protuberant foils and (b) effect of $R_{b1}$ and $R_{b2}$ on bearing load

Effect of the relative position between protuberances of the middle foil and that of the bottom foil is also be analyzed for its influence on the stiffness characteristics of the support points. As shown in Fig.5(a), the supporting effect of the bottom foil on protuberances of the middle foil is different when $R_{b2} < R_{m2}$ or $R_{b2} > R_{m2}$. In order to conduct further analysis for the effect of $R_{b1}$ and $R_{b2}$ on bearing load, the bearing load with $R_{b1}$ and $R_{b2}$ is sampled and presented in Fig.5(b), when $R_{m1}=9.7$mm and $R_{m2}=14.3$mm. As mentioned above, Fig.5(b) represents the data point of $R_{m1}=9.7$ and $R_{m2}=14.3$ in Fig.4(b), that is to say, every data point of Fig.4(b) is derived from the maximal calculated value of these figures that similar to Fig.5(b). The stiffness characteristics of the support points is determined by the distribution of protuberances on both protuberant foils. Indeed, the two stacking-up layers of protuberant foils can be approximated as a distributed non-linear spring under the top foil, which is affected by the location of all protuberances. The distribution of the spring is the location of protuberances on the middle foil ($R_{m1}$ and $R_{m2}$). Supporting stiffness characteristics of middle foil changes with $R_{b1}$ and $R_{b2}$ with a certain value of $R_{m1}$ and $R_{m2}$. When $R_{b2} < R_{m2}$, inner protuberances on the middle foil are supported by inner and outer protuberances on the bottom foil, and outer protuberances on the middle
foil are only supported by outer protuberances on bottom foil. There is an obvious difference between the supporting effect on inner and outer protuberances. When $R_{b1}$ is constant, the supporting effect on inner protuberances of the middle foil is weakened with an increase of $R_{b2}$; while it is opposite for outer protuberances of the middle foil. The bearing load increases with a decrease of the difference between supporting effect on inner and outer protuberances. The bearing load reaches a maximal value when the difference between the supporting effect on inner and outer protuberances reaches the minimal value. Similarly, the bearing load decreases with a rise of the difference between supporting effect on inner and that on outer protuberances.

5. Comparison of parameters of pre-optimized and optimized bearing

Effect of protuberances distribution on bearing load has been analyzed, which needs to be proved by the detailed numerical results of the bearing with different protuberances distribution. Comparative analysis between two bearings with different protuberances distribution is conducted to analyze the effect of protuberances distribution on bearing performance intuitively. The double-decked protuberant foil thrust bearing that has been tested is named as pre-optimized bearing (Zheng, 2017), and the foil thrust bearing that achieves maximal bearing load in the numerical calculation is named as optimized bearing. Structural parameters of the pre-optimized and optimized bearing are listed in Tab.2. Dimensionless foil deformation, film thickness and pressure are analyzed to compare the static characteristics of these two foil bearings. Comparison of dimensionless top foil deformation, dimensionless middle foil deformation, dimensionless film thickness and dimensionless pressure between the pre-optimized bearing and optimized bearing are shown from Fig.6 to Fig.9.

| Variables                          | Pre-optimized | Optimized |
|------------------------------------|---------------|-----------|
| Inner radius of foils /mm          | 6             | 6         |
| Outer radius of foils /mm          | 17            | 17        |
| Radius of inner protuberances on middle foil /mm | 10.4          | 9.7       |
| Radius of outer protuberances on middle foil /mm | 14.8          | 14.3      |
| Radius of inner protuberances on bottom foil /mm | 8.2           | 7.8       |
| Radius of outer protuberances on bottom foil /mm | 12.6          | 13.3      |
| Bearing load /N                    | 2.23          | 3.93      |

Comparison of the dimensionless deformation of middle foil is shown in Fig.6. For these two bearings, axial displacement of the inner protuberances is small when the film pressure is applied on foils. It can be inferred that the inner protuberances on the middle foil are all supported with larger stiffness. For the pre-optimized bearing, the radial location of outer protuberances on the bottom foil is smaller. The supporting effect of outer protuberances is less than that of inner protuberances on the middle foil. Therefore, the axial displacement of outer protuberances on the middle foil is larger compared with that of inner protuberances. For the optimized bearing, a uniform axial displacement could be achieved in the inner and outer protuberances when the film pressure is applied to the foils, owing to the uniform stiffness distributed in the whole bearing surface. As shown in Fig.7, the deformation of the top foil increases continuously from the middle area to the outer area in the pre-optimized bearing. The difference of the axial displacement between the inner and outer protuberances on the middle foil leads to a large difference in the deformation between the inner and outer area on the top foil. However, obvious deformation only appears on the area without support in the optimized bearing. A comparison of the dimensionless film thickness is shown in Fig.8. For the pre-optimized bearing, thin film thickness area only appears at the inner area due to the nonuniform deformation in radial direction. When the film thickness of the inner area approaches the nominal clearance, the film thickness of the other areas is still large. Under this circumstances, bearing wear may be caused due to the smaller film thickness at the other area. For the optimized bearing, a thin film thickness area could appear on both inner and outer area simultaneously thanks to the evenness of the top foil deformation. As shown in Fig.9, high film pressure only exists at the inner area for its film thickness distribution, the other area can not generate effective high pressure in the pre-optimized bearing. However, the high pressure area is enlarged to almost the entire bearing surface resulting from the enlargement of thin film thickness area in the optimized bearing.
Fig. 6 Dimensionless middle foil deformation of (a) pre-optimized and (b) optimized bearings

