NATURAL VIBRATION PROPERTY ANALYSIS OF HYDRAULIC GENERATING UNIT WITH DYNAMIC CHARACTERISTICS OF GUIDE BEARING

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Abstract: A simplified FEM kinetic model of unit shaft system including the rotor, turbine runner and three guide bearings is established. According to the Lyapunov first approximation theory, under the unbalanced magnetic pull of rotor and nonlinear sealing force of turbine, the change of the critical instable speed of different shaft system (with or without considering of the dynamic characteristics of guide bearings) was compared. The dynamic characteristics of guide bearings are taken into account to study the first critical speed calculation of shaft system with flexible support, on this basis, the effect of the clearance of the bearings on the first critical speed was discussed. The results presented that there is no stable operating point for vertical shaft system when the dynamic characteristics of guide bearing was considered. The relationship between the bearing clearance and the critical speed showed nonlinear characteristics.

Keyword: Unit shaft system, critical instable speed, bearing clearance, critical speed

1. INTRODUCTION

The operation stability of hydraulic generating unit is an important problem, and it is concerned extensively and significantly by researchers. The critical instable speed is defined as the highest speed for the rotor could operate at a stable point which can be calculated by Lyapunov first approximation theory. The resonance may happen when the operating speed is closed to the basic natural frequency of shaft system, and this speed is called first critical speed. The bearing flexibility plays a predominant role in locating the both speed as mentioned before. The speed will increases with the bearing to be more rigid, but there is a limit to the increasing[1]. It is significantly to accurately predict the value of the critical instable speed and first critical speed.

The rotor speed is an important parameter for dynamic properties of bearing. In the range of high speed the value of parameters will increase with the speed increasing[2]. Recent studies have given results that the first critical speed can be reduced under the unbalanced magnetic pull and fluid sealing force [3]. The influence to basic frequency and first critical speed of shaft by the dynamic characteristics of flexible support was discussed[4]. A nonlinear rotor-bearing system was established to study the whirling and whip instabilities

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of system[5].Two kinds of stiffness are distinguished between shaft and bearings, the researchers considered that there are two forms of instabilities for rotor-bearing system with the stiffness value of bearing on different level [6].

In the past, most research works on the critical instable speed were emphasized on the horizontal unit, but the large scale hydraulic generating set almost all are vertical unit. The bearing system has a very small lateral initial stiffness because of less eccentricity which without load from gravity for vertical unit. So, the critical instable speed is a minimum value and the operation of shaft is unstable[7]. In the conventional studies of the first critical speed, the stiffness of the support is a fixed value in normal. In actually, the stiffness and damping of bearing are dynamic parameters with the rotor speed, furthermore, the first critical speed of the shaft system is changed with the base frequency.

A short journal bearing model is employed in this paper. The shaft-bearing model with shaft (include rotor and turbine) and three bearing support on the upper, middle and lower location of shaft is established. The dynamic behaviors of bearing are considered as an effective stiffness and damping coefficient. These characteristics are combined in the model’s stiffness, mass and damping matrix in the numerical model. And numerical method is used here to certificate that the dynamic behaviors of bearing support have significant influence on critical instable speed of vertical unit and the calculation of first critical speed of system. On this basis, the gradual change of first critical speed of system was analyzed with switching clearances of three bearings by numerical calculation.

2. MODEL OF SHAFT AND BEARING

The model is simplified as five nodes, and each node has two single degrees of freedom(SDOF) in horizontal x and y directions. As illustrated in Fig.1,the shaft is divided into four beam elements which is annular cross section, only the lateral degrees of unit is considered, while the vertical and angle degrees are ignored. The element displacement vector is expressed as $x=\{x_1, x_2, x_3, x_4\}^T$, which the x_1, x_2 are the displacements of x and y on node i, and x_3, x_4 are displacements on node j.

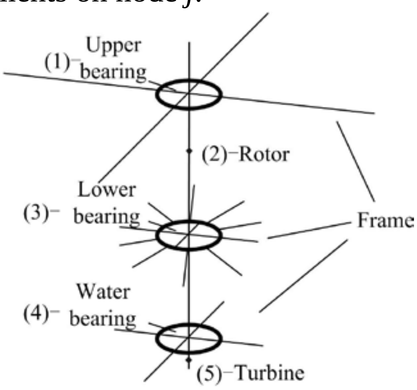


Fig.1 Shaft System Outline Drawing

The stiffness matrix of beam element can be expressed as

$$K^e = \begin{pmatrix} k_x & 0 & -k_x & 0 \\ 0 & k_y & 0 & -k_y \\ -k_x & 0 & k_x & 0 \\ 0 & -k_y & 0 & k_y \end{pmatrix} \tag{1}$$

where, k_x=12EI$_x$/l^3, k_y=12EI$_y$/l^3, I$_x$=I$_y$=$\frac{\pi(r_1^4-r_2^4)}{4}$, so, $k_x=k_y$.

