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气体润滑弹性波箔片轴承动态特性分析

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摘 要: 通过摄动方法并结合气体润滑压力控制 Reynolds 方程和气膜压力、气膜厚度及箔片变形等变量的 Taylor 展开式, 推导出弹性波箔片轴承气膜动力特性系数计算微分方程组。同时, 考虑气体可压缩性等因素, 建立气膜厚度、箔片变形和气膜压力三者之间关系的数学模型方程组。运用 Newton-Raphson 迭代法和有限差分法耦合求解气膜动力特性系数计算微分方程组和气膜厚度数学模型方程组, 得到弹性波箔片轴承气膜刚度和阻尼等动力特性系数随偏心率、转速、扰动频率及轴承长径比等参数变化的规律特性, 为箔片轴承设计、优化和弹性波箔片轴承-转子系统理论及试验分析奠定基础。计算结果表明, 随着偏心率和转速的增大, 润滑气膜主刚度均有不同程度的提高, 同时, 润滑气膜刚度和阻尼的大小与扰动频率关系密切。

关 键 词: 气体润滑; 弹性波箔片轴承; 气膜厚度数学模型; 耦合解; 动态特性

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Dynamic characteristics analysis of bump foil journal bearings with air lubrication

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Abstract: Differential equations of air film in elastic bump foil journal bearing were deduced by the perturbation method with air lubrication Reynolds equation and series expansions of air film, air film thickness and foil deformation. The mathematics model of the relations among air film thickness, bump foil deformation and air film pressure was developed in consideration of air compressibility. Both finite difference method and Newton-Raphson method were applied to solve the coupled differential equations and air film thickness mathematics model equations, while the variation rules of air film stiffness and damping coefficients were achieved with different eccentricities, speeds, perturbation frequencies and different length-diameter ratios. This provides a basis for bump foil journal bearing design, optimization and theoretical and test analysis of rotor-elastic foil bearing system. The numerical results show that the main air film stiffness is enhanced when eccentricity and speed increase, and the value of air film stiffness and damping of bump foil journal bearing are connected strongly with the perturbation frequencies.

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Key words: air lubrication; elastic bump foil journal bearing;
air film thickness mathematics model; coupling solution;
dynamic characteristics

弹性波箔片轴承是一种自作用式弹性箔片气体动压轴承,由于弹性波箔片轴承具有低噪声、长寿命、高转速、耐高温/低温、低能量损失及优良的动力学特性等优点^[1-4]而被成功地运用在航空、航天^[5]等领域,如波音 747,波音 777,SAAB-2000,B-2 等各种高速轻载旋转机械中^[6]。随着研究和试验的不断深入,箔片轴承可望在船舶机械等领域的应用实现突破。

箔片轴承结构复杂及作用过程的自适应特点,使得箔片轴承的理论研究相较于油润滑等刚性表面轴承研究进展缓慢,同时,理论分析也滞后于试验研究(包括箔片内表面镀层等试验)^[7-12]。

在弹性波箔片轴承动态特性研究方面,2004 年和 2005 年,西安交通大学虞烈教授等先后发表了关于弹性波箔片空气轴承的完全气弹耦合解^[13-14]的文章,通过引入柔性箔片的静、动变形,联立求解气体润滑 Reynolds 方程和弹性波箔片动静变形方程,获得完全气弹润滑耦合解。在国外的箔片轴承气膜动态特性研究中,摄动法也是主要分析方法之一^[15]。

本文通过摄动法并结合气体润滑 Reynolds 方程和气膜压力、厚度及波纹箔片变形等变量的 Taylor 展开式推导出弹性波箔片轴承气膜动力特性系数计算微分方程组。考虑箔片变形和气体润滑介质可压缩性等因素建立起气膜厚度、箔片变形及气膜压力三者之间关系的数学模型方程组。通过 Newton-Raphson 迭代法和有限差分法耦合求解微分方程组和气膜厚度数学模型方程组,得到弹性波箔片轴承气膜刚度和阻尼等动力特性系数随偏心率、转速、扰动频率及轴承长径比等参数变化的规律特性。

1 理论分析

弹性波箔片轴承结构及波纹箔片几何结构示意图如图 1 所示。箔片轴承由轴承壳体、波纹箔片及平箔片组成,波纹箔片和平箔片的前缘点焊在轴承壳体上,后缘为自由端,波纹箔片为单一平箔片提供支承。波箔片类似于弹簧为轴承提供刚度,光滑的平箔片提供轴承表面。

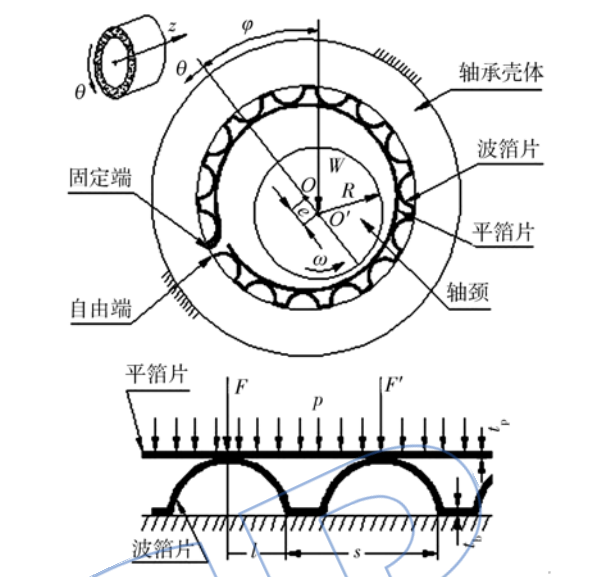


图 1 弹性波箔片轴承及波箔片几何结构图

Fig. 1 Elastic bump foil journal bearing and bump foil structure

1.1 Reynolds 方程及摄动法处理

箔片轴承气体润滑压力控制 Reynolds 方程表达式如下:

$$\frac{1}{R^2} \frac{\partial}{\partial \theta} \left(\frac{ph^3}{\mu T} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left(\frac{ph^3}{\mu T} \frac{\partial p}{\partial z} \right) = 6\omega \frac{\partial}{\partial \theta} \left(\frac{ph}{T} \right) + 12 \frac{\partial}{\partial t} \left(\frac{ph}{T} \right) \quad (1)$$

