

滑动轴承最佳稳定性的优化方法*

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【摘要】 在求解短轴承非线性油膜力作用下轴心涡动轨迹的基础上, 给出了轴颈瞬时涡动能量的表达式, 建立了以涡动能量的时间积分为最小的具有最佳稳定性的滑动轴承参数优化模型, 并采用最优控制理论对这一模型进行了求解, 结合算例说明轴承最佳参数的确定方法。

主题词: 滑动轴承 稳定性 最佳控制
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多年来大型汽轮发电机组、空压机等许多旋转机械上屡次发生滑动轴承油膜振荡事故, 影响机组正常运行造成机组损坏。要想从根本上消除油膜振荡, 必须从轴承的设计阶段入手, 设计出具有足够稳定性能的滑动轴承。目前在设计过程, 设计参数的选取很大程度上仍要靠经验资料, 这样设计出的轴承在承载力及润滑性能上一般还能满足要求, 但是否满足稳定性的要求往往没有把握, 因此在设计过程中迫切需要以稳定性为目标函数, 对轴承参数进行优化设计。关于滑动轴承优化设计, 国内外曾作过许多研究, 文献[1-5]表现了其中具有代表性的研究工作, 但这些文章中都没有把轴承稳定性作为一个优化目标来考虑。文献[6]以机组的稳定裕量为优化目标, 但研究是在线性条件下进行的。本文在非线性条件下轴承油膜涡动轨迹计算的基础上, 以轴承稳定性为优化目标, 提出一种采用最优控制理论进行轴承参数的优化设计方法, 本文给出一个算例, 说明轴承最佳参数的确定方法。

1 模型建立、问题求解

图 1 为滑动轴承结构示意图, 轴颈在径向和周向两方向上的运动微分方程为

$$\begin{aligned} \ddot{\epsilon} - \epsilon \ddot{\Phi} &= -\alpha \bar{F}_\epsilon + \beta \cos \Phi \\ \ddot{\Phi} + 2\epsilon \dot{\Phi} &= \alpha \bar{F}_\Phi - \beta \sin \Phi \end{aligned} \quad (1)$$

式中: $\alpha = \mu u L / (M C \psi \omega^2)$, $\beta = g / C \omega^2$, μ 为润滑的粘度, u 为轴颈表面速度 $u = R \omega$, ω 为轴颈半径, M 为转子质量的一半, C 为轴承间隙, ψ 为轴承间隙比 $\psi = C / R$, ϵ 和 Φ 分别为轴承的偏心率和偏位角。 $\bar{F}_\epsilon$ 和 $\bar{F}_\Phi$ 分别为两方向油膜力的无量纲形式, 引入代换: $X_1 = \epsilon$, $X_2 = \Phi$, $X_3 = \dot{\epsilon}$, $X_4 = \dot{\Phi}$ 则(1)

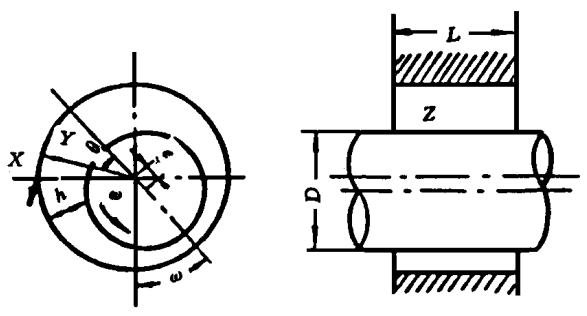


图 1 轴承结构及旋转坐标系

式降阶为:

$$\begin{aligned} \dot{X}_1 &= X_3, & \dot{X}_2 &= X_4 \\ \dot{X}_3 &= f_1(X_1, X_2, X_3, X_4, \alpha, \beta), & \dot{X}_4 &= f_2(X_1, X_2, X_3, X_4, \alpha, \beta) \end{aligned} \quad (2)$$

代入适当边界条件, 求解上述微分方程组可获得任一时刻的轴心位置 (ϵ, Φ) 及涡动速度 $(\dot{\epsilon}, \dot{\Phi})$ 。轴颈瞬时涡动能量可表示如下(引入无量纲 $\bar{t} = \omega t$):

$$T = (1/2)M(\dot{\epsilon}^2 + \dot{\epsilon}^2\Phi^2) = (1/2)M\omega^2 C^2(\dot{\epsilon}^2 + \dot{\epsilon}^2\Phi^2) \quad (3)$$

轴颈所具有的涡动能量越大, 说明轴承工作稳定性越差, 为了获得最佳稳定性, 建立如下轴承参数优化模型:

$$J = \int_0^{\bar{t}} (\dot{\epsilon}^2 + \dot{\epsilon}^2\Phi^2) d\bar{t} = \int_0^{\bar{t}} (X_1^2 X_4^2 + X_3^2) d\bar{t} \quad (4)$$

$$\text{约束条件} \quad st \begin{cases} \dot{X}_1 = X_3 \\ \dot{X}_2 = X_4 \\ \dot{X}_3 = f_1(X_1, X_2, X_3, X_4, \alpha, \beta) \\ \dot{X}_4 = f_2(X_1, X_2, X_3, X_4, \alpha, \beta) \end{cases} \quad (5)$$

其中: st 为约束条件, $\bar{t}$ 为无量纲时间 $\bar{t} = \omega t$, $[\bar{t}_0, \bar{t}_f]$ 为人为设定的积分区间。

式(4)和(5)所表示的优化问题属有约束最优控制问题, $X = (X_1, X_2, X_3, X_4)^T$ 为状态向量, $U = (\alpha, \beta)^T$ 为控制向量, 它与轴承参数有关, 需找到最佳的 U^* 以便式(4)之值最小, 令:

$$L(X, U, \bar{t}) = X_1^2 X_4^2 + X_3^2 \quad (6)$$

则(4)、(5)两式化为:

$$\min J(U) = \int_0^{\bar{t}} L(X, U, \bar{t}) d\bar{t}, \quad \dot{X} = f(X, U, \bar{t}) \quad (7)$$

并确定边界条件 $X(\bar{t}_0) = 0$, 引入伴随向量 $\lambda(\bar{t}) = [\lambda_1(\bar{t}), \lambda_2(\bar{t}), \lambda_3(\bar{t}), \lambda_4(\bar{t})]^T$ 。

采用 Lagrange 乘子法构造增广泛函 J' , 将有约束最优控制问题无约束化:

$$J' = \int_0^{\bar{t}} [L(X, U, \bar{t}) + \lambda^T(\bar{t})f(X, U, \bar{t}) - \lambda^T(\bar{t})\dot{X}(\bar{t})] d\bar{t} \quad (8)$$

令: $H(X, U, \lambda, \bar{t}) = L(X, U, \bar{t}) + \lambda^T(\bar{t})f(X, U, \bar{t})$ 。这里 H 是标量函数, 称为 Hamilton 函数, 则式(8)为:

$$J' = \int_0^{\bar{t}} [H(X, U, \lambda, \bar{t}) + \lambda^T(\bar{t})\dot{X}(\bar{t}) - \lambda^T(\bar{t})\dot{X}(\bar{t})] d\bar{t} \quad (9)$$

对上述泛函分别求 X 和 U 的变分, 并令其为零:

$$\delta J' = \int_0^{\bar{t}} \left[\left(\frac{\partial H}{\partial U} \right)^T \delta U + \left(\frac{\partial H}{\partial X} \right)^T \delta X + \lambda^T \delta X \right] d\bar{t} - \lambda^T \delta X \Big|_0^{\bar{t}_f} \quad (10)$$

由于 δX 和 δU 是相互独立的任意向量, 故有:

$$\partial H / \partial U = 0, \quad \partial H / \partial X + \dot{\lambda} = 0 \quad (11)$$

将(11)式与(5)式联立得如下微分方程组:

$$\begin{aligned} \dot{X}_1 &= X_3, & \dot{X}_2 &= X_4 \\ \dot{X}_3 &= f_1(X_1, X_2, X_3, X_4, \alpha, \beta), & \dot{X}_4 &= f_2(X_1, X_2, X_3, X_4, \alpha, \beta) \\ \lambda_3(\partial f_1 / \partial \alpha) + \lambda_4(\partial f_2 / \partial \alpha) &= 0, & \lambda_3(\partial f_1 / \partial \beta) + \lambda_4(\partial f_2 / \partial \beta) &= 0 \\ \dot{\lambda}_1 &= -[2X_1 X_4^2 + \lambda_3(\partial f_1 / \partial X_1) + \lambda_4(\partial f_2 / \partial X_1)] \end{aligned}$$

$$\begin{aligned}
 \dot{\lambda}_2 &= - [\lambda_3(\partial f_1/\partial X_2) + \lambda_4(\partial f_2/\partial X_2)] \\
 \dot{\lambda}_3 &= - [2X_3 + \lambda_1 + \lambda_3(\partial f_1/\partial X_3) + \lambda_4(\partial f_2/\partial X_3)] \\
 \dot{\lambda}_4 &= - [2X_4 + \lambda_2 + \lambda_3(\partial f_1/\partial X_4) + \lambda_4(\partial f_2/\partial X_4)] \quad (12)
 \end{aligned}$$

式中: $\partial f_1/\partial \alpha$, $\partial f_2/\partial \alpha$, $\partial f_1/\partial \beta$, $\partial f_2/\partial \beta$, $\partial f_1/\partial X_i$, $\partial f_2/\partial X_i$, ($i = 1, 2, 3, 4$), 12 个偏导数可根据式(2)计算, 式(12)中有 10 个方程, 正好有 10 个未知量($X_1, X_2, X_3, X_4, \lambda_1, \lambda_2, \lambda_3, \lambda_4, \alpha, \beta$), 故方程可解, 又因其中含有 8 个对 $\bar{t}$ 求导的微分方程, 故需 8 个边界条件, 设 X 和 λ 两向量的初值均为零, 即: $X_i(\bar{t}_0) = 0, \lambda_i(\bar{t}_0) = 0, (i = 1, 2, 3, 4)$ 。只要反复迭代求解方程组(12), 即可获得最佳的 α^* 和 β^* 值, 这里选择变尺度法作为搜索方法, 它既有共轭梯度法程序简单、容易实现及收敛速度快等优点, 又克服了牛顿法计算工作量大的缺点, 在现代控制理论中广为应用, 由于过程冗长, 在此不便细述。

2 算 例

在前面理论分析的基础上编制相应计算程序, 并对某圆轴承(其结构参数: 长度为 26 mm, 直径为 38 mm, 载荷为 490 N, 半径间隙为 0.028 mm, 油粘度(30°C)为 0.025 Pa·s, 油密度为 987 kg/m³)进行了计算, 取转速为 3200 r/min, 计算得当前参数下的 α 和 β 值: $\alpha = 12.12$, $\beta = 3.117$ 。以这一组 α 和 β 值作为初值, 优化求解一组最佳的控制参数 α^* 和 β^* , 但值得注意的是在式(1)中 F_ϵ 和 F_ϕ 中含有当量雷诺数及长径比两个与轴承参数有关的量, 因此需根据 α^* 和 β^* 重新计算当量雷诺数和长径比, 再一次求解控制参数, 直到前后两次的结果基本相等为止。代入程序反复迭代计算出一组最佳控制参数: $\alpha^* = 9.763$, $\beta^* = 3.025$ 。半径间隙可确定为: $C = g/\beta^* \omega^2 = 0.02885$ mm。在 α 的表达式中, 若保持 M 和 μ 不变, 即可得轴承半径与长度应满足如下关系: $R^3 L = 1.5712 \times 10^{-7} (m^4)$ 。

当然除结构尺寸外, 其它参数如 M 和 μ 也可作为自变量进行优化。

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tolerant design are combined and a structure life evaluation model is developed. The formulas to determine the service life and the inspection periods of a structure with the high reliability are presented. Then the model is applied to evaluating the inspection periods for aircraft powerplant dynamic components, and an example is given.

Key Words Fatigue Fracture Lifetimes Model

DETERMINATION OF AXIAL PRELOADS OF ROTORS WITH CURVIC COUPLINGS

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ABSTRACT

Rotors with curvic couplings, which are separately prestressed from the front part and the rear part, are often applied to modern turboprop and turboshaft engines. It is very important but difficult to determine the values of preloads for a rotor. The prerequisite of the axial relaxed/pressed forces caused by variety of rotor loads in operation is available, the criteria to determine the axial preloads of rotors are proposed in this paper, which takes into account synthetically the factors, such as tightness margin, strength margin, assembling relationship and etc. Hence the optimization model of the axial preloads is established, and the corresponding solution is presented. The computer program is developed according to the presented criteria and method. Finally, an example is given.

Key Words Rotors Axial preloads Optimization

AN OPTIMUM METHOD OF JOURNAL BEARING STABILITY

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ABSTRACT

The expression of journal instantaneous whirl energy is formulated, based on solution for shaft center whirl orbit of a short bearing under the nonlinear oil film force. An optimization model of journal bearing parameters with optimum stability is established with the aim of minimizing time integration of whirl energy. The model has been solved with the optimal control method. In the last, a numerical example illustrates the determination of the optimal parameters of a given bearing.

Key Words Plain bearings Stability Optimal control

A NEW APPROACH TO COMBINED FATIGUE EXPERIMENT OF TURBINE DISK MORTISES AT ELEVATED TEMPERATURE

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ABSTRACT

Crack and fracture fault emerged in the mortises of a given aero-engine second-stage turbine disk for a long time and seriously endangered the aircraft's regular flight. The fractography analysis indicated that the faults were caused by combined fatigue at elevated temperature. So we were engaged in this experiment to study and research