Field Dynamic Balance Method Study for the AMB - Flexible Rotor System

Li Hongwei[1)], Xu Yang[1)], Gu Huidong[2)], Zhao Lei[1)]

1) Institute of Nuclear and New Energy Technology, Tsinghua University, Beijing, China
2) Department of Engineering Physics, Tsinghua University, Beijing, China

ABSTRACT

To exceed the first two flexible critical rotational speeds, the rotor suspended by active magnetic bearings must be controlled very well. Besides the good control method, the field dynamic balance is also a very efficient way to reduce the rotor's vibration. Therefore this paper studied the field dynamic balance method for the flexible rotor suspended by active magnetic bearings and analyzed the influence of the field dynamic balance when the rotor exceeds critical rotational speeds through experiments. Two conventional balance methods are combined the field dynamic balance method in the paper. The influence coefficient method is applied to balance the rigid modals while the modal balance method is applied to the flexible modals. In practical experiments, the study object is a rotor with three balance disks supported by active magnetic bearings. The results show that the balanced rotor's vibration amplitude is obviously reduced and the maximum reduction is 83.6%. After balanced, the rotor can successfully exceed the first two flexible critical speeds and can rotate stably at 800Hz. With the same conditions, the system obtains better performance after the rotor is balanced. These show that the field dynamic balance takes an important part in increasing the secure operation rotational speed of the rotor and the stability, security and cost of the AMBs' system.

INTRODUCTION

The HTR-10GT, an important project in the 863 Program, will employ the helium gas turbine instead of the steam gas turbine to generate electricity more efficiently. The rating rotational speed (15,000rpm) of the turbine rotor will be above the second flexible critical rotational speed (8,496rpm) [1]. To meet this request and satisfy the clean helium condition, the ideal method to support the turbine rotor and the generator rotor is by active magnetic bearings (AMB).

As known, the rotor vibrates severely when exceeding critical rotational speeds. It is a key problem that how to reduce the rotor vibration efficiently supported by AMBs in the helium gas turbine. The rotor vibration is excited by many factors. The rotor's imbalance is the principal factor to excite the synchronous vibration and this will lead to the great amplitude, strain and counterforce. In order to compensate the rotor's imbalance force, the power amplifier may be saturated under high rotational speed in AMBs system. This will make the rotor could not exceed the critical rotational speeds or bate the system stability, even unsteady the system. Therefore, reducing the rotor's imbalance is an efficient way to reduce the rotor's vibration and increase the system stability, besides the good control methods.

After assembling the AMBs system, the previous balance effect of the rotor may be bated. The factors include the erection error, the difference of measure planes and the difference of the supporting planes and so on. Therefore the field dynamic balancing method will be applied to the rotor. Compared with the rotor supported by common mechanical bearings, the field dynamic balance for the rotor supported by AMBs doesn't need the balancing apparatus and the special sensors to measure the vibration signals for the AMB system has its own displacement sensors.

FIELD DYNAMIC BALANCE METHOD

The field dynamic balance method for the rotor supported by mechanical bearings includes influence coefficient method, modal balancing method and optimized influence coefficient method and so on [2-4].

The rigid rotor's rotational speed is much lower than the first flexible critical rotational speed. Compared with the eccentricity, the elastic deformation of the rigid rotor is too small and can be ignored. The balancing effect is not influenced by the rotational speed. Once the rotor's speed reaches or exceeds the flexible critical rotational speed, the elastic deformation could not be ignored. And the balancing effect will be influenced by the rotational speed. The rotor balanced well at one flexible critical rotational speed may become unbalanced at another critical rotational speed.

The influence coefficient balance method is to get the smallest vibration under the balancing rotational speed and don't consider other modals. This will damage the balance effect of other modals. Besides that, to get the influence coefficient matrix, this method needs many times to measure vibrations at various rotational speeds and many times switching between start and stop the rotor's rotating.

The modal balancing method aims at the smallest vibration at the critical rotational speed. There is not influence of the balance effect between different modals. This method can reduce the measure times and the start-stop times. But this method needs to know the critical rotational speeds and the modal shapes previously.

