Parameter Identification for Stiffness and Damping in AMB-Flexible Rotor System

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Abstract

To describe the support characteristic of an Active Magnetic Bearing(AMB) in rotor system via the traditional rotor dynamic theory, the equivalent stiffness and the equivalent damping are the frequently-used and important parameters. Some methods have been proposed for measuring the support parameters of AMB. However, most of them were based on the single degree of freedom(DOF) model or the multi-DOF rigid rotor model. In this study, a measuring method is proposed to identify the equivalent stiffness and the equivalent damping of the rotor radial support, which is employed in the flexible rotor system levitated on AMB. By means of the condensation of DOF in finite element rotor model, which includes the static force equivalence or the dynamic force equivalence, the whole rotor DOFs can be reduced to some certain DOFs and the number of the unknowns in the condensed model is decreased. The relation between the excitation and the response is extracted from the condensed rotor model. The identification equations are established in condition of the AMB excitation. By numerical simulation, the parameters of the equivalent stiffness and the equivalent damping of the AMB support are effectively identified and the possible problems are analyzed.

Keywords : flexible rotor; active magnetic bearing; condensation of degrees of freedom; equivalent stiffness; equivalent damping

1. Introduction

Active Magnetic Bearing(AMB) is a slowly emerging technology with tremendous potential for a variety of rotating applications with high performance. The AMB can not only provide the contactless support for the rotor without mechanical friction, but also offer possible solutions for applying a real-time electromagnetic force to regulate the rotor dynamic behaviors and control the rotor vibration actively.

To describe the support characteristic of AMB in a rotor system via the traditional rotor dynamic theory, the equivalent stiffness and the equivalent damping are the frequently-used and important parameters, which are decided by the AMB size as well as its control strategy and control parameter. Because of the complexity and diversity of control, it is difficult to make certain the parameters of the equivalent stiffness and the equivalent damping of AMB, which can not be simply evaluated by theoretical calculation.

In recent years, there have been a great number of investigations into measuring and analyzing the AMB support characteristic. Many methods for measuring the equivalent stiffness and the equivalent damping of AMB have been proposed. Tsai et al. [1] applied the wavelet transform algorithm to identify the magnetic damping and magnetic stiffness coefficients of an AMB system and analyzed their nonlinear order. Additionally, he found that the identified damping coefficient maybe negative and hence implies that under specific displacement and speed, the dynamic of the AMB system is unstable. Lim et al. [2] identified the equivalent stiffness and the equivalent damping for the AMB with the PID control strategy and analyzed the change rule of the support characteristic under the different PID parameters. Furthermore, Lim et al. [3] also presented the parameter identification with multi-frequency excitation, in which the Schroeder Phased Harmonic Sequence (SPHS) is

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adopted to avoid the magnetic force saturation due to the superimposing peak of multi-frequency component. Bauomy et al. [4] investigated the response of AMB-rotor system subjecting to a periodically time-varying stiffness. The stability of system near the simultaneous combined and sub- harmonic resonance was analyzed by the 4th Rung–Kutta method. Sayyad et al. [5] presented the use of a variable stiffness type magnetic vibration absorber to control the vibration of a beam structure. Its principle is similar to the rotor vibration adjusting of AMB with variable stiffness in single degree of freedom(DOF). Jiang et al. [6,7,8] proposed the identification methods for the equivalent stiffness and the equivalent damping of AMB under three conditions, which are based on the single DOF levitation model, the multi-DOF rigid rotor model and multi-frequency excitation, respectively.

There are a few researches for the measurement of AMB support parameter in the case of the flexible rotor system. Tiwari et al. [9,10] estimated the AMB dynamic parameters along with inherent unbalances in a flexible rotor that is fully levitated by AMBs. The number of unknowns in the identification model is deduced by the static force equivalence. Sun et al. [11] presented an identification method for an AMB system with a flexible rotor, which adopted the measured frequency-response model and the transfer function matrix model to decompose the identification procedure into a few steps and estimate its unknown parameters.

