Nonlinear Analysis of Rotor-stator-bearing System Unsteady Oil Film Force

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Abstract: According to the LaGrange energy equation, the establishment of the unsteady oil film force of the rotor - stator - bearing system dynamics model, the application of numerical method, the system with the speed change time domain waveforms, amplitude spectra and Axis Orbit, the results show that: the unsteady oil film force and speed of change is closely related to changes in the process of oil film force oil whirl Oil Whip. Provide a theoretical basis for effective diagnosis of rotor - stator - bearing oil film instability fault research.

Keywords*: *Rotor-stator-bearing system; unsteady oil film force; oil whirl; oil whip

Ⅰ. INTRODUCTION

For large-scale high-speed rotor system, the bearing oil film force caused by the motion of the rotor instability phenomena of the outstanding problems, directly related to the unit collapse prevention and safety and reliable operation [1]. Along with rotating machinery to keep the high-speed, large-scale development, the vibration and stability of the rotor system has an increasingly important impact on the performance of the machinery and equipment. In the study of the dynamics of the rotor system, the oil film not only serves as a load bearing, to reduce friction and eliminate wear, vibration, suppress instability and bearing cooling role and dynamic coefficients of oil film (oil film stiffness coefficient and damper coefficient of oil film) is a direct impact on the rotor system dynamics calculations and stability analysis [2,5]. In recent years, the study reveals a complex nonlinear dynamical behavior characteristics [6~9]. Jeffcott rotor model with nonlinear oil film force bending vibration The presence of these nonlinear failure factors will have a greater low-frequency vibration after the instability of the system and the periodic motion, with the same frequency cycle movement superposition causes the system to produce non-harmonized movement shaft alternating stress generated. The rotor instability state long-term existence of the consequences are very serious[10~14].

Currently, the nonlinear behavior of the rotor system research has a certain foundation, but there are some deficiencies, such as the model is too simple, the rotor system is often split rotor system as " turntable + shaft " combination of 2 quality 4 degree of freedom mechanical model. This paper during the course of the study of the key issues of the rotor-stator-bearing system, according to the Lagrange equation to establish the unsteady oil film force of rotor stator bearing system with 8 degrees of freedom 4 quality of mechanical model, the model is more close to the actual production, thus the theoretical value is particularly highlight. From the analysis of the nonlinear characteristics of the system oil film force start, laid the foundation for the subsequent analysis of the dynamics of other fault systems.

Ⅱ.TO DESCRIBE THE LAGRANGE EQUATION

With n particle system of particles, by the complete ideal constraints, with N degrees of freedom, and its position by N generalized coordinates equation. Then

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial L}{\partial \dot{q}_i}\right)-\frac{\partial L}{\partial q_i}+\frac{\partial R}{\partial \dot{q}_i}=Q_i, \quad i=(1,2,\cdots,N) \tag{1}$$

Where, $L= T-V$-Lagrange function; T-the kinetic energy of the system function; V-system potential energy function; R-and the damping of the system corresponding to the dissipation function; Q_i -role in the system of generalized force; q_i-system of independent generalize coordinates; N -the total degree of freedom number.

Ⅲ.UNSTEADY OIL FILM ROTOR-STATOR-BEARING SYSTEM DYNAMICS MODEL

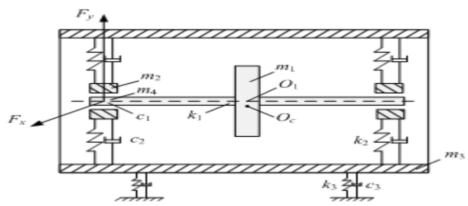


Fig.1 system-the mechanical Model of rotor-stator-bearing with unsteady oil film force

As shown in Fig.1 for the unsteady oil film force of rotor-stator-bearing system dynamics model, the rotor system as two identical oil film bearing support of a rigid disk. In

Fig.1,*oxy* is fixed coordinate system; O_1-rotor geometry center; O_c-for rotor center of mass; m_1-for rotor quality; m_2-bearing quality; m_3-as the stator quality for shaft bearing; m_4-in the equivalent concentrated quality, units are kg; k_1-rotation stiffness coefficient;k_2-bearing supporting the stiffness coefficient; k_3-as the stator support stiffness coefficient, the unit is N• m^{-1} ;c_1-rotating shaft of damping coefficient; c_2-bearing supports of the structural damping coefficient; c_3-based on stator rotor damping coefficient, units of N•sm^{-1}; x_i, y_i (i = 1,2,3,4)-for respectively the quality of the displacement coordinate.

