NEW METHOD FOR ASSISTING ROTOR TO PASS THROUGH CRITICAL SPEEDS AND DYNAMIC BALANCING

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A new method to assist a rotor-bearing system to pass through critical speeds and to perform dynamic balancing is presented. It uses rotating electromagnetic forces to suppress the excessive vibration of the rotor-bearing system while the rotor is passing through critical speeds. The forces play the same effect as centrifugal force generated by correcting weights in a traditional balancing. Imbalance of the rotor-bearing system is then calculated from the electromagnetic forces recorded in the whole running range. By adding or removing the calculated weight at correcting axial locations and angular positions, the rotor-bearing system can be balanced possibly in one time. This method will be able to save much time and runs than the traditional balancing process. Also, the method is an effective way to suppress rotor vibration when balance technology is not used. In numerical simulation, a rigid rotor with elastic supports at ends rotated from zero speed till certain critical speed was past. When the vibration amplitude became large near the critical speed, electromagnetic forces were applied till the vibration being suppressed to an accepted level. The forces were changeable depending on the vibration. They became zero if the vibration is small enough. The forces which were functions of time were used to reconstruct initial imbalance of the rotor. The correction weights were deduced from the calculated initial imbalance. Experimental equipment was set up on a standard rotor test rig. Eight coils were used for passing two critical speeds. Electrical controller was developed from active-magnet-bearing technology. Experimental study was carried out. The results showed that the method was promising.

Keywords: rotor-bearing system, electromagnetic force, critical speed, dynamic balance, vibration suppression

1. Introduction

The resonance caused by imbalance is a common malfunction in a rotor system. In some situations, traditional balancing methods such as influence coefficient method [1] and modal balancing [2] are inefficient and costly because of the need of trial weights [3]. In order to improve the balance efficiency, balancing methods without trial weights [4,5] and online active balancing methods [6,7] were proposed. However, the balancing accuracy of the methods without trial weights still needs to be improved, and the online active balancing devices have to work in the whole cycle of the rotor, which limit the application and development of the two methods.

In recent years, researches have been made on vibration suppression by controllable electromagnetic forces using active magnetic bearings [8,9]. As a kind of non-contact force, electromagnetic force has a great advantage in the vibration control of a rotor. Combining the advantages of online active balancing methods and traditional balancing methods, a new method to assist a rotor-bearing system to pass through critical speeds and to make dynamic balancing is presented. This method uses controllable rotating electromagnetic forces to suppress the excessive vibration of the rotor-bearing system. The forces play the same effect as centrifugal force generated by correcting weights in a traditional balancing. Imbalance of the rotor-bearing system is then calculated from the electromagnetic forces recorded in the whole running range. By adding or removing the calculated amounts of weight at correct axial locations and angular positions, the rotor-bearing system can be balanced theoretically in one time. This method will be able to save much time and runs than the traditional balancing process. In addition, this method can also be used to assist an unbalanced rotor to pass through several critical speeds directly.

In this paper, the quantitative control of electromagnetic force is realized by the method of calibration. The process of assisting a rotor to pass through critical speeds and to make dynamic balance was simulated using MATLAB. In the simulation, a rigid rotor with an elastic support at each end rotated up from zero speed until certain critical speed was past. The simulation results show that this method is feasible. Experimental equipment is set up on a standard rotor test rig. The experimental results show that this method is effective and practical.

2. Control of electromagnetic force

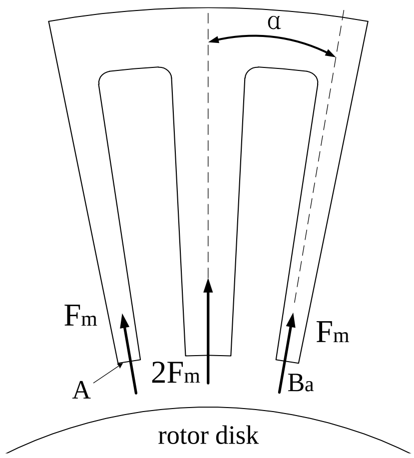

Figure 1: Schematic diagram of magnetic attraction between “E” type magnetic pole and rotor disk.