2. Problem Formulation

Several aforementioned studies have been devoted to the elaboration of the method to measure the stiffness and damping of AMB. However, most of them were based on the single degree of freedom(DOF) model or the multi-DOF rigid rotor model. In this study, the attention will be focused on the parameter identification of the equivalent stiffness and the equivalent damping of AMB, which is employed in the AMB-flexible rotor system. Compared with rigid rotor system, the parameter identification would be done more difficultly in a flexible rotor system.

One difficulty is, even though a practical flexible rotor can be discretized by the finite element modeling (FEM), which possesses a finite number of DOFs. It will result in great difficulty to solve the model equations because of the large number of unknowns.

The other is, for a rigid rotor, the suspending rotor posture is ascertained only by the displacement coordinates of any two positions in rotor. However, it is impossible to measure the whole posture of a flexible rotor, but the finite number of node displacements in the flexible rotor.

In this study, by the condensation of DOF in rotor finite element model, the whole DOFs can be reduced to some certain DOFs. The relation between the excitation and the response in AMB DOFs is extracted from the condensed rotor model. The identification equations are established and the expected parameters of AMB support are solved out.

3. Identification Process

The proposed method for parameter identification involves four procedures.

3.1 Procedure 1: FEM of flexible rotor system

By the finite element method, which is commonly used for modeling of flexible rotor system, a practical rotor with the irregular shape is discretized into the smaller elements, such as elastic shaft section, rigid disk, bearing support, lumped mass, et al. The appropriate number of elements is determined depending on the order of vibration modes expected to be known and geometry of the shaft and mounting of discs. The motion of the practical rotor in infinite number of DOFs is simplified as the node motions in finite number of DOFs.

All motion equations from each sub-elements are reassembled to get the global equations of motion, which are gived as

$$[M_{\text{global}}]\ddot{u}_{\text{global}} + \left([D_{\text{global}}] - \omega[G_{\text{global}}]\right)\dot{u}_{\text{global}} + [K_{\text{global}}]u_{\text{global}} = f_{\text{global}} \tag{1}$$

$$u_{\text{global}} = \begin{bmatrix} u_{\text{global},x} \\ u_{\text{global},y} \end{bmatrix}, \quad u_{\text{global},x} = \begin{bmatrix} u_{x1} \\ u_{x2} \\ \cdots \\ u_{xN} \end{bmatrix} = \begin{bmatrix} x_1 \\ \theta_{y1} \\ x_2 \\ \theta_{y2} \\ \cdots \\ x_N \\ \theta_{yN} \end{bmatrix}, \quad u_{\text{global},y} = \begin{bmatrix} u_{y1} \\ u_{y2} \\ \cdots \\ u_{yN} \end{bmatrix} = \begin{bmatrix} y_1 \\ -\theta_{x1} \\ y_2 \\ -\theta_{x2} \\ \cdots \\ y_N \\ -\theta_{xN} \end{bmatrix}$$

where, u_{global} and f_{global} are called the displacement vector and force vector of all nodes in the rotor. x and y are the moving displacement of translational DOF in direction x and y, θ_x, θ_y are the angular displacement of rotational DOF around axis x and axis y. The subscripts $1 \sim N$ are the node numbers. $[M_{\text{global}}]$, $[D_{\text{global}}]$, $[G_{\text{global}}]$ and $[K_{\text{global}}]$ are the global mass, damping, gyroscopic and stiffness matrices of the whole rotor, respectively. For the detail of the FEM for flexible rotor, please see the textbook about rotor dynamics. It will not be covered again here for limited paper space.

3.2 Procedure 2: modeling of AMB support

The AMB support is denoted by the classical stiffness-damping model. Without consideration of cross stiffness and damping, the supporting forces at each AMB are assumed of the following form,

$$\begin{bmatrix} kx_1 & & & & \cdots & 0 \\ & \cdots & & & & \vdots \\ & & kx_i & & & \\ & & & ky_1 & & \\ \vdots & & & & \cdots & \\ 0 & \cdots & & & & ky_i \end{bmatrix} \begin{bmatrix} u_{amb1,x} \\ \cdots \\ u_{ambi,x} \\ u_{amb1,y} \\ \cdots \\ u_{ambi,y} \end{bmatrix} + \begin{bmatrix} dx_1 & & & & \cdots & 0 \\ & \cdots & & & & \vdots \\ & & dx_i & & & \\ & & & dy_1 & & \\ \vdots & & & & \cdots & \\ 0 & \cdots & & & & dy_i \end{bmatrix} \begin{bmatrix} \dot{u}_{amb1,x} \\ \cdots \\ \dot{u}_{ambi,x} \\ \dot{u}_{amb1,y} \\ \cdots \\ \dot{u}_{ambi,y} \end{bmatrix} = f_{amb} \tag{2}$$

