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带挤压油膜阻尼器刚性转子的双稳态特性

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摘　　要

本文通过理论计算和实验，研究了带定心弹性支承的挤压油膜阻尼器刚性转子的双稳态特性。采用阻尼器轴颈偏心率、传递率和落后相位角的变化表示这种特性。

一、引　　言

挤压油膜阻尼器因结构简单、重量轻、减振效果好，已在航空发动机和其它旋转机器中广泛应用。这些阻尼器是非线性的。油膜刚度随挤压作用而增大，挤压作用越大时非线性程度越高，很容易产生双稳态现象。偏心率、传递率等发生突然跳变，于工作不利。此现象常发生于阻尼器中有气涡的半油膜情况。增大供油压力使之成为全油膜就不致发生[3]。但增大供油压力会带来其他问题，并不可取。

国内、外常提到双稳态现象[1～4]，但系统研究的文章少见，且有的说法不一，未详细说明出现双稳态现象时的落后角如何变化。本文从理论计算和实验两方面，对带定心弹性支承的刚性转子的双稳态现象作了分析，进一步了解这种特性，以便控制其发生。

二、模型及理论分析

采用的模型如图1(a)所示，左支点带低刚性定心弹性支承的挤压油膜阻尼器，右支点为刚性铰支。转子刚性高，过一阶临界时，转子基本不弯曲，变形主要发生在左支承。现代航空发动机转子不少采用此种类型结构。图1(b)为简化的计算模型。根据工作时，转子各部分惯性力对右支点的力矩总和与简化模型集中质量的惯性力对该处力矩相等求出集中质量。计算中略去轴颈偏斜的影响。

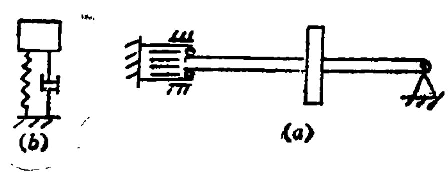

图1　刚性转子分析模型
(a)原转子模型　(b)计算模型

计算模型$m=7.5(kg)$，弹支刚度k取了几种，给出的图中$k=2480(kg/cm)$，阻尼器长、径比$L/D \doteq 0.25$，阻尼器间隙比c/R取了三种：0.002、0.0375和0.05。供油压低，两端不封油。滑油动粘度取为常数$\mu=0.0213Pa\cdot s$。

由润滑理论，按短轴承半油膜近似解得出作用于阻尼器轴颈径向和周向的油膜力分别为

$$F_r=\frac{\mu\omega RL^3}{2c^2}\left[\frac{4\varepsilon^2}{(1-\varepsilon^2)^2}\right]=\frac{\mu\omega RL^3}{2c^2}f_r \quad (1)$$

本文于1987年10月收到。

$$F_t=\frac{\mu\omega RL^3}{2c^2}\left[\frac{\pi\varepsilon}{(1-\varepsilon^2)^{3/2}}\right]=\frac{\mu\omega RL^3}{2c^2}f_t \tag{2}$$

式中$f_r=4\varepsilon/(1-\varepsilon^2)^2$，$f_t=\pi\varepsilon/(1-\varepsilon^2)^{3/2}$。等效油膜刚度和阻尼系数分别为

$$k_0=\frac{\mu\omega RL^3}{c^3}\left[\frac{2\varepsilon}{(1-\varepsilon^2)^2}\right],\quad d_0=\frac{\mu RL^3}{c^3}\left[\frac{\pi}{2(1-\varepsilon^2)^{3/2}}\right] \tag{3}$$

忽略其它阻尼。等效系统中各作用力如图 2 所示。O_b、O_j、O_m分别为轴承、轴颈和质量中心。ρ为质量偏心距，$P_\rho=m\omega^2\rho$为质量的不平衡力。$P_j=m\omega^2e$为质量惯性力。e为轴颈偏心，β为落后相位角。由径向力平衡和周向力平衡得

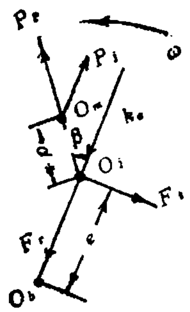

图 2　等效系统中作用的力

$$F_r+ke-m\omega^2e=P_\rho\cos\beta,\quad F_t=P_\rho\sin\beta \tag{4}$$

由此求出

$$P_\rho=\sqrt{(F_r+ke-m\omega^2e)^2+F_t^2} \tag{5}$$

无量纲化，取偏心率$\varepsilon=e/c$，不平衡偏心比$U=\rho/c$，转速比$\delta=\omega/\omega_c$，临界角速度

$\omega_c=\sqrt{k/m}$，轴承参数$B=\mu RL^3/(2m\omega_e c^3)$，（5）式化为

$$U\delta^2=\sqrt{(B\delta f_r+\varepsilon-\varepsilon\delta^2)^2+(Bf_t\delta)^2} \tag{6}$$