式中 R 为轴颈半径(m); θ 为圆周方向坐标(rad); p 为气膜压力(Pa); h 为气膜厚度(m); μ 为润滑气体黏度(Pa·s); T 为润滑气体温度(K); z 为轴向坐标(m); ω 为转轴转动角速度(rad/s); t 为运行时间(s)。

无量纲参数定义如下:

$$\bar{z} = \frac{z}{0.5L}; \quad \bar{h} = \frac{h}{C}; \quad \bar{p} = \frac{p}{p_a};$$

$$\bar{T} = \frac{T - 273}{T_0 - 273}; \quad \bar{\mu} = \frac{\mu}{\mu_0}; \quad \bar{t} = \omega t \quad (2)$$

式中 L 为轴承长度(m); C 为名义半径间隙(m); p_a 为环境气体压力(Pa); μ_0 为环境气体黏度(Pa·s); T_0 为环境气体温度(K); ω 为涡动角速度(rad/s)。

将式(2)代入式(1),得到无量纲形式的 Reynolds 方程为

$$\frac{\partial}{\partial \theta} \left(\frac{\bar{p} \bar{h}^3}{\mu T} \frac{\partial \bar{p}}{\partial \theta} \right) + \left(\frac{2R}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(\frac{\bar{p} \bar{h}^3}{\mu T} \frac{\partial \bar{p}}{\partial \bar{z}} \right) = \Lambda_x \frac{\partial}{\partial \theta} \left(\frac{\bar{p} \bar{h}}{T} \right) + 2\gamma \Lambda_x \frac{\partial}{\partial \bar{t}} \left(\frac{\bar{p} \bar{h}}{T} \right) \quad (3)$$

其中

$$\Lambda_x = \frac{6R^2 \omega \mu_0}{p_a C^2}; \quad \gamma = \frac{v}{\omega} \quad (4)$$

假设轴颈在转速为 ω 时的平衡位置 O 的坐标为 (x_0, y_0) , 坐标建立如图2所示. 给定轴颈在平衡位置一个小扰动量, 其中小扰动位移和小扰动速度分别为 $(\Delta x, \Delta y)$ 和 $(\Delta \dot{x}, \Delta \dot{y})$, 小扰动量无量纲化如下:

$$\left. \begin{aligned} \Delta X &= \frac{\Delta x}{C} = |\Delta X| e^{i\bar{t}} \\ \Delta Y &= \frac{\Delta y}{C} = |\Delta Y| e^{i\bar{t}} \end{aligned} \right\} \quad (5)$$

对式(5)求导整理可得

$$\left. \begin{aligned} \Delta \dot{X} &= |\Delta X| e^{i\bar{t}} i = \Delta X i \\ \Delta \dot{Y} &= |\Delta Y| e^{i\bar{t}} i = \Delta Y i \end{aligned} \right\} \quad (6)$$

$$\left. \begin{aligned} \Delta \ddot{X} &= |\Delta X| e^{i\bar{t}} i \cdot i = -\Delta X \\ \Delta \ddot{Y} &= |\Delta Y| e^{i\bar{t}} i \cdot i = -\Delta Y \end{aligned} \right\} \quad (7)$$

箔片变形 u 无量纲化定义为

$$u = \bar{u}C \quad (8)$$

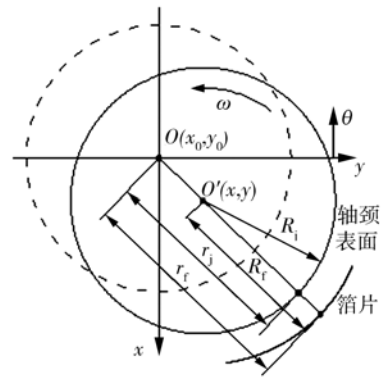


图2 轴颈平衡位置坐标

Fig. 2 Coordinate of journal at equilibrium position

将气膜压力 $\bar{p}$ 、气膜厚度 $\bar{h}$ 和箔片变形 $\bar{u}$ 在平衡位置 $\bar{p}_0, \bar{h}_0, \bar{u}_0$ 处对各扰动参数按 Taylor 形式展开如下:

$$\left. \begin{aligned} \bar{p} &= \bar{p}_0 + \bar{p}_x \Delta X + \bar{p}'_x \Delta \dot{X} + \bar{p}_y \Delta Y + \bar{p}'_y \Delta \dot{Y} \\ \bar{h} &= \bar{h}_0 + \bar{h}_x \Delta X + \bar{h}'_x \Delta \dot{X} + \bar{h}_y \Delta Y + \bar{h}'_y \Delta \dot{Y} \\ \bar{u} &= \bar{u}_0 + \bar{u}_x \Delta X + \bar{u}'_x \Delta \dot{X} + \bar{u}_y \Delta Y + \bar{u}'_y \Delta \dot{Y} \end{aligned} \right\} \quad (9)$$