Where, u_{amb} and f_{amb} are the displacement vector and supporting force vector in AMB DOF. The subscripts 1-i are the serial numbers of AMB; kx, ky, dx and dy *are* the equivalent stiffness and the equivalent damping in the corresponding DOF of AMB support, which just are what we want to solve out.

By the Laplace transform, the frequency domain equations of Eq.(2) can be obtained as

$$\begin{bmatrix} kx_1 + j\omega dx_1 & & & & \cdots & 0 \\ & \cdots & & & & \vdots \\ & & kx_i + j\omega dx_i & & & \\ & & & ky_1 + j\omega dy_1 & & \\ \vdots & & & & \cdots & \\ 0 & \cdots & & & & ky_i + j\omega dy_i \end{bmatrix} \begin{bmatrix} U_{amb1,x} \\ \cdots \\ U_{ambi,x} \\ U_{amb1,y} \\ \cdots \\ U_{ambi,y} \end{bmatrix} = F_{amb} \tag{3}$$

$$Z_{kc}\ U_{amb} = F_{amb}$$

Where, U_{amb} and F_{amb} are the frequency domain transform of the displacement vector and supporting force vector. Z_{kc} is a diagonal matrix.

If the cross stiffness and damping are considered, the Z_{kc} in Eq.(2) will not be the diagonal matrix, but the following form

$$\begin{bmatrix} kxx_1 + j\omega dxx_1 & & & kxy_1 + jdxy_1 & \cdots & 0 \\ & \cdots & & & & \vdots \\ & & kxx_i + j\omega dxx_i & & & kxy_i + jdxy_i \\ kyx_1 + jdyx_1 & & & kyy_1 + j\omega dyy_1 & & \\ \vdots & & & & \cdots & \\ 0 & \cdots & kyx_i + jdyx_i & & & kyy_i + j\omega dyy_i \end{bmatrix} \begin{bmatrix} U_{amb1,x} \\ \cdots \\ U_{ambi,x} \\ U_{amb1,y} \\ \cdots \\ U_{ambi,y} \end{bmatrix} = F_{amb} \tag{4}$$

$$Z_{kc}\ U_{amb} = F_{amb}$$

3.3 Procedure 3: Condensation of Degrees of Freedom

3.3.1 Dividing Degrees of Freedom

The FEM of flexible rotor has a great number of the DOFs. Some non-critical DOFs may be reduced in order to decrease the number of unknowns in motion equation, as well as overcome the limitation of number of measurements that can be made in practical rotors. Therefore, the all rotor DOFs will divide into two parts as the following principle. The master DOF will be retained and the slave DOF will be discarded or, it might be said, condensed into the interior of the model.

1: All of the rotational DOFs will be regarded as the slave DOF, because their displacements are unknowns in general. It is difficult to measure the angle displacement of the rotational DOFs.

2: For parameter identification of AMB support, the attention will be focused on the relation between the AMB exciting force and AMB nodes vibration. Therefore, the translational DOFs in AMB nodes should be designate as the master DOFs. All other translational DOFs also are the slave DOFs.

3: If the master DOFs are few, it is consequently that the order of the condensed model will be too low to satisfy the analysis of the higher order mode. If need be, the sensors for detecting displacement would be added so that the order of the condensed model can be increased. Then, the added DOFs with displacement detecting would be regarded as the master DOFs. The displacement of the master DOF must be measurable and known.

To simplify analyzing, it is considered that the position where the magnetic force is applied by AMB winding is in the same node of the position where the displacement is detected by AMB sensor.