Fig. 7 Dimensionless top foil deformation of (a) pre-optimized and (b) optimized bearings

Fig. 8 Dimensionless film thickness of (a) pre-optimized and (b) optimized bearings
In order to analyze the supporting effect of the protuberant foil on the top foil quantitatively, the nodes where protuberances are located on the middle foil is marked as No.1 to No.6, as shown in Fig.10(a). Actual bearing stiffness is determined by its structural parameters, which could be obtained through the test of static loading (load-deflection). In this paper, static superficial stiffness \(K_{sp}\) is used to represent the supporting effect of protuberant foil on top foil. The superficial stiffness of these nodes is calculated with the displacement and corresponding action force \(K_{sp} = F/X\), the displacement of these nodes are derived from the middle foil deformation, while the corresponding action force is derived from the reaction force of the contact points between the top foil and middle foil under the given operation condition. The static stiffness of the elastic support points under the steady state (which is approximated as a distributed non-linear spring under the top foil) could be represented through the static superficial stiffness \(K_{sp}\). The value of corresponding \(K_{sp}\) could be obtained according to the specific numerical results of the given steady-state condition. The spring coefficients of the protuberant foil could be defined by \(K_{sp}\) of these nodes where protuberances are located on the middle foil, for they are the contact points between the top foil and protuberant foil. In other words, the distribution of the spring is the location of protuberances on the middle foil, while the stiffness characteristics of the spring is represent by \(K_{sp}\) (which is determined by the distribution of protuberances on both protuberant foils). Displacement and static superficial stiffness of these nodes are shown in Fig.10(b). Compared with the axial displacement of these points in the pre-optimized bearing, that of the optimized bearing is more uniform. Besides, there is a large difference in the value of \(K_{sp}\) between No.1, 2 and No.3, 4, 5, 6 in the pre-optimized bearing, while the static superficial stiffness of all points is more uniform in the optimized bearing. In general, a large difference in static superficial stiffness distribution has a negative effect on the static characteristics, which is in accordance with the comparative analysis of the numerical results between the two bearings.

Afterwards, the performance of the pre-optimized bearing and the optimized bearing is compared in different rotating speed and nominal clearance. Bearing load of these two bearings with rotating speed and nominal clearance is
shown in Fig.11. In small nominal clearance, bearing load improvement of the optimized bearing is evident in all speed ranges. In large nominal clearance, bearing load improvement of the optimized bearing is not obvious in low speed range, and bearing load difference between the two bearings grows gradually with rotating speed. In low rotating speed, the bearing load improvement of the optimized bearing is more evident with decrease of the nominal clearance, which means the optimization for bearing performance is more applicable in smaller nominal clearance when the rotating speed is low. Besides, effect of the rotating speed on the bearing load difference between the two bearings is weaker with a decline of the nominal clearance. It is worth noting that the bearing load of these two bearings tends to different asymptotic values with rotating speed.

![Fig.11](image1)

**Fig.11** Bearing load of (a) pre-optimized and (b) optimized bearings

### 5.1 Effects of structural parameters on parametric study

Effect of structural parameters on parametric study is also researched for further study on bearing performance. Foil thickness and foil area are two important parameters that should be concerned in engineering design. Further parametric study of the bearing is conducted considering the effect of foil thickness. When the foil thickness is changed from 0.07mm to 0.1mm with other parameters being constant, bearing load with different foil thickness is shown in Fig.12. The trend of the optimized results is similar with different foil thickness. The optimized radial location of protuberances is almost the same when it reaches the maximal bearing load. Therefore, the thickness of foils has no significant impact on the optimal radial location of protuberances. Under the same gas film pressure, foil deformation decreases for the foil with thicker foil thickness. The high pressure area enlarges due to smaller local and overall deformation of the foil, which is beneficial to the bearing load.

![Fig.12](image2)

**Fig.12** Effect of $R_{m1}$ and $R_{m2}$ on bearing load with different thickness

Thrust bearing area is another important parameter for the bearing load. The bearing is named as the original bearing when its size is unchanged, and the bearing is named as the larger bearing when its inner and outer diameters of
foils are increased to 18mm and 40mm, respectively. The foils of the original bearing and the larger bearing are shown in Fig.13(a) and Fig.13(b), respectively. The effect of $R_{m1}$ and $R_{m2}$ on the bearing load with different foil size is shown in Fig.13(c). The foil size has a slight influence on the optimized results. For the larger bearing, the effect of $R_{m2}$ on the bearing load is reduced compared with that of the optimized bearing. Moreover, the area of the larger bearing is increased by 26% compared with the optimized bearing. However, the maximal bearing load is just increased by 5% and the maximal bearing load per unit area is decreased by 17% through conversion.

