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Imbalance–misalignment–rubbing coupling faults in hydraulic turbine vibration☆

X.Q. Fu*, W.T. Jia, H. Xu, S.L. Song

School of Power and Mechanical Engineering, Wuhan University, Wuhan 430072, China

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ABSTRACT

To solve the turbine generator set's fault, a nonlinear dynamics model of the imbalance–misalignment–rubbing coupling fault is established. The dynamic behavior of the system with the rotor speed is also studied with numerical integration methods. The nonlinear dynamic response of generator rotor and turbine rotor at coupling fault is analyzed systematically by a series of methods, such as bifurcation diagrams, Poincaré map, Orbit, time-domain waveform and amplitude spectrum. Combined with the experimental simulation on the experiment table, it is demonstrated that the system model can fit in current situation very well, which can be applied to the dispose analysis of the coupling fault.

Rotating Machinery plays an irreplaceable role in many areas (e.g. energy exploration, transportation and defense and military fields). And in the process of fault diagnosis, rotor system misalignment fault is the most common phenomenon. According to the relevant statistical data, there is over 60% of the total number of modern rotating machinery faults belonged to rotor system misalignment faults [1]. And with the increasing efficiency of rotating machinery, the gap between rotor and stator reduces gradually, which makes the possibility of the rubbing faults caused by the misalignment of the coupling greatly amplifies. Therefore, it is of great significance to study the rotor system of misalignment–rubbing coupling faults mechanism.

In the literature [1], through the establishment of misalignment constraints relationship between the two rotors, the rotor misalignment system characteristics is analyzed with dimensionless factor as an argument, and the motion path of quasi-periodic to chaotic path and chaos motion burst into periodic motion is analyzed. The literature [2] confirms the frequency and transient response of the dimensionless, which takes advantage of the numerical misalignment method based on the combination of Newmark and Newton iteration to solve nonlinear equations. The literature [3] studies rotor-sliding bearing system's axis orbit and load response with the finite element by the combination of the modal reduction method, and establishes a simplified model to calculate the motion characteristics of the system with the numerical solution. The literature [4] gets the relevant diagnostic features about parallel misalignment and angles misalignment Rotor System through studying the double-disc rotor system vibration response of the misalignment fault. The literature [5] studies the dynamic response of flexible rotor–ball bearing system under the unbalance–rubbing–misalignment coupling faults, and establishes the coupling faults flexible multi-body system dynamic control model of rotor–ball bearing system based on the finite element combined with the numerical calculation simulation method.

This paper established vertical structure bearing–rotor system unbalance–misalignment–rubbing coupling faults dynamic models, deduces differential equations according to the generator rotor rubbing problems caused by the integrated misalignment of Rub hydroelectric generating set coupling, and analyzes the nonlinear dynamical behavior of the rotor system with the parallel misalignment and rotate speed as the control parameter. The misalignment–rubbing coupling faults experiment is carried out on the multi-purpose rotor experiment table, which tests the validity of the model.

1. Rotor–bearing system unbalance–misalignment–rubbing coupling faults dynamic model

Fig. 1 is a hydraulic turbine guide bearing–Generator rotor and stator–impeller system. According to the unbalance–misalignment–rubbing coupling faults that the relevant unit may exist, the dynamic model is established, and O,O_1 and O_2

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* Corresponding author.
E-mail address: rflwrbz@163.com

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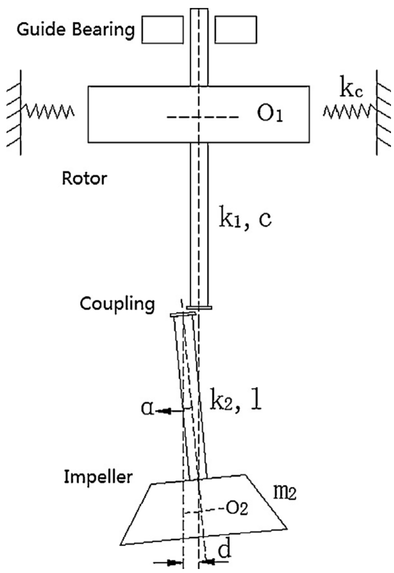

Fig. 1. Mechanical model of the hydropower rotors system.

respectively represent the position of hydro turbine rotor axis, hydro-generator rotor center and the turbine impeller center under the balanced condition; m_1 and m_2, respectively, represent the quality of the axis neck of hydro-generator and rotor equivalent to the rotating shaft center point; e is defined as the mass eccentricity of the hydro-generating unit's generator rotor, while the turbine impeller's mass eccentricity, the radial and torsional vibration generated during rotation are ignored.

According to Newton's second law, the motion system's differential equations are as follows:

$$\begin{cases} m_1\ddot{x}_1 + c_1\dot{x}_1 + k(x_1 - x_2) + F_{1x} = F_x \\ m_1\ddot{y}_1 + c_1\dot{y}_1 + k(y_1 - y_2) + F_{1y} = F_y \\ m_2\ddot{x}_2 + c_2\dot{x}_2 + 2k(x_2 - x_1) + F_{2x} = m_2e\omega^2\cos(\omega t) + P_x + F_{cx} \\ m_2\ddot{y}_2 + c_2\dot{y}_2 + 2k(y_2 - y_3) + F_{2y} = m_2e\omega^2\sin(\omega t) + P_y + F_{cy} \end{cases} \tag{1}$$

