Vibration Problems in a Turbo Molecular Pump with Active Magnetic Bearings

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Abstract

In experimental tests of a proto-type 2000L turbo molecular pump suspended by 5-DOF AMBs, complex vibration problems had been met. They were caused by different reasons including rotor bending modes, rotor gyroscopic effects, pump blade modes and a vibration mode created by interaction of the dynamics of the rotor and the pump motor. They influenced the pump stability and had to be considered carefully in AMB controller designs to run the rotor to a target speed of 24000 rpm. Different methods were used to deal with these vibration problems. The gyroscopic effects were dealt with using cross feedback control methods. Other vibrations, including the bending mode vibrations, the pump blade vibrations and so on, were dealt with using phase shaping control methods. Experiment results concerning them were provided. With the suitable controller design and the appropriate consideration of the dynamic problems, the rotor was successfully accelerated to its target speed.

1 Introduction

Turbo molecular pumps are important in obtaining and maintaining high vacuum, and widely used in a variety of high vacuum applications. Active magnetic bearings (AMBs) are attractive for their special advantage compared with traditional ball bearings and fluid film bearings, such as contactless, no wear, no oil, low power consumption, low maintenance cost, and controllability of bearing dynamic characteristics and of rotor unbalance response. But AMBs have a lower dynamic stiffness compared with traditional ball bearings [1].

AMBs are very suitable for turbo molecular pumps to use. Because pump loads in operation are small and it is not necessary for pump bearings to provide high dynamic stiffness except for some extreme conditions. More importantly, for an application which is sensitive to grease contamination, such as semiconductor etch processes, pumps with AMBs are popular [2].

In experiment tests of a proto-type 2000L 5-DOF AMB turbo molecular pump, complex vibration problems had been met. They were caused by different reasons, including the pump rotor’s bending modes, influence of the rotor gyroscopic effects, the pump blade modes and a vibration mode created by interaction of the dynamics of the rotor and the pump motor. They influenced the pump stability and had to be considered carefully in AMB controller designs.

The 1st bending mode frequency of the rotor was higher than the rotor’s maximum speed. But it was not high enough for the mode to be ignored in the controller design. Furthermore, the gyroscopic effects would decrease its corresponding backward whirling frequency. The bending mode vibration should be efficiently damped throughout the whole operation speed range within which the mode frequency would shift in a certain range. In fact, without a careful controller design, the bending mode would destroy the AMB stability even in a static suspension.

Because the rotor inertia ratio was large, the rotor modes were greatly influenced by the gyroscopic effects. Besides the bending modes, the rigid modes were influenced more greatly. Without an appropriate consideration about it in controller designs, the nutation (the forward whirling of the rotor’s conical rigid mode) and the precession (the backward whirling of the rotor’s cylindrical rigid mode) would destroy the rotor stability even at a not so high rotation speed.

Pump blades in the pump wheel were thin and not so stiff and their natural vibration frequencies were mainly located in the mid-frequency region. The rotor had to run above critical speeds corresponding to such eigen frequencies. When the speed was close to the blade eigen frequencies, the eigen vibration would be excited by unbalance vibrations, if there were no suitable treatment.

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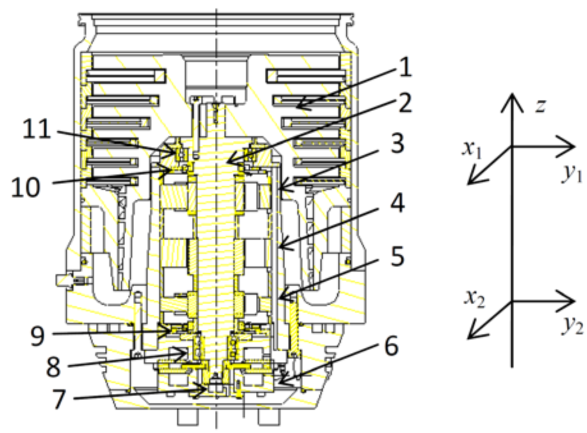
The most inconspicuous but annoying vibration was caused by interaction of the dynamics of the rotor and the pump DC motor. This vibration was closely related with the DC motor. In a primitive test pump system equipped with a three phase squirrel-cage motor, the vibration could not be observed [3]. The interaction mode vibration was not obvious and its vibration character changed when the rotor speed increased. When the nutation frequency was close to its frequency, it would be dangerous. Its stability would be weakened and the system stability would be destroyed.

