Effects of the Wet Steam Non-Equilibrium Condensation on the Dynamics of the Excitation-Relying Bearing-Rotor system

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Abstract. This paper investigated the effects of the wet steam non-equilibrium condensation on the dynamic characteristics of the bearing as well as the bearing-rotor system by constructing and analyzing a non-linear coupled model of the bearing-rotor system. An excitation-relying dynamic model of bearing is established based on the finite difference method, in which the excitation is converted from the pressure pulsation on the surface of rotor blades generated from the non-equilibrium condensation process. The Raccia transfer matrix method is implemented to analyse the dynamic behavior of this coupled system. Results show that the wet steam non-equilibrium condensation process would greatly reduce the bearing stiffness and damping and result in more intense vibration of the system, besides, its induced pulsed displacement would drive the excitation-relying bearing-rotor system more unstable.

1 Introduction

The instability of the bearing-rotor system would induce serious accidents of turbines. Especially for the wet steam turbine such as nuclear turbine and the last stage of low pressure gas turbine, the non-equilibrium condensation of wet steam (NECS) not only causes the pressure pulsation on the blade surface, but also affects the stiffness and damping of bearing by loading torque to the bearing. Therefore, it has great engineering significance to analyze the dynamics of the turbines bearing-rotor system considering the effects of NECS.

Currently, in the design of bearing, only the gravity of the rotor is regarded as the bearing support reaction force to determine the stiffness and damping of bearing, a lot of studies on this subject used the computational fluid mechanics or the finite difference method have been carried out [1-5]. However, the calculated bearing stiffness and damping using above approaches is insufficient to reflect the reality, which is the bearing stiffness and damping will change during the NECS process. The effects of the NECS on the dynamic characteristics of the bearing as well as the bearing-rotor system have not been clarified clearly.

The aim of this paper is to investigate such effects by constructing and analyzing a non-linear coupled model of the bearing-rotor system considering the effects of NECS. An excitation-relying dynamic model of bearing is established based on the finite difference method (FDM), in which the excitation is equivalent from the pressure pulsation on the surface of rotor blades generated from the

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NECS process. The Raccia transfer matrix method (RTMM) is applied to investigate the dynamic behavior of this coupled system under the NECS excitation.

2 Calculation models

Taking a low pressure turbine that contains ten cascades as an example, the excitation-relying bearing-rotor system working in wet steam as shown in Figure 1. Inertial coordinate system ($O-XYZ$), bearing-rotor system coordinates (o_s-xyz) and blade coordinate system ($o_i-x_iy_iz_i$) are defined to describe the coupled system.

Figure 1. Dynamic model of the excitation-relying bearing-rotor system

The differential equations of motion of the excitation-relying bearing-rotor system can be expressed as

$$[M_r]\ddot{U}(t) + \{[C_r] + [C_o(F_w, t)]\}\dot{U}(t) + ([K_r] + [K_o(F_w, t)])U(t) = F_w(t) \quad (1)$$

where M_r , C_r , K_r is the mass, damping and stiffness of the rotor (include the shaft and the blade). $F_w(t)$ is the NECS excitation. C_o and K_o are the excitation-relying damping and stiffness of the bearing, which can be deduced from the bearing support reaction force (i.e. the oil film force F_o) as given by

$$K_o = \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix} = \begin{bmatrix} \frac{\partial F_o \cos \Phi}{\partial x} & \frac{\partial F_o \cos \Phi}{\partial y} \\ \frac{\partial F_o \sin \Phi}{\partial x} & \frac{\partial F_o \sin \Phi}{\partial y} \end{bmatrix}; \quad C_o = \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} = \begin{bmatrix} \frac{\partial F_o \cos \Phi}{\partial \dot{x}} & \frac{\partial F_o \cos \Phi}{\partial \dot{y}} \\ \frac{\partial F_o \sin \Phi}{\partial \dot{x}} & \frac{\partial F_o \sin \Phi}{\partial \dot{y}} \end{bmatrix} \quad (2)$$