3.3.2 Rotor Motion Equation With AMB Excitation

For dividing of DOF, Eq.(1) is split as subvectors and submatrices relating to the master DOFs and the slave DOFs and can be represented as

$$\begin{bmatrix} M_{mm} & M_{ms} \\ M_{sm} & M_{ss} \end{bmatrix}\begin{bmatrix} \ddot{u}_{\mathrm{m}} \\ \ddot{u}_s \end{bmatrix} + \left(\begin{bmatrix} D_{mm} & D_{ms} \\ D_{sm} & D_{ss} \end{bmatrix} - \omega \begin{bmatrix} G_{mm} & G_{ms} \\ G_{sm} & G_{ss} \end{bmatrix}\right)\begin{bmatrix} \dot{u}_{\mathrm{m}} \\ \dot{u}_s \end{bmatrix} + \begin{bmatrix} K_{mm} & K_{ms} \\ K_{sm} & K_{ss} \end{bmatrix}\begin{bmatrix} u_{\mathrm{m}} \\ u_s \end{bmatrix} = \begin{bmatrix} f_{\mathrm{amb}} + f_{imp} \\ 0 \end{bmatrix} \tag{5}$$

Where, the subscripts of m and s represent the master DOFs and the slave DOFs; u_{m}, u_s represent the displacements of the master DOFs and the slave DOFs respectively. Here, f_{imp} is the AMB excitating force(expressed in Newton) which is known; f_{amb} is the AMB supporting force(expressed by stiffness and damping) which is unknown and just need identifying. f_{imp} and f_{amb} are both in the master DOFs in the case of AMB Excitation. The sequence of DOF is rearranged and the global mass, damping, gyroscopic and stiffness matrices are split into four submatrices as shown in Eq.(5).

3.3.3 Static Condensation of DOF

According to the rotor deformation under a static force, the static force equivalence is adopted to eliminate the slave DOFs from the finite element equations. The static force equations of Eq.(5) are

$$\begin{bmatrix} K_{mm} & K_{ms} \\ K_{sm} & K_{ss} \end{bmatrix}\begin{bmatrix} u_{\mathrm{m}} \\ u_s \end{bmatrix} = \begin{bmatrix} f_{\mathrm{amb}} + f_{imp} \\ 0 \end{bmatrix} \tag{6}$$

The second row of Eq.(6) is

$$u_s = -K_{ss}^{-1} K_{sm} u_{\mathrm{m}} \tag{7}$$

Therefore, the following relation can be obtained,

$$\begin{bmatrix} u_{\mathrm{m}} \\ u_s \end{bmatrix} = \begin{bmatrix} \mathrm{I} \\ -K_{ss}^{-1} K_{sm} \end{bmatrix} u_{\mathrm{m}} = \mathrm{T} u_{\mathrm{m}} \tag{8}$$

Where, T is just the transform matrix between the master DOFs and the slave DOFS.

Eq. (5) is rebuilt by substitution of Eq. (8). Meanwhile, for the regular equation form, both sides of equation are multiplied by the transform matrix T,

$$[M]\ddot{u}_m + ([D] - \omega[G])\dot{u}_m + [K]u_m = f_m \tag{9}$$

Where, $[K] = \mathrm{T}^{\mathrm{T}}\begin{bmatrix} K_{mm} & K_{ms} \\ K_{sm} & K_{ss} \end{bmatrix}\mathrm{T}$, $[D] = \mathrm{T}^{\mathrm{T}}\begin{bmatrix} D_{mm} & D_{ms} \\ D_{sm} & D_{ss} \end{bmatrix}\mathrm{T}$, $f_m = \mathrm{T}^{\mathrm{T}}\begin{bmatrix} f_{\mathrm{amb}} + f_{imp} \\ 0 \end{bmatrix}$, $[M] = \mathrm{T}^{\mathrm{T}}\begin{bmatrix} M_{mm} & M_{ms} \\ M_{sm} & M_{ss} \end{bmatrix}\mathrm{T}$, $[G] = \mathrm{T}^{\mathrm{T}}\begin{bmatrix} G_{mm} & G_{ms} \\ G_{sm} & G_{ss} \end{bmatrix}\mathrm{T}$

In Eq.(9), it is visible that the motion equation of rotor is expressed only by the master DOFs. The slave DOFs are eliminated and condensed into the interior of the model.

