

航空双转子发动机动静件 碰摩振动特征研究*

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【摘要】 双转子发动机有两个不同转速的转子,因而有两个不同频的激振源,故发生动静件碰摩时发动机的振动特征不同。从理论上分析了这种特征,通过模型转子系统实验验证,并与某航空双转子发动机发生碰摩故障时的实测振动特征相比较,表明理论分析与实测结果有良好的一致性。

主题词: 双转子喷气发动机 碰撞 振动分析

分类号: V 214. 33

1 双转子发动机动静件碰摩力学模型

1. 1 动静件碰摩和双转子发动机动静件碰摩力学模型

1982 年 Muszynska 曾提出一种动静件碰摩力学模型,1989 年^[1]又提出了修改模型:

$$M \ddot{z} + C \dot{z} + Kz = Fe^{i\omega t} - f_r [\delta/k + K_r(\delta - c)]e^{i\beta} \quad (1)$$

式中符号说明见文献[1]。1981 年 Childs D W^[2]提出另外一种模型:

$$M \ddot{z} + C \dot{z} + Kz = Fe^{i\omega t} - C_r \dot{\delta} + K_r(\delta - c)(1 - i\mu)e^{i\beta} \quad (2)$$

式中: C_r 和 μ 分别为动静件接触阻尼和摩擦系数,其余符号说明与文献[1]的相同。

这两种模型较适用于计算点碰摩和局部偏摩的情况。1993 年杨巍^[3]在上述基础上提出了同时适于整圈碰摩的力学模型:

$$M \ddot{z} + C \dot{z} + Kz = Fe^{i\omega t} + [f \varphi(z)]\{\Lambda\}, \quad \dot{\alpha}(z, t, \Lambda) = 0 \quad (3)$$

上述几种模型只能解释接触刚度较大的情况,还不适用于航空发动机。据赫芝^[4]接触理论依据两个接触体的形状、接触力与位移存在如下关系: $F = Kx^{3/2}$ 。航空发动机机匣刚度低,可将静子部件看作弹性体,动静件碰摩中情况更为复杂。碰摩时动静件间接触力与径向位移可视为成奇次幂函数关系: $F = K_1x + K_3x^3 + \dots$ 。由此可得如下力学模型:

$$M \ddot{z} + C \dot{z} + Kz = Fe^{i\omega t} - \sum_{j=1}^n F_j(\Omega t) [K_r(\delta - c)^{q/r} (1 - i\mu) e^{iq\beta/r}] e^{j\theta_j} \quad (4)$$

式中: $F_j(\Omega t)$ 为周期接触函数, $e^{j\theta_j}$ 为旋转因子。 q 和 r 为常数,根据接触形式可取 1, 2, 3, …; j 是碰摩的点数或区域数。

$$F_j(\Omega t) = 1 - \left\{ \frac{\Omega t_i}{2} - \sum_{k=1}^n \frac{2}{k} \sin \frac{k\Omega t_i}{2} \cos \left(k\Omega \frac{t - t_i}{2} \right) \right\} / \pi \quad (5)$$

为得出简便的数学模型假设: (1) 转子材料的内阻和轮盘的陀螺力矩影响都很小, 两者均忽略不计; (2) 碰摩时动静件接触处互有力作用和变形, 接触刚度相对很小。根据以上假设得出某型双转子发动机的动静件碰摩简化力学模型如图 1 所示。

图中 3 条线从下而上依次代表高压转子、低压转子、机匣。 m_1, m_3 和 m_5 分别代表前机匣、低压压气机转子和高压压气机转子的质量。 m_2, m_4 和 m_6 分别代表后机匣、低压涡轮转子和高压涡轮转子的质量。 k 和 c 分别表示各有关部分的刚度和阻尼。