As shown in Fig. 1, the electromagnetic force F_m between the side magnetic pole and the rotor disk is as follows [10,11]:

$$F_m = \frac{B_a^2 A}{2\mu_0}. \tag{1}$$

In Eq. (1), B_a is the magnetic induction intensity between magnetic pole and rotor disk, A is the cross section of the air gap between side pole and rotor disk, μ_0 is the permeability of vacuum. In Fig. 1, the front area of the middle pole is two times of that of the side pole, so the total force between the “E” type magnetic pole and the rotor disk is F:

$$F = 2F_m\left(1+\cos\alpha\right) = \frac{B_a^2 A}{\mu_0}\left(1+\cos\alpha\right). \tag{2}$$

α is the angle shown in Fig. 1. The specific relationship between magnetic induction intensity B_a and electromagnetic force F is determined by calibrating the Eq. (2). Experimental measurements show that the calibration curves of different magnetic poles with the same rotor disk are basically the same, however, the calibration curves will vary with rotor disks. This is because the magnetic pole structure is consistent, but different rotor disk has different thickness which will affect the cross section A. The fitting formula of F with rotor disk 1 is shown in Eq. (3). Thus, the electromagnetic force generated by the magnetic pole can be controlled by controlling the value of the magnetic induction intensity B_a of the gap.

$$F = 61.7190B_a^2 - 1.1075B_a + 0.0205 \quad R^2 = 0.9991 \tag{3}$$

The gap magnetic induction intensity B_a has a linear relationship with the pole current I, and the DSP order value d has a linear relationship with the pole current as well, so the DSP order value d is linear with the gap magnetic induction intensity B_a. The d-B_a fitting formula of pole 1 and rotor disk 1 is shown in Eq. (4). The magnetic pole current is controlled by a differential method. Each rotor disk needs four poles to generate the electromagnetic force which is synchronous with the rotor speed. The control equation of d for each magnetic pole with rotor disk 1 is shown in Eq. (5). ΔF is the magnitude of the rotating electromagnetic force, φ is the initial phase of ΔF, ω is the rotor speed. In order to assist a rotor to pass through its 1st and 2nd order critical speed, electromagnetic forces on two rotor disks are necessary. The control equations of d of rotor disk 2 are similar to Eq. (5), which is no longer given here.

$$d = 2880.9B_a + 17.410 \quad R^2 = 0.9994 \tag{4}$$

$$\text{rotor disk 1:} \begin{cases} \#1: \ d_1 = 474.3 + 47.37\Delta F \sin(\omega t + \varphi) \\ \#2: \ d_2 = 506.9 - 51.68\Delta F \sin(\omega t + \varphi) \\ \#3: \ d_3 = 492.9 + 49.78\Delta F \cos(\omega t + \varphi) \\ \#4: \ d_4 = 510.4 - 50.98\Delta F \cos(\omega t + \varphi) \end{cases} \tag{5}$$

3. MATLAB simulation

The rigid rotor model used in MATLAB simulation is shown in Fig. 2.

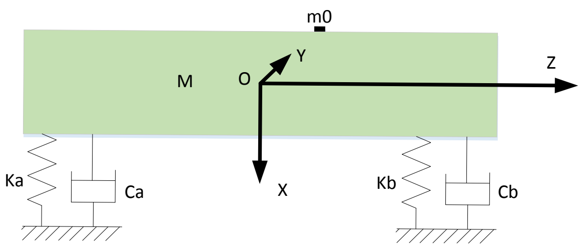

Figure 2: Rigid rotor model used in MATLAB simulation.

The rotor is L=0.6m in length and R=0.1m in radius. Its material density is ρ=7800kg/m^3. The coordinate system is established at the geometry center of the rotor. The imbalance mass located at the outer surface of the rotor is m_0=0.8kg. Inside the XY plane, the angle between the projection of m_0 and the X axis is 60°. The z-axis coordinate value of m_0 is U_z=0.07m. The rigid rotor with an elastic support at each end rotates around the Z axis. The stiffness is K_b= K_a =1×10^{7}N/m, and the damping is C_b= C_a =1000N·s/m.