The lumped mass matrix, namely:

$$M^e=\text{diag}\{m_i,\ m_i,\ m_j,\ m_j\} \tag{2}$$

Where, m_i=0.5ρS(l_i+ l_{i-1}), m_j=0.5ρS(l_j+ l_{j+1}). S is the area of element cross section.

The global stiffness matrix and lumped mass matrix is respectively assembled by element matrix, as:

$$K_G=\begin{pmatrix} k_1 & 0 & -k_1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & k_1 & 0 & -k_1 & 0 & 0 & 0 & 0 & 0 & 0 \\ -k_1 & 0 & k_1+k_2 & 0 & -k_2 & 0 & 0 & 0 & 0 & 0 \\ 0 & -k_1 & 0 & k_1+k_2 & 0 & -k_2 & 0 & 0 & 0 & 0 \\ 0 & 0 & -k_2 & 0 & k_2+k_3 & 0 & -k_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & -k_2 & 0 & k_2+k_3 & 0 & -k_3 & 0 & 0 \\ 0 & 0 & 0 & 0 & -k_3 & 0 & k_3+k_4 & 0 & -k_4 & 0 \\ 0 & 0 & 0 & 0 & 0 & -k_3 & 0 & k_3+k_4 & 0 & -k_4 \\ 0 & 0 & 0 & 0 & 0 & 0 & -k_4 & 0 & k_4 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & -k_4 & 0 & k_4 \end{pmatrix} \tag{3}$$

$$M_G=\text{diag}\{m_1,\ m_1,\ m_2+m_r,\ m_2+m_r,\ m_3,\ m_3,\ m_4,\ m_4,\ m_5+m_t,\ m_5+m_t\} \tag{4}$$

Where m_r is the rotor lumped mass, m_t is the turbine lumped mass.

The spring elements are used to model the dynamic stiffness of bearings in two directions, which express as k_u, k_d, k_w.

The short journal bearing model is employed to simulate the flexible support of shaft[8],which is obtained by solving the Reynold equation. The support force is defined as Eq.(6), and its coordinate reference system is shown in Fig.2.

P_ε and P_Ψ are the oil film pressure in radial direction and tangential direction, given as,

$$\begin{cases} P_\varepsilon = \frac{\pi S_0}{2}\left(\frac{L_b}{R_b}\right)^2 \overline{P}_\varepsilon = \frac{\pi S_0}{2}\left(\frac{L_b}{R_b}\right)^2 \left[(\omega-2\dot{\psi})G_1(\varepsilon)+2\dot{\varepsilon}_b G_2(\varepsilon)\right] \\ P_\psi = \frac{\pi S_0}{2}\left(\frac{L_b}{R_b}\right)^2 \overline{P}_\psi = \frac{\pi S_0}{2}\left(\frac{L_b}{R_b}\right)^2 \left[(\omega-2\dot{\psi})G_3(\varepsilon)+2\dot{\varepsilon}_b G_4(\varepsilon)\right] \end{cases} \tag{5}$$

Where ω is the rotor angular speed, the bearing parameter are bearing length L_b, bearing radius R_b, bearing clearance c_b and oil viscosity μ.The bearing axis eccentricity is e_b, the axis displacement in lateral direction x_b and y_b, $e_b=\sqrt{x_b^2+y_b^2}$, the relative eccentricity is ε_b= e_b/c_b, $\dot{\varepsilon}_b=\left(x_b\dot{x}_b+y_b\dot{y}_b\right)/e_b c_b$ .The bearing axis deviation angle is Ψ=arctan(y_b/x_b)as shown in Fig.2, so, $\dot{\psi}=\left(x_b\dot{y}_b-\dot{x}_b y_b\right)/e_b^2$.$S_0=\mu R b^3 L_b/\pi c_b$.

