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Stability and bifurcation analysis of rotor-bearing-seal system

G Y Ying[1] S L Liu[2] R Ma[3] S Y Zheng[3]

[1] Electric Power Scientific Research Institute of Zhejiang Province, State Grid Corporation of China, Hangzhou, China
[2] School of Mechanical and Automotive Engineering, Zhejiang University of Science and Technology, Hangzhou, China.
[3] Institute of Chemical Machinery, Zhejiang University, Hangzhou, China

E-mail: liushul21@aliyun.com
Zhengshuiying@zju.edu.cn

Abstract. Labyrinth seals were extensively used in turbine units, and the seal fluid forces may induce self-excited vibrations of rotor under certain conditions. It has become the main factor to instability of rotor system. In this paper Muszynska seal fluid force model is used to investigate the stability of the rotor system. Nonlinear equations are numerically solved by Newmark integration method. The effect of different seal clearances and differential pressures on system stability is studied. The calculation results show that the dominant vibration component leading to instability changes with different seal clearance. With the differential pressure increased, the unstable speed is reduced. Then the bifurcation behavior of the system with and without seal force is calculated. Results show that the rotor vibration becomes severe and complicated, and the bifurcation behavior of the system has been changed when seal force is considered.

1.Introduction

Labyrinth seals were extensively used in turbine units. The purpose of a seal in turbo-machinery is to prevent the leaking of gas from a high--pressure region to a low-pressure region and to maintain the pressure difference between components rotating at high speed. It's well known that the seal fluid forces may induce self-excited vibrations of rotor under certain conditions. With the pressure increasing of medium in turbine units the fluid excitation problems become increasingly prominent. It has become the main factor to instability of rotor system, even induce the destabilization. A lot of literatures have focused on seal fluid force models and its influence on rotor system, which generally propose that seal forces are a product of asymmetric pressure in seal caused by rotor deflection and the seal forces are linearized under the hypothesis of small disturbances. The force model is expressed by eight-parameters [1-3]. In small disturbance conditions the eight-parameter theory can be used to describe motion characteristics of the rotor system, but there is a lot limitation under large disturbance circumstance. It can not reflect the nonlinear characteristics of sealing force. The distributed pressure and fluid characteristics can be better acquired by simulating the flow inside labyrinth seal with the computational fluid dynamic (CFD) analysis [4-5], by the method it is crucial how to couple the gas excitation force into the rotor system. Muszynska seal fluid force model [6-7], not only inherits the characteristics of the

liushul21@aliyun.com

traditional eight parameters in certain degree, but also reflect some nonlinear characteristic of Alford force. So Muszynska seal fluid force model is mostly used to study the stability of the rotor system.

The Muszynska model of seal fluid forces is applied in this paper to investigate stability of rotor. There are two types of model parameters in the Muszynska model. Some parameters are related to seal structure and media parameters, others are experience parameters which are related to loss and obtained from the experiment or the experience. This paper will focus on the nonlinear bifurcation characteristics under the seal fluid forces, and the effect of Muszynska model parameters on the stability of rotor system is analyzed.

2 Seal Force Muszynska Model

According to Muszynska model, the nonlinear character of seal fluid force may be expressed as follow[6～8] in fixed reference coordinates X, Y.

$$\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = -\begin{bmatrix} K_s - m_f\tau^2\omega^2 & \tau\omega d \\ -\tau\omega D & K_s - m_f\tau^2\omega^2 \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{bmatrix} D & 2\tau\omega m_f \\ -2\tau\omega m_f & D \end{bmatrix}\begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} - \begin{bmatrix} m_f & 0 \\ 0 & m_f \end{bmatrix}\begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} \quad (1)$$

In Eq(1), K_s is the stiffness of rotor disturbed by fluid; D is the damping; m_f is the inertial effect. τ, K and D are the nonlinear function of disturbing displacements, x and y, respectively.