To get good balance effect and simplify the balance process, the influence coefficient method and the modal balance method are respectively used to balance the rigid modals and the flexible modals in AMBs-flexible rotor system.

THE FIELD DYNAMIC BALANCE EXPERIMENTAL SETUP

According to the helium gas turbine rotor, an experimental setup (code name AMB-P, see Fig. 1) is designed to study the method to control the rotor's vibration. The field dynamic balance experiment utilizes its own electric eddy current displacement sensors to measure the vibrations at two sides and utilizes the infrared sensor to measure the rotational speed. When the rotational speed is steady, the computer gets the rotational speed signal and the vibration signal via the data acquisition card. Through FFT, it is easily to obtain the amplitude and phase of the synchronous vibration. According to the results, the balance mass in different balance planes are calculated.

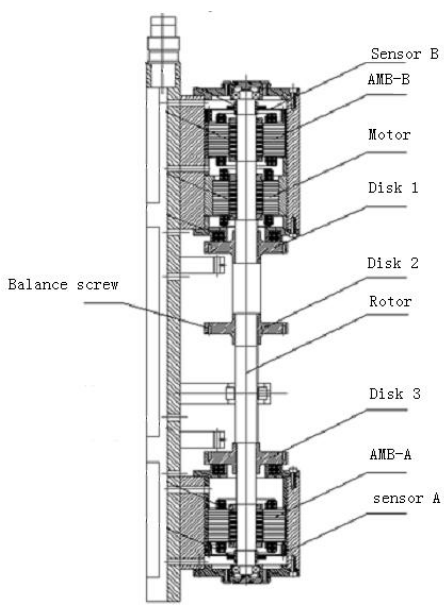

Fig. 1 the System of AMB-P

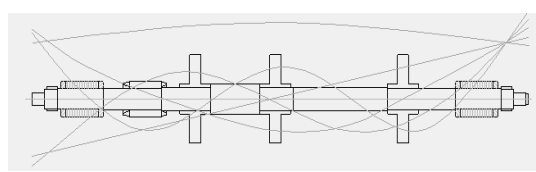

Fig. 2 the Modal Shapes of the Rotor

The three disks are the balance planes in this experimental setup. There are 8 equispaced M3 screws along a circle in every disk. In experiments, various mass M3 bolts are prepared according to the calculated results. One balance mass

in an arbitrary angle can be divided into two balance mass bolts in two close screws.

THE FIELD DYNAMIC BALANCE EXPERIMENTS

The nature frequencies of the AMB-P's rotor are shown in Table 1 and the modal shapes are shown in Fig. 2.

Table 1. the Critical Speeds of AMB-P's Rotor

Modal	1st Rigid Modal	2nd Rigid Modal	1st Flexible Modal	2nd Flexible Modal	3rd Flexible Modal
Nature Frequency/Hz	23.7	46.0	296.7	699.3	1206.4

Field Dynamic Balance Experiments for the Rigid Rotor

The rotor can exceed the rigid modals successfully under the designed controller. To reduce the vibration of the rigid modals more, the rigid modals balancing experiments are applied at the rotational speed about 200Hz. The balance disks are the disk 1 and 3 in the rigid balancing experiments. The balancing process is shown in table 2. Fig. 3 shows the vibrations before and after the balancing experiments. To avoid the disturbance of the motor, the vibration and rotor's rotational speed signal are measured after the motor is closed. Therefore the measured rotational speeds are tiny different.

Table 2 shows that the side B's vibration declined by 28.8% and side A's declined by 62.4%. After rigid balancing experiments, the rotor can successfully reach 300Hz in the influence region of the 1st flexible modal. But now the vibration amplitude is too large. In order to guarantee security, the balancing experiments for the 1st flexible modal must be carried out.

