The Dynamic Characteristic of Rotor-Active Magnetic Bearings System on Flexible Base subjected to Base Vibration

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Abstract: Magnetic bearings are the core and key technology of a multi-electric engine. The aero engine will have no lubrication system, less weight, high reliability and efficiency, and low maintenance costs if magnetic bearings be used instead of rolling bearings. Many papers had been published research results on rotor-AMB system on ground based support, which can not better describe the real condition of aero engine. This paper models the rotor-active magnetic bearing (AMB) system hanging from a cantilever beam to simulate the effect of flexible wing on the rotor dynamic characteristics. Besides, in order to observe the effect of wing vibration on the engine rotor, the beam's free end receives excitation and the frequency response of rotor is analyzed. From simulation analysis result, it can be found that change of hanging position results in change of the base spatial modal properties (M and K). Consequently, the modal frequency of some rotor shapes changes irregularly. Under the same excitation conditions, imposed on the flexible base (as opposed to a rigid foundation), the rotor frequency response slightly decreased except in the beam type modal frequencies.

Keywords: Flexible Base, Base Excitation, Rotor-Active Magnetic Bearings

Introduction

In 1992, the United States proposed and started the research project of multi-electric aircraft. One of its core technologies is the multi-electric engine. And high-temperature magnetic bearing is the core technology of the multi-electric engine. In 1999, the European Community successfully developed the 550°C high temperature magnetic bearing system. Next year, the British Government started a multi-electric engine and wing system plan. And one of the objectives is to develop high temperature magnetic bearing. Research on high temperature magnetic bearing carried out in China since 1992. Many papers had been published research results on High temperature magnetic bearing and rotor-active magnetic bearings (AMB) system on ground based support [1-5]. But there is less research on the dynamic characteristics of the engine rotor under hanging in the wings, and wing vibration state.

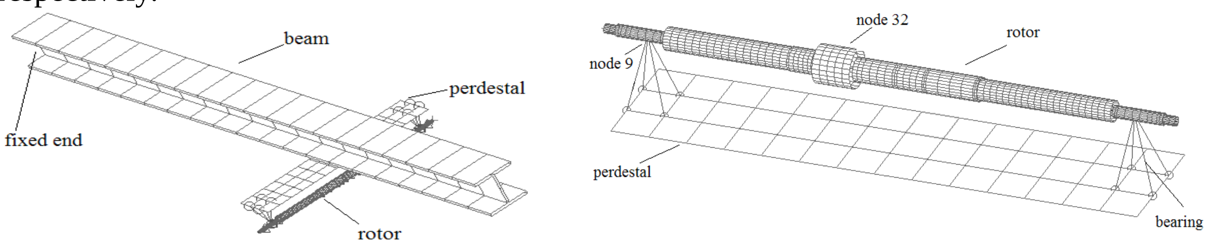
Nowadays, engines of jumbo jet hang under the wings due to their large weight. The wings of aircrafts behave like an elastic support rather than rigid support. More so, the rotor in aero engine usually gets excitation from the wing vibration caused by air current, a situation which modeled system on the ground can not simulate.

In fact, domestic and foreign researchers have done some work on the effect of base movement on the rotor-active magnetic bearings system. [6] established the mathematical model of rotor-bearing-base system and successfully carried out the numerical analysis of base movement. But the model did not consider the bearing support stiffness and damping. An Sung Lee et al[7] [8] researched on the impact of acceleration excitation on the vibration of the rotor, They used Lagrange's and the finite element methods to establish relatively complete mathematical models and compared the amplitude changes under different base excitation when the rotor speeds at 0 rpm and 6000 rpm. [9] gave the experiment analysis of rotor vibration when the base is transversely excited. Also the vibration responses caused by different speeds, different exciting forces and different exciting frequencies are compared respectively. Professor A. Berlioz et al [10] [11] investigated the dynamic behavior of flexible rotor systems subjected to base excitation (support movements) theoretically and experimentally. Lin Fusheng and Meng Guang investigated the dynamic characteristics of rotor system in a maneuvering aircraft [12]. This paper models the rotor-AMB system hanging from a cantilever beam to simulate the effect of flexible wing on the rotor dynamic characteristics. Besides, in order to observe the effect of wing vibration on the engine rotor, the beam's free end receives excitation and the frequency response of rotor is analyzed.