According to the coordinates described physical sense, be unsteady oil film force of rotor stator bearing system of the differential equations of motion:

$$\begin{cases} m_1\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_4) + k_1(x_1 - x_4) = m_1 e\omega^2 \cos\omega t \\ m_1\ddot{y}_1 + c_1(\dot{y}_1 - \dot{y}_4) + k_1(y_1 - y_4) = m_1 e\omega^2 \sin\omega t - m_1 g \\ m_2\ddot{x}_2 + c_2(\dot{x}_2 - \dot{x}_3) + k_2(x_2 - x_3) = -F_x(x_2 - x_4, y_2 - y_4, \dot{x}_2 - \dot{x}_4, \dot{y}_2 - \dot{y}_4) \\ m_2\ddot{y}_2 + c_2(\dot{y}_2 - \dot{y}_3) + k_2(y_2 - y_3) = -F_y(x_2 - x_4, y_2 - y_4, \dot{x}_2 - \dot{x}_4, \dot{y}_2 - \dot{y}_4) - m_2 g \\ m_3\ddot{x}_3 + c_2(\ddot{x}_3 - \ddot{x}_2) + k_2(x_3 - x_2) + c_3\dot{x}_3 + k_3 x_3 = 0 \\ m_3\ddot{y}_3 + c_2(\dot{y}_3 - \dot{y}_2) + k_2(y_3 - y_2) + c_3\dot{y}_3 + k_3 y_3 + m_3 g = 0 \\ m_4\ddot{x}_4 + c_1(\dot{x}_4 - \dot{x}_1) + k_1(x_4 - x_1) = F_x(x_2 - x_4, y_2 - y_4, \dot{x}_2 - \dot{x}_4, \dot{y}_2 - \dot{y}_4) + m_4 b\omega^2 \cos\omega t \\ m_4\ddot{y}_4 + c_1(\dot{y}_4 - \dot{y}_1) + k_1(y_4 - y_1) = F_y(x_2 - x_4, y_2 - y_4, \dot{x}_2 - \dot{x}_4, \dot{y}_2 - \dot{y}_4) - m_4 g + m_4 b\omega^2 \sin\omega t \end{cases} \tag{2}$$

In the formula, b is shaft eccentricity, the unit is mm; F_x, F_y, respectively for the horizontal and vertical direction of the oil film strength, in units of N, and other parametric as above.

Nonlinear Oil Film Force model using the short bearing assumption Capone nonlinear oil film force model [15], the model has better accuracy and convergence[16], Dimensionless Reynolds equation under the assumption that in the short-bearing oil film force:

$$\left[\frac{R}{L}\right]^2 \frac{\partial}{\partial z}\left(h^3 \frac{\partial p}{\partial z}\right) = x\sin\theta - y\cos\theta - 2(\dot{x}\cos\theta + \dot{y}\sin\theta) \tag{3}$$

Where, R-bearing radius; L- bearing width; h=$\overline{h}$ /C -for the dimensionless oil film thickness; $\overline{h}$ - for the oil film thickness; C -bearing radial clearance; $z = \overline{z}/L$ -for the dimensionless axial displacement; $p = \dfrac{\overline{p}}{6\mu\omega(R/C)^2}$ -as the dimensionless pressure of oil film; $\overline{p}$ -for the oil film pressure; μ -for the oil film coefficient of viscosity Number; ω -as the rotor rotational speed; x for the dimensionless x axial center direction; y for the dimensionless y axial center direction; $\dot{x}$ for the dimensionless x axial center direction velocity component; $\dot{y}$ for the dimensionless y axial center direction velocity component;

Based on the formula can be dimensionless oil film pressure

$$p = \frac{1}{2}\left[\frac{L}{D}\right]^2 \frac{(x - 2\dot{y})\sin\theta - (y + 2\dot{x})\cos\theta}{(1 - x\cos\theta - y\sin\theta)^3}(4z^2 - 1) \tag{4}$$

Wherein, D is a bearing diameter; Dimensionless nonlinear oil film force ultimately can be expressed as:

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\frac{[(x - 2\dot{y})^2 + (y + 2\dot{x})^2]^{1/2}}{1 - x^2 - y^2} \times \begin{Bmatrix} 3xV(x,y,\alpha) - \sin\alpha G(x,y,\alpha) - 2\cos\alpha S(x,y,\alpha) \\ 3yV(x,y,\alpha) + \cos\alpha G(x,y,\alpha) - 2\sin\alpha S(x,y,\alpha) \end{Bmatrix} \tag{5}$$

Where

$$V(x,y,\alpha) = \frac{2 + (y\cos\alpha - x\sin\alpha)G(x,y,\alpha)}{1 - x^2 - y^2}$$

$$S(x,y,\alpha) = \frac{x\cos\alpha + y\sin\alpha}{1 - (x\cos\alpha + y\sin\alpha)^2}$$

$$G(x,y,\alpha) = \frac{2}{(1 - x^2 - y^2)^{1/2}}\left[\frac{\pi}{2} + \operatorname{arctg}\frac{y\cos\alpha - x\sin\alpha}{(1 - x^2 - y^2)^{1/2}}\right]$$

$$\alpha = \operatorname{arctg}\frac{y + 2\dot{x}}{x - 2\dot{y}} - \frac{\pi}{2}\operatorname{sign}\left[\frac{y + 2\dot{x}}{x - 2\dot{y}}\right] - \frac{\pi}{2}\operatorname{sign}(y + 2\dot{x})$$

Where, R is the bearing radius, L is the length of the bearing.

For studying the problem of convenience, the above equation dimensionless.

Set up $\tau = \omega t$, $x_1 = x_1/\delta_2$, $y_1 = y_1/\delta_2$, $x_2 = x_2/\delta_2$, $y_2 = y_2/\delta_2$, $x_3 = x_3/\delta_2$, $y_3 = y_3/\delta_2$, $x_4 = x_4/\delta_2$, $y_4 = y_4/\delta_2$, $\dot{x}_1 = \dot{x}_1/\omega\delta_2$, $\dot{y}_1 = \dot{y}_1/\omega\delta_2$, $\dot{x}_2 = \dot{x}_2/\omega\delta_2$, $\dot{y}_2 = \dot{y}_2/\omega\delta_2$, $\dot{x}_3 = \dot{x}_3/\omega\delta_2$, $\dot{y}_3 = \dot{y}_3/\omega\delta_2$, $\dot{x}_4 = \dot{x}_4/\omega\delta_2$, $\dot{y}_4 = \dot{y}_4/\omega\delta_2$, $\ddot{x}_1 = \ddot{x}_1/\omega^2\delta_2$, $\ddot{y}_1 = \ddot{y}_1/\omega^2\delta_2$, $\ddot{x}_2 = \ddot{x}_2/\omega^2\delta_2$, $\ddot{y}_2 = \ddot{y}_2/\omega^2\delta_2$, $\ddot{x}_3 = \ddot{x}_3/\omega^2\delta_2$, $\ddot{y}_3 = \ddot{y}_3/\omega^2\delta_2$, $\ddot{x}_4 = \ddot{x}_4/\omega^2\delta_2$, $\ddot{y}_4 = \ddot{y}_4/\omega^2\delta_2$, $\rho = e/\delta_2$, $\tilde{b} = b/\delta_2$, $\xi_1 = c_1/m_1\omega$, $\xi_2 = c_2/m_2\omega$, $\xi_3 = c_3/m_3\omega$, $\xi_{14} = c_1/m_4\omega$, $\xi_{23} = c_2/m_3\omega$, $\eta_1 = k_1/m_1\omega^2$, $\eta_2 = k_2/m_2\omega^2$, $\eta_3 = k_3/m_3\omega^2$, $\eta_{14} = k_1/m_4\omega^2$, $\eta_{23} = k_2/m_3\omega^2$, $G = g/\delta_2\omega^2$, $M_2 = \delta_2 m_2\omega^2$, $M_4 = \delta_2 m_4\omega^2$;