3.3.4 Dynamic Condensation of DOF

If the dynamic force is considered, Eq.(6) will be added by damping term and inertia term, which are the first order differential term and the second order differential term of the rotor displacement, respectively. The dynamic condensation should be implemented in frequency domain. Then, Eq.(6) is transformed into

$$\begin{gathered}\left(-\omega^2[M_{\text{global}}]+j\omega\left([D_{\text{global}}]-\omega[G_{\text{global}}]\right)+[K_{\text{global}}]\right)U_{\text{global}}=F_{\text{global}}\\ ZU_{\text{global}}=F_{amb}+F_{imp}\end{gathered} \tag{10}$$

Where, $Z=-\omega^2[M_{\text{global}}]+j\omega\left([D_{\text{global}}]-\omega[G_{\text{global}}]\right)+[K_{\text{global}}]$; U_{global} and F_{global} are the frequency domain transformation of u_{global} and f_{global} , respectively.

Similarly, Eq.(10) also is split as subvectors and submatrices relating to the master DOFs and the slave DOFs as follows,

$$\begin{bmatrix} Z_{mm} & \mid & Z_{ms} \\ - & + & - \\ Z_{sm} & \mid & Z_{ss} \end{bmatrix}\begin{bmatrix} U_m \\ - \\ U_s \end{bmatrix}=\begin{bmatrix} F_{amb}+F_{imp} \\ - \\ 0 \end{bmatrix} \tag{11}$$

Provided that the number of the master DOFs is “a” and the slave is “b”, Eq(10) can be deduced as

$$\begin{gathered}\{Z_{mm}U_m\}_{a\times 1}+\{Z_{ms}U_s\}_{a\times 1}=\{F_{amb}+F_{imp}\}_{a\times 1}\\ \{Z_{sm}U_m\}_{b\times 1}+\{Z_{ss}U_s\}_{b\times 1}=\{0\}_{b\times 1}\end{gathered} \tag{12}$$

Where, the braces{ } and the subscript represent the size of the matrix.

From the second row of Eq.(12), the transform relation between master DOFs and slave DOFS can be obtained in condition of dynamic force,

$$U_s=-Z_{ss}^{-1}Z_{sm}U_m \tag{13}$$

Eq.(13) is similar to Eq.(7) in form.

Then, the frist row of Eq.(12) is rebuilt by substitution of Eq.(13) to eliminate the slave DOFs.

$$\{Z_{mm}-Z_{ms}Z_{ss}^{-1}Z_{sm}\}_{a\times a}\{U_m\}_{a\times 1}=\{F_{amb}+F_{imp}\}_{a\times 1} \tag{14}$$

Eq.(14) is the rotor motion equation in frequency domain with dynamic condensation of DOF.

3.4 Procedure 4: Creating Identification Equations

In Eq.(2), the displacements of the AMB nodes are replaced by the master DOFs,

$$F_{amb}=Z_{kc}U_m \tag{15}$$

In the condition of AMB excitation, Eq.(14) is rebuilt by substitution of Eq.(15) to create the identification equations as follows

$$\begin{gathered}(Z_{mm}-Z_{ms}Z_{ss}^{-1}Z_{sm}-Z_{kc})\ U_m=F_{imp}\\ (Z'_{mm}-Z_{kc})\ U_m=F'_m\\ Z_{kc}U_m=Z'_{mm}U_m-F'_m\end{gathered} \tag{16}$$

Where, $Z'_{mm}=Z_{mm}-Z_{ms}Z_{ss}^{-1}Z_{sm}$, $\quad F'_m=F_{imp}$.

Eq.(16) is the identification equation under AMB excitation. It expresses the relation of excitation, response and identification parameter. In the identification equation, Z'_{mm}, F'_m and U_m all are known. It is desired that Z_{kc} would be identified and solved out.

4. Simulation

4.1 Introduce for Simulation example

The simulation example of AMB-rotor system is composed of five elastic shaft sections with different diameter, three rigid discs and two AMBs, whose structure and size are shown in Fig.1. The out diameters of the rigid discs are 250mm and the thickness 30mm; the rotor parts of AMB are also look as the rigid disc, whose out diameters are 200mm and the thickness 80mm; By finite element modeling, the rotor system is divided into 11 nodes. Two AMBs are located in the No.4 and No.8 nodes respectively. There are 44 DOFs in the global rotor model. Each shaft section is modeled according to the Timoshenko beam mode.