$K_s = K_0(1-e^2)^{-n}$ $\quad D = D_0(1-e^2)^{-n}$ $\quad \tau = \tau_0(1-e)^b$

$e = \sqrt{x^2+y^2}/c_s$, e is the relative eccentricity displacement of rotor, c_s is the seal clearance.

$$\boldsymbol{K}_0 = \mu_3\mu_0; \qquad D_0 = \mu_3\mu_1; \qquad m_f = \mu_3\mu_2 T$$

$$\mu_0 = \frac{2\sigma^2}{1+\xi+2\sigma}E(1-m_0); \qquad \mu_1 = \frac{2\sigma^2}{1+\xi+2\sigma}\left(\frac{E}{\sigma}+\frac{B}{2}\left(\frac{1}{6}+E\right)\right);$$

$$\mu_2 = \frac{\sigma}{1+\xi+2\sigma}\left(\frac{1}{6}+E\right); \qquad \mu_3 = \frac{\pi R\Delta P}{\lambda}$$

$$T = \frac{l}{v_a}; \qquad \sigma = \frac{\lambda l}{c};$$

$$E = \frac{1+\xi}{1+\xi+2\sigma}; \qquad B = 2 - \frac{(\frac{R_v}{R_a})^2 - m_0}{(\frac{R_v}{R_a})^2+1}$$

$$\lambda = n_0 R_a^{m_0}\left[1+(\frac{R_v}{R_a})^2\right]^{\frac{1+m_0}{2}}; \qquad R_v = \frac{Rc\omega}{\upsilon}; \quad R_a = \frac{2v_a c}{\upsilon}$$

n=0.5～3, $0<b<1$, $\tau_0<0.5$

τ_0 is the average circumferential speed ratio; ξ is the inlet port loss coefficient of gas flow; n, τ_0 , n_0, and m_0 are the experience parameter and determined by experiment and the seal structure; ΔP is the axial differential pressure; R is the radius of the seal; λ is the factor of friction; l is the seal length; va is the axial flow velocity in seal clearance; Ra is the axial flow Reynolds numbers; Rv is the circumferential flow Reynolds numbers; ω is the angular velocity of rotor, ν is the fluid viscosity coefficient.

3. Rotor-Bearing-Seal System

The numerical calculations were performed in the model, as illustrated in Figure.1 .The rotor is supported by two sliding bearings. The seal rig is designed in the left of the rotor. All the disks along the axle are in 170mm diameter, 25mm width. The diameter of the axle is 44mm, and the journal diameter is 32 mm. The whole rotor weights 37.02 kg. The sliding bearings are cylindrical bearings. And the short bearing theory is applied to calculate the oil forces.

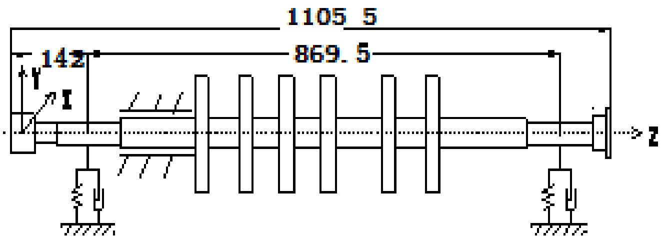


Figure.1 Rotor-bearing-seal model

The bearing parameters: width L = R, clearance c= 0.004×R, and unbalance located on the second disks. The seal construct parameters: seal diameter R=44mm, length l=130mm. Muszynska experience parameters by Child experiment are showed in Tab1 [9]. The axial flow velocity in seal clearance va is 5m/s; The fluid viscosity coefficient ν is 1.48e-5; The Differential pressure is 0.65MPa.

Table 1 Muszynska experience parameters

m_0	n_0	τ_0	n	b	ξ
0.24	0.079	0.28	2.11	0.3	0.093

4. Dynamic Equations of Rotor-Bearing-Seal System

The dynamic equations of rotor-bearing-seal system were built up by d'Alemdert principle, and a continuous shaft was treated as infinite thin disks that may be considered as rigid bodies. For simplification, the dynamic equations of system were treated by Rize way. So it is easy to deal with the integral and derivation of the dynamic equations for the three orders polynomial being selected as the location function. Because of the overlap of the three-order polynomial in the axial direction, not only the displacement and velocity but also the bending moment and transverse force are continuous. Then the nonlinear oil film forces based on steady short bearing model, unbalanced excitation. The dynamic equations of the rotor-bearing system derived in inertia coordinate system and could be written as:

$$\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{C}\dot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = \boldsymbol{B}_1\boldsymbol{F}_1 + \boldsymbol{B}_2\boldsymbol{F}_2 + \boldsymbol{B}_3\boldsymbol{F}_3 + \boldsymbol{B}_4\boldsymbol{F}_4 \quad (2)$$