Modal Analysis

There will be some change of engine rotor dynamic characteristics (modal frequency and mode shape) when aircraft engine hang on the wing. And the wing and rotor-AMB need to be discussed and analyzed as a whole system.

A rotor-AMB hanging from a cantilever beam is modeled in Patran finite element software (figure 1.a). The first natural frequency of the cantilever beam is about 37 Hz. The rotor is modeled as beam elements with lumped masses; bearing pedestal is modeled as shell elements which have very high Young modulus (E), a condition it can be regarded as a rigid body. The cantilever beam is modeled as beam elements, ignoring its torsion characteristics. Finally, the magnetic is modeled as stiffness and damping coefficient of 1e6N/m and700N*s/m respectively.



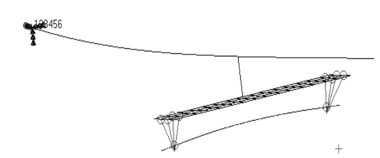
(a) rotor-AMB hanging from a cantilever beam (b) rotor-AMB sited on the ground

Figure.1 finite element model of system

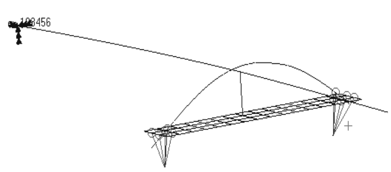
The hanging position of the rotor-AMB system was varied and the bending stiffness value calculated. These results are shown in Table 1. It can be seen from the result that these values are close to AMB stiffness values. Figure 2 show the mode shapes corresponding mode frequencies (Table 2).

Table1. Bending rigidity of different position

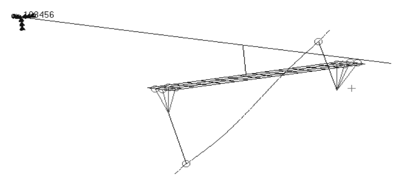
position（m）	0.8	1.2	1.6	2
stiffness（N/m）	9.92e6	2.94e6	1.24e6	0.63e6



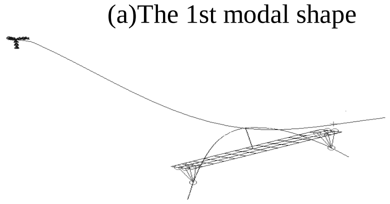
(a)The 1st modal shape



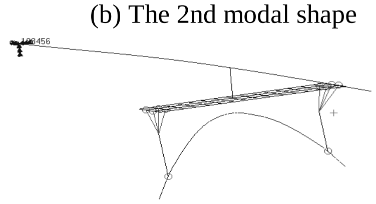
(b) The 2nd modal shape



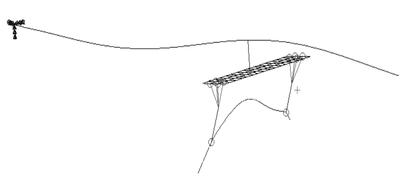
(c) The 3rd modal shape



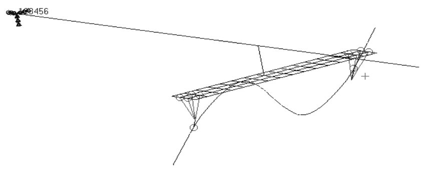
(d) The 4th modal shape



(e) The 5th r modal shape



(f) The 6th modal shape



(g) The 7th modal shape

Figure.2 Modal shapes of system

The mode shapes seen above can be classified into two; rotor type and beam type [13], depending on which part's (rotor or beam) deformation mode is predominant. From Table 2 it can be seen that the frequencies of the beam type modal (a), (d), (f) increases with the decrease in hanging position (direction towards the fixed end). The frequencies of rotor type modal (c), (g) are almost constant at 157 Hz and 555 Hz respectively. This may be caused by that weak coupling between rotor and beam under these two modal shapes. The frequencies of rotor type modal (b), (e) do not have obvious regular pattern; this may be due to the changes of the distribution of vibration mass caused by the different hanging position. Neglecting the mass of the beam, it can be seen from Figure 3, that the modal frequencies of rotor (b), (e) are low as well as the bending rigidity.