以某型发动机为例列出碰摩时的动力

$$\begin{aligned}
 \text{方程: } m_1 \ddot{z}_1 + (c_1 + c_3 + c_4) \dot{z}_1 + (k_1 + k_3 + k_4) z_1 - c_3 \dot{z}_2 - c_4 \dot{z}_3 - k_3 z_2 - k_4 z_3 &= 0 \\
 m_2 \ddot{z}_2 + (c_2 + c_3 + c_5) \dot{z}_2 + (k_2 + k_3 + k_5) z_2 - c_3 \dot{z}_1 - c_5 \dot{z}_4 - k_3 z_1 - k_5 z_4 &= F(\Omega t) k(\delta - c)^{3/2} (1 - i\mu) e^{i\beta} \\
 m_3 \ddot{z}_3 + (c_4 + c_6 + c_7) \dot{z}_3 + (k_4 + k_6 + k_7) z_3 - c_4 \dot{z}_1 - c_6 \dot{z}_4 &= F_1 \cos \Omega t + F_2 \cos \Omega t \\
 m_4 \ddot{z}_4 + (c_5 + c_6 + c_9) \dot{z}_4 + (k_5 + k_6 + k_9) z_4 - c_5 \dot{z}_2 - c_6 \dot{z}_3 &= F_1 \cos \Omega t + F_2 \cos \Omega t \\
 m_5 \ddot{z}_5 + (c_7 + c_8) \dot{z}_5 + (k_7 + k_8) z_5 - c_7 \dot{z}_3 - c_8 \dot{z}_6 - k_7 z_3 - k_8 z_6 &= F_1 \cos \Omega t + F_2 \cos \Omega t \\
 m_6 \ddot{z}_6 + (c_8 + c_9) \dot{z}_6 + (k_8 + k_9) z_6 - c_8 \dot{z}_5 - c_9 \dot{z}_4 - k_8 z_5 - k_9 z_4 &= F_1 \cos \Omega t + F_2 \cos \Omega t - F(\Omega t) k(\delta - c)^{3/2} (1 - i\mu) e^{i\beta_1}
 \end{aligned} \quad (6)$$

2 双转子发动机碰摩振动响应特征

2.1 转子一点与机匣整圈碰摩

这种碰摩多发生在发动机磨合试车过程和转子振幅较大的突加不平衡或过临界转速的时候。这相当于转子上受到较大的不平衡力和摩擦力故此时 $F(\Omega t)$ 中的 $\Omega t_i = 0$ 。转子的振动特性与不平衡力激起的振动一样, 难以区别。以某双转子的简化模型为例按式(6)计算。得出图 2 所示的这种碰摩的频谱图。图上可看到两个转子的频率 N_1 和 N_2 , 这和不平衡力作用下的频谱图相同。故这种碰摩现象在测试中容易当作不平衡量较大而放过。

2.2 转子一圈与机匣一点碰摩

当机匣局部发生变形内凹时出现这种碰摩。这时偏心旋转的转子碰摩点上还受到一个定

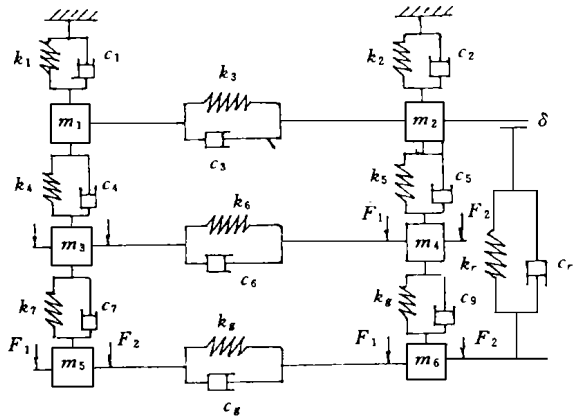


图 1 动静件碰摩简化力学模型

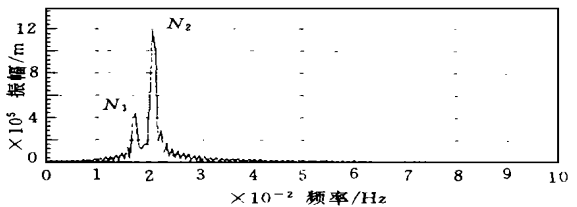


图 2 碰摩频谱图

向的力。这和横置不平衡转子在重力作用下的情况相似,可有零频振动谱。而且因机匣的刚性相对较小,作用力很小,所以动静件碰摩特征在测试中也难区分出来。

2.3 转、静子偏摩

这是指转子局部和静子局部接触相摩。图 3 中(a)和(b)分别给出某双转子发动机一处碰摩和三处碰摩算得的振动频谱图。图中大振动是碰摩的高压转子的频率和其倍频,其两侧的是复合频率,其他还有次谐波频率。