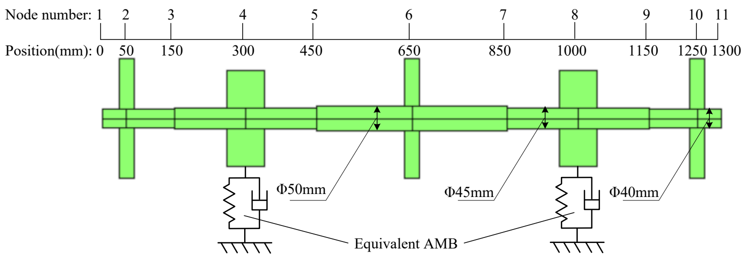

Fig.1 Structure and Size of Simulation Example for AMB-Rotor System

4.2 Simulation for PD AMB control

In order to verify the validity of parameter identification for rotor supporting, the PD control is applied in AMB at first. It is well known that a practical AMB system seldom adopts the pure PD control without integral and time delay. However, to adopt the PD control would bring about that the equivalent stiffness and the equivalent damping are both constants, which is more intuitive to check up the identification results and discover the potential mistakes from the proposed method.

The main parameters for simulation include that, the current stiffness coefficient is 480N/A, the displacement stiffness coefficient is $2\times10^{6}\,\mathrm{N/M}$; the proportional coefficient of AMB control is 2, the differential is 0.001; the gain of whole control loop is 3125. According to the aforementioned parameter, by numerical evaluation, the result of the equivalent stiffness should be $1\times10^{6}\,\mathrm{N/M}$, the equivalent damping 1500 $\mathrm{N.s/M}$. Both of them are constants.

The AMB magnetic excitation is imposed from the left AMB with the rotational vector. The rotor unbalance force can be simulated by the rotational vector excitation, namely, applying the sinusoidal exciting force into the *x* direction of rotor motion and the cosine the *y* direction. The amplitude of rotational vector excitation is proportional to rotational speed squared. The equivalent mass-radius product is *em*= $20\times10^{-3}\,\mathrm{kg}\cdot\mathrm{m}$ and its phase is in the positive direction of *y* axis.

In the range of the excitation angular frequency 1~1000rad/s, the amplitude-frequency characteristic of the rotor vibration in the positions of the left and right AMBs are carefully measured and plotted in Fig(2) and Fig(3), respectively.

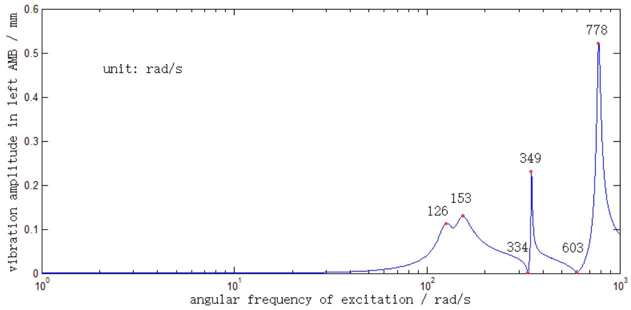

Fig.2 vibration response of *x* direction in position of left AMB

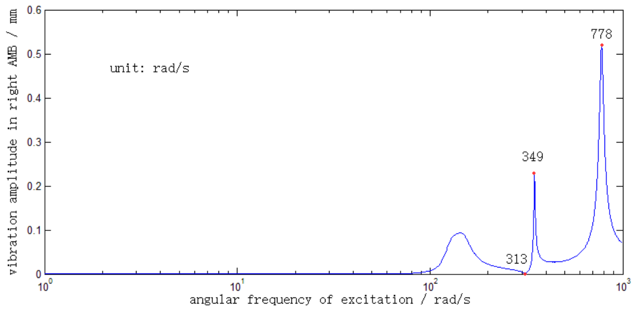

Fig.3 vibration response of *x* direction in position of right AMB

In the range of 1~1000rad/s, it can be observed that there are four vibration amplitude peaks in Fig.2, which are located in 126, 153, 349 and 778 rad/s (In aftermentioned contents, the units of angular frequency are all rad/s). According to the rotor dynamic theory, they respectively correspond with the critical frequency (resonance frequency) of the cylindrical mode, the conical mode, the first order bending mode and the second order bending mode of rotor motion. Because the AMB excitation is imposed by single side, namely, only by the left AMB, the cylindrical mode is inconspicuous relatively, so that the unobvious vibration peak of cylindrical mode in the right AMB as shown in Fig.3. Affirmatively, the peak positions of the first order bending mode and the second order bending mode in the right AMB are consistent with that of the left AMB.