The oil film force in x and y directions are generated by the P_ε and P_Ψ,as shown in Eq.(6),

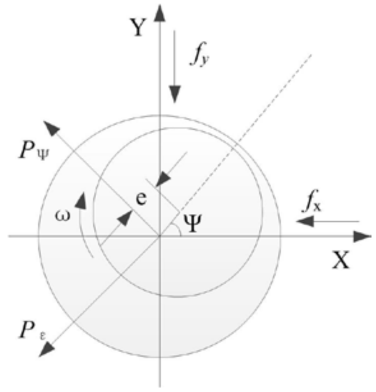


Fig.2 Oil Film Force for Bearing

$$\begin{pmatrix} f_x \\ f_y \end{pmatrix} = \frac{\pi S_0}{2e_b}\left(\frac{L_b}{R_b}\right)^2 \begin{pmatrix} -\overline{P}_\varepsilon & -\overline{P}_\psi \\ \overline{P}_\psi & -\overline{P}_\varepsilon \end{pmatrix}\begin{pmatrix} x_b \\ y_b \end{pmatrix} = A_{01}\begin{pmatrix} -\overline{P}_\varepsilon & -\overline{P}_\psi \\ \overline{P}_\psi & -\overline{P}_\varepsilon \end{pmatrix}\begin{pmatrix} x_b \\ y_b \end{pmatrix} \tag{6}$$

The four stiffness coefficients (k_{xx}, k_{yy} , k_{xy}, k_{yx}) of short bearing can be defined as the coefficients in front of the bearing axis displacement vector in Eq.(6).

The unbalanced magnetic pull is acted on rotor[9],[10], given as Eq.(7),

$$\begin{cases} f_{umpx} = \dfrac{R_r L_r \pi\mu_0 F_\delta^2}{\delta_0^2}\dfrac{e_r}{\delta_0}\left(\dfrac{1}{2}+\dfrac{5}{8}\varepsilon_r^2\right)\dfrac{x_r}{e_r} = A\left(\dfrac{1}{2}+\dfrac{5}{8}\varepsilon_r^2\right)x_r = A_{02}x_r \\ f_{umpy} = \dfrac{R_r L_r \pi\mu_0 F_\delta^2}{\delta_0^2}\dfrac{e_r}{\delta_0}\left(\dfrac{1}{2}+\dfrac{5}{8}\varepsilon_r^2\right)\dfrac{y_r}{e_r} = A\left(\dfrac{1}{2}+\dfrac{5}{8}\varepsilon_r^2\right)y_r = A_{02}y_r \end{cases} \tag{7}$$

The rotor parameters are rotor length L_r, rotor radius R_r. x_r and y_r are displacements of rotor center, δ_0 is the mean air-gap length when the rotor is centered, e_r is the eccentricity of rotor center, $e_r = \sqrt{x_r^2 + y_r^2}$,the relative eccentricity $\varepsilon_r = e_r/\delta_0$ .F_δ is the amplitude of fundamental magnetic force, μ_0 is the air gap permanence.

The sealing force is generated by Muszynska nonlinear model, which is expressed as Eq.(9),

$$\begin{pmatrix} f_{sx} \\ f_{sy} \end{pmatrix} = -\begin{pmatrix} K - m_f\tau_f^2\omega^2 & \tau_f\omega D \\ -\tau_f\omega D & K - m_f\tau_f^2\omega^2 \end{pmatrix}\begin{pmatrix} x_s \\ y_s \end{pmatrix} - \begin{pmatrix} D & 2\tau_f m_f\omega \\ -2\tau_f m_f\omega & D \end{pmatrix}\begin{pmatrix} \dot{x}_s \\ \dot{y}_s \end{pmatrix} - \begin{pmatrix} m_f & 0 \\ 0 & m_f \end{pmatrix}\begin{pmatrix} \ddot{x}_s \\ \ddot{y}_s \end{pmatrix} \tag{9}$$

Where ω is turbine angular speed. x_s and y_s are the center displacement of sealing structure, the eccentricity $e_s = \sqrt{x_s^2 + y_s^2}$,c_s is the clearance of sealing.And more details about the above expressions can be seen in reference[11].