where, $\Phi = \theta + \varphi$ is the opening angle of the oil film, θ is the angle between the bearing concentric line and the oil film force. x , y is the NECS excitation induced offset of the bearing center (O_b). $F_o \cos \Phi$ and $F_o \sin \Phi$ are the components of the oil film force in x and y deriection, can be expressed as functions of the oil film pressure (p_o) and the radius of the shaft journal (r) as follows

$$F_o \sin \Phi = \int_{-L/2}^{L/2} \int_0^{2\pi} -p_o \sin \Phi r dr d\Phi dz; \quad F_o \cos \Phi = \int_{-L/2}^{L/2} \int_0^{2\pi} -p_o \cos \Phi r dr d\Phi dz \quad (3)$$

The change of the oil film force (F_o) will result in the alter of the thickness h and the pressure p_o of oil film, the motion of oil film is governed by the Reynolds equation [6] as follows

$$\frac{1}{r^2} \frac{\partial}{\partial \Phi} \left(\frac{h^3}{\mu} \frac{\partial p_o}{\partial \Phi} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial p_o}{\partial z} \right) = 6\omega \frac{\partial h}{\partial \Phi} + 12(\dot{y} \cos \Phi + \dot{x} \sin \Phi) \quad (4)$$

where, c and e are the radius gap and the eccentricity. μ and ω are the lubricant dynamic viscosity and the journal rotation angular velocity. The oil film force F_o can then be calculated by iterately solving Eqs.(3)-(4) use the FDM until the balance, which is the oil film force F_o is equal to the sum of the NECS excitation ($F_w(t)$) and the rotor gravity (F_G) as given by

$$F_o = F_w(t) + F_G \quad (5)$$

For a certain turbine, the rotor gravity can be easily calculated, however, the NECS excitation is hard to be obtained. The approach applied here is to equivalent the pressure (p) generated from the NECS on the blade surface into the torque (T) acting on the axis [6] (as shown in Fig.1(c)), and then, convert the torque (T) into the NECS excitation ($F_w(t)$) by introducing the torque factors ($\alpha_x=3.5$ and $\alpha_y=1.42$) [7-8], the detailed deduction given as follows. The torque acting on the axis can be expressed as the resultant force moment of the y_i direction forces of each blade (Eq.(6)) because the x_i direction forces can be offset over the full cascade.

$$T(t) = \sum_i^n \iint p_i dS_i \cos \beta_{yi} (d_i + R_d) \quad (6)$$

where, for the blade i , d_i is the distance between the blade centroid and hub centroid (o_i). R_d is the radius of the shaft. S_i is the surface area of blade. β_{yi} is the angle between the y_i of the blade coordinates $o_i-x_i y_i z_i$ and y of the bearing coordinates o_s-xyz . $n=10$ is the number of cascade in the wet steam turbine. p_i is the pressure on the blade surface which can be obtained from the numerical simulation of the NECS performed by ANSYS CFX based on the non-isothermal classical nucleation theory [9-10] and the droplet growth model [11]. Then, the NECS excitation $F_w(t)$ can be calculated as gave by

$$F_w(t) = T \sqrt{\alpha_x^2 + \alpha_y^2} \quad (7)$$

Finally, the excitation-relying damping and stiffness of the bearing can be calculated by substituting Eq.(7) into Eq.(5) and combining with Eqs.(2)-(4). Once the NECS excitation $F_w(t)$ as well as the bearing stiffness and damping $K_o(F_w, t)$ and $C_o(F_w, t)$ are determined, the dynamic characteristics of the coupled bearing-rotor system can be obtained by calculating Eq.(1) based use the RTMM.

3 Calculation results and discussion

Take the blade at the last stage as an example, the time-varying pressure on the blade surface at different blade spans during the NECS process is obtained as shown in Figure 2. It can be found that the pressure shows great unsteady characteristic in the middle and lower area of the blade (3%~50% span), this pulsation will gradually disappear at the blade tip (90% span).