要清楚地看出常见的双转子发动机偏摩振动的频谱特征,按式(6)计算太繁,下面用分析法来导算。为了简化,主要抓住双转子发动机有两个激振源这一特点,其他方面就不作过细的考虑。以简化为具有双激振源的单质量系统为例进行分析,并设定某种碰摩接触形式使得碰摩接触点给转子增加了较小的立方非线性刚性,列出动力学方程:

$$\ddot{u} + 2\eta\dot{u} + \omega_0^2 u + \xi u^3 = F_1 \cos \omega_1 t + F_2 \cos \omega_2 t \quad (7)$$

式中: u 为振动位移, η 为阻尼比, ξ 为小参数, ω_0 为系统无碰摩时的固有频率, ω_1 和 ω_2 为两激振力的角频率。式(7)就成为简单的常见非线性方程式并可用小参数摄动法求解,令:

$$u(t, \xi) = u_0(t) + \xi u_1(t) + \xi^2 u_2(t) + \dots \quad (8)$$

将式(8)代入式(7),忽略 ξ 的高次项得如下方程组:

$$\ddot{u}_0 + 2\eta\dot{u}_0 + \omega_0^2 u_0 = F_1 \cos \omega_1 t + F_2 \cos \omega_2 t, \quad \ddot{u}_1 + 2\eta\dot{u}_1 + \omega_0^2 u_1 = -u_0^3 \dots \quad (9)$$

逐次求解上述方程得:

$$u_0 = A_1 \cos(\omega_1 t - \varphi_1) + A_2 \cos(\omega_2 t - \varphi_2) \quad (10)$$

$$\begin{aligned} u_1 = & B_{13} \cos(\omega_1 t - \varphi_1) + B_{23} \cos(\omega_2 t - \varphi_2) + B_{13} \cos(3\omega_1 t - \varphi_3) + B_{23} \cos(3\omega_2 t - \varphi_3) \\ & + B_{1+} \cos[(2\omega_1 + \omega_2)t - \varphi_{+2}] + B_{2+} \cos[(2\omega_2 + \omega_1)t - \varphi_{+1}] \\ & + B_{1-} \cos[(2\omega_1 - \omega_2)t - \varphi_{-2}] + B_{2-} \cos[(2\omega_2 - \omega_1)t - \varphi_{-1}] \end{aligned} \quad (11)$$

式(10)和(11)中 A_1, B_{13}, B_{23} 与两激振力及各种频率值有关的系数,一般随激振力增大并视各振动成分频率是否接近固有频率而有所不同, φ 为相位角。

由式(10)和(11)中可以看出所设碰摩形式的振动中除有 ω_1 和 ω_2 基频振动外并出现了 $3\omega_1, 3\omega_2$ 等倍频振动; $2\omega_1 - \omega_2, 2\omega_1 - \omega_2, 2\omega_2 + \omega_1, 2\omega_2 + \omega_2$ 等复合频率振动。又当转速 $\omega = \omega_1$ 或 $\omega_2 = \omega_2$ 时, A_1, B_{13} 和 A_2, B_{23} 分别达最大, u_0, u_1 从而振幅 u 最大。即分别发生频率为 ω_1 和 ω_2 的共振。当 $3\omega_1, 3\omega_2, 2\omega_2 + \omega_1, 2\omega_2 + \omega_2, 2\omega_1 - \omega_2, 2\omega_1 - \omega_2$ 分别等于 ω_0 时,同样出现 u 值较大的共振。可想而知当上述倍频或复合频率接近 ω_0 时 u 值也比较大。因此双转子发动机发生动静件碰摩时,可出现倍频和复合频率的振动。究竟何种频率成份的振动较大易测出,这与它们是否同转子的固有频率接近有关。

3 模型双转子系统实验结果和碰摩实测振动频谱图

用一结构类似某型双转子发动机的简化模型系统进行了碰摩实验,模拟该发动机的实际情况,碰摩是发生在高压转子与机匣之间,振动是在机匣上测量。实验结果如图 4(a), (b) 和