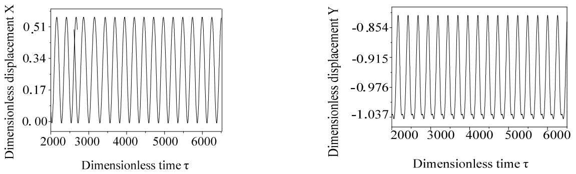
The above equation is transformed into:

$$\begin{cases} \ddot{x}_1 = -\xi_1(\dot{x}_1 - \dot{x}_4) - \eta_1(x_1 - x_4) + \rho\cos\tau + P_x/m_1\omega^2\delta_2 \\ \ddot{y}_1 = -\xi_1(\dot{y}_1 - \dot{y}_4) - \eta_1(y_1 - y_4) + \rho\sin\tau - \mathrm{G} + \mathrm{P_y}/m_1\omega^2\delta_2 \\ \ddot{x}_2 = -\xi_2(\dot{x}_2 - \dot{x}_3) - \eta_2(x_2 - x_3) = -f_x/M_2 \\ \ddot{y}_2 = -\xi_2(\dot{y}_2 - \dot{y}_3) - \eta_2(y_2 - y_3) = -f_y/M_2 - G \\ \ddot{x}_3 = -\xi_{23}(\ddot{x}_3 - \ddot{x}_2) - \eta_{23}(x_3 - x_2) - \xi_3\dot{x}_3 - \eta_3 x_3 - P_x/m_3\omega^2\delta_2 \\ \ddot{y} = -\xi_{23}(\dot{y}_3 - \dot{y}_2) - \eta_{23}(y_3 - y_2) - \xi_3\dot{y}_3 - \eta_3 y_3 - G - P_y/m_3\omega^2\delta_2 \\ \ddot{x}_4 = -\xi_{14}(\dot{x}_4 - \dot{x}_1) - \eta_{14}(x_4 - x_1) + f_x/M_4 + \tilde{b}\cos\tau \\ \ddot{y}_4 = -\xi_{14}(\dot{y}_4 - \dot{y}_1) - \eta_{14}(y_4 - y_1) + f_y/M_4 - G + \tilde{b}\sin\tau \end{cases} \tag{6}$$

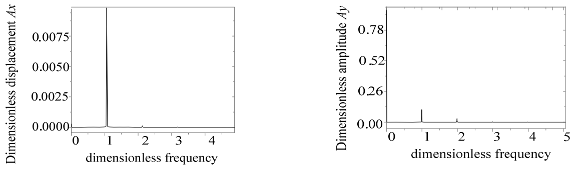
This paper discusses the rotor-stator-bearing system only in the unsteady film force motion characteristics. In view of the nonlinear rotor system complexity and the bearing oil film force, the analytical method has been powerless, only numerical solution, where the use of a fourth-order Runge-Kutta method to solve the equation (6).

Ⅳ. NUMERICAL SIMULATION OF UNSTEADY OIL FILM

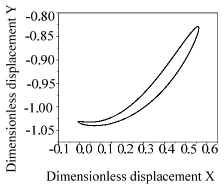
In the calculation in order to be able to quickly get stable solution, in the calculation of the time allowed under the circumstances, should be selected as small as possible and the step cycle enough. In order to eliminate the impact of transient response, at least 2000 data points to take out after the 4500 data points. Calculation of trajectory map from 10 to 20 cycles after. Select the system parameters are as follows: $m_1 = 4.0kg$, $m_2 = 32.1kg$, $m_3 = 50.0kg$, $m_4 = 20.0kg$, c_1=1050N·sm^1, c_2=2100N·sm^{-1}, c_3=2100N·sm^{-1}, k_1=2.5e5Nm^{-1}, k_2=2.5e5N·m^{-1}, k_3=2.5e7N·m^{-1}, k_r=1.0e7N·m^{-1}; R=0.025m, L=0.570m, δ_1=0.2mm, δ_2=0.2mm; η=0.018MPa; R_1=0.015m; f=0.2; The speed ratio S=□/□$_0$, get the system response, as shown in Fig. 2~ Fig. 4.