将式(9)代入式(3), 并按扰动微量归并整理可得如下5个微分方程:

$$\frac{\partial}{\partial \theta} \left(\frac{\bar{p}_0 \bar{h}_0^3}{\mu T} \frac{\partial \bar{p}_0}{\partial \theta} \right) + \left(\frac{2R}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(\frac{\bar{p}_0 \bar{h}_0^3}{\mu T} \frac{\partial \bar{p}_0}{\partial \bar{z}} \right) = \Lambda_x \frac{\partial (\bar{p}_0 \bar{h}_0)}{\partial \theta} \quad (10)$$

$$\frac{\partial}{\partial \theta} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}_x \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}_x \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}_x}{\partial \theta} \right) + \left(\frac{2R}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}_x \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}_x \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}_x}{\partial \bar{z}} \right) = \Lambda_x \frac{\partial}{\partial \theta} (\bar{p}_0 \bar{h}_x + \bar{p}_x \bar{h}_0) - 2\gamma \Lambda_x (\bar{p}_0 \bar{h}'_x + \bar{p}'_x \bar{h}_0) \quad (11)$$

$$\frac{\partial}{\partial \theta} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}'_x \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}'_x \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}'_x}{\partial \theta} \right) + \left(\frac{2R}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}'_x \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}'_x \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}'_x}{\partial \bar{z}} \right) = \Lambda_x \frac{\partial}{\partial \theta} (\bar{p}_0 \bar{h}'_x + \bar{p}'_x \bar{h}_0) + 2\gamma \Lambda_x (\bar{p}_0 \bar{h}_x + \bar{p}_x \bar{h}_0) \quad (12)$$

$$\frac{\partial}{\partial \theta} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}_y \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}_y \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}_y}{\partial \theta} \right) + \left(\frac{2R}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}_y \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}_y \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}_y}{\partial \bar{z}} \right) = \Lambda_x \frac{\partial}{\partial \theta} (\bar{p}_0 \bar{h}_y + \bar{p}_y \bar{h}_0) - 2\gamma \Lambda_x (\bar{p}_0 \bar{h}'_y + \bar{p}'_y \bar{h}_0) \quad (13)$$

$$\frac{\partial}{\partial \theta} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}'_y \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}'_y \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \theta} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}'_y}{\partial \theta} \right) + \left(\frac{2R}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(3 \bar{p}_0 \bar{h}_0^2 \bar{h}'_y \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}'_y \bar{h}_0^3 \frac{\partial \bar{p}_0}{\partial \bar{z}} + \bar{p}_0 \bar{h}_0^3 \frac{\partial \bar{p}'_y}{\partial \bar{z}} \right) = \Lambda_x \frac{\partial}{\partial \theta} (\bar{p}_0 \bar{h}'_y + \bar{p}'_y \bar{h}_0) - 2\gamma \Lambda_x (\bar{p}_0 \bar{h}_y + \bar{p}_y \bar{h}_0) \quad (14)$$

1.2 箔片变形与气膜厚度及气膜压力关系的数学模型

如图2所示,假设轴承半径用 R_j 表示,箔片初始半径用 R_f 表示,轴颈表面位置距坐标圆点为 r_j , 箔片表面位置距坐标圆点为 r_f , 平衡位置 (x_0, y_0) 受到小位移 $(\Delta x, \Delta y)$ 扰动后,坐标变为

$$(x, y) = (x_0 + \Delta x, y_0 + \Delta y) \quad (15)$$

各结构参数之间的关系如下:

$$\left. \begin{aligned} r_j - R_j &= -x \sin \theta + y \cos \theta \\ r_f - R_f &= u = u_0 + \Delta u \end{aligned} \right\} \quad (16)$$

式(16)中 Δu 为受到扰动后箔片径向变形增量. 气膜厚度可表达如下:

$$h = r_f - r_j = (R_f + u_0 + \Delta u) - (R_j - x \sin \theta + y \cos \theta) \quad (17)$$

将式(15)代入式(17)并无量纲化得

$$\bar{h} = \bar{h}_0 + (\Delta X \sin \theta - \Delta Y \cos \theta + \Delta \bar{u}) \quad (18)$$

结合 Taylor 展开式(9)可得

$$\left. \begin{aligned} \bar{h}_x &= \bar{u}_x + \sin \theta; & \bar{h}'_x &= \bar{u}'_x; \\ \bar{h}_y &= \bar{u}_y - \cos \theta; & \bar{h}'_y &= \bar{u}'_y \end{aligned} \right\} \quad (19)$$

设箔片刚度为 K_b , 阻尼为 B , 则气膜压力与箔片变形关系表达式有

$$p = K_b u + B \dot{u} = K_b u + B \frac{\partial u}{\partial t} \quad (20)$$

对应的无量纲形式为

$$\begin{aligned} \bar{p} &= \frac{K_b C}{p_a} \bar{u} + \frac{B \omega C}{p_a} \gamma \frac{\partial \bar{u}}{\partial \bar{t}} = \bar{K}_b \bar{u} + \bar{B} \gamma \frac{\partial \bar{u}}{\partial \bar{t}}; \\ \bar{K}_b &= \frac{K_b C}{p_a}; & \bar{B} &= \frac{B \omega C}{p_a} \end{aligned} \quad (21)$$

将式(9)和式(19)代入式(21)可得气膜压力与气膜厚度关系表达式

$$\bar{p}_0 = \bar{K}_b (\bar{h}_0 - 1 - \epsilon \cos \theta) \quad (22)$$

$$\bar{p}_x = \bar{K}_b \bar{h}_x - \bar{K}_b \sin \theta - \bar{B} \gamma \bar{h}'_x \quad (23)$$

$$\bar{p}'_x = \bar{K}_b \bar{h}'_x + \bar{B} \gamma \bar{h}_x - \bar{B} \gamma \sin \theta \quad (24)$$

$$\bar{p}_y = \bar{K}_b \bar{h}_y + \bar{K}_b \cos \theta - \bar{B} \gamma \bar{h}'_y \quad (25)$$

$$\bar{p}'_y = \bar{K}_b \bar{h}'_y + \bar{B} \gamma \bar{h}_y + \bar{B} \gamma \cos \theta \quad (26)$$