Different methods were used to deal with the problems mentioned above. Gyroscopic effects were dealt with using cross feedback control methods. Other vibrations, including bending mode vibration, pump blade vibrations and so on, were dealt with using phase shaping control methods. Experiment results concerning those problems were provided. With the suitable controller designs and the appropriate consideration of the dynamic problems, the rotor was successfully accelerated to 24 000 rpm.

2 Pump Rotor System

The structure of the turbo molecular pump was shown in Figure 1, where (x_1, y_1) were the coordinates of the upper radial AMB plane and (x_2, y_2) were coordinates of the lower radial AMB.



1.pump wheel 2.spindle 3.upper radial AMBs
4. DC motor 5.lower radial AMBs 6.axial AMB
7. axial sensor 8.lower backup bearings 9.lower radial sensors
10. upper radial sensors 11.upper backup bearing

Figure 1: the pump structure and the AMB coordinates

The rotor was composed by a steel spindle with a DC motor rotor and an aluminum wheel with blades. The pump wheel was connected with the spindle using 5 screws. Some pump parameters were listed in Table 1.

Parameters	Value
Bearing Bias Current I_0 (A)	0.5
Rotor Mass m (kg)	11
Inertia Ratio of the Rotor	0.45
Maximum Rotation Speed Ω (rpm)	24 000
Motor Power (kw)	0.5
Upper Radial AMB Capacity (N)	550
Lower Radial AMB Capacity (N)	340
Axial AMB Capacity (N)	1100

Table 1: pump parameters

There were complex dynamic characters in the pump rotor, including the characters of the vibration modes from the wheel blades, flexible modes and the rotor's gyroscopic effects. The rotor's dynamic behaviors were studied by

FEM. For simplicity, the blade modes and vibration modes of the whole rotor were analyzed separately. The blades with different sizes were modeled as a cantilever structure and their frequencies were about 330, 370 and 730 Hz respectively. Then in the FEM model construction for the whole rotor, the pump wheel was replaced with a simulation wheel without blades whose gravity center position, polar moment of inertia and transverse moment of inertia were all similar with the real wheel.

The simplified rotor model had similar rotor dynamic behaviors with the pump rotor and the analysis results could be used in controller designs. In analysis, the model was further simplified with 2D Fourier elements [4]. 2D Fourier elements could model a rotor using 2D axisymmetrical volume finite elements whose displacement field was developed in Fourier series. With this approach, the modeling of the rotor with complex meridian cross sections and obvious gyroscopic effects could be achieved. The 2D model for the simplified pump rotor was shown in Figure 2, where the rotor was supported by two springs (stiffness = 500 N/mm) at the center positions of the radial AMBs.

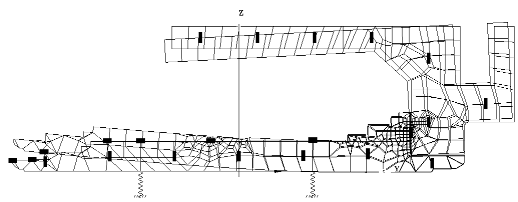


Figure 2: model of the rotor

The rotor’s first 2 bending modes were 538 Hz and 1730 Hz respectively. The rotor’s Campbell diagram was shown in Figure 3. Both the rigid modes and the bending modes of the rotor were greatly influenced by the gyroscopic effects. When the rotor ran to a high rotation speed, the eigenfrequency corresponding to the precession decreased markedly (even close to zero), and on the contrary, the nutation eigenfrequency rose rapidly. Similar behavior also appeared in the forward and the backward whirling frequencies of the 1st bending mode. Since the 1st bending mode frequency was higher than the rotor’s maximum rotation speed in the whole speed range, the rotor could be considered as working in subcritical operation. But the 1st bending mode frequency was not high enough and its value changed with the rotor speed. Its damping in the whole speed range should be taken into account in the controller design.