取

$$E=U\delta^2-\sqrt{(B\delta f_r+\varepsilon-\varepsilon\delta^2)^2+(Bf_t\delta)^2} \tag{7}$$

传递率为外传力与不平衡力之比，为

$$T=\frac{F_t}{m\omega^2\rho}=\frac{\sqrt{(F_r+ke)^2+F_t^2}}{m\omega^2\rho}=\frac{1}{U\delta^2}\sqrt{(B\delta f_r+\varepsilon)^2+(B\delta f_t)^2} \tag{8}$$

$$\beta=tg^{-1}[B\delta f_t/(B\delta f_r+\varepsilon-\varepsilon\delta^2)] \tag{9}$$

计算各种δ值时的ε、T 和β值。取定δ值后，假设一ε算出f_r、f_t，代入（7）式算出E。如 E接近零或在控制精度范围，则ε即为所求。否则重设ε重复上述计算，直到求出ε为止。然后由（8）、（9）式分别算出相应的T 和β。算出的典型结果绘出于图 3、4、5。

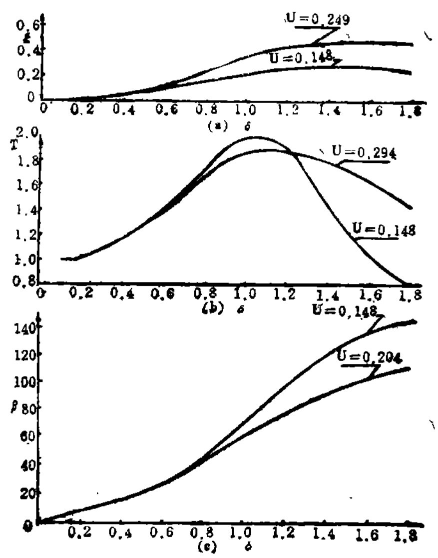

图 3　计算的带阻尼器转子动力特性之一

图 3 上示出的特性无跳变，即无双稳态现象。图 4、5 中三种特性均有跳变现象。可见B 值偏小或不平衡偏心比过大均容易发生双稳态现象。同样 B 值时，不平衡量越大发生跳变的转速比越高。发生特性跳变时，三种特性均在同转速下发生。升速时，ε和T下跳的转速比高于降速时上跳的转速比。而β则是升速时上跳的转速比高于降速时下跳的转速比。β角的跳变通称重心转向。上跳时β角由小于或稍大于90°上跳到远大于90°，降速时则作反向跳动。可见在高速时，数值较大的ε和T同较小的β角相对应；数值较小的ε和T 同较大的β角相对应。