1.3 气膜刚度和阻尼系数求解表达式

联立求解式(10)~(14)和式(22)~(26)共10个方程,可得到 $\bar{p}_x$, $\bar{p}'_x$, $\bar{p}_y$, $\bar{p}'_y$ 以及 $\bar{h}_x$, $\bar{h}'_x$, $\bar{h}_y$, $\bar{h}'_y$ 等. 气膜刚度和阻尼系数表达式如下:

$$\left. \begin{aligned} \left\{ \begin{aligned} K_{xx} \\ K_{yx} \end{aligned} \right\} &= \left\{ \begin{aligned} \frac{\partial F_x}{\partial x} \\ \frac{\partial F_y}{\partial x} \end{aligned} \right\} = \\ &= -\frac{0.5 p_a R L}{C} \int_0^{2\pi} \int_{-1}^1 \bar{p}_x \begin{Bmatrix} \sin \theta \\ -\cos \theta \end{Bmatrix} d\theta d\bar{z}; \\ \left\{ \begin{aligned} \bar{K}_{xx} \\ \bar{K}_{yx} \end{aligned} \right\} &= \frac{C}{0.5 p_a R L} \left\{ \begin{aligned} K_{xx} \\ K_{yx} \end{aligned} \right\} \\ \left\{ \begin{aligned} K_{xy} \\ K_{yy} \end{aligned} \right\} &= \left\{ \begin{aligned} \frac{\partial F_x}{\partial y} \\ \frac{\partial F_y}{\partial y} \end{aligned} \right\} = \\ &= -\frac{0.5 p_a R L}{C} \int_0^{2\pi} \int_{-1}^1 \bar{p}_y \begin{Bmatrix} \sin \theta \\ -\cos \theta \end{Bmatrix} d\theta d\bar{z}; \\ \left\{ \begin{aligned} \bar{K}_{xy} \\ \bar{K}_{yy} \end{aligned} \right\} &= \frac{C}{0.5 p_a R L} \left\{ \begin{aligned} K_{xy} \\ K_{yy} \end{aligned} \right\} \end{aligned} \right\} \quad (27)$$

$$\left. \begin{aligned} \left\{ \begin{aligned} D_{xx} \\ D_{yx} \end{aligned} \right\} &= \left\{ \begin{aligned} \frac{\partial F_x}{\partial \dot{x}} \\ \frac{\partial F_y}{\partial \dot{x}} \end{aligned} \right\} = \\ &= -\frac{0.5 p_a R L}{C v} \int_0^{2\pi} \int_{-1}^1 \bar{p}'_x \begin{Bmatrix} \sin \theta \\ -\cos \theta \end{Bmatrix} d\theta d\bar{z}; \\ \left\{ \begin{aligned} \bar{D}_{xx} \\ \bar{D}_{yx} \end{aligned} \right\} &= \frac{C v}{0.5 p_a R L} \left\{ \begin{aligned} D_{xx} \\ D_{yx} \end{aligned} \right\} \\ \left\{ \begin{aligned} D_{xy} \\ D_{yy} \end{aligned} \right\} &= \left\{ \begin{aligned} \frac{\partial F_x}{\partial \dot{y}} \\ \frac{\partial F_y}{\partial \dot{y}} \end{aligned} \right\} = \\ &= -\frac{0.5 p_a R L}{C v} \int_0^{2\pi} \int_{-1}^1 \bar{p}'_y \begin{Bmatrix} \sin \theta \\ -\cos \theta \end{Bmatrix} d\theta d\bar{z}; \\ \left\{ \begin{aligned} \bar{D}_{xy} \\ \bar{D}_{yy} \end{aligned} \right\} &= \frac{C v}{0.5 p_a R L} \left\{ \begin{aligned} D_{xy} \\ D_{yy} \end{aligned} \right\} \end{aligned} \right\} \quad (28)$$

2 试验结果及分析

2.1 计算域网格划分

由于轴承气膜厚度和气膜压力分布均关于轴承长度方向的中面对称,所以,在进行气膜压力和气膜厚度理论分析时,仅取半轴承长度进行计算. 由于气膜厚度相对于轴颈半径大小可忽略不计,所以,计算中仅取轴颈外表面进行网格划分,如图3所示. 对 z 向的半轴承长度进行 n_z 等分,对 θ 向的圆周长度进行 n_θ 等分,因此, z 向节点总数为 $n_z + 1$, θ 向的节点总数为 $n_\theta + 1$. 定义轴

承求解域为 $(0:n\theta; 0:nz)$,即横纵坐标求解域分别为 $0\sim n\theta$ 和 $0\sim nz$.

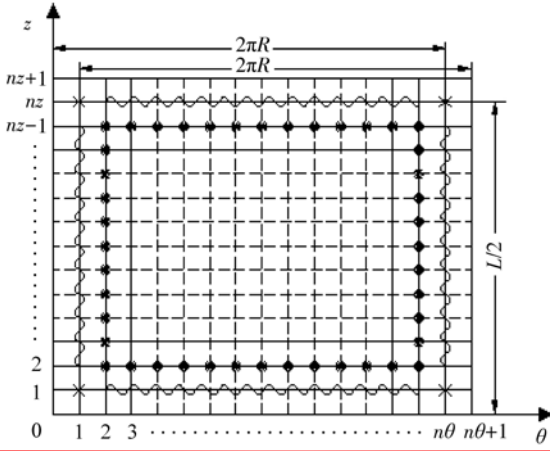


图3 计算域网格划分

Fig. 3 Grid plotting of the computational area

2.2 动力特性系数求解边界条件

2.2.1 定常求解边界条件

$$\bar{p}_0(0:n\theta, 0) = 1$$

$$\bar{p}_0(0, 0:nz) = \bar{p}_0(n\theta, 0:nz)$$

$$\frac{\partial \bar{p}_0(i, j)}{\partial \bar{z}} = 0 \quad (i=0:n\theta, j=nz)$$

$$\bar{p}_0(1, 0:nz) = \bar{p}_0(n\theta+1, 0:nz)$$

$$\bar{p}_0(0:n\theta, nz-1) = \bar{p}_0(0:n\theta, nz+1)$$

(29)

