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Rotordynamic Behavior and Rigid Mode Vibration Control by Hybrid Foil-Magnetic Bearing System

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Abstract

In a hybrid foil-magnetic bearing (HFMB), an air foil bearing (AFB) serve as auxiliary bearings for an active magnetic bearing (AMB). This enables design simplification and weight reduction, and it can reduce the energy consumption of the AMB. Therefore, an HFMB has higher efficiency and reliability. This study conducts vibration control tests of an HFMB at increasing rotor speeds for various AMB control parameters. AMB control parameters are experimentally determined for achieving an appropriate load distribution between the AMB and the AFB. Furthermore, to minimize the vibration and energy consumption, the ON/OFF period control of the AMB is adjusted properly according to the operating speed. Accordingly, the HFMB can afford the high load capacity of the AFB at high speed as well as the high static stiffness to prevent initial rubbing and the control capability of the AMB to suppress excessive vibration. Therefore, the HFMB improves the performance of rotating machinery significantly in terms of range of operation and reliability. In addition, the effects of the load distribution between the AMB and the AFB are investigated by changing the control characteristics to produce different stiffness and damping effects. The HFMB for a high-speed rotor is experimentally optimized by altering the load distribution of the two bearings in terms of rigid mode vibration.

Keywords : Hybrid bearing, Active magnetic bearing, Air foil bearing, Rotordynamics, Vibration control

1. Introduction

Recent developments in compact, oil-free rotating machinery have led to demands for high reliability despite high-speed operation. Air-foil bearings (AFBs) satisfy most of these requirements in that they offer a higher degree of reliability than rolling-element bearings. AFBs eliminate the need for complex oil lubrication and sealing systems, and they have excellent high-speed characteristics because they are driven without friction as the rotor is floated on an air film generated at high speed. However, Agrawal (1997) was presented that AFBs suffer from initial rubbing and a low damping effect prior to reaching the critical speed of the rotor-bearing system. Active magnetic bearings (AMBs) offer the advantage of low-speed operation because metal-to-metal contact is avoided. In addition, dynamic balancing is possible, and both their stiffness and degree of damping are electronically adjustable. However, despite these advantages, AMBs employ conventional ball bearings as an auxiliary bearing in case of electrical failure. Conventional ball bearings support limited rotating speed, and ball bearing failure may occur at high rotating speed. AFBs show superior high-speed performance, and therefore, better performance is expected for the conventional auxiliary bearing role of AMBs. Heshmat et al. (2000) proposed a hybrid bearing in which an AFB and AMB support a rotor. And Swanson et al. (2002), they studied the load distribution between the bearings, proposed a monitoring control algorithm that can provide proper stability to the system stiffness and speed change, and experimentally verified that the performance was excellent. However, because the AFB and AMB structures supported different rotor positions, the system had large size, and it could not eliminate the need for the AMB auxiliary bearing. Furthermore, this system had limited maximum vibration reducing effect. To overcome these disadvantages, the present study proposes a hybrid bearing that has a structure that combines an AFB and an AMB as one bearing, thus reducing the volume of the bearing. The proposed system is expected to reduce the vibration and improve the stability because each bearing can support the same rotor position. Figure 1 shows the vibration reduction strategy used in the HFMB. This strategy is expected to reduce the vibration and improve the stability. The rotor supported by the AFB generates friction with a bearing and rubbing owing to the increase in the vibration amplitudes at critical speed or the occurrence of excitation. In this critical mode vibration, the HFMB provides a magnetic effect by turning on the AMB's controller and floats up the rotor. In this way, it can reduce the vibration caused by the critical mode. At high speed, it reduces the AMB's supporting load by turning off the AMB and supports the rotor using only the AFB. Moreover, it maximizes the AFB's performance at high speed and can reduce the electric power consumption. As a result, it can reduce the vibration occurrence as well as the total power consumption at critical speed.

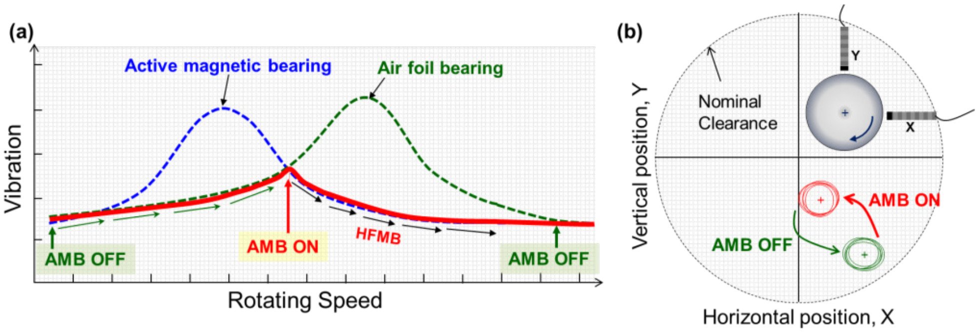

Fig. 1 Strategy concept for vibration control with HFMB: (a) Vibration curve of each bearing, (b) Change of rotor center as AMB controller is switched ON/OFF