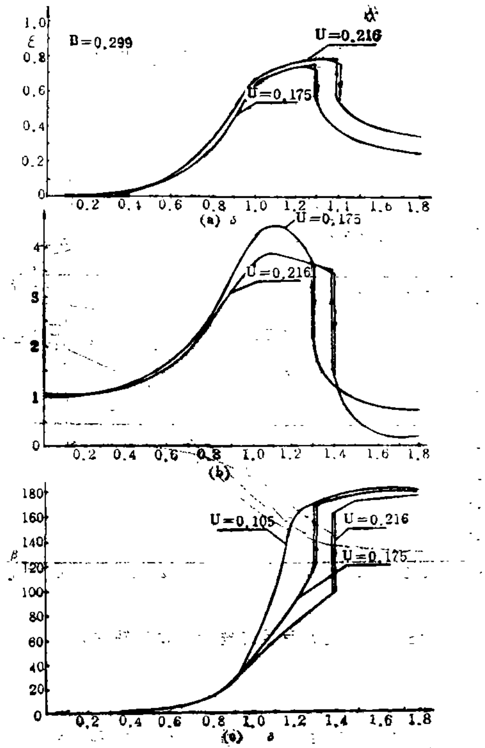

图 4　计算的带阻尼器转子动力特性之二

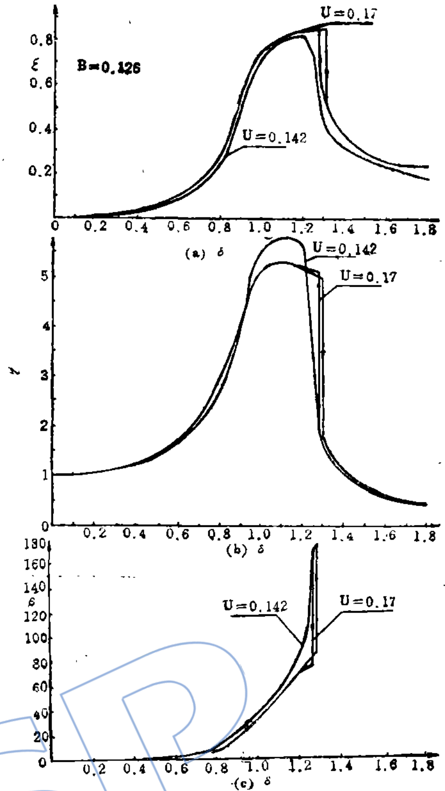

图 5　计算的带阻尼器转子动力特性之三

三、试 验 结 果

用计算的原转子进行了试验，所得典型结果示于图 6 、 7 和 8 。因未测外传力故图中缺传递率特性线。

图 6 同图 3 比较，转子数据相同。曲线数值上虽有些差异，而无双稳态现象是一致的。图 7 与图 4 转子，图 8 与图 5 转子数据分别相同。试验曲线与理论计算曲线比较，可得到关于双稳态现象的相同结论。实验曲线中ε下跳的转速高于计算值，上跳的转速低于计 算 值。β曲线有相应关系。

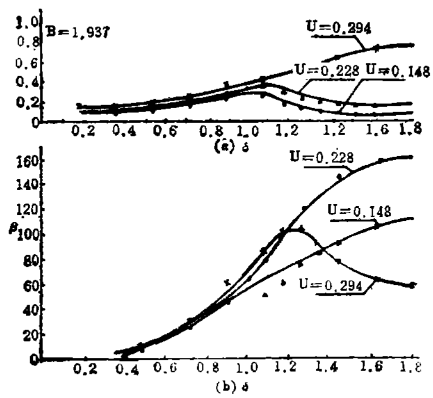

图 6　试验的带阻尼器转子动力特性之一

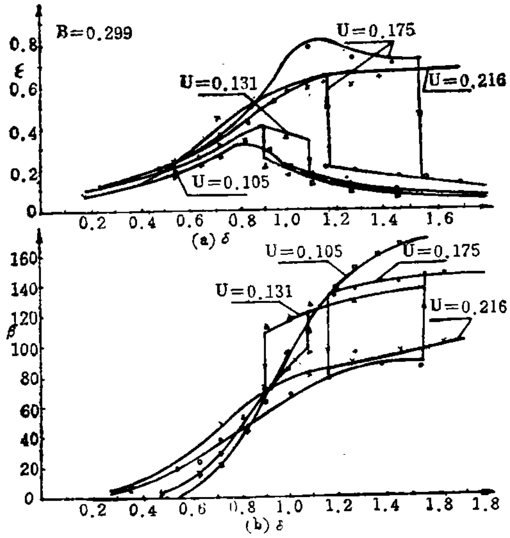

图 7 试验的带阻尼器转子动力特性之二

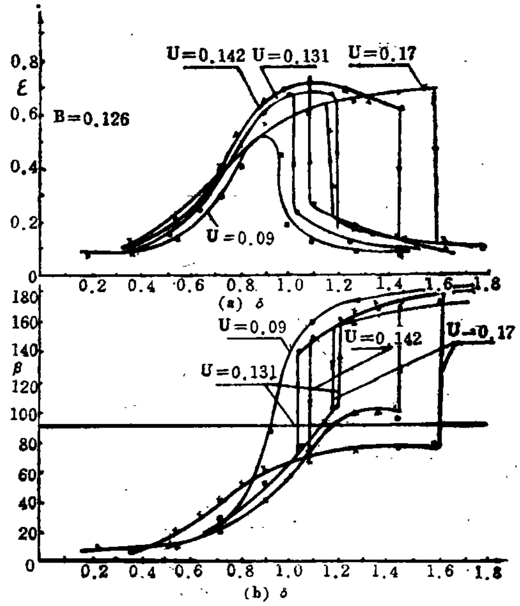

图 8 试验的带阻尼器转子动力特性之三

试验曲线与计算曲线数值上有差别的原因分析来自两个方面：计算中按短轴承（实际L/D=0.25不算短）半油膜近似，实际供油压约一个大气压以上，且未计轴颈倾斜。这些因素给计算造成一定误差。另一方面试验中装配、测试误差，特别是开车时不可避免的加、减速等影响必给试验带来误差。实验中跳变转速与计算的有较大差异的原因据国外资料认为是实验中跳变转速对阻尼敏感，阻尼略变跳变转速变化较大。尽管如此，从研究双稳态特性的发生规律、影响因素来看，试验结果与理论计算有良好的一致性。