According to the proposed method, the identification results of the equivalent stiffness and the equivalent damping are solved out and shown as from Fig.4 to Fig 7. The following analyses are for identification results in x direction, that of y direction is exactly the same.

It is clear that the identification results of the equivalent stiffness and the equivalent damping are not what we expected. In theory, the equivalent stiffness should be 1×10^{6} N/M and the equivalent damping 1500 N.s/M . In practice, in relatively low frequency, the identification results are correct; however, as the frequency increases, there are some sudden changes of the identification results in relatively high frequency, as shown in Fig4. and Fig.5.

By means of checking data, it is found that the sudden changes of the stiffness identification results of the left AMB occur at four frequencies of 328, 334, 482, [592,611] and that of the right AMB at three frequencies of 313, 328, 482. Among them, at 328 and 482, the sudden changes simultaneously occur in two AMBs position; At other frequencies of 334, [592,611] and 313, the sudden changes occur in only one AMB, left or right.

As shown in Fig6. and Fig.7, the sudden changes of the damping identification results of the left AMB occur at the same frequency as that of stiffness.

To make sure the reason for the identification sudden changes, it is intuitional consideration that the rotor vibration may suddenly change at the corresponding frequencies. Firstly, the vibration amplitude changes are checked in the range of excitation frequency, which just are shown in Fig.2 and Fig.3. Obviously, in both the left and right AMB, the vibration amplitude peaks in the critical frequency do not cause the identification sudden changes.