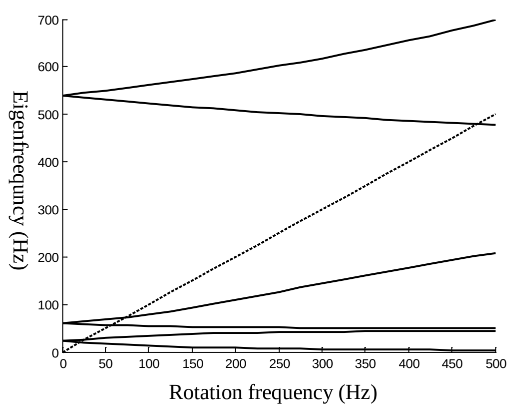


Figure 3: Campbell diagram

3 Interaction mode vibration from the rotor and the DC motor

The motor used in the pump was a 2-pole permanent motor. There was a 2-pole permanent motor rotor fixed on the rotor with a stainless steel sleeve outside. The permanent material used for the motor was not stable enough and the remanence of permanent magnet was changed when the rotor ran at its maximum speed for some hours first time. Then an inconspicuous vibration at about 220 Hz would appear in radial AMBs, especially the lower bearings. It was hard to observe and identify the vibration mode. In static suspension, it could not be excited by radial AMBs. But when sine excitation signals were applied to the corresponding frequency range at a rotation speed above 280 rps, a small peak in the frequency range could be found. When the rotor ran above 350 rps, the vibration peak would appear at near 220 Hz in FFT results of the lower radial displacements.

When the rotor reached its maximum speed, the vibration near 220 Hz was stable at first. In a few hours, the maximum value of the FFT peak corresponding to the vibration would increase slowly but continually. Finally, the vibration would cause the rotor instability.

It was hard to explain the phenomenon. But many experiments for it had been done. The mode was not from the pump housing and not a rotor or blade mode either. It had something to do with the status of the permanent magnet of the motor rotor and the nutation of the rotor. At a high rotation speed, when the rotor nutation frequency was close to this strange mode frequency, the nutation would couple with the 220 Hz mode. Complex but unconspicuous vibration phenomenon would appear.

4 Controller Designs

In a general controller design, there was a great deal research work on suppression of rotor flexible modes, blade modes, rotor nutation and precession. For this pump system, besides these problems, the strange 220 Hz mode vibration was to be solved at the same time. Obtaining a suitable controller for the AMBs became difficult.

For a complex controller design, robust methods, such as H_∞ and μ synthesis, were attractive. But for an industrial pump, they were too complex and more detail information for the pump, such as the strange 220 Hz mode, was needed when they were to be used in the controller design. So some phase shaping methods were used for the AMB controllers. The phase shaping methods could make use of different kind of filters to change magnitude and phase of a controller in a local frequency domain and effectively restrain a mode vibration in the frequency domain. These filters were usually added to a PID controller to change its performance in a local frequency domain and their parameters were easy to be adjusted according to different vibration modes.

4.1 The flexible mode vibration suppression

For an AMB rotor whose operation speed surpassing or approaching its bending mode eigenfrequencies, the suppression of the mode vibration had to be considered carefully, or the self-excitation would happen easily when disturbance forces with a frequency near the eigenfrequencies acted on the rotor, such as synchronous unbalance force. A Phase Shaping method could be used to improve the bending mode damping effectively.

The Phase Shaping method used here was realized by adding suitable filters into the basic PID controller of the radial bearings and changing its amplitude and phase characters in a local frequency domain. Bode diagram of the result filter used was shown in Figure 4.

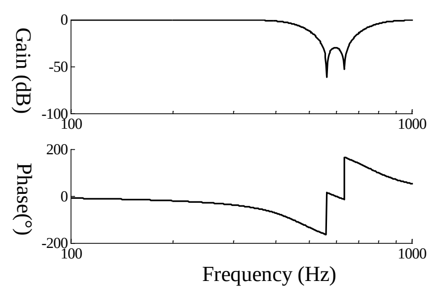


Figure 4: Bode diagram of the filter for 1st flexible mode

It was a second order notch filter and it changed the damping character of the original PID controller obviously. The phase of the controller in the frequency range near the 1st bending mode was obviously increased and it could provide sufficient damping for the mode vibration, even when the backward whirling eigenfrequencies of 1st bending mode were varied obviously due to the gyroscopic effects at a high rotation speed.