Table 2. Balance Process of the Rigid Rotor

Balancing process	Balance mass for disk 1 /mg∠°	Balance mass for disk 3 /mg∠°	Side B's vibration /μm∠°	Side A's vibration /μm∠°	Rotational speed /Hz
Initial vibration			91.19∠117.64	25.48∠286.71	195.10
Test mass	280∠0		100.15∠82.98	20.65∠288.06	194.91
Test mass		680∠0	126.02∠153.01	49.55∠343	195.94
Calculated Balance mass	444.2 ∠274.6	381.2 ∠286.6			
Actual balance mass	420 ∠270	380 ∠270	41.24∠112.26	54.17∠295.20	195.19
Calculated Balance mass	210∠285.5	524.6∠94.9			
Actual balance mass	210∠270	520∠90	64.89∠296.53	9.57∠308.19	195.86

Field Dynamic Balance Experiments for the 1st Flexible Modal

According to the N+2 planes dynamic balance method, the three disks along the rotor are all utilized to carry out the balance experiments for the 1st flexible modal. The characteristics of the three disks are shown in table 3. z is the distance from the balance disk to the center of AMB-B and φ is the size of the modal shape at balance disk.

The equation set for the 1st flexible modal balance is as following:

$$\begin{cases} -0.1871P_1 - 0.4739P_2 + 0.0139P_3 = -c_1M_1 \\ P_1 + P_2 + P_3 = 0 \\ 0.1395P_1 + 0.2385P_2 + 0.3960P_3 = 0 \end{cases} \quad (1)$$

$$\Longrightarrow \begin{cases} P_1 = -1.6851 c_1 M_1 \\ P_2 = 2.7443 c_1 M_1 \\ P_3 = -1.0592 c_1 M_1 \end{cases} \quad (2)$$

Table 3. Parameters of the Balance Disks

Parameter	Disk 1	Disk 2	Disk 3
z_i	0.1395	0.2385	0.396
φ_{1i}	-0.1871	-0.4739	0.0139
φ_{2i}	0.5970	-0.1053	-0.5169

According to the Eq. (2), the results show that the balance masses at the three disks increase with a certain proportion. Therefore only the balance mass at the disk 2 is recorded. The same is to the balance experiments for the 2nd flexible modal.

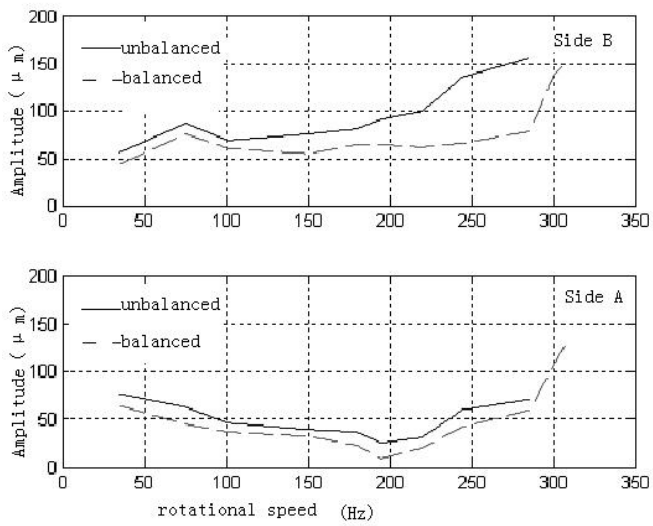

Fig. 3 the Amplitude-Frequency Response of the Rotor before and after Rigid Modal Balance

The process of the1st flexible modal balance is shown in table 4. Fig. 4 shows the amplitude-frequency response of the rotor's vibrations before and after the balance experiments for the 1st flexible modal.

Table 4. Balance Process for the 1st Flexible Modal

Balancing process	Balance mass for disk 2 /mg∠°	Side B's vibration /μm∠°	Side A's vibration /μm∠°	Rotational speed /Hz
Initial vibration		137.59∠265.34	107.55∠280.15	298.25
Test mass	180∠0	31.49∠141.00	17.60∠341.32	302.87

The table 4 shows that the side B's vibration declined by 77.1% and side A's declined by 83.6%. After the balance experiments for the1st flexible modal, the rotor can successfully exceed the 1st flexible critical rotational speed and reach

640Hz in the influence region of the 2nd flexible modal. But now the vibration is too large. In order to guarantee security, the balancing experiments for the 2nd flexible modal must be carried out.