Figure 2. Pressure on the blade surfaces at different spans

The NECS excitation ($F_W(t)$) can be obtained by substituting the pressure on the blade surface into Eqs.(6)-(7) as shown in Figure 3. It can be found that the NECS excitation in x direction is fluctuating between about $6.3\times10^5\sim7.3\times10^5$ N, and the NECS excitation in y direction is fluctuating between about $4.5\times10^5\sim5.2\times10^5$ N.

Figure 3. Components of the NECS excitation in x and y direction

Unsteady excitation-relying damping C_o and stiffness K_o of the bearing can be calculated use the FDM by substituting the NECS excitation $F_W(t)$ into Eq.(5) and combining with Eqs.(2)-(4). The eight components of the non-dimensional C_o and K_o are shown in Figure 4. Besides, in order to estimate the effects of NECS, the steady bearing stiffness and damping that calculated without the NECS excitation (i.e. $F_W(t)=0$) are shown in Table 1.

Figure 4. Components of the non-dimensional excitation-relying damping and stiffness of bearing

Table 1. Steady non-dimensional stiffness and damping of the bearing that calculated without the NECS excitation

k_{xx}	k_{xy}	k_{yx}	k_{yy}
0.0076	0.0246	0.049	0.019
c_{xx}	c_{xy}	c_{yx}	c_{yy}
0.0467	0.0021	0.0309	0.0974

The comparison between the bearing stiffness and damping calculated with and without the NECS excitation shows that most components of the excitation-relying damping and stiffness have great decrease (about 70%) except the k_{yy} which is increased by about 30%. Besides, the decreasing amplitude of the bearing stiffness is larger than the bearing damping, which means that the former is more sensitive to the NECS excitation.

Once the excitation-relying damping and stiffness of the bearing are obtained, the dynamic motion Eq.(1) can then be calculated by the RTMM. The axes contrails are drawn as shown in Figure 5 to show the stability of the bearing-rotor system.

Figure.5 Axes contrails of the bearing-rotor system. Case a: with the NECS excitation, Case b: without the NECS excitation.

The comparison between Case a and Case b shows that the displacement of the excitation-relying bearing-rotor system is almost four times than the non-relying bearing-rotor system in which the effects of the NECS excitation on the bearing are ignored. Furthermore, axes contrails of Case a is pulsating with about 10-4m during the rotation of the system, in which the pulsation displacement in x direction (P_x) is slightly more than in y direction (P_y). While axes contrails of Case b is steady during the rotation of the system. It can be concluded that when the effects of NECS excitation on the bearing are considered. The vibration response would greatly increase and the system has more tendency to instability.

In a sum, the NECS excitation would greatly reduce the bearing stiffness and damping and result in more intense vibration of the system. Besides, the pulsation displacement caused by NECS would drive the excitation-relying bearing-rotor system more unstable.

4 Conclusions

This paper constructed a non-linear coupled model of the excitation-relying bearing-rotor system considering the non-equilibrium condensation of wet steam to analyze the effects of wet steam excitation. An excitation-relying dynamic model of bearing was established based on the FDM, in which the excitation is converted from the pressure on the surface of rotor blades generated from the NECS process. The RTMM is applied to investigate the dynamic behavior of this coupled system. The following conclusions are draw:

(1) NECS could decrease the stiffness and damping of the bearing. In the calculated case shown in this paper, mosts components of the bearing stiffness and damping are decreased by about 70%.

(2) When the effects of NECS excitation on the bearing are considered, the vibration response would greatly increase and the system has more tendency to instability. In the calculated case shown in this paper, the vibration response of the excitation-relying bearing-rotor system is almost four times than the non-relying bearing-rotor system, and the pulsation displacement (about 10^{-4} m) was generated in the former.

Acknowledgment:

This work was supported by the key research and development program of Shandong Province (Grant No. 2017GGX40116)

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