2.2.2 微分求解边界条件

循环边界条件

$$\left. \begin{aligned} \bar{p}_x(0, 0:nz) &= \bar{p}_x(n\theta, 0:nz), \\ \bar{p}_x(1, 0:nz) &= \bar{p}_x(n\theta+1, 0:nz) \\ \bar{p}'_x(0, 0:nz) &= \bar{p}'_x(n\theta, 0:nz), \\ \bar{p}'_x(1, 0:nz) &= \bar{p}'_x(n\theta+1, 0:nz) \\ \bar{p}_y(0, 0:nz) &= \bar{p}_y(n\theta, 0:nz), \\ \bar{p}_y(1, 0:nz) &= \bar{p}_y(n\theta+1, 0:nz) \\ \bar{p}'_y(0, 0:nz) &= \bar{p}'_y(n\theta, 0:nz), \\ \bar{p}'_y(1, 0:nz) &= \bar{p}'_y(n\theta+1, 0:nz) \end{aligned} \right\} \quad (30)$$

环境边界条件

$$\left. \begin{aligned} \bar{p}_x(0:n\theta, 0) &= \bar{p}'_x(0:n\theta, 0) = 0 \\ \bar{p}_y(0:n\theta, 0) &= \bar{p}'_y(0:n\theta, 0) = 0 \end{aligned} \right\} \quad (31)$$

对称边界条件

$$\left. \begin{aligned} \frac{\partial \bar{p}_x(i, j)}{\partial \bar{z}} &= \frac{\partial \bar{p}'_x(i, j)}{\partial \bar{z}} = 0 \\ &(i=0:n\theta, j=nz) \\ \bar{p}_x(0:n\theta, nz-1) &= \bar{p}_x(0:n\theta, nz+1) \\ \bar{p}'_x(0:n\theta, nz-1) &= \bar{p}'_x(0:n\theta, nz+1) \\ \bar{p}_y(0:n\theta, nz-1) &= \bar{p}_y(0:n\theta, nz+1) \\ \bar{p}'_y(0:n\theta, nz-1) &= \bar{p}'_y(0:n\theta, nz+1) \\ \frac{\partial \bar{p}_y(i, j)}{\partial \bar{z}} &= \frac{\partial \bar{p}'_y(i, j)}{\partial \bar{z}} = 0 \\ &(i=0:n\theta, j=nz) \end{aligned} \right\} \quad (32)$$

2.3 收敛判断

耦合求解数值计算的收敛判断公式如下:

$$\sqrt{\sum_{i=1}^{n\theta} \sum_{j=1}^{nz} \left(\frac{\Delta \bar{Q}_{i,j}}{\bar{Q}_{i,j}} \right)^2} \leq 10^{-6} \quad (33)$$

其中 Q 分别代表 p_0, p_x, p_y, p'_x 和 p'_y .

2.4 箔片轴承气膜刚度和阻尼系数求解框图

箔片轴承气膜刚度和阻尼系数程序求解框图如图4所示.

3 计算结果分析

定常耦合求解可参考文献[16].

气体润滑弹性波箔片轴承刚度和阻尼动力特性系数计算轴承模型结构参数及润滑气体参数如表1所示.

图5(a)和(b)分别为气体润滑弹性波箔片轴承在 4×10^4 r/min 转速时不同偏心率对应的动力特性系数.由图5可知,随着轴承偏心率的增加,气膜主刚度系数 $\bar{K}_{xx}$ 和 $\bar{K}_{yy}$ 不断增加,且主刚度系数 $\bar{K}_{yy}$ 的值与增加幅度都比主刚度系数 $\bar{K}_{xx}$ 大,这是由于静态载荷在垂直方向作用的缘故.

交叉刚度系数 $\bar{K}_{xy}$ 和 $\bar{K}_{yx}$ 随着偏心率增大呈现下降趋势,主阻尼系数 $\bar{D}_{xx}$ 和 $\bar{D}_{yy}$ 有上升趋势,特别是主阻尼系数 $\bar{D}_{xx}$ 逐渐趋近于零,主阻尼系数 $\bar{D}_{yy}$ 增大梯度较小,交叉阻尼系数 $\bar{D}_{xy}$ 和 $\bar{D}_{yx}$ 随着偏心率的增大则有下降趋势,规律与文献[17]基本一致.偏心率增大,气膜压力升高,轴承载荷能力提高,同时主阻尼增大,消耗更多的多余能量,有利于改善轴承运转稳定性.由图5可知,偏心率为0.6时出现主刚度系数小于交叉刚度系数的情况,表明在给定结构、转速和扰动频率等参数