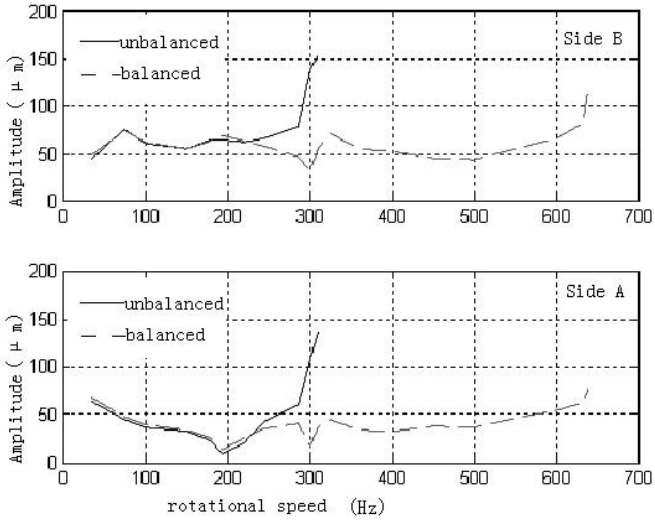

Fig. 4 the Amplitude-Frequency Response of the Rotor before and after the Balance for the 1st Flexible Modal

Field Dynamic Balance Experiment for the 2nd Flexible Modal

As there are only 3 balance disks, the balance experiments for the 2nd flexible modal could not directly use the N+2 planes balance method, which requires improved algorithm. The data of the 2nd flexible modal shape has been shown in Table 3.

To avoid influence the previous balance effect, the equation set for the 2nd flexible modal balance is as the following:

$$\begin{cases} 0.5970P_1 - 0.1053P_2 - 0.5169P_3 = -c_2M_2 \\ -0.1871P_1 - 0.4739P_2 + 0.0139P_3 = 0 \\ P_1 + P_2 + P_3 = 0 \\ 0.1395P_1 + 0.2385P_2 + 0.3960P_3 = 0 \end{cases} \tag{3}$$

But generally the Eq. (3) has no roots for there are 3 arguments and 4 equations in it. If the 2nd flexible modal balance must be carried out, the previous balance effect will be influenced. To reduce the influence, the least square solution of the Eq. (3) is employed to calculate the balance masses of the 3 disks.

The Eq. (3) is written in the form of matrix.

$$V_0 + \boldsymbol{A}P = 0 \tag{4}$$

The solution is

$$P = -(\boldsymbol{A}^{*T}\boldsymbol{A})^{-1}\boldsymbol{A}^{*T}V_0 \tag{5}$$

And $P = [P_1, P_2, P_3]^T$, $V_0 = [c_2M_2, 0, 0, 0]^T$, $\boldsymbol{A} = \begin{bmatrix} 0.5970 & -0.1053 & -0.5169 \\ -0.1871 & -0.4739 & 0.0139 \\ 1 & 1 & 1 \\ 0.1395 & 0.2385 & 0.3960 \end{bmatrix}$.

The balance masses of the 3 disks are

$$\begin{cases} P_1 = -1.0756 c_2 M_2 \\ P_2 = 0.4954 c_2 M_2 \\ P_3 = 0.5234 c_2 M_2 \end{cases} \tag{6}$$

The Eq. (7) gives the modal imbalance remainders of the rigid modals and the first two flexible modals.

$$V = [0.0351 c_2 M_2, -0.0263 c_2 M_2, -0.0567 c_2 M_2, 0.1732 c_2 M_2]^T \tag{7}$$

The process of the 2nd flexible modal balance is shown in table 5. Fig. 5 shows the amplitude-frequency response of the rotor's vibrations before and after the balance experiments for the 2nd flexible modal.

From the table 5, there are 4 times balance experiments to realize that the rotor successfully exceed the 2nd flexible critical rotational speed. The safe operation rotational speed of the rotor can be improved after each balance, but the improvement is getting smaller and the balance becomes difficult. Finally, through adjusting the control parameters, the rotor can successfully exceed the 2nd flexible critical rotational speed and can rotate at 800Hz. The vibration at this speed is shown in Fig. 6.