4.2 The blade mode vibration suppression

Because of the pump wheel's complex shape (there were many blades on it), its dynamics influence the AMB stability. The blade modes whose frequencies were about 330, 370 and 730 Hz respectively would be excited by synchronous unbalance force of the rotor or its harmonic components.

For these blade modes, notch filters similar with the notch filter for the rotor's 1st flexible mode were used and they could effectively restrain the modes. But a notch filter near 330 Hz would obviously influence the phase lead characteristic of the controller below 330 Hz.

The Bode diagram of a notch filter used to restrain the 330 Hz blade mode was shown as the solid curve in Figure 5. Though the notch filter near 330 Hz could decrease the magnitude of the controller there, it also influenced the controller phase near 175 Hz and caused about 17 degree phase shift. It would do harm to the suspension of the rotor nutation at a high rotation speed (when the rotor speed was 400 rps, the nutation frequency would be about 175 Hz).

So another kind of filter, a 2nd filter with the form of Equation (1) was used to replace the notch filter. Bode diagram of a sample filter was shown as the dash curve in Figure 5. The filter could adjust the damping character in a local frequency domain around the 330Hz mode and it didn't cause obvious controller phase delay below 330 Hz.

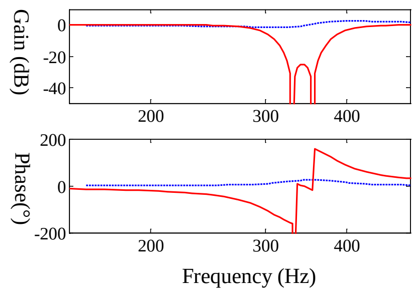


Figure 5: Bode diagram of the filter for 330 Hz blade mode

$$f = \frac{[s-2\pi(a_1+jb_1)][s-2\pi(a_1-jb_1)]}{[s-2\pi(a_2+jb_2)][s-2\pi(a_2-jb_2)]} \quad (1)$$

4.3 The 220 Hz mode vibration suppression

Though the origin of the 220 Hz mode vibration was unclear yet, experimental results showed that it was sensitive to the magnitude of the controller near 220 Hz. If the magnitude was large, it was hard to restrain the vibration whatever the controller phase was. So the most critical requirement for the mode suppression was to decrease the controller magnitude near 220 Hz.

Since the nutation frequency of the rotor at 400 rps was about 175 Hz, when phase shaping near 220 Hz was applied to the controller, additional consideration should be given to the nutation damping. When the final controller was designed, the mode suppression, the nutation and precession suppression were considered carefully at the same time.

4.4 The nutation and precession suppression

Generally, a PID controller could provide a basic performance for an AMB rotor system. But it couldn't satisfy the stability requirement of rotors with strong gyroscopic effects at a high rotation speed.

When a simple PID controller was used to control the pump AMBs, it worked well at low rotation speeds. But the rotor's rigid modes including the precession and the nutation would be unstable when the speed increased. Because, when the rotation frequency was close to the frequency range where the integrator of the PID controller changed the phase characteristic of the AMBs obviously, the precession damping would be destroyed. Furthermore, it was difficult for the PID controller to restrain the nutation at a high speed. The gyroscopic effects would change the nutation frequency to a much higher value and make it harder for the controller to damp the nutation in a higher frequency range, especially for the pump AMBs whose bandwidth was limited by the application characters, such as the limited AMB size, the limited current and voltage for the AMB coils in vacuum, and so on.

There were different ways to restrain nutation and precession of AMB rotors. A direct way was to adjust a PID controller's magnitude and phase frequency characteristics by adding more complex filters to it. When the rotation speed was not so high, increasing a controller's amplitude and damping ratio was reasonable by this way. But the bandwidth would be increased to satisfy requirement of damping in a wide frequency range, the magnitude of the controller would be high and vibration modes without strong damping would be easier to be excited. When the rotation speed was high enough, the precession frequency was close to zero and the nutation was increased markedly, only using local channels of the controller could hardly dealt with the nutation and precession.