2. Design of hybrid foil-magnetic bearing and rotor

As shown in Figure 2(a), a common AMB is usually installed with ball bearings as auxiliary bearings. This study introduces the compact hybrid bearing structure shown in Figure 2(b). It simplifies the structure by using the AFB to replace the auxiliary bearing of the AMB, and the AMB compensates the damping force. The structure uses the respective functions of the bearings in a hybrid type. To insert the AFB in the gap between the inner diameter of the AMB and the rotor, the space where the coils are wound inside the AMB was designed to perform the role of the AFB's housing by molding it with epoxy having no magnetic properties. A displacement sensor was mounted to control the rotor's position according to the AMB. Thus, the AFB and AMB of an HFMB can support a rotor at the same position, and the vibration performances of each bearing can be compared and analyzed; in this hybrid bearing system, therefore, the AFB and AMB supplement each other's weaknesses. Lee (2010) has this design patents. And the detailed dynamic analysis of HFMB has been published by Jeong et al. (2015).

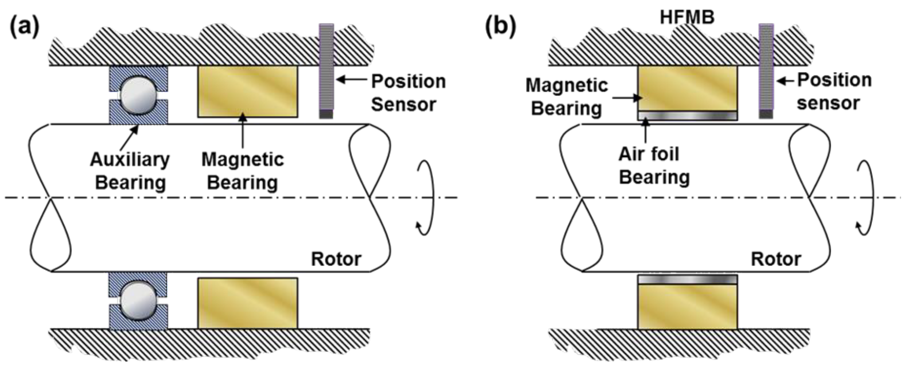

Fig. 2 Schematic view of (a) Conventional magnetic bearing structure and (b) Hybrid foil-magnetic bearing.

2.1. Design of active magnetic bearing and air foil bearing

An AMB offers the advantage of low-speed operation because metal-to-metal contact is avoided. Furthermore, both the stiffness and the damping can be electronically controlled. This bearing is being studied for applications to smart and intelligent machines because a specific air gap can be maintained between the rotor and the bearing by controlling the magnetic force through real-time feedback control by measuring the distance from a rotor using a displacement sensor. However, despite these advantages, the AMB and rotor levitation may be difficult to maintain in case of a fault owing to electrical failure or sensor failure. Typically, the AMB is designed to support the rotor through an auxiliary bearing or backup bearing. Usually, simple retainer rings or special ball bearings are used as auxiliary bearings. In this study, the AMB structure is designed such that the AFB can play the role of an auxiliary bearing. The design dimensions of the AMB are summarized and presented in Figure 3 and Table 1. The AMB was designed as a hetero-polar radial type, which is the most widely used, and comprised four electromagnets.

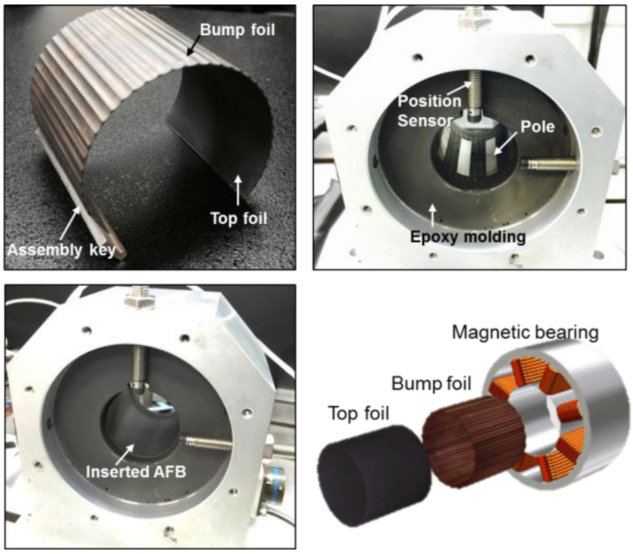

Fig. 3 Manufactured HFMB by combined AFB and AMB.

Table 1 Geometry and dimensions of AMB.

Parameters	Values
Permeability of free space	$4\pi\times10^{-7}$ H/m
Bias current	1.2 A
Area of pole	210 mm^2
Coil turns	240
Air gap thickness	0.7 mm
Pole angle	22.5°
Resistance of coil	1.4 Ω
Induction of coil	0.6 mH
Current stiffness	34.4 N/A
Static stiffness	54.5 kN/m

The design dimensions of an AFB are summarized and presented in Figure 4 and Table 2. An AFB eliminates complex oil lubrication and sealing systems and reduces friction at high rotation speed, heat generation, and power losses. It comprises a bump foil and a top foil. It has excellent high-speed characteristics because it is driven without friction as the rotor is floated on an air film generated at high speed.