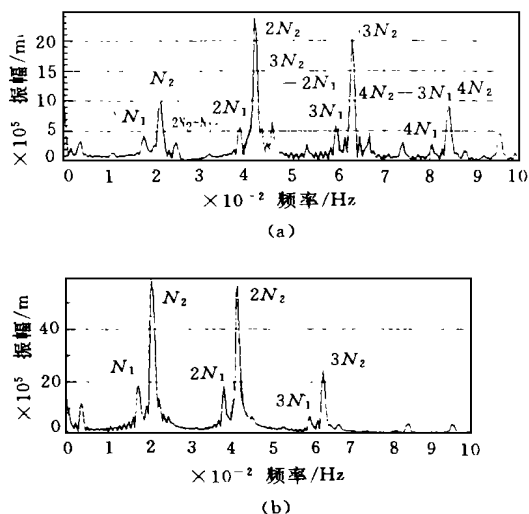


图 3 一处碰摩和三处碰摩算得的振动频谱图

(c)所示。从图上看到当低压转子不平衡力相对较小,无碰摩时仅出现高压转子频率的振动,

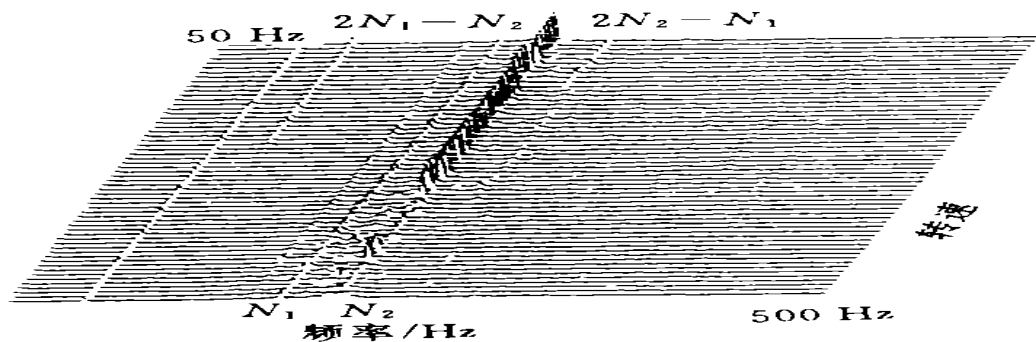


图5 实测发动机振动频谱图和三维瀑布图

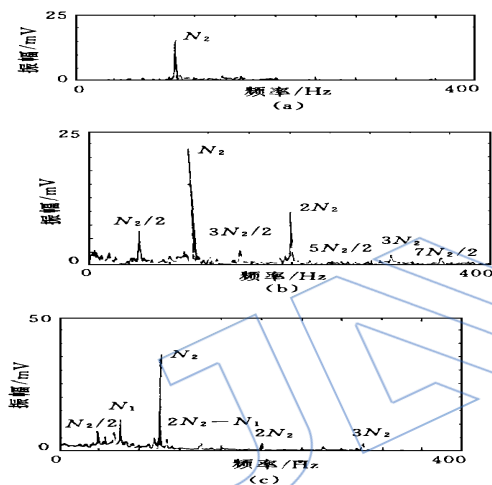


图4 实验结果

碰摩时也只出现高压转子频率及其倍频和分频振动; 低压转子不平衡力加大后碰摩时, 则出现两转子频率及其倍频、分频和复合频率的振动。实验结果与前面理论分析所得的振动特征是一致的。

某双转子发动机高压转子与机匣碰摩时实测出发动机振动频谱图和三维瀑布图如图5所示。图上可见出现 N_1 , N_2 , $2N_2 - N_1$ 和 $2N_1 - N_2$ 等基频和复合频率的振动。在另一次实测的瀑布图上则除出现上述频谱外还有更多的倍频、分频和复合频率的振动。各次测试结果略有不同的原因是各次实测中两转子的不平衡力和具体碰摩情况不尽相同。图上未出现 $2N_2 + N_1$ 和 $2N_1 + N_2$ 等组合频率振动的原因是其值很大, 超出所测振动的频率范围。由此可见理论分析与实测结果一致。表明前面的理论分析包括其中的假设是合理的、可行的。而理论分析式中未得出分频和高阶复合频率振动则又说明理论分析中所设定的振动模式与发动机中发生的具体碰摩模式还有微小的差别。