![Fig.13 Foil of (a) the original bearing and (b) the larger bearing and (c) effect of $R_{m1}$ and $R_{m2}$ on bearing load with different foil size](image)

Fig.14 Dimensionless (a) middle foil deformation, (b) top foil deformation, (c) film thickness and (d) pressure field of the larger foil bearing

![Fig.14 Dimensionless](image)
Foil deformation, film thickness and pressure field of the larger bearing with its maximal bearing load are shown in Fig.14. As shown in Fig.14(a) and Fig.14(b), larger deformation appears on the top foil, compared with the optimized bearing. Although the axial displacement of nodes where the inner and outer protuberances are located on the middle foil is almost the same, the deformation of the top foil at the protuberant supporting locations is similar. However, there is a big difference in deformation between the area supported by protuberances and the area without support. Nonuniformity of foil deformation is more obvious in the entire bearing surface. The maximal deformation of the top foil in the larger foil bearing is doubled compared with that of the optimized bearing. Appropriate local deformation is beneficial to the characteristics of the foil bearing, but excessive deformation of the top foil has a negative influence on the static load capacity. With a certain number of protuberances, nonuniformity of foil deformation increases with bearing area. As is shown in Fig.14(c) and Fig.14(d), larger film thickness appears in the inner and middle area of the bearing due to larger local displacement. The high-pressure zone of the bearing is more dispersive compared with the pressure field of the optimized bearing, which causes the decline of load capacity. It is necessary to increase the bearing stiffness by increasing the number of protuberances on each protuberant foil, which is effective to alleviate the excessive deformation of the top foil and improve the static performance of the bearing. However, too many protuberances lead to undue stiffness of the bearing, which is unbeneﬁcial to the bearing performance.

6. Conclusion

Parametric studies of the protuberant foil thrust bearing are conducted to improve its load capacity, which is important for its engineering applications. A numerical model of the double-decked protuberant foil thrust bearing is applied to predict the static characteristics of this kind of foil thrust bearing. Effect of the protuberances radial location on static characteristics of the bearing is analyzed through the results of the parametric study. In addition, further parametric studies of the bearing are conducted considering the effect of foil thickness and foil size. The results can be summarized as follows.

1. Bearing load of the optimized protuberant foil thrust bearing is improved by 76% compared with the pre-optimized bearing through parametric study, which improves the bearing performance prominently.
2. Bearing performance is determined by the distribution of protuberances on both the middle foil and bottom foil, due to the stacking of two layers of protuberant foils. Bearing characteristics is affected directly by the protuberances of the middle foil. Moreover, the stiffness characteristics of these support points is another important influential factor for the bearing, which is decided by the relative position between protuberances of the middle foil and that of the bottom foil
3. Stiffness distribution of the bearing is determined by both distribution and stiffness characteristics of the support points. It is necessary to ameliorate the large unevenness between stiffness distributions by combination of protuberant distributions on both protuberant foils.
4. Foil thickness has no significant impact on the optimal radial location of protuberances. However, maximal bearing load per unit area of the larger bearing is decreased by 17% compared with the original bearing, resulting from its larger local deformation. It is necessary to increase the stiffness of the larger bearing by increasing the number of protuberances on its protuberant foil, which is effective to alleviate the inordinate local deformation.

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 NOMENCLATURE

\( b \)  taper land ratio \\
\( c \)  height of taper [\( \mu \text{m} \)]
$E$ modulus of elasticity [GPa]
$\{f\}$ generalized force vector
$F$ action force of nodes
$h$ film thickness [μm]
$h_0$ normal clearance [μm]
$h_{taper}$ height of taper [μm]
$H$ dimensionless film thickness
$H_0$ dimensionless normal clearance
$K_{sp}$ static superficial stiffness of nodes
$[K]$ foil structural stiffness matrix
$l$ thickness of foil [mm]
$p$ film pressure [Pa]
$p_a$ ambient pressure [Pa]
$P$ dimensionless film pressure
$p(x,y)$ load in vertical direction [N]
$q$ displacements at one node [m]
$\{q\}$ generalized displacements vector
$r$ coordinate in radial direction
$R_1$ inner radius of foils [mm]
$R_2$ outer radius of foils [mm]
$R_{m1}$ radius of inner protuberance on middle foil [mm]
$R_{m2}$ radius of outer protuberance on middle foil [mm]
$R_{b1}$ radius of inner protuberance on bottom foil [mm]
$R_{b2}$ radius of outer protuberance on bottom foil [mm]
$t$ time [s]
$\tau$ dimensionless time
$w$ foil deflection of top foil [m]
$W$ dimensionless foil deflection of top foil
$W_2$ dimensionless foil deflection of middle foil
$x$ coordinate in $x$ direction
$y$ coordinate in $y$ direction
$X$ displacement of nodes
$\gamma$ whirl ratio
$\theta$ coordinate in circumferential direction
$\mu$ viscosity [Pa s]
$\beta$ angular extent of bearing pad
$\nu$ Poisson ratio
$\omega$ rotational speed [rad/s]
$\Lambda$ bearing number
$\zeta$ relaxation factor

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