Where $\boldsymbol{M}$ is a mass matrix; $\boldsymbol{C}$ and $\boldsymbol{K}$ are the damping matrix and stiffness matrix respectively; $\boldsymbol{F}_1$ is the unbalanced force; $\boldsymbol{F}_2$ is the gravity, F3 is the oil force; $\boldsymbol{F}_4 = [F_{gx}, F_{gy}]$ is the gas force; B_i are the location matrix of different forces.

Considering the change of the shaft diameter and the existence of additional mass, the shaft is divided into 47 portions. That is to say there are 47 position functions. Negligible axial vibrations, there are approximate 100 degrees of freedom in the system. For simplification purposes, it is necessary to decrease the degrees of freedom by modal transformation on the basis of accuracy assurance. Such system equations are local nonlinear and have high order. The equations are solved by Newmark integration method.

5. Results and discussion

5.1. Effect of Differential Pressure on Stability

The unstable speeds changing with differential pressure are shown in Figure.2 where the diameter clearance equals to 1.0 mm and eccentric distance located on the second disk is 50μm located. With the differential pressure increased, the unstable speed is reduced. When the differential pressure is small the unstable speed reduces very small with differential pressure changing, but when the differential pressure to a certain value the unstable speed drops acceleratly. The unstable speed changing with differential pressure shows obvious nonlinear characteristics. It indicates that the differential pressure is the main factor to affect the gas flow forces.

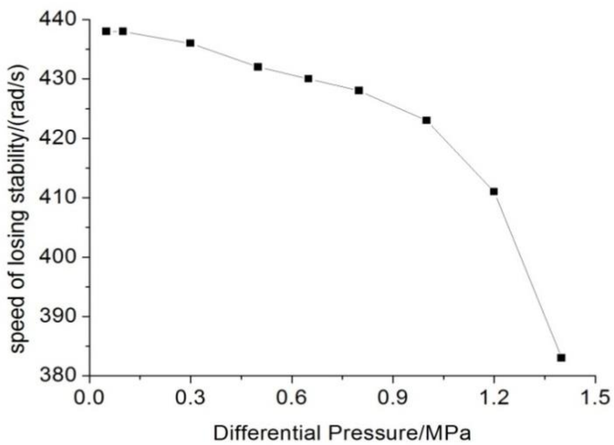
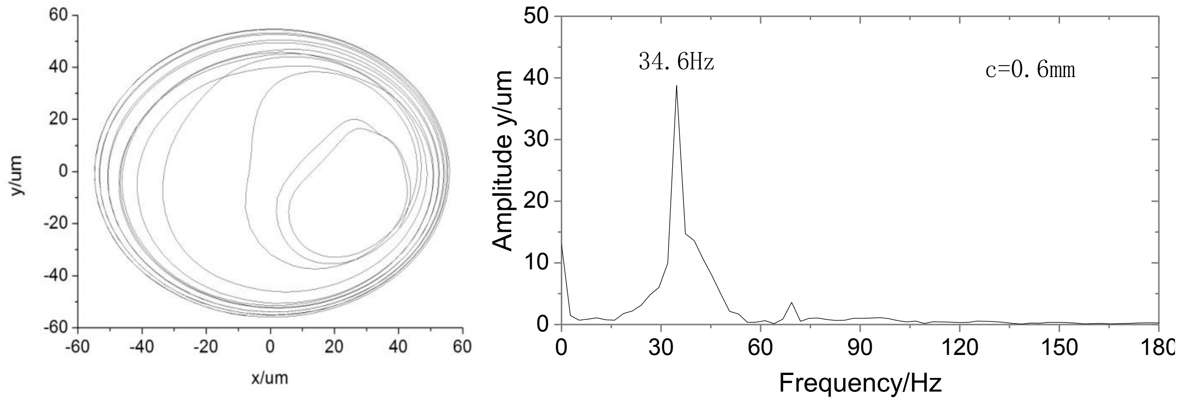