(a) x, y direction of the time domain waveform diagram

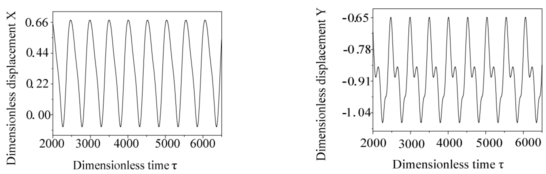


(b) x, y direction of spectrum diagram

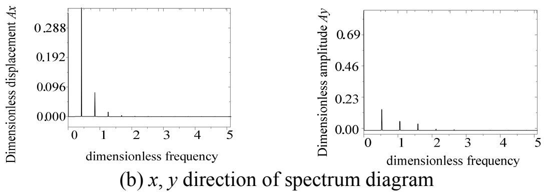


(c) axis path

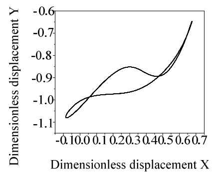
Fig. 2 Speed ratio of S=1 with the response of the system



(a) x, y direction of the time domain waveform diagram

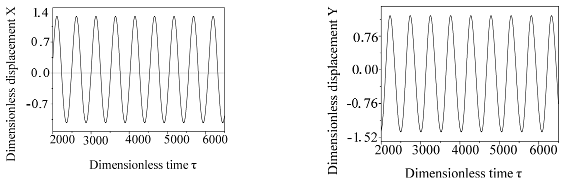


(b) x, y direction of spectrum diagram

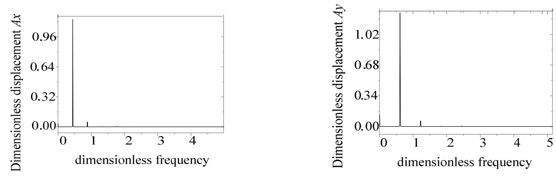


(c) axis path

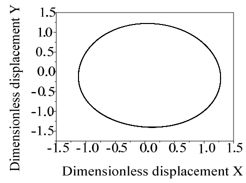
Fig. 3 Speed ratio of S=2 with the response of the system



(a) x, y direction of the time domain waveform diagram



(b) x, y direction of spectrum diagram



(c) axis path

Fig. 4 Speed ratio of S=4 with the response of the system

Ⅴ. CONCLUSION

When the rotational speed ratio as the only variable, the observed rotor system of the x domain diagram in the y-direction, x, y directions spectrum diagram and Orbit, the following conclusions:

(1) From a time domain waveform diagram of view, when the rotational speed equals the critical speed, the x-direction of the waveform showing a single peak of the sine waveform, to reflect the system to the fundamental frequency component based, y direction waveform reflects the combination frequency; When speed is 2times the critical speed, the rotor the as table film with the vibration frequency changes occur in the system oil whip, x, y direction of the time domain waveform are changed, then the system half-frequency amplitude is higher than the base frequency the waveform of the amplitude, x direction showing mainly half-amplitude of the sine wave shaped, y direction waveform showing the M superimposed waveform of the two components; when the speed is higher than 2-fold more than the critical speed, x, y direction of the time domain waveform are sine wave.

(2) From the spectrum chart of view, when the rotor system is equal to the critical speed, the system *x*-direction does not have the low-frequency components exist, the system is often accompanied by a frequency of 1×, the vibration stability, the system y direction occurs in 1× and 2× a combination, apparent low-frequency component; when the rotational speed of the rotor system is equal to 2 times the critical speed, the system characterized in a frequency of (0.42~0.48)×, often accompanied by a frequency of the combination frequency, vibration stability is poor, a short time amplitude of a sudden increase direction of vibration performance for the *x* direction, the system by the oil whirl into the state of the oil film oscillation; when the rotor system is higher than 2-fold above the critical speed, low frequency components are still in existence, the characteristic frequency (0.42~0.48)×, but the size of the *x*-direction vibration amplitude does not vary with speed change, the system is often accompanied by a frequency (0.42~0.48)× and 1× a combination of frequency, the vibration direction of performance for the amplitude of the y direction increases suddenly.

(3) Orbit view from below the first critical speed, as the speed increases, orbit increasing the eddy center go up with the speed, speed each 50rad/s sliding scale, the center of the eddy increased 2~4μm, but when the speed is just over the first critical speed, particularly due to the decrease of the journal vibration decreases in the vertical direction, so that the center of the eddy and some subsidence; as the speed continues increase, the journal whirl trajectory began floating, and appeared half-frequency eddy center position by the amplitude of the amplitude, speed; additional down speed rotor - bearing system in the oil film force instability in stability, spread to the entire region of the axis of the rotor track, The eddy center go up to the most vertices.

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