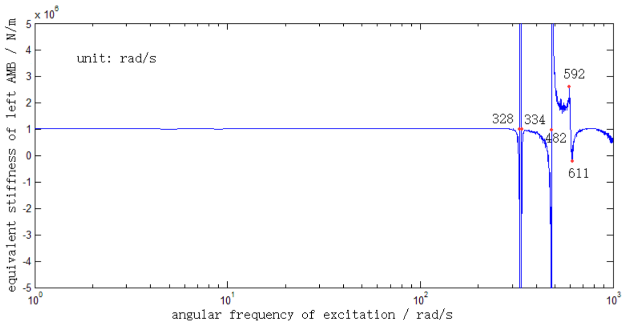

Fig.4 equivalent stiffness in x direction of left AMB

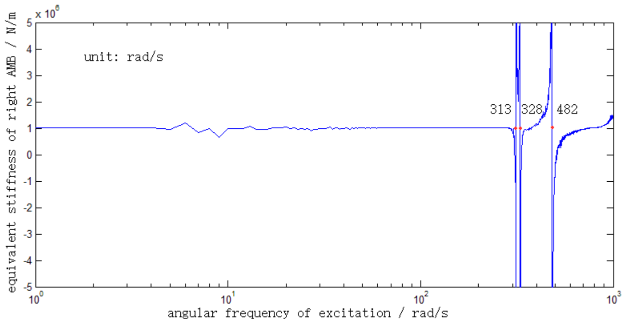

Fig.5 equivalent stiffness in x direction of right AMB

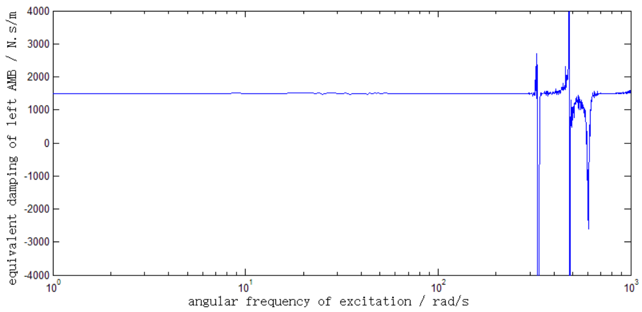

Fig.6 equivalent damping in x direction of left AMB

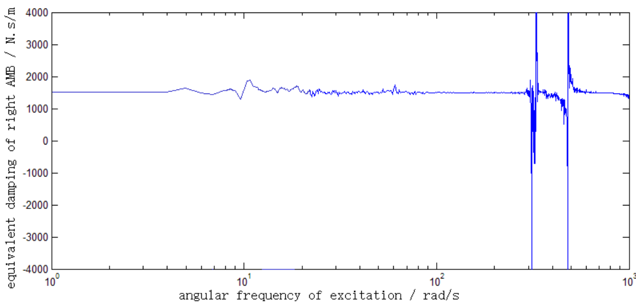

Fig.7 equivalent damping in x direction of right AMB

Next, the vibration phase is checked. The vibration phase changes in two AMB positions are plotted in Fig.8, where, the exciation phase in x direction of the left AMB is regarded as the zero phase. From Fig.8, there are two phase sudden changes at 334 rad/s and [592,611] rad/s solely occuring in the left AMB position and one phase sudden change at 313 rad/s in right. They are consistent with that of identification sudden changes solely occuring in the left or right AMB position.

Then, To analyse the identification sudden changes simultaneously occuring in both AMBs position, the vibration phase difference between left and right AMB, namely, the difference between two lines in Fig.8, is plotted in Fig.9. By this way, it is found that the angle frequency points of 328 rad/s and 482 rad/s, in which the identification sudden changes simultaneously occur in both AMBs positions, just correspond to the phase diffrence of π and 2π , respectively. Therefore, it can be analysed that, the phase range of complex operation is $[-\pi,+\pi]$, however, it is entirely possible that the vibration phase difference in a flexible rotor exceeds this range, so that the error occurs. For example, when the phase difference exceeds $+\pi$, the complex operation considers that the phase difference suddenly changes form $+\pi$ to $-\pi$. Consequently, the identification sudden change occurs.

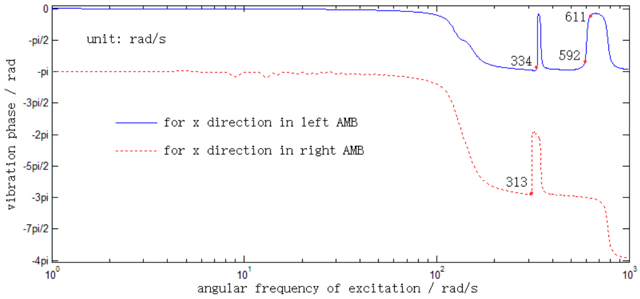

Fig.8 vibration phases of x direction in two AMB

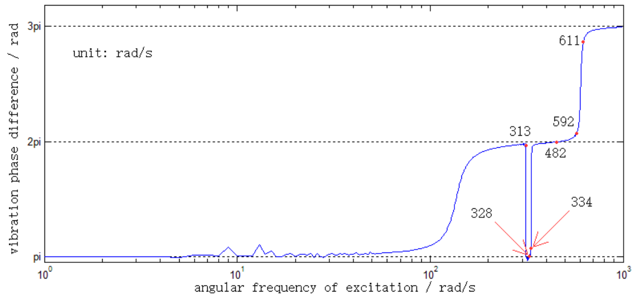

Fig.9 vibration phase difference of x direction between two AMB

5. Conclusion

In this study, a parameter identification method, which is based on the finite element rotor model and its DOF condensation, is proposed to identify the support parameter of the equivalent stiffness and the equivalent damping in the AMB-flexible rotor system. From the numerical simulation, it is can be concluded that,

1: In normal condition, the proposed method can effectively identify the parameters of the equivalent stiffness and the equivalent damping of AMB support in the flexible rotor system.

2: The vibration sudden changes may occur due to the rotor dynamic, so that the identification result changes suddenly with it. The identification result will get well when the vibration sudden change goes away.

3: For a flexible rotor system, the vibration phase difference between two different positions in rotor may exceed the range of $[-\pi,+\pi]$, which is the phase range of complex operation. It will cause the identification error.

4: For a flexible rotor system, the vibration sudden changes and the vibration phase difference are decided by the rotor's flexibility and support condition. Therefore, the different flexible rotor system has very different dynamic characteristic, namely, the identification result from the different rotor system may has very different graphic feature.

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