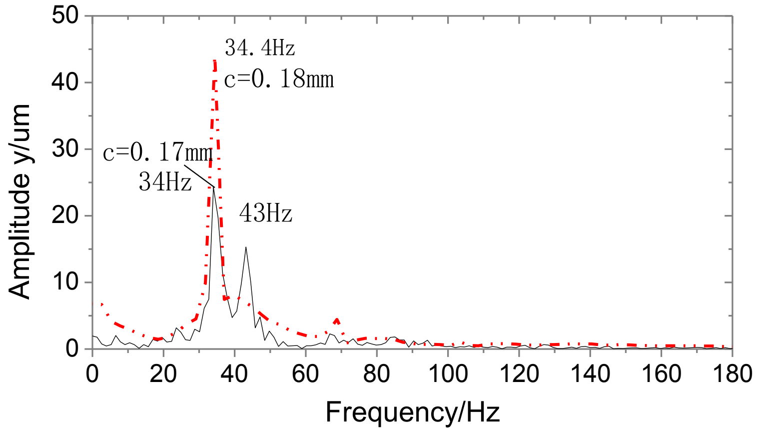
Figure.2 Effect of differential pressure on speed of losing stability

5.2. Effect of seal clearance on Stability

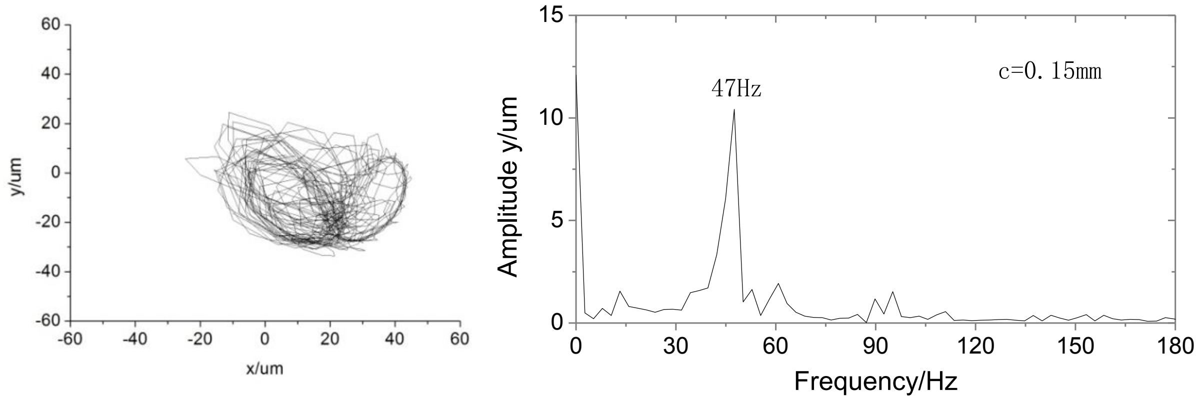
The effect of different seal clearances on system stability are shown in Figure.3 where the differential pressure equals to 0.65MPa and eccentric distance located on second disk is 50μm. When the seal clearance is 0.6mm, the threshold speed of stability is 429rad/s (Figure. 3(a)). It indicates that the trajectory of journal is already too large and amplitude of the half frequency vibration is much larger than that of the rotating frequency vibration. The half-frequency vibration plays the major role which is also known as oil whip. Changing the seal clearance value until to 0.18mm, the half-frequency vibration is still dominant (Figure. 3(b)). When the seal clearance is as small as 0.17mm, the threshold speed of stability is 421rad/s, but the vibration becomes complicated, i.e. there are two kinds of frequency vibrations (Figure. 3(b)). When the seal clearance continuously reduces to 0.15mm, the threshold speed of stability drops to 290rad/s. The working-frequency vibration now dominates. The vibration characteristics are quite different.