Table 5. the Balance Process for the 2nd Flexible Modal

Balancing process	Balance mass for disk 2 /mg∠°	Side B's vibration /μm∠°	Side A's vibration /μm∠°	Rotational speed /Hz	Maximum rotational speed /Hz
Initial vibration		411.99∠176.17	310.60∠328.89	633.63	
Test mass	90∠0	199.04∠197.76	148.66∠314.78	618.89	680
Calculated Balance mass	341∠7.4				
Actual balance mass	130∠180	136.58∠216.47	105.77∠301.28	600.55	700
Actual balance mass	150∠180	202.56∠195.78	72.47∠289.16	703.74	710
Actual balance mass	130∠180	265.82∠176.54	230.15∠281.24	720	800

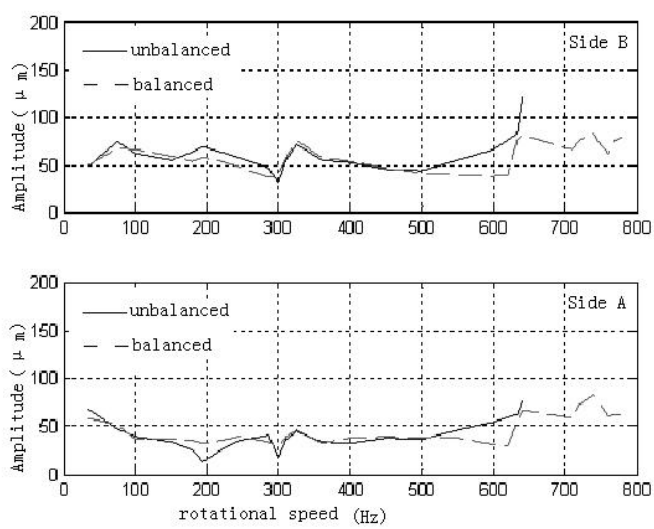

Fig. 5 the Amplitude-Frequency Response of the Rotor before and after the Balance for the 2nd Flexible Modal

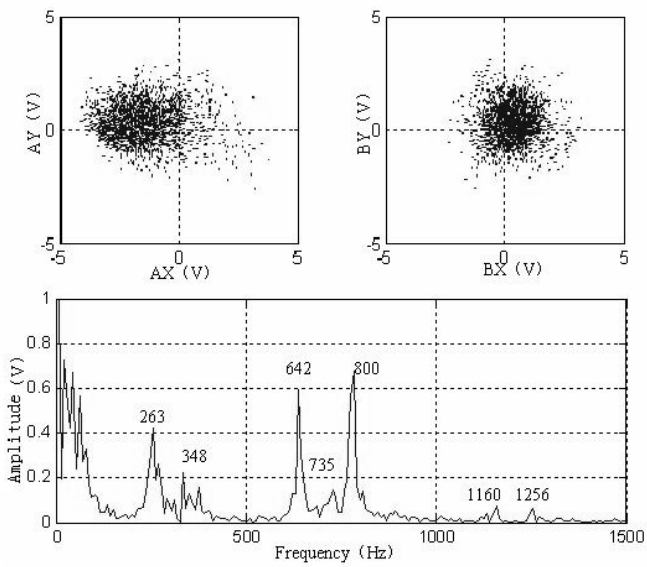

Fig. 6 the Vibration of the Rotor Rotating at 800Hz

CONCLUSION

The field dynamic balance experiments were carried out on the special designed AMBs - flexible rotor system. Two balance methods are combined together. The influence coefficient method is applied to balance the rigid modals and the modal balance method is applied to the flexible modals. After balanced, the rotor supported by AMBs can exceed the first two flexible critical rotational speeds smoothly and can rotate at 800Hz stably. The experimental results indicate that the field dynamic balance method used in this paper is a good method to reduce the rotor's vibration and increase the rotor's maximum secure rotational speed and can boost the performance, safety and cost of the AMBs' system.

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