3. ANALYSIS OF CRITICAL INSTABLE SPEED

The displacement vector of global model is $x=\{x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8, x_9, x_{10}\}^T$, where the displacement of upper bearing are x_1, x_2, the displacement of lower bearing are x_5, x_6, the displacement of water bearing are x_7, x_8,the displacement of rotor center are x_3, x_4,the displacement of turbine center are x_9, x_{10},the kinetic equation of system is shown as Eq.(10),

$$M_G\ddot{x}+C_G\dot{x}+K_Gx=F \tag{10}$$

$$F=\{f_{ubx}, f_{uby}, f_{erx}+f_{umpx}, f_{ery}+f_{umpy}, f_{dbx}, f_{dby}, f_{wbx}+f_{sx}, f_{wby}+f_{sy}, f_{etx}, f_{ety}\}^{T} \tag{11}$$

Where, f_{ub}， f_{db} and f_{wb} are force which is generated from the dynamic properties of three bearings, respectively, f_s is the sealing force. f_{er}, f_{et} stands respectively the quality eccentric force of rotor and turbine. There is no gravity on the horizontal direction because the hydraulic generating set is arranged in vertical.

The Eq.(12) can be obtained after moving load vector to the left of Eq.(10):

$$M_G\ddot{x}+C_G\dot{x}+K_Gx-F=0 \tag{12}$$

Substituting Eq.(6), (8), (9) into (12), yield:

$$M_{G0}\ddot{x}+C_{G0}\dot{x}+K_{G0}x=0 \tag{13}$$

The state equation of Eq.(13) can be generated:

$$f(x,\dot{x},\omega)=\begin{pmatrix}\ddot{x}\\ \dot{x}\end{pmatrix}=\begin{pmatrix}-M_{G0}^{-1}C_{G0} & -M_{G0}^{-1}K_{G0}\\ E & 0\end{pmatrix}\begin{pmatrix}\dot{x}\\ x\end{pmatrix} \tag{14}$$

When the nonlinear system which is presented by Eq.(14) is balanced, the acceleration and rotating speed equal zero. The solution vector x_0 of the Eq.(15) is the balance point of the system:

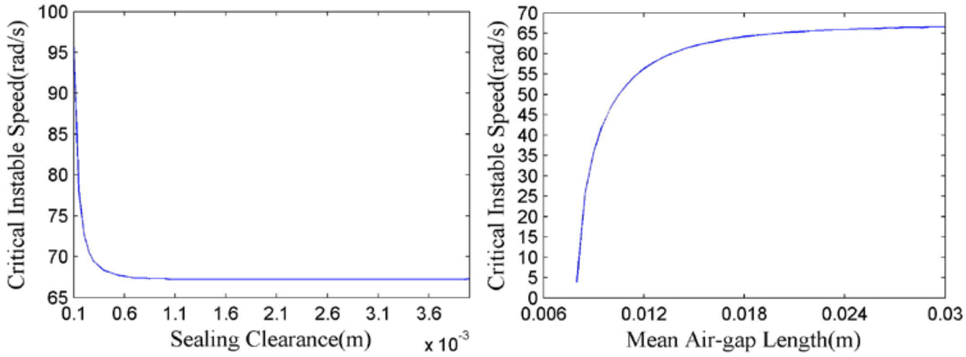
$$K_{G0}x=0 \tag{15}$$

The Jacobian equation can be derived from the linearization of nonlinear system using Taylor expansion on the equilibrium position, as shown in Eq.(16).

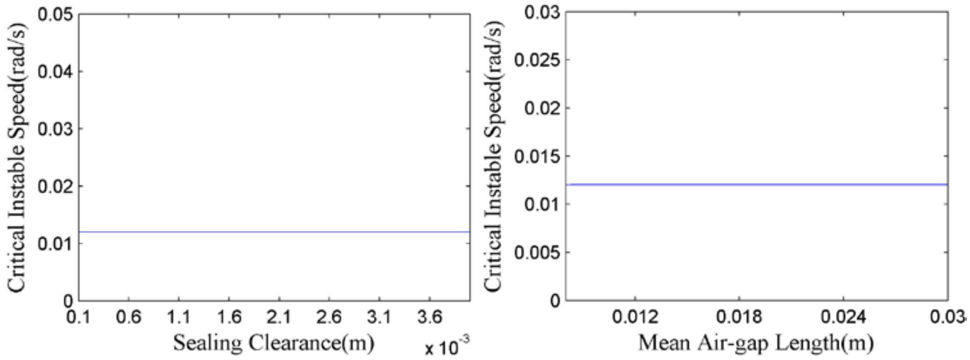
$$J=\left.\begin{pmatrix}\frac{\partial f_1}{\partial \dot{x}_1} & \cdots & \frac{\partial f_1}{\partial \dot{x}_{10}} & \frac{\partial f_1}{\partial x_1} & \cdots & \frac{\partial f_1}{\partial x_{10}}\\ \vdots & \ddots & \vdots & \vdots & \ddots & \vdots\\ \frac{\partial f_{10}}{\partial \dot{x}_1} & \cdots & \frac{\partial f_{10}}{\partial \dot{x}_{10}} & \frac{\partial f_{10}}{\partial x_1} & \cdots & \frac{\partial f_{10}}{\partial x_{10}}\\ & E_{10\times10} & & & 0_{10\times10} & \end{pmatrix}\right|_{(0,x0)} \tag{16}$$