四、结 论

1. 带挤压油膜阻尼器转子如参数匹配不当或不平衡力较大，容易出现双稳态现象。

2. 转子动力特性发生跳变时，偏心率、传递率与落后相角均同时发生跳变。

3. 轴承参数过小或（和）不平衡偏心比过大容易发生双稳态现象。同样B值发生双稳态现象时，不平衡偏心比U越高，升速时偏心率下跳的转速也越高。

4. 适当选择轴承参数B和控制转子不平衡量可在工作转速范围内避免出现双稳态现象。具体设计中可通过适当选取阻尼器长度、间隙和弹性支承刚性来增大阻尼器轴承参数B。

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quired in both the blade and the disk are more accurate. 2.In order to avoid the assumption for certain boundary conditions,the cyclic symmetric relations are employed at the corresponding boundaries of the disk sector. 3.Several different kinds of elements are applied to modeling the structure. For comparison five other simplified models are also evaluated.

AN EXPERIMENTAL INVESTIGATION ON THE BISTABLE BEHAVIORS OF A FLEXIBLE ROTOR-SQUEEZE FILM DAMPER SYSTEM

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Abstract

The bistable behaviors of resonance type occurred near main resonance speed of the system is investigated,based on experiments of a flexible rotor-squeeze film damper system. The influences of unbalance speed,radial clearance ratio, centralizing spring stiffness and acceleration process on the behaviors are considered. It is shown that the squeeze film damper is effective only within a certain unbalance range. If the unbalance is large enough, the bistable operation may occur. The larger the unbalance, the centralizing spring stiffness or the radial clearance ratio is, the more the possibility of occurance of the bistable operation.

THE BISTABLE BEHAVIORS OF A RIGID ROTOR WITH SQUEEZE FILM DAMPER

Yan Litang, Zhang Shiping, Li Qihan, Zhao Fuan

(Beijing Institute of Aeronautics and Astronautics)

Abstract

The bistable behaviors of the rigid rotor with centralizing spring and squeeze film damper are studied by means of calculations and experiments. The bistable behaviors are characterized by the variations of eccentricity, transmissibility and Lagging phase angle. The results calculated and experimental data are in a good agreement. The bistable operation of the rotor can

be eliminated at operating speeds if only the bearing parameters including the length and clearance of the film damper,and the stiffness of the centralizing spring are chosen properly over a certain unbalance range of the rotor.

A LOW CYCLE FATIGUE LIFE PREDICTION METHOD FOR BOLT HOLES OF DISK

Wang Lusheng, Gong Mengxian, Liang Cairong, Ru Meixiu

(Gas Turbine Establishment)

Abstract

A LCF(Low Cycle Fatigue)Life prediction method is provided on the basis of systematic experimental investigation, in which the median and reliability LCF Lives to crack initiation for the bolt holes of rotating disk can be predicted precisely by statistic data of smooth round bar specimen LCF lives to failure. The study shows that the LCF life to crack initiation for bolt holes of disk usually depends on the maximum strain history applied to the critical points of the bolt holes, and the strain gradient influence on the LCF life to crack initiation can be neglected as usual. In addition, the study also indicates that the median LCF life usually depends on the steel chemical composition and its thermal treatment.

THE EFFECTS OF PRESTRESSING OVERLOAD ON LOW CYCLE FATIGUE AND da/dN

Hou Jingyung, Liu Shaolun, Xie Jizhou

(Beijing Institute of Aeronautical Materials)

Abstract

The effects of prestressing overload on dow cycle fatigue and fatigue crack growth rates da/dN of titanium alloy TC-11 and Ni-base superalloy GH33A for gas turbine disks were investigated at room temperature and 633K. The experimental results show that the low cycle fatigue crack initiation life is increased by prestressing overload treatment within the range of low cycle fatigue life 5×10^3 to 5×10^4, but the effect of the prestressing overload on fatigue crack propagation life Np is not obvious. In addition, an empirical expression is formulated to estimate the fatigue life of side edge notched (SFN) specimen under prestressing overload as $S(N_f/r)=c$. When this expression is employed to estimate the fatigue life of a notched specimen prestressed, it is only necessary to know the conventional fatigue S-N curve.