The vibration at both ends of the rotor assembly is solved by MATLAB for the duration of speeding up. When the vibration amplitude exceeds the pre-set limit R_0, the rotor speed is maintained constant, and the electromagnetic devices start to work. Two electromagnetic devices are located at z=0.25m and z=-0.25m respectively. Due to the symmetry of the rotor and the location of electromagnetic devices, the electromagnetic forces produced by both electromagnetic devices are the same in magnitude and phase when supressing the 1st order vibration. Firstly, we keep the two in-phase electromagnetic forces small, and then change their phase to find the phase value that minimizes the vibration. Finally, we fix this phase, and slowly increase the magnitude of electromagnetic forces until the vibration is less than the limit value (here is 0.1R_0). After that, we continue to increase rotating speed, keeping the magnitude and phase of the electromagnetic force unchanged. If the vibration amplitude reaches the limit R_0 again, then a small force will be superimposed on the current electromagnetic force and the process of searching for the proper magnitude and phase of electromagnetic force will be repeated. The above process will be contiued until the 1st order critical speed is passed.

After passing the 1st order critical speed, the electromagnetic forces will be removed when the vibration is small enough. The rotor continues to speed up. When the vibration once again reaches R_0, the rotor enters the 2nd order resonance region. At this time, the electromagnetic forces produced by both electromagnetic devices are the same in magnitude and opposite in phase. The process of assisting rotor to pass through the 2nd order critical speed is similar to that of passing through the 1st order critical speed.

Set R_0=0.005m, the process of assisting rotor to pass through the **1**st and **2**nd critical speeds is shown in Fig. 3.

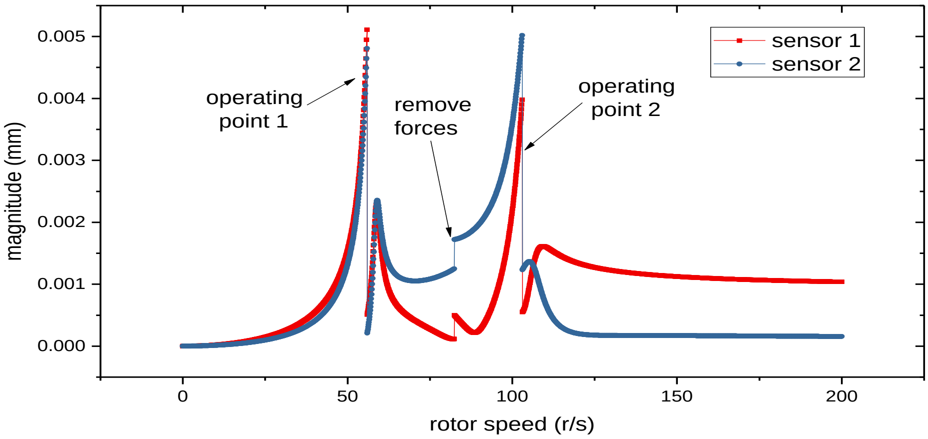

Figure 3: Amplitude-frequency curve of assisting rotor to pass through the 1st and 2nd critical speeds.

The recorded information of the electromagnetic force at the z=0.25m during assisting rotor to pass through the **1**st and **2**nd critical speeds is shown in Table. 1.

Table 1: Recorded electromagnetic force information of passing through the 1st and 2nd critical speeds in simulation.

Operating point	Rotor speed [r/s]	Electromagnetic force magnitude [N]	Electromagnetic force phase [°]
1	55.84	4576.46	240
2	102.97	4175.73	239

The data in Table 1 is converted to a balanced quality procedure as follows:

1) Operating point 1 is in the 1st order resonance region, so the equivalent mass m1 of the electromagnetic force fm1 at z=0.25m is m_1:

$$f_{m1} = m_1 r \omega_1^2 \rightarrow 4576.46 = m_1 \times 0.1 \times (55.84 \times 2 \times \pi)^2$$

m_1=0.3718kg, and its phase is 240°.

2) Operating point 2 is in the 2nd order resonance region, so the equivalent mass m2 of the electromagnetic force fm2 at z=0.25m is m_2:

$$f_{m2} = m_2 r \omega_2^2 \rightarrow 4175.73 = m_2 \times 0.1 \times (102.97 \times 2 \times \pi)^2$$

m_2=0.0998kg, and its phase is 239°.

3) The actual weight should be added at z = 0.25m is m_b:

$$m_b = m_1 \angle 240° + m_2 \angle 239°$$

m_b=0.4715kg∠239.8°.

4) The actual weight should be added at z = -0.25m is m_a:

$$m_a = m_1\angle 240° + m_2\angle 59°$$

m_a=0.2720kg∠240.4°.

After adding the calculated correcting weights, the 1st and 2nd order resonance amplitudes of the rotor are reduced by 92.9% and 88.9% respectively. The simulation shows that this new method is feasible.