The Cross Feedback method was competitive for its good adaptation for rotor models and simplicity in realization [5, 6].The controller structure was shown in Figure 6. The control of axial degree was considered decoupled with the radial motion and not considered here. The Cross Feedback control was achieved by adding cross feedback channels with suitable phase shaping filters to a controller with only local PID control channels. It could weaken the influence of the gyroscopic effects and increase the nutation and precession damping.

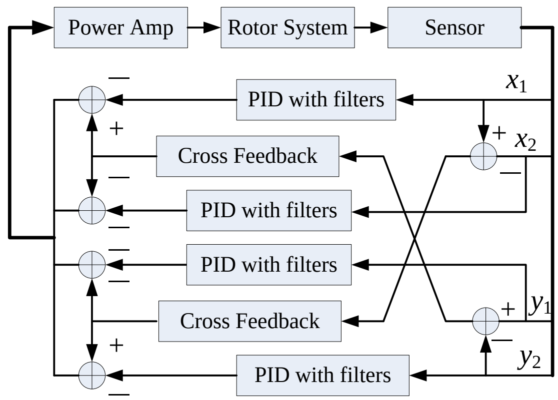


Figure 6: cross feedback controller structure

For a rigid rotor with obvious gyroscopic effects, the transfer function of a cross feedback channel could be simply composed by a one order low pass filter (for precession) and a one order high pass filter (for nutation) with opposite sign. But for the pump rotor with the complex dynamics, the 220 Hz mode and the blade modes should also be considered in the transfer function design for the cross feedback channels.

The basic transfer function of the cross channel was as Equation (2). It had a simple structure similar with a PID controller.

$$k_{cl}/(T_{lp}s+1)+k_{ch}s/(T_{hp}s+1) \tag{2}$$

Based on the transfer function as Equation (2), some filters were added to deal with the 220 Hz mode and the blade modes. The main idea was to decrease the magnitude of the cross feedback channel in the frequency ranges corresponding to the 220 Hz mode and the blade modes without obviously decreasing its leading phase for the nutation. At last, the filters used were a notch filter and some filters similar with Equation (1) and the detail parameters were obtained through experimental tests.

5 The final controller

With careful consideration as discussed above, the final controller was obtained and its effectiveness was tested in experiments.

For the pump rotor, the center of its upper radial AMB rotor was almost coincided with the rotor's mass center. So the suspension of the vibration modes discussed above depended almost completely on the lower radial AMBs. The filters for the phase shaping were mainly added to the lower AMB controller.

The Bode diagram of the local channel controller for x_2 was shown in Figure 7 (the controller for y_2 was the same). It could be seen that the magnitude near 220 Hz was purposely low while a suitable leading phase was ensured, and the magnitude near the blade modes and the bending mode was obviously low.

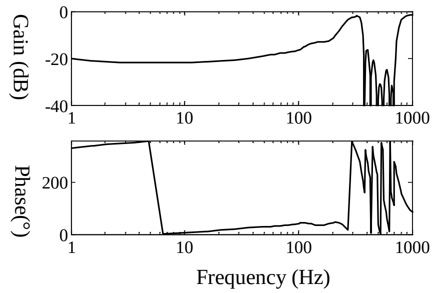


Figure 7: Bode diagram of the controller for x_2 and y_2 local channels

The Bode diagram of the cross feedback channel was shown in Figure 8. The magnitude near 220 Hz was also purposely low while a suitable phase lead was ensured. The influence of the controller to the blade modes were weakened by a notch filter.

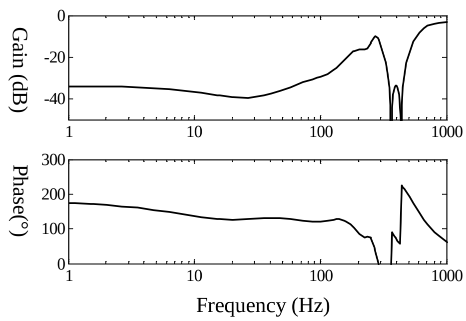


Figure 8: Bode diagram of the cross feedback channel

6 Experimental results

Some important experimental results related to the previous discussion were provided below.