Table.2 Modal frequencies (Hz) on different position

position(m)	2m	1.6m	1.2m	0.8m
Modal shape(a)	28.34	31.56	34.47	36.48
Modal shape (b)	97.43	95.11	90.34	86.44
Modal shape (c)	156.92	156.92	156.92	156.91
Modal shape (d)	192.17	210.79	204.12	199.24
Modal shape (e)	251.37	247.82	249.55	250.25
Modal shape (f)	477.12	516.77	506.52	514.02
Modal shape (g)	554.66	554.68	554.67	554.68

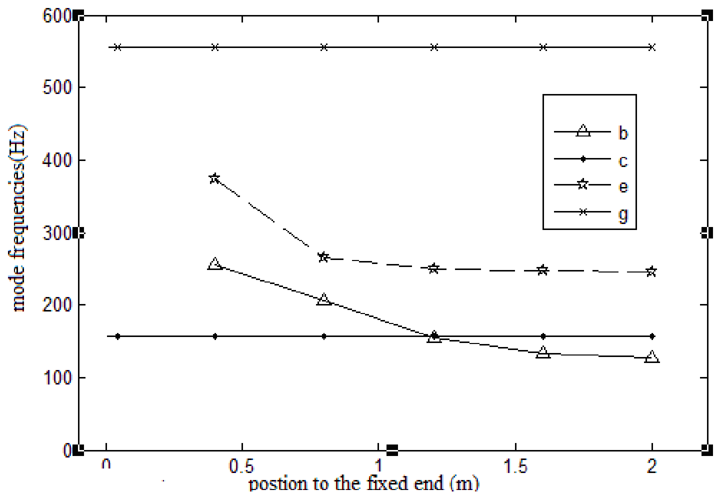


Figure.3 change of rotor modal frequencies

Frequency response analysis

1) Beam Excitation

Here we investigate the response to varying frequency excitation supplied to the beam. The rotor, rotating at speed (say 6000rpm) hung at 1.2m away from the fixed end of the beam. The beam's free end receives varying-frequency excitation and the response at the 32^{nd} node (i.e. *disc node*) measured. This displacement is plotted against excitation frequencies. Similar plots are produced for different speeds of the rotor (18000, 3000 and 42000 rpm) and compared in Figure 4. No significant change in vibration amplitude is observed for the different speeds at excitation frequencies below 800 Hz. (Fig 4). This may due to there is no imbalance mass in the model. However at higher excitation frequencies, the amplitude increases with rotation speed (Figure 5).