4. Experimental results

Schematic diagram of the rotor system used in experimental study is shown in Fig. 4. Three rotor disks are evenly fitted between two bearings. Each of disk 1 and disk 2 is surrounded by four magnetic poles to produce rotating electromagnetic force. Two rotating electromagnetic forces cooperate with each other to assist the rotor to pass through 1st and 2nd critical speeds and to make dynamic balancing. An eddy current displacement sensor is arranged on one side of disk 1 and disk 2 respectively. Sensor 1 is used to measure the vibration in the vertical direction. Sensor 2 is used to measure the vibration in the horizontal direction.

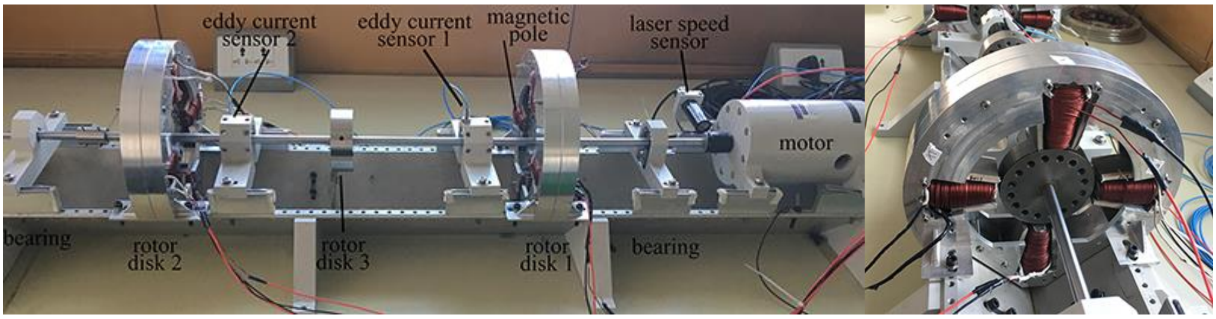

Figure 4: Schematic diagram of the rotor system used in experimental study.

The synchronous rotating electromagnetic force can be used to counteract the centrifugal force produced by the imbalance of the rotor system. When vibration amplitude reaches the vibration limit, electromagnetic forces will be applied till the vibration being suppressed to an accepted level. These electromagnetic forces information are recorded in the whole running range. Imbalance of the rotor-bearing system is then calculated from the recorded information. By adding or removing the calculated amounts of weight at correctting axial locations and angular positions, the rotor-bearing system can be balanced efficiently. The initial vibration amplitude - frequency curve of the experimental rotor system is shown in Fig. 5.

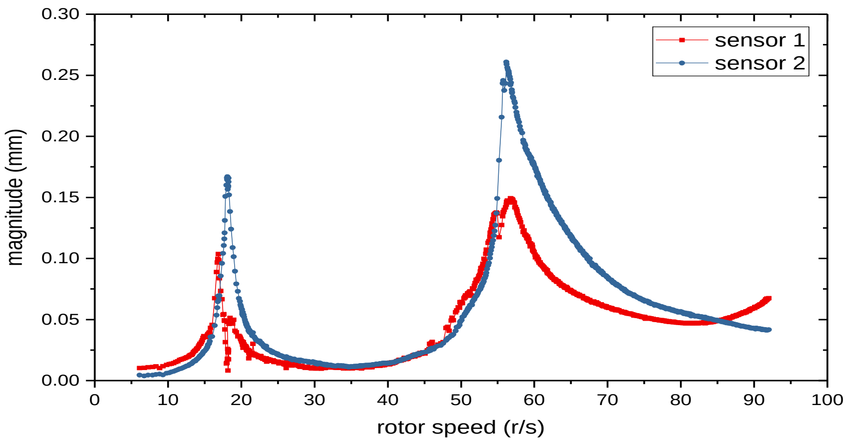

Figure 5: Initial vibration amplitude-frequency curve of the experimental rotor system.

The process of assisting experimental rotor to pass through critical speeds by means of synchronous rotating electromagnetic forces is shown in Fig. 6.