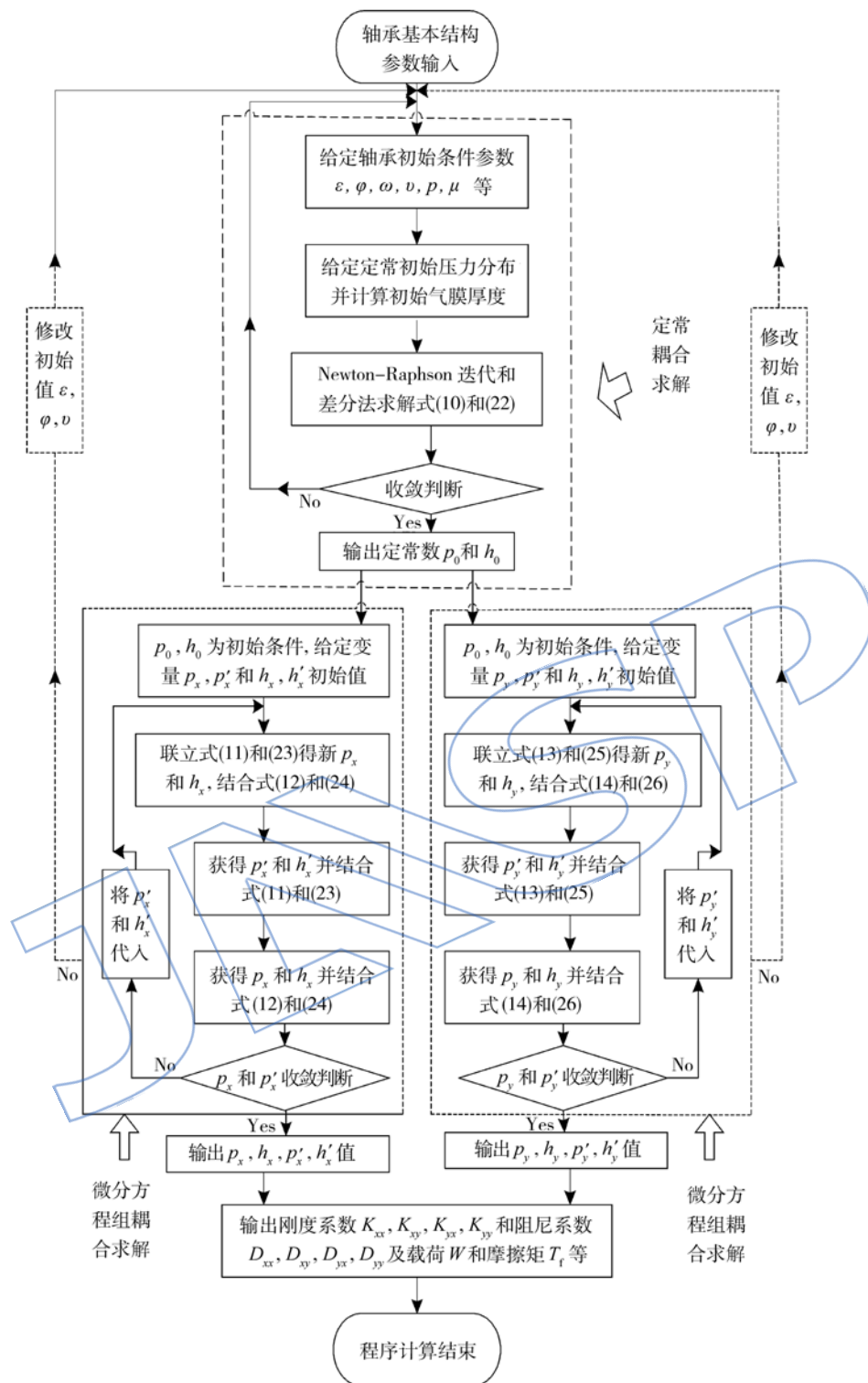


图4 气膜刚度和阻尼系数程序求解框图

Fig. 4 Flowchart solution of the air film stiffness and damping coefficient program

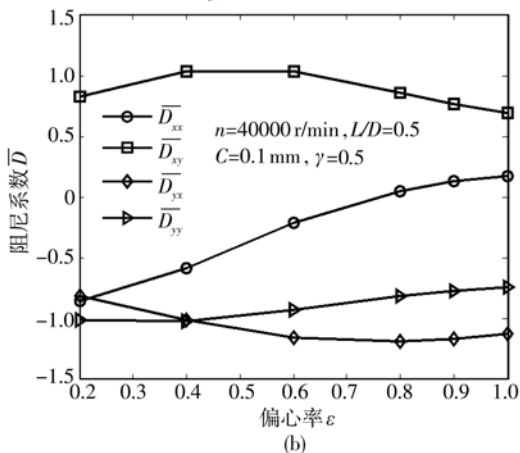
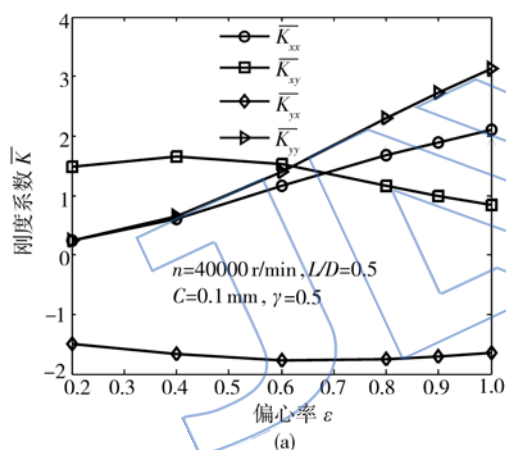
条件下,系统的稳定性较差.在阻尼系数曲线中,当偏心率较小时,交叉阻尼系数大于正向阻尼系数,但随着偏心率增大,主阻尼系数上升同时交叉阻尼系数下降,说明偏心率小时,轴承工作稳定性差,随着偏心率增大,稳定性提高.需要说明的是,轴承壳体及箔片几何结构参数、名义半径间隙、箔

片变形系数、运行参数如转速及扰动频率等因素都对动力特性系数参数有重要影响.

图6(a)和(b)分别为不同扰动频率时对应的气膜刚度和阻尼系数关系曲线,从图6可知,在低扰动频率时,特别是在半频扰动前,刚度和阻尼系数值变化较小,当扰动频率 γ 接近1时,正向刚度

表1 箔片轴承结构和润滑气体参数^[3]Table 1 Parameters of foil bearing and lubricant air^[3]

参数	数值
轴颈半径 $R/10^{-3}$ m	50
轴承长度 $L/10^{-3}$ m	75
名义半径间隙 $C/10^{-6}$ m	100
平箔片厚度 $t_p/10^{-6}$ m	76.2
波纹箔片厚度 $t_b/10^{-6}$ m	101.6
波纹箔片高度 $h_b/10^{-3}$ m	0.63
波纹箔片单元长度 $s/10^{-6}$ m	4.064
波纹箔片波纹长度 $2l/10^{-6}$ m	3.434
波纹箔片弹性模量 $E/10^9$ (N/m ²)	207
波纹箔片泊松比 ν	0.3
润滑气体黏度 $\mu/10^{-5}$ (Pa·s)	1.932
润滑气体密度 ρ /(kg/m ³)	1.1614
环境气体压力 $p_a/10^5$ Pa	1.01325

图5 4×10^4 r/min 转速时不同偏心率对应的气膜刚度和阻尼系数Fig. 5 Air film stiffness and damping coefficients with different ϵ at 4×10^4 r/min

系数和交叉刚度系数变化明显,这主要是受到给定轴承壳体及箔片结构尺寸等参数的影响,但交叉刚度系数和交叉阻尼系数值都小于主刚度系数和主阻尼系数值,这与文献[14]的结论和规律基本一致.总的来说,箔片轴承气膜刚度和阻尼系数与扰动频率密切相关.