where F_{1x} and F_{1y} are the force at the left end of the bearing; P_x and P_y are the two components of rubbing force between the rotor and stator in the x and y directions; F_{cx} and F_{cy} are two components of misalignment force at coupling in the x and y directions. Let $X_1 = x_1/c, Y_1 = y_1/c, X_2 = x_2/c, Y_2 = y_2/c, \tau = \omega t$, c is the average oil film thickness, the dimensionless motion equations for the system are: (still using x_1 represents X_1, y_1 represents Y_1, x_2 represents X_2, y_2 represents Y_2)

$$\begin{cases} \ddot{x}_1 + \frac{c_1}{\omega m_1}\dot{x}_1 + \frac{k}{\omega^2 m_1}(x_1 - x_2) + \frac{c^2 f_{1x}}{\omega^2 m_1} = \frac{\sigma f_x}{c\omega^2 m_1} \\ \ddot{y}_1 + \frac{c_1}{\omega m_1}\dot{y}_1 + \frac{k}{\omega^2 m_1}(y_1 - y_2) + \frac{c^2 f_{1y}}{\omega^2 m_1} = \frac{\sigma f_y}{c\omega^2 m_1} \\ \ddot{x}_2 + \frac{c_2}{\omega m_2}\dot{x}_2 + \frac{2k}{\omega^2 m_2}(x_2 - x_1) + \frac{c^2 f_{2x}}{\omega^2 m_2} = \frac{e\omega^2\cos(\omega t)}{c\omega^2} + \frac{p_x}{c\omega^2 m_2} + \frac{f_{cx}}{c\omega^2 m_2} \\ \ddot{y}_2 + \frac{c_2}{\omega m_2}\dot{y}_2 + \frac{2k}{\omega^2 m_2}(y_2 - y_1) + \frac{c^2 f_{2y}}{\omega^2 m_2} = \frac{e\omega^2\sin(\omega t)}{c\omega^2 m_2} + \frac{p_y}{c\omega^2 m_2} + \frac{f_{cy}}{c\omega^2 m_2} \end{cases} \tag{2}$$

1.1. Misalignment force model

The model in this paper mainly considers the coupling integrated misalignment, which includes two cases of angle misalignment and parallel misalignment. When the master and slave rotor axis connected through the coupling exists misalignment, coupling housing is forced to make circular motion around its center by the limit of the two halve-couplings rotating around their respective axes. Its trajectory can be described by the following formulas:

$$\begin{cases} x = \frac{1}{2}\Delta e \cdot \sin\left(2\omega t - \theta\right) \\ y = \frac{1}{2}\Delta e \cdot \cos\left(2\omega t - \theta\right) \end{cases} \tag{3}$$

where ω is the rotate speed, θ is initial phase of misalignment, Δe is the equivalent amount of misalignment, which is determined by the coupling spacing Δl, parallel misalignment amount Δd and misalignment angle, $\Delta\alpha \cdot \Delta e = \Delta d + \Delta l \cdot \tan\left(\Delta\alpha/2\right)$.

Through Newton's second law of motion and acceleration of couplings can obtain equivalent misalignment force:

$$\begin{cases} F_{cx} = m_c \cdot \Delta e \cdot \omega^2 \cdot \sin\left(2\omega t - \theta\right) \\ F_{cy} = m_c \cdot \Delta e \cdot \omega^2 \cos\left(2\omega t - \theta\right) \end{cases} \tag{4}$$

where m_c is the quality of the coupling housing.

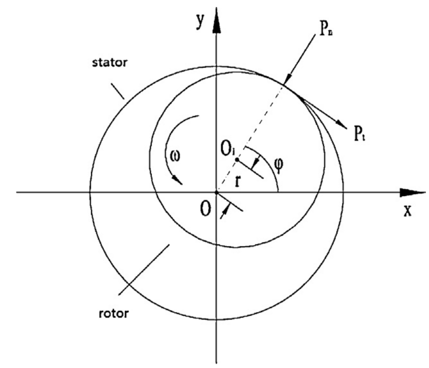

Fig. 2. Dynamic model of rubbing faults.

1.2. Rubbing force model

The model assumes that there is elastically rubbing between the rotor and stator, and the rubbing force is determined by the rotor and stator and the stiffness of spindle, and the thermal effects under the rubbing force is ignored. As shown in Fig. 2, f is assumed as the friction coefficient about hydroelectric generating set, k_c is the rigidity of the stator, δ is the gap between generator rotor and stator, so the radial rubbing force of the system P_n and the tangential rubbing force of the system can be described as

$$\begin{cases} P_n = \left(r - \delta\right)k \\ P_t = fP_n \end{cases} \tag{5}$$

where $r = \sqrt{x^2 + y^2}$ is the radial displacement of hydro-generating unit of the rotor relative to the geometric center O of the hydroelectric generator stator, the rotor axis position coordinates is (x,y) at this condition.

The rubbing force of the system is broken down to the component in x and y directions of Cartesian coordinate system as

$$\begin{Bmatrix} P_x \\ P_y \end{Bmatrix} = \begin{bmatrix} -\cos\varphi & \sin\varphi \\ -\sin\varphi & -\cos\varphi \end{bmatrix} \begin{Bmatrix} P_n \\ P_t \end{Bmatrix} \tag{6}$$

Through formulas (5) and (6), the rubbing force between the rotor and stator of Turbine can be expressed as

$$\begin{Bmatrix} P_x \\ P_y \end{Bmatrix} = L\left(r - \delta\right)\frac{(r - \sigma)k_c}{r}\begin{bmatrix} 1 & -f \\ f & 1 \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} \tag{7}$$