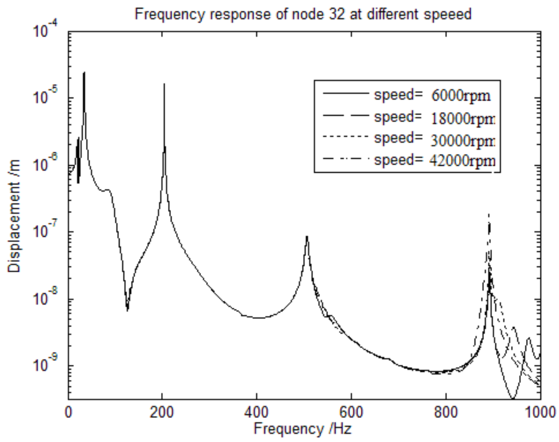


Figure.4 Frequency response of node 32 at different speed

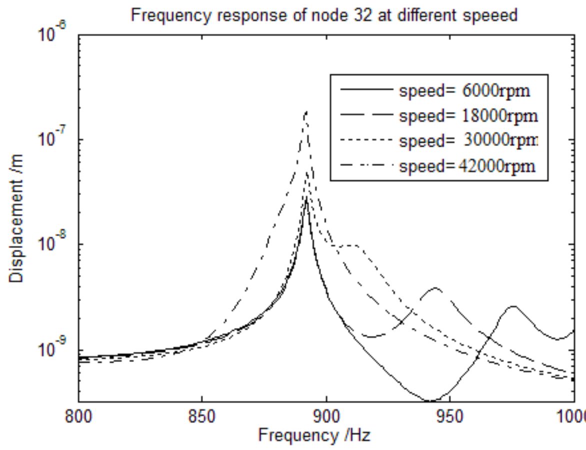


Figure.5 Frequency response of node 32 at high speed

2) Beam excitation versus Pedestal excitation

The rotor system is detached from the beam and sited on the ground (figure 1.b). Varying frequency excitation is supplied to the pedestal base while the rotor rotates at 6000 rpm. The response amplitude at the 32^{nd} node is measured and compared to that of 1) of same speed. The result is shown in Figure 6. As show in figure 6 we can see that response amplitude is higher for beam than pedestal when the excitation frequencies are close to the modal frequencies of beam type modal.

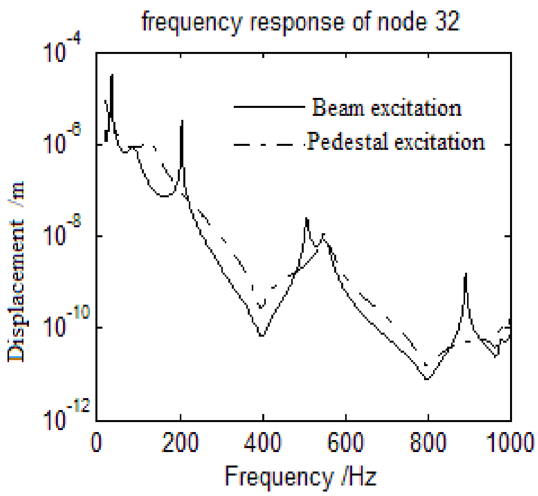


Figure.6 Frequency response of node 32

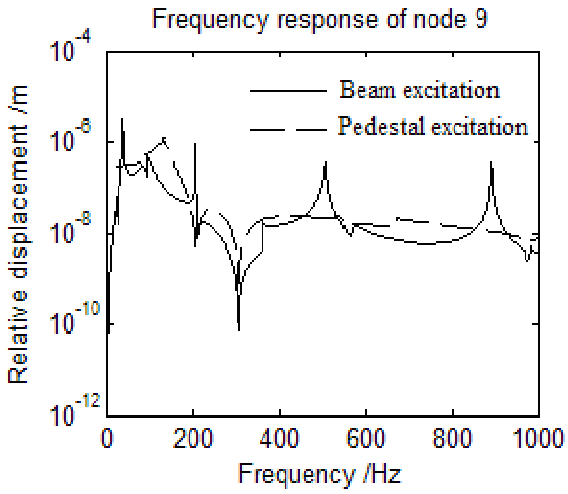


Figure.7 relative Frequency response of node 9

The gap between rotor and magnetic bearings are generally very small, in the order of 10^{-4}m, therefore, the relative vibration of the rotor and bearings must be given attention. Figure 7 show the relative vibration of node 9 (node of rotor close to the bearing) under the cases above. More so, it is also observed that the relative vibration is high for beam than pedestal when the excitation frequencies close to the beam type modal frequencies.

Summary

From the above simulation, the conclusion can be drawn that changes of hanging positions not only affect the bending rigidity but also the distribution of vibration mass, so that the changes of some rotor modal frequencies are not very regular.

In addition, it can be found that the frequency response of rotor vibration is small except the excitation frequencies close to the modal frequencies of base type modal when rotor system on flexible base than on rigid base.

These conclusions will provide some reference value for controller design and stability judgment of magnetic bearings used in the aero engine.

Acknowledgement

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