表2为相应偏心率在不同转速下对应的气膜刚度和阻尼系数.表明随着转速的升高,轴承偏位角 φ 降低,同时,箔片轴承主刚度系数 $\overline{K}_{xx}$ 和 $\overline{K}_{yy}$ 都不断增大,承载能力增强,相对而言,阻尼系数值都较小,且随着转速的升高,阻尼系数值不断下降.

表3为 4×10^4 r/min 转速、扰动频率 $\gamma=0.5$ 时不同轴承长径比对应的气膜刚度和阻尼系数.从表中可以看出,随着轴承长度增加,主刚度系数和交叉刚度系数都不同程度增加,体现出承载能力的提高.表4为不同箔片柔度系数对应的气膜

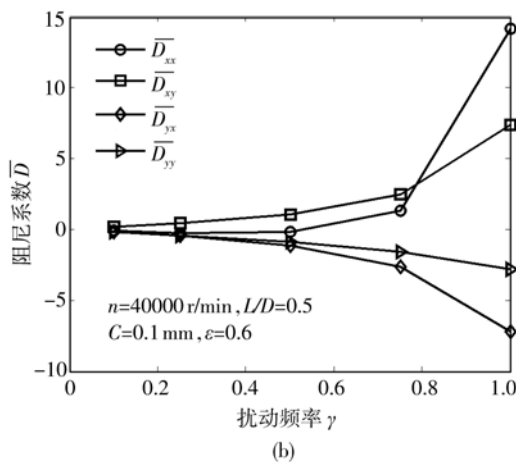
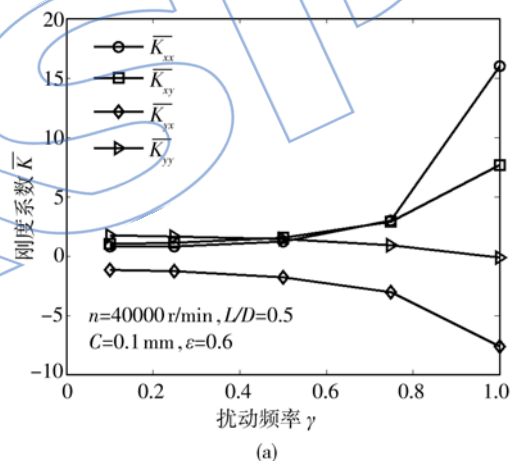
图6 4×10^4 r/min 转速时不同扰动频率对应的气膜刚度和阻尼系数Fig. 6 Air film stiffness and damping coefficients with different γ at 4×10^4 r/min

表2 不同偏心率对应的气膜刚度和阻尼系数

Table 2 Air film stiffness and damping coefficients with different eccentricities

ϵ	$n/10^4$ (r/min)	$\varphi/(^{\circ})$	无量纲刚度系数				无量纲阻尼系数			
			$\overline{K_{xx}}$	$\overline{K_{yy}}$	$\overline{K_{yx}}$	$\overline{K_{xy}}$	$\overline{D_{xx}}$	$\overline{D_{yy}}$	$\overline{D_{yx}}$	$\overline{D_{xy}}$
0.3	40	67.2	0.384	1.579	-1.57	0.391	-0.74	0.935	-0.90	-1.02
	60	59.2	0.867	1.667	-1.82	1.020	-0.44	0.830	-0.88	-0.59
	80	52.8	1.279	1.690	-1.95	1.461	-0.24	0.708	-0.80	-0.34
	100	47.5	1.632	1.694	-2.01	1.798	-0.11	0.600	-0.70	-0.18
0.5	40	58.9	0.868	1.627	-1.73	0.992	-0.39	1.070	-1.10	-0.98
	60	51.8	1.211	1.395	-1.75	1.629	-0.25	0.728	-0.87	-0.51
	80	46.6	1.528	1.307	-1.80	1.975	-0.14	0.551	-0.73	-0.30
	100	42.4	1.810	1.274	-1.82	2.214	-0.07	0.442	-0.63	-0.18
0.7	40	47.7	1.421	1.348	-1.77	1.846	-0.06	0.959	-1.18	-0.87
	60	42.7	1.567	1.010	-1.63	2.358	-0.11	0.555	-0.82	-0.44
	80	39.1	1.799	0.906	-1.61	2.600	-0.08	0.391	-0.65	-0.27
	100	36.2	2.023	0.875	-1.62	2.755	-0.05	0.303	-0.55	-0.18
0.9	40	36.3	1.894	0.982	-1.70	2.719	0.125	0.770	-1.16	-0.77
	60	33.7	1.907	0.667	-1.47	3.090	-0.05	0.403	-0.75	-0.39
	80	31.6	2.077	0.584	-1.43	3.242	-0.06	0.272	-0.58	-0.25
	100	29.8	2.258	0.565	-1.43	3.333	-0.04	0.206	-0.48	-0.17

表3 不同长径比对应的气膜刚度和阻尼系数

Table 3 Air film stiffness and damping coefficients with different L/D

参数	L/D	无量纲刚度系数				无量纲阻尼系数			
		$\overline{K_{xx}}$	$\overline{K_{yy}}$	$\overline{K_{yx}}$	$\overline{K_{xy}}$	$\overline{D_{xx}}$	$\overline{D_{yy}}$	$\overline{D_{yx}}$	$\overline{D_{xy}}$
$C=0.1\text{ mm}$	0.50	0.432	1.235	-1.054	0.859	-0.414	0.705	-0.603	-0.965
$\epsilon=0.6$	0.75	1.150	1.517	-1.774	1.404	-0.210	1.039	-1.161	-0.930
$n=4\times 10^4\text{ r/min}$	1.00	2.086	1.620	-2.272	1.859	0.327	1.181	-1.572	-0.747
$\gamma=0.5$	1.20	3.288	1.468	-2.509	2.187	1.290	1.045	1.773	-0.562

表4 箔片不同柔度系数对应的气膜刚度和阻尼系数

Table 4 Air film stiffness and damping coefficients with different foil deformation numbers α