As discussed above, the AMB stability was obviously influenced by the gyroscopic effects of the rotor. When only the local channel transfer function of the controller as Figure 6 was used, the controller could not provide enough damping for the nutation and the precession. It was shown in Figure 9 that the nutation vibration was continually increased and would soon lose its stability when the rotor only ran to 237 rps. When the transfer functions of the cross channels with the filters for the nutation were added, the nutation could be restrained and the rotor could run to a higher speed. Soon, the precession became a problem as seen in Figure 10. When the rotor first ran to 283 rps, the precession destroyed the stability of the rotor. So the filters for the precession were added to the cross channels and the precession was well restrained after that.

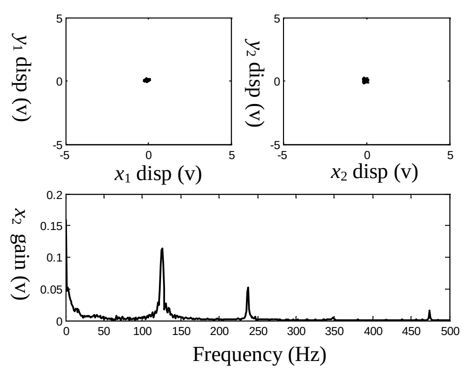


Figure 9: orbits and the FFT analysis results of the AMBs at 237 rps

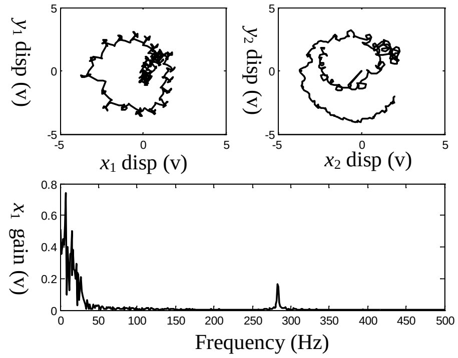


Figure 10: orbits and the FFT analysis results of the AMBs at 283 rps

When the nutation and the precession were well restrained and the rotor ran at a speed of 380 rps, without careful consideration about the 220 Hz mode in the controller, the vibration corresponding to the mode increased rapidly as shown in Figure 11. It was seen that, the peak amplitude of the 220 Hz mode became much higher than the synchronous rotor vibration and the AMB rotor would soon lose its stability.

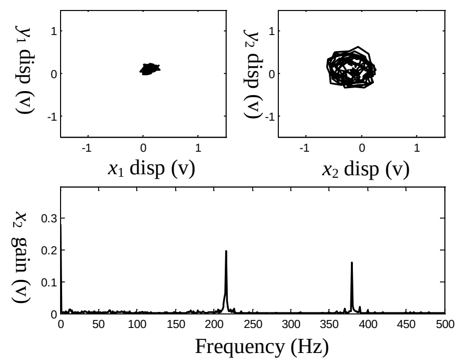


Figure 11: orbits and the FFT analysis results of the AMBs at 380 rps

With the final controller designed above, the rotor was successfully accelerated to 24 000 rpm. The corresponding orbits and the FFT analysis results were shown in Figure 12.

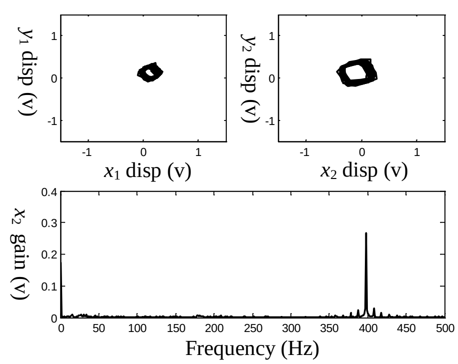


Figure 12: orbits and the FFT analysis results of the AMBs at 24 000 rpm

7 Conclusion

For the AMB turbo molecular pump, the complex vibration modes on the rotor greatly influenced the system stability. The modes included the bending mode, the nutation and precession, the blade modes and the strange 220 Hz mode. With the controller carefully designed according to the characteristics of all these modes, the good experimental results were obtained. The gyroscopic effects were dealt with using cross feedback control methods. The bending mode vibrations, the blade mode vibrations and the strange 220 Hz mode were all dealt with using phase shaping control methods. By using the controller designed, the rotor successfully ran to its target speed 24 000 rpm.

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