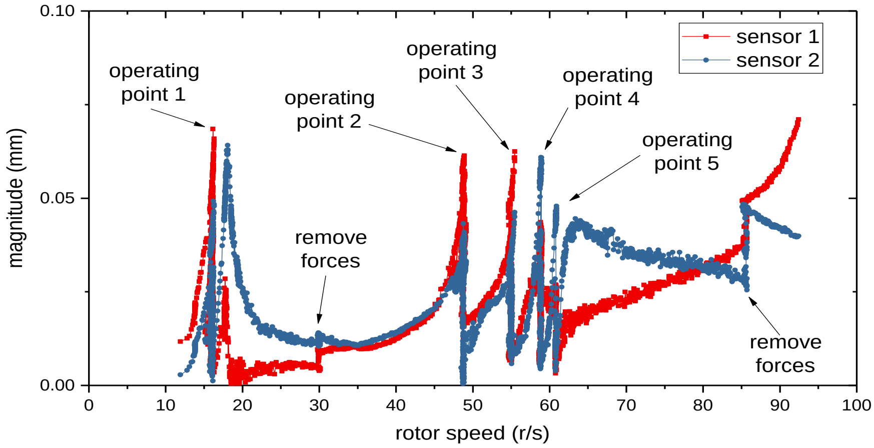

Figure 6: The process of assisting experimental rotor to pass through critical speeds.

The recorded information of the electromagnetic force of disk 1 during the process of assisting rotor to pass through the 1st and 2nd critical speeds is shown in Table. 2.

Table 2: Recorded electromagnetic force information of passing through the 1st and 2nd critical speeds in experiment.

Critical speed	Operating point	Rotor speed [r/s]	Electromagnetic force magnitude [N]	Electromagnetic force phase [°]
1st	1	16.02	0.11	10
2nd	2	48.78	1.1	320
	3	54.99	1.7	320
	4	58.88	2.1	320
	5	60.79	2.3	320

Correcting weights are added at 0.03m in radius of disk 1 and disk 2. The result calculated by the information of operating point 1 and 5 is: 0.80g∠340° (disk 1) and 0.40g∠96° (disk 2). After adding the calculated correcting weights, the amplitude-frequency curve of the rotor system is shown in Fig. 7. Compared with the initial vibration, the 1st and 2nd order resonance amplitudes of the balanced rotor are reduced by about 40% and 75% respectively. Only one operating point makes the reduction of the 1st order resonance amplitude less than that of the 2nd.

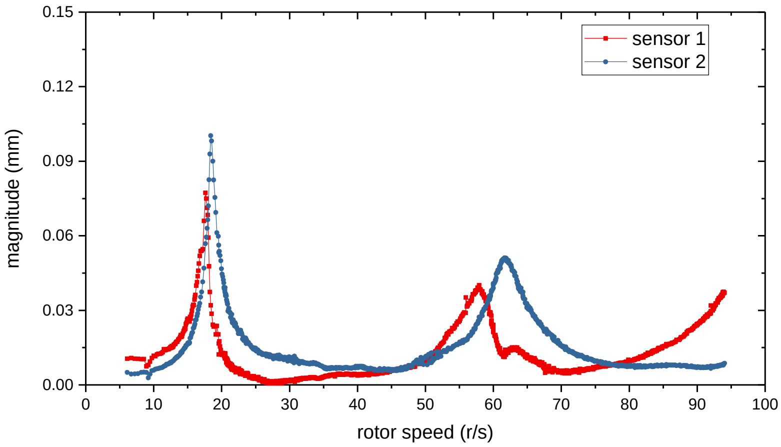

Figure 7: The amplitude-frequency curve of the experimental rotor after adding correcting weights.

5. Conclusion

Based on controllable rotating electromagnetic force, this paper presents a new method to assist a rotor-bearing system to pass through critical speeds and to make dynamic balancing. Numerical simulation was undertaken using MATLAB, which proves the feasibility of this method. In the experiment, through the cooperation of two electromagnetic forces, the 1st and 2nd critical speeds are passed successfully. The correcting weights are also calculated by the force information recorded when passing through critical speeds. Rotor vibration is reduced obviously by adding the calculated correcting weights. Experimental results show that this method is efficient in practice.

Theoretically, multiple resonances of a rotor can be balanced in one time. But in actual application, the control error of electromagnetic force, the acquisition error of vibration information and the nonlinearity of rotor system and other factors will affect the balance effect. Sometimes, more than one time of balancing is needed. Further work will be focussed on how to reduce these errors.

In conclusion, both simulation and experiment prove that this method is effective and practical in assisting a rotor to pass through critical speeds and to make dynamic balancing.

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