参数	α	无量纲刚度系数				无量纲阻尼系数			
		$\overline{K_{xx}}$	$\overline{K_{yy}}$	$\overline{K_{yx}}$	$\overline{K_{xy}}$	$\overline{D_{xx}}$	$\overline{D_{yy}}$	$\overline{D_{yx}}$	$\overline{D_{xy}}$
$C=0.1\text{ mm}$	0.10	1.159	1.947	-2.192	2.293	-0.516	1.047	-1.596	-1.489
$\epsilon=0.6$	0.20	1.148	1.743	-2.024	1.867	-0.392	1.045	-1.405	-1.250
$n=4\times 10^4\text{ r/min}$	0.30	1.146	1.615	-1.894	1.602	-0.298	1.042	-1.273	-1.079
$L/D=1.20$	0.40	1.149	1.526	-1.787	1.423	-0.220	1.039	-1.173	-0.946
$\gamma=0.5$	0.50	1.156	1.461	-1.693	1.296	-0.152	1.035	-1.092	-0.836

刚度和阻尼系数,表明箔片柔度系数越大,气膜主刚度系数和交叉刚度系数越低,这是由于箔片刚度本身的降低所造成,进一步说明提高箔片刚度,可以有效提高轴承的载荷能力。

4 结束语

结合气体润滑 Reynolds 方程和气膜压力、气膜厚度及箔片变形等变量的 Taylor 展开式,通过摄动法推导出弹性波箔片轴承气膜动力特性计算微分方程组。进一步考虑气体可压缩性等因素,建立气膜厚度、箔片变形及气膜压力之间关系的数学模型方程组。运用 Newton-Raphson 迭代法和有限差分法耦合求解气膜动力特性计算微分方程组和气膜厚度数学模型方程组,得到弹性波箔片轴承气膜刚度和阻尼等动力特性系数随偏心率、转速、扰动频率及轴承长径比等参数变化的规律特性,为箔片轴承设计、优化和弹性波箔片轴承-转子系统理论及试验分析奠定基础。计算结果表明,随着偏心率和转速的增大,气膜主刚度均有不同程度的提高,同时,气膜刚度和阻尼的大小与扰动频率关系密切。

参考文献:

- [1] Ku C P P, Heshmat H. Compliant foil bearing structural stiffness analysis Part 1: theoretical model including strip and variable bump foil geometry[J]. ASME Journal of Tribology, 1992, 114(2): 394-400.
- [2] Peng Z C, Khonsari M M. Hydrodynamic analysis of compliant foil bearings with compressible gas flow[J]. ASME Journal of Tribology, 2004, 126(3): 542-546.
- [3] Peng Z C, Khonsari M M. A thermohydrodynamic analysis of foil journal bearings[J]. ASME J. Tribol., 2006, 128(3): 534-541.
- [4] 刘占生, 许怀锦. 波箔型动压气体径向轴承的应用与研究进展[J]. 轴承, 2008(1): 39-43.
LIU Zhansheng, XU Huaijin. Application and research progress of bump-type foil aerodynamic journal bearings[J]. Bearing, 2008(1): 39-43. (in Chinese)
- [5] Howard S A, DellaCorte C. Gas foil bearings for space propulsion nuclear electric power generation[R]. NASA TM-2006-214115, 2006.
- [6] Agrawal G L. Foil air/gas bearing technology—an overview[R]. ASME 97-GT-347, 1997.
- [7] DellaCorte C. A new foil air bearing test rig for use to 700 °C and 70 000 rpm[R]. NASA TM-1997-107405, 1997.
- [8] Stanford M K, DellaCorte C. Friction and wear characteristics of Cu-4Al foil bearing coating at 25 and 650 °C [R]. NASA TM-2004-212972, 2004.
- [9] Heshmat H, Hryniewicz P, Walton J F, II, et al. Low-friction wear-resistant coating for high-temperature foil bearings[R]. WTC 2005-63684, 2005.
- [10] Balić E E, Blanchet T A. Thrust-washer tribological evaluation of PS304 coating against Rene 41 [J]. Wear, 2005, 259(7-12): 876-881.
- [11] Howard S A. Misalignment in gas foil journal bearings: an experimental study[J]. ASME Journal of Engineering for Gas Turbines and Power, 2009, 131(3): 02250(7 pages).
- [12] Radil K, Zeszotek M. An experimental investigation into the temperature profile of a compliant foil air bearing[J]. NASA TM-2004-213100, 2004.
- [13] 虞烈. 弹性箔片轴承的气弹润滑解[J]. 西安交通大学学报, 2004, 38(3): 327-330.
YU Lie. Solution of elasto-aerodynamic lubrication for compliant foil bearings[J]. Journal of Xi'an Jiaotong University, 2004, 38(3): 327-330. (in Chinese)
- [14] 虞烈, 戚社苗, 耿海鹏. 弹性箔片空气动压轴承的完全气弹润滑解[J]. 中国科学 E 辑, 2005, 35(7): 746-760.
YU Lie, QI Shemiao, GENG Haipeng. Solution of whole elasto-aerodynamic lubrication for compliant foil bearings [J]. Science in China, Ser. E, 2005, 35(7): 746-760. (in Chinese)
- [15] Peng J P, Carpino M. Calculation of stiffness and damping coefficients for elastically supported gas foil bearings[J]. ASME Journal of Tribology, 1993, 115(2): 20-27.
- [16] 许怀锦, 刘占生, 张广辉, 等. 波箔片气体动压径向轴承气膜压力分布特点[J]. 轴承, 2008(6): 23-27.
XU Huaijin, LIU Zhansheng, ZHANG Guanghui, et al. Characteristics of film pressure distribution of bump foil gas bearings[J]. Bearing, 2008(6): 23-27. (in Chinese)
- [17] Heshmat H, Walowit J A, Pinkus O. Analysis of gas-lubricated foil journal bearings[J]. ASME Journal of Lubrication Tribology, 1983, 105(10): 647-655.