Taking use of numerical algorithms, such as iterative method, the equilibrium Eq.(15) could be solved. After getting the numerical solutions, substituting them into Jacobian matrix, then the stability of system could be determined according to the eigenvalues of Jacobian matrix. According to the Lyapunov first approximation theory, the critical instable speed can be obtained[8].

As in Eq.(15), it is known that the vector x=0 is the only solution for equilibrium equations when substituting the vector into Jacobian matrix (16).



(a) The stiffness coefficients of bearing is a constant



(b) The stiffness coefficients of bearing is a variable

Fig.3 Critical Instable Speed Changed in Different Models

As a conventional method, the stiffness coefficients of three bearing are 5e8 N/m. Keep other parameters unchanged, as shown in Fig.3(a). With the increase of c_s, the system critical instable speed decreases firstly and then keep unchanged, as seen on the left of Fig.3(a). While the critical instable speed increased with the increasing of δ_0 continuously, as depicted on the right of Fig.3(a).In this paper, From the results depicted in Fig.3(b), the value of critical instable speed is approximately equal to zero, the increasing of c_s and δ_0 did not appeared to have any influence on the critical instable speed.

In actually, the initial pressure of bearing oil film is very small when the rotor has a very small initial eccentricity for the vertical shaft because there is no gravity acted on the rotor in horizontal direction. When the rotor speed is increased from a very low level, the unbalanced force such as sealing force and unbalanced magnetic pull began to form, but there is on support force to counteract, the rotor will leave the static stable position as soon as possible as the speed increasing.

4. ANALYSIS OF FIRST CRITICAL SPEED

When the rotor speed is considered as a variable, the Compell method is used to calculate the first critical speed of the shaft model. On this basis, bearing clearance c_b is selected to discuss its effect on the first critical speed, and the result is shown in Fig.4. As a comparison, a normal analysis in the past is illustrated. The rotor speed is considered as a constant in the calculation of the first critical speed, mostly is the rated speed (in this paper, it is 150r/min), and the results of the c_b-switched was shown in Fig.5.

Depending on the analysis of the effect of bearing parameters on critical instable speed and first critical speed, the main conclusions are as follows.

When the bearing is given an initial stiffness coefficient the critical instable speed is influenced by unbalanced magnetic pull and sealing clearance obviously. But when the dynamic properties of journal bearing was considered, the initial stiffness of bearing is very small, so the critical instable speed approximate to zero, and the two factors as mentioned above have almost no influence on the speed value.