(a)The trajectory and spectrum(c=0.6m, rotating speed=429rad/s)



(b) The spectrum(c=0.17m rotating speed=421rad/s and c=0.18m rotating speed=426rad/s)

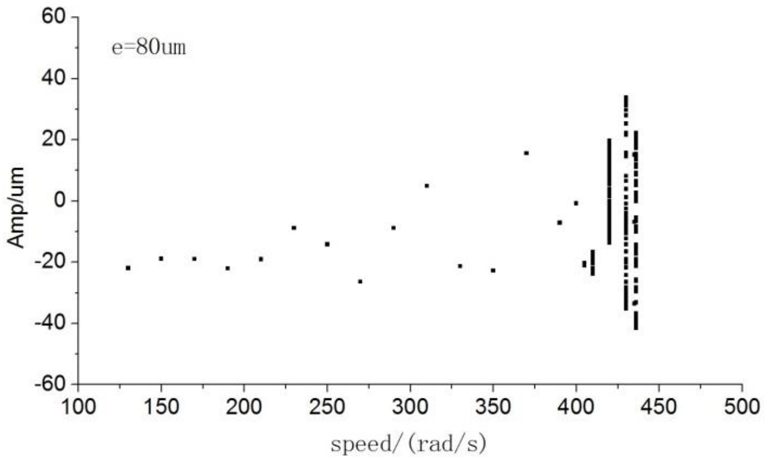


(c)The trajectory and spectrum(c=0.15m, rotating speed=290rad/s)

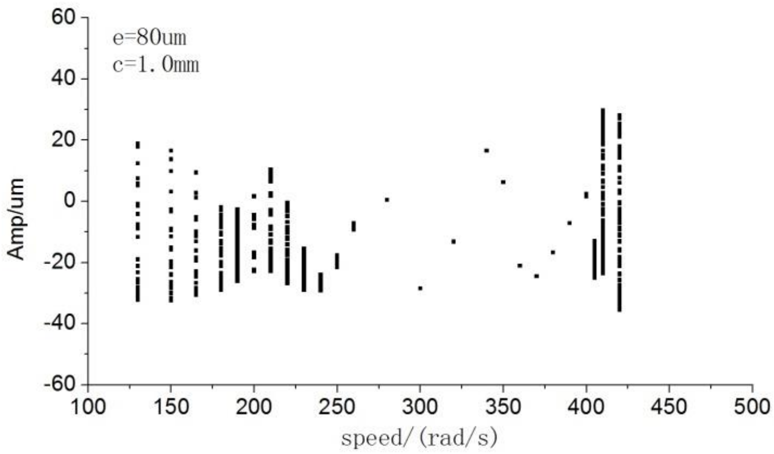
Figure.3 The trajectory and spectrum under different seal clearance

5.3. Nonlinear bifurcation characteristic with and without the seal force

In order to illuminate the relation between rotation speed and nonlinear dynamic characteristic of rotor with and without the seal force, and the bifurcation map of the right journal displacement in the vertical direction y is plotted(Figure.4 where the diameter clearance equals to 1.0mm, eccentric distance located on the second disk is to 80μm). Figure.4(a) reveals that the rotor first experiences periodic vibration, then comes into quasi-periodic vibration and finally lost stability when there are no seal force on the rotor system. At the same condition if the seal force is considered, Figure.4(a) shows that the rotor come from quasi-periodic vibration into periodic vibration, then returns to quasi-periodic vibration again eventually losing stability. The unstable speed with the seal force is lower. The calculations show that the rotor vibration becomes severe and complicated when there is the seal force acting on the rotor system, and often becomes quasi-periodic.



(a) Bifurcation map of the vertical direction displacement without seal force



(b) Bifurcation map of the vertical direction displacement after seal force

Figure.4 Bifurcations map of the right bearing displacement in vertical direction y varying with speed

6. Conclusions

(1) The calculations show that the rotor vibration becomes severe and complicated when there is the seal force acting on the rotor system, and often become quasi-periodic. The bifurcation behavior of the system has been changed when seal force is considered.

(2)With the differential pressure increased, the unstable speed is reduced. The differential pressure is the main factor to affect the gas flow forces.

(3) The seal clearance has much influence on vibration characteristics. When the seal clearance is large the half-frequency vibration plays the major role in inducing to instability, but when the seal clearances reduces to a certain value, the characteristic of instability is dominated by the working frequency vibration.

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