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挤压油膜阻尼器性能的油气两相流分析与试验*

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【摘要】 基于对油膜气穴试验结果的考察和理论分析，本文给出了预计挤压油膜阻尼器力学性能的油气两相流分析方法。润滑膜压力分布的计算结果与测试结果有良好的一致性。本文还提供了阻尼器的油气两相流性能数据，且与纯 π 膜的计算结果作了比较，这些数据可供工程上计算分析采用。所提出的分析方法，也可拓宽用于其它带类似油气两相流润滑膜轴承。

关键词： 挤压油膜阻尼器　油气两相流　分析　试验

1 前　言

在挤压油膜阻尼器的传统分析方法中，滑油视为不可压缩流，且常采用 π 油膜边界条件来简化油膜的气穴现象。然而，油膜气穴试验的考察和分析表明[1~7]，不仅空气气穴存在于阻尼器的低压区，且空气小气泡气穴能在高压区经受压缩而持续存在，这表明阻尼器承载润滑膜具有油气两相可压流特性；此外，蒸汽气穴也会出现在润滑膜的负压峰值附近[4~6]。因此，传统分析方法是有待修正的。本文将在文献[1]的研究基础上，给出挤压油膜阻尼器力学性能的油气两相流分析方法。其中，除考虑油气两相可压流润滑膜粘度和密度的流变效应，并计及负压的影响。进行压力分布测试以进一步验证理论分析的合理性。用油气两相流分析方法所预计的性能数据与纯油 π 膜的计算结果进行了比较。这些数据可供工程上计算分析时采用。此外，所提供的分析方法，也可拓宽用于其它带有类似油气两相流润滑膜轴承。

2 理论分析

进动挤压油膜阻尼器轴承的几何特性及坐标系，如图 1 所示，轴承采用中央供油沟槽导油润滑。

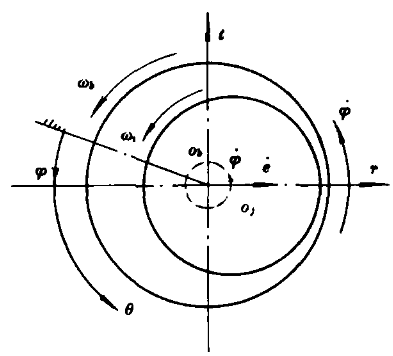

图 1　几何特性及坐标系

对于 $\omega_j=\omega_b=0$ 的稳态圆进动等温可压流挤压膜，无量纲 Reynolds 方程为[8]

$$\frac{\partial}{\partial\theta}\left(\frac{\bar{\rho}_e\bar{h}^3}{\bar{\eta}_e}\frac{\partial\bar{p}}{\partial\theta}\right)+\left(\frac{R_j}{L}\right)^2\frac{\partial}{\partial\bar{Z}}\left(\frac{\bar{\rho}_e\bar{h}^3}{\bar{\eta}_e}\frac{\partial\bar{p}}{\partial\bar{Z}}\right)=-12S_0\frac{\partial(\bar{\rho}_e\bar{h})}{\partial\theta}\qquad(1)$$

这里 $S_o=\eta_l\Phi/(p_o\psi^2)$，$\bar{p}=p/p_0$，$\bar{Z}=Z/L$，$\bar{h}=h/c=1+\varepsilon\cos\theta$。其中 η_l 为连续相滑油粘度，p_o 为大气压力，$\psi=c/R_j$，R_j、L 和 c 分别为阻尼器轴承半径、

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半宽和半径间隙。(1)式中无量纲等效粘度和等效密度分别为[1]：$\bar{\eta}_e=\eta_e/\eta_l=1+1.5\beta[1+1.75(R/h)^3]$ 和 $\bar{\rho}=\rho_e/\rho_l=1-\beta$。式中 β 为气泡体积浓度，R 为气泡壁半径，ρ_l 为连续相滑油密度。气泡体积浓度为[1,9] $\beta=\beta_o(\bar{R})^3$，$\bar{R}=R/R_o$。略去惯性项和粘性项后，无量纲气泡壁半径 $\bar{R}$ 为[1,9]

$$\bar{p}\bar{R}^{3n}+\frac{2\sigma}{p_oR_o}\bar{R}^{3n-1}-\left(1+\frac{2\sigma}{p_oR_o}\right)=0 \tag{2}$$