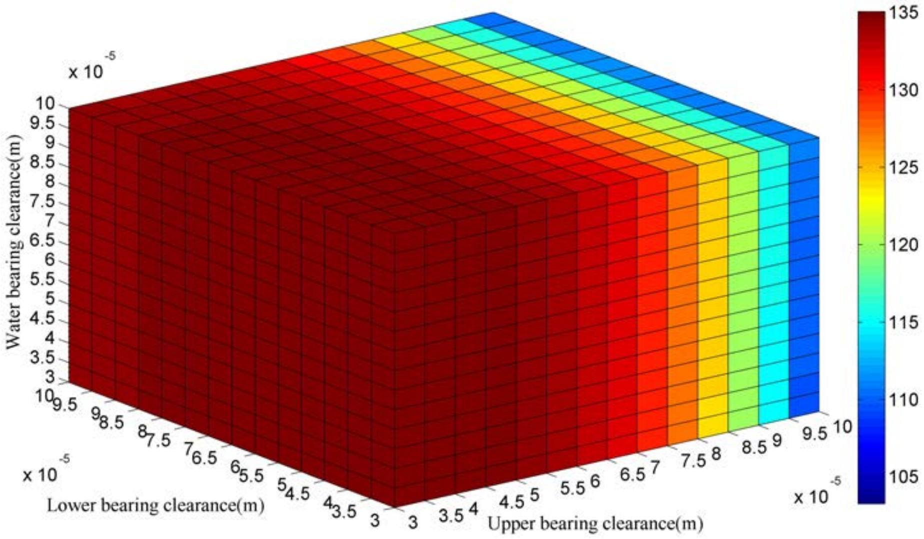


Fig. 4 First Critical Speed with Dynamic Stiffness Coefficients of Bearing

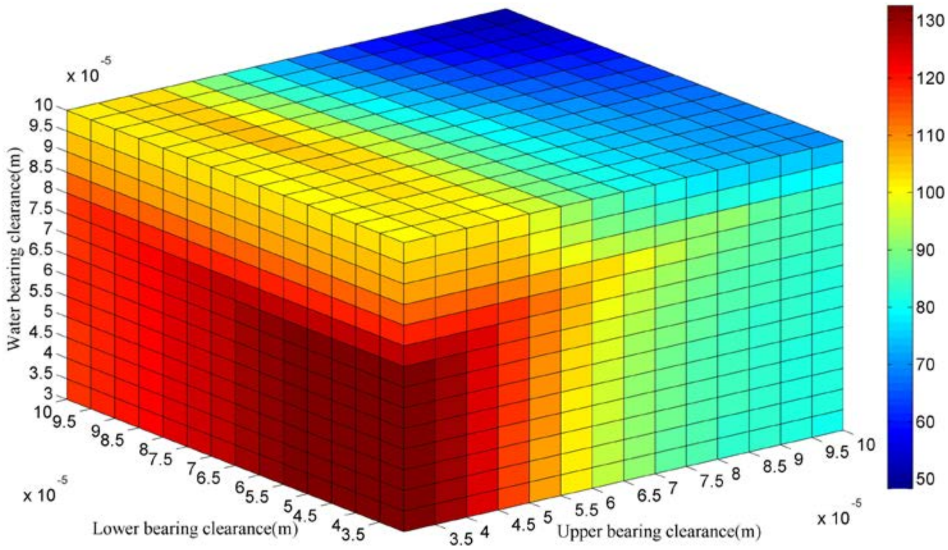


Fig.5 First Critical Speed with Invariable Stiffness Coefficients of Bearing

The nature frequency of bearing-shaft system would change with the rotor angular speed when the bearing dynamic characteristics are considered. So, if the bearing stiffness is a constant with a speed (such as rated rotor speed), the calculation of first critical speed is not precisely.

REFERENCE

[1] Bai Yannian. Design and Calculation of Hydro-electric Generating Set [M]. Beijing: China Machine Press, 1982.

[2] Lin Shengqiang.The effect of Oil film thickness of Sliding bearing on the Rotor stability

and Vibration [D]. Jilin: Northeast Dianli University, 2013.

[3] Song Zhiqiang. Research on Coupling Vibration Characteristics of Generator Set and Hydropower House [D]. Dalian: Dalian University of Technology, 2009.

[4] Ma Hongda. Stability Analysis of Rotor System in Hydro-generate Unit [D]. Tangshan: Hebei United University, 2012.

[5] Helio Fiori de Castroa, Katia Lucchesi Cavalca, et al. Whirl and whip instabilities in rotor-bearing system considering a nonlinear force model [J]. Journal of Sound and Vibration, Vol.317 (2008) ,Issues1-2, 273-294.

[6] A. Muszynska, D.E. Bently. Fluid-generated instabilities of rotors [J], Bently Nevada Corporation ORBIT, Vol.10 (1989), Issues 1, 6–14.

[7] Zhang Zhengsong, Mu Huaping. Limit cycle characteristics analysis of oil film instability whirl with Hopf bifurcation method [J]. Journal of Tsinghua University, Vol.36 (1996) No.7, 30-35.

[8] Wen Bangchun. Senior Rotor Dynamics [M]. Beijing: China Machine Press, 1999.

[9] Xu Yong. Dynamic Modeling and Simulation Analysis of the Shaft System for Large Hydro-turbine Generator Units [M]. Wuhan: Huazhong University of Science & Technology, 2012.

[10] Guo D, Chu F, Chen D. The unbalanced magnetic pull and its effects on vibration in a three-phase generator with eccentric rotor [J]. Journal of Sound and Vibration, Vol.254 (2002), Issues2, 297-213.

[11] A. Muszynska, D.E. Bently. Frequency-swept rotating input perturbation techniques and identification of the fluid force models in rotor/bearing/seal systems and fluid handling machines [J]. Journal of Sound and Vibration, Vol.143 (1990), Issues1, 103-124.