结论 理论分析、模型实验和某机实测结果均表明, 双转子发动机因有两个不同频的激振源, 发生动静件碰摩时, 除出现转子的基频振动外, 当发生偏摩时还会出现多种倍频率和分频振动以及两转子转速频率的多种复合频率的振动。

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NONSYNCHRONOUS MOTION AND JUMP PHENOMENON OF ROTOR- FLUID FILM BEARING SYSTEM WITH SQUEEZE FILM DAMPER

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ABSTRACT The dynamical behaviors of a flexible rotor- fluid film bearing system with squeeze film damper are investigated. Owing to the nonlinearity of the film forces, there are many complex phenomena in the system, such as: jump, nonsynchronous motion etc. As the parameters are varied, the system can undergo a series of bifurcations. It is found that the secondary Hopf bifurcation, a series of period-doubling bifurcation and the saddle-node bifurcation result in quasi-periodic motion, subharmonic adoption motion and jump phenomenon respectively. The results of this study show that the introduction of squeeze film dampers in this system may lead to undesirable nonsynchronous vibrations.

Key Words Bearings Rotors Nonlinear Systems Stability

AN ITERATIVE PRECISE INTEGRATION METHOD FOR CALCULATION OF TRANSIENT RESPONSE IN ROTOR- BEARING SYSTEM

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ABSTRACT An iterative precise integration method is proposed to calculate the transient response of rotor-bearing systems. The state transfer matrices are calculated with the precise integration method and the precise solutions of homogeneous equations can be obtained. Therefore an implicit integration procedure is given in the light of the characteristics of oil film forces. The matrices and vectors are partitioned and the dimension of the nonlinear equations is reduced greatly. The integration method presented is unconditionally stable and can assure high precision and efficiency.

Key Words Rotor Transient response Iteration methods Integration methods

VIBRATION FEATURES FROM RUBBING BETWEEN ROTOR AND CASING FOR A DUAL - SHAFT AEROENGINE

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ABSTRACT The vibration features from rubbing between rotor and casing for single-shaft gas turbine engine are well known and have been studied by many researchers. The dual-shaft engine has two shafts with different frequencies in the engine system. An improved model and calculation method are provided, based on the mathematical models derived

by the former researcher on rubbing of rotor and stator. The vibration characteristics for some types of rubbing for the dual- shaft engine are analyzed and discussed. The theoretical results are consistent with data measured on a dual- shaft aircraft engine.

Key Words Double spool jet engines Collisions Vibration analysis

STRESS ANALYSIS OF TRAPEZOID TOOTH BELT WITH FEM

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ABSTRACT The geometric nonlinearity of synchronous trapezoid tooth belt is taken as the point of departure. A nonlinearity finite element solution of the synchronous belt is provided on the basis of elastic linear finite element method to solve two- dimensional stress distribution problem. The level stress distribution for a tooth of the synchronous belt is researched with help of this method. Finally, for an example, a maximum stress point and stress distributions of a synchronous belt tooth have been found out and compared with its elastic linear solution.

Key Words Gears stress Distribution Nonlinearity Finite element analysis

NUMERICAL PREDICTION OF COLD FLOW FIELD BEHIND EBM C FLAMEHOLDER

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ABSTRACT A numerical prediction of the cold flow field behind a novel flameholder- Edge Blowing Mixture Curtain (EBM C) in two- Dimension duct was completed by using a modified k- ϵ model and S M P L E R method. The influence of the airflow parameters and the flameholder geometrical parameters on the recirculation zone was also studied. The prediction provides basis for designing the novel flameholder and studying its combustion performance.

Key Words Aerodynamic stabilization Flame holder Flow field Prediction

EXPERIMENTAL INVESTIGATION ON EBM C FLAMEHOLDER

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ABSTRACT A novel flameholder is presented, as named Edge Blowing Mixture Curtain (EBM C) flameholder. It has a streamline form head, small base width, with air and fuel mixture flowing from the slots at its edges into the main flow. It has small cold pressure loss without reheat and variable blowing on reheat so as to enlarge recirculation zone and improve combustion performance. An experimental investigation of this flameholder was carried