式中 σ 为气泡表面张力，R_o 和 β_o 为相应于 p_o 下的变量。

现取阻尼器油槽边界压力为供油压力，外端边界压力为大气压力。在轻载（低润滑膜压力）下，应将饱和蒸汽压作为油膜所能经受的最大负压并为数值模拟负压分布[9]。在解算中，对方程(1)采用有限差分超松弛迭代法，方程(2)采用牛顿下山法，由此联解式(1)、(2)，可获得整个压力场的压力分布。进而可以计算供油槽一侧的无量纲润滑膜径向的和切向的反力

$$\begin{Bmatrix}\bar{F}_r\\ \bar{F}_t\end{Bmatrix}=\int_0^1\int_0^{2\pi}\bar{p}(\theta,\bar{Z})\begin{Bmatrix}\cos\theta\\ \sin\theta\end{Bmatrix}d\theta d\bar{Z} \tag{3}$$

其中定义 $\bar{F}_r=F_r/(R_jLp_o)$，$\bar{F}_t=F_t/(R_jLp_o)$，$\bar{k}_o=k_o(\psi^3/\eta_l\Phi L)$ 及 $\bar{d}_o=d_o(\psi^3/\eta_lL)$，所求无量纲油膜等效刚度和阻尼分别为 $\bar{k}_o=\bar{F}_r(p_o\psi^2/\eta_l\Phi\varepsilon)$，$\bar{d}_o=\bar{F}_t(p_o\psi^2/\eta_l\Phi\varepsilon)$。

3　压力分布测量与理论比较

试验装置的主轴旋转时，通过偏心套使阻尼器轴颈在轴承中产生圆进动挤压运动。试验的偏心率可通过更换偏心套来改变。供油压力可调。测量系统原理示于图 2 中[9]。

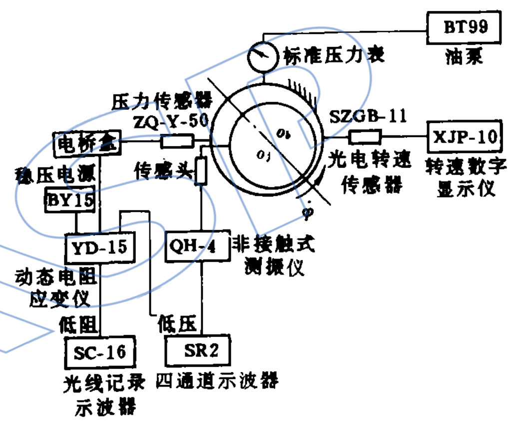

图 2　测量系统原理框图

由于圆进动和压力场随主轴作同步旋转的特点，只需在挤压油膜轴承上嵌入一个压力传感器并经光线示波器，便可持续测量出周向压力分布。

试验中所用原始数据为：阻尼器轴承直径 D 为 0.18m；一侧半宽 L 为 0.016m；半径间隙比 $\psi=c/R_j=0.00185$；偏心率 ε 为

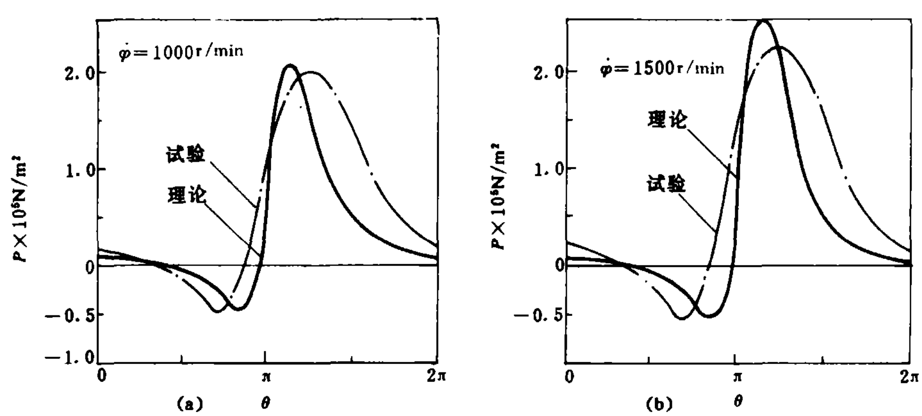

图 3　试验与理论压力分布比较

0.7；润滑油选用 10\# 机械油。理论分析中，取 10\# 机械油在 30℃下，粘度 $\eta_l=1.8\times10^{-2}\mathrm{N\cdot s/m^2}$，密度 $\rho_l=0.9\times10^3\mathrm{kg/m^3}$；又取其饱和蒸汽压约为 $-0.53\times10^5\mathrm{N/m^2}$，表面张力约为 $3\times10^{-2}\mathrm{N/m}$；理论分析中，在大气压 p_0 下气泡体积浓度 β_0 及气泡壁半径 R_0 分别取为 0.10 和 3μm[1]；气泡内状态变化按等温($n=1$)。试验在不同运转条件组合下进行，图 3 给出所测得的两条压力分布曲线，并与油气两相流分析方法的计算结果进行了比较。由图 3 可见，在峰值压力和总的变化趋势上，理论与试验结果具有良好的一致性。尤其是所测量得的负压极值，大都在 $-(0.5\sim0.6)\times10^5\mathrm{N/m^2}$，因含气泡而略高于纯油饱和蒸汽压。试验进一步说明了油气两流分析方法的合理性。

4　计算结果

图 4 是油气两相流分析方法和纯油 π 膜边条预计阻尼器周向压力分布的比较。可见，油气两相流分析中，由粘度和密度的流变效应使润滑膜具有较高的正压力；同时也计及了负压效应，这对于轻载工况是重要的。图 5a 和 b 分别给出了油气两相流分析方法预计的进动挤压油膜阻尼器无量纲油膜刚度和阻尼，并与文献[10]的纯油 π 膜边条预计结果进行了对比。

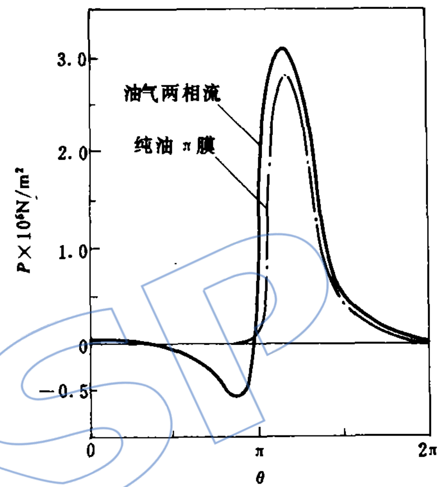

图 4　理论计算的周向压力分布比较

油气两相流分析方法中，粘度和密度的流变效应以及负压效应都是增大阻尼的因素，因而阻尼器具有较高的阻尼；粘度和密度的流变效应是使刚度增大的因素，而负压效应是使刚度减小的因素，所以，当 L/D 和 ε 值较小时，负压效应可观，使具有较小的刚度，而当 L/D 和 ε 较大时，油气两相流方法所预计的刚度相对略大。

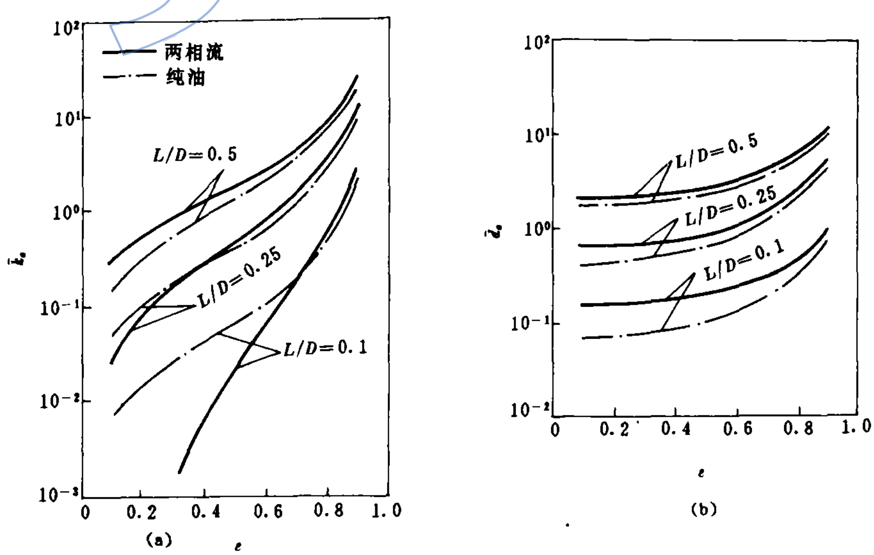

图 5　油膜刚度和油膜阻尼(a. 刚度　b. 阻尼)

表1 无量纲油膜刚度和阻尼

L/D	0.10				0.25				0.50			
ε	纯油 π 膜		油气两相流		纯油 π 膜		油气两相流		纯油 π 膜		油气两相流	
	k_e	d_e	k_e	d_e	k_e	d_e	k_e	d_e	k_e	d_e	k_e	d_e
0.10	0.00806	0.0635	−0.00149	0.0755	0.0476	0.388	0.0256	0.653	0.158	1.45	0.313	1.71
0.20	0.0171	0.0664	−0.000447	0.154	0.101	0.406	0.0848	0.680	0.332	1.50	0.547	1.80
0.30	0.0286	0.0720	0.00247	0.178	0.167	0.437	0.171	0.689	0.544	1.60	0.783	1.89
0.40	0.0446	0.0810	0.00488	0.201	0.258	0.489	0.281	0.713	0.828	1.76	1.090	2.02
0.50	0.0697	0.0973	0.0188	0.239	0.398	0.573	0.472	0.770	1.240	2.01	1.580	2.28
0.60	0.114	0.121	0.0843	0.245	0.640	0.714	0.764	0.871	1.930	2.42	2.400	2.69
0.70	0.209	0.169	0.206	0.252	1.130	0.974	1.360	1.130	3.210	3.13	3.990	3.44
0.80	0.472	0.282	0.570	0.347	2.420	1.560	2.940	1.750	6.280	4.59	7.530	4.50
0.90	1.85	0.721	2.370	0.833	8.370	3.550	10.500	4.010	18.200	8.25	21.600	9.52

油膜刚度和阻尼的变化，将影响阻尼器所在转子系统的振动特性。尤其是，两相流的阻尼的较大提高，将有利于降低转子系统不平衡响应的振幅，现实条件下刚度的减小，又可缩小非线性振动不稳定区。为工程上计算应用方便，表1中列出了用油气两相流分析方法所提供的等效刚度和阻尼系数，并同时给出了纯油 π 膜的有关数据[10]，以作比较。

5 结 论

(1)给出的进动挤压油膜阻尼器的油气两相分析方法，考虑了润滑膜粘度和密度的流变效应，并计及了负压影响。试验结果进一步验证了分析方法的合理性。(2)按油气两相流分析方法对比传统方法所预计的结果来说，阻尼器具有较高的阻尼；且当：L/D 和 ε 值较小时刚度较小，而它们较大时刚度略大。(3)本文为工程上计算分析提供了有关润滑膜刚度和阻尼系数的性能数据。同时，所提出的分析方法，也可拓宽用于其它的具有类似油气两相流润滑膜轴承。

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3-D FINITE ELEMENT CYCLIC SYMMETRIC AND CONTACT STRESS ANALYSIS FOR A COMPLETE GEAR TRAIN

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ABSTRACT A complete gear train of a reduction gearbox is the object of finite element stress analysis. One of the basic segments of the complete gear train is taken as the computational model in the light of the cyclic symmetry of the gear train, meanwhile the contact transmission forces between the corresponding meshed teeth are considered in the analysis of the model. For simplicity the corresponding meshed lines are used instead of the actual contact surfaces. Both torque and centrifugal loads are involved in the analysis. The stresses in all the parts of a complete gear train can be determined by one analysis. The computed results show that the contact force on a meshed tooth is correlative not only to the length of the meshed line, but also to its position, and, moreover, the neglect of the stress resulted from centrifugal load is inappropriate to a high speed gear train.

OPTIMUM BALANCING OF FLEXIBLE ROTOR

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ABSTRACT A new criterion for optimum balancing of a flexible rotor is introduced, which is the minimum rotor strain energy criterion with constrains. This criterion meets the requirements for integral design of structure strength and performance of a gas turbine aeroengine. In this criterion the balance correction masses (quantity and orientation) are the optimum design variable, the objective function of optimization is the total strain energy of the rotor and its supports. The response amplitudes at several cross-section along the rotor are taken as its constraints. The Powell's method and the net random-ray method are chosen as optimum methods. A program is written with FORTRAN 77 for the whole optimization procedure. The program has been testified by a numerical example in comparison with its experiments.

STUDY ON PERFORMANCE OF SQUEEZE FILM DAMPER WITH OIL-AIR TWO-PHASE FLUID

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ABSTRACT This paper presents an analytical method for lubrication film performance of squeeze film damper with oil-air two-phase fluid on the basis of the testing results and numerical analysis of oil film cavitation. The calculational results for the pressure distribution of lubrication oil film are in good agreement with the experimental results. The damper performance data obtained from two-phase fluid analysis are also compared with the results calculated by pure oil film damper. It is

proven that the data can be used for the engineering analyses. The analytical method can be extended for the other bearings with similar lubrication film of oil-air two-phase fluid.

A FEM OF THERMOELASTO—PLASTIC ANALYSIS WITH DUE REGARD FOR LOADING PATH

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ABSTRACT Until now a lot of investigations have been done on the thermoelasto-plastic analysis. Its genaral approach is to add up all the external forces on the structure to form a so-called load vector and to divide the element states into elastic, plastic and elastplastic elements, but the variation of the element states during the loading procedure and the loading path are always ingnored. In fact these two factors do affect the reliability of the analytic results. In order to improve the accuracy of the thermoelasto-plastic analysis, two algorithms are proposed which imitate the loading path and take the shifting of the element states into account. The external forces are expressed as the functions of time respectivley. Thereby the analysis procedure corresponds to the loading procedure. The element states are classified into six kinds according to the different stress-strain states. Thus a program system is developed. Two examples are given to show the difference between the analysis results under different loading path.

A NEW MECHANICAL MODEL AND ALGORITHM FOR FINITE ELEMENT ANALYSIS OF CONTACT PROBLEMS

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ABSTRACT A new mechanical model for FEM analysis of contact problems is presented, in which a nodal point on the surface of contact body comes into contact with an arbitrary point on the surface of target body instead of the nodal point pair model. It is convenient to construct FE nets. The boundary conditions such as stick、slid with friction and open contact were given in terms of this model. The contact boundary conditions as Lagrangian multipliers are substituted into the functional according to the principle of modified potential energy and therefrom the symmetrical nonlinear governing equations are deduced for the static and dynamic contact problems with material and/or geometric nonlinearities. Due to the symmetrization of equations, the general effective solution methods of nonlinear equations can also be used to the contact governing equations. In the paper the judgement criteria of contact conditions after each iteration are presented, which can decide whether and how the contact conditions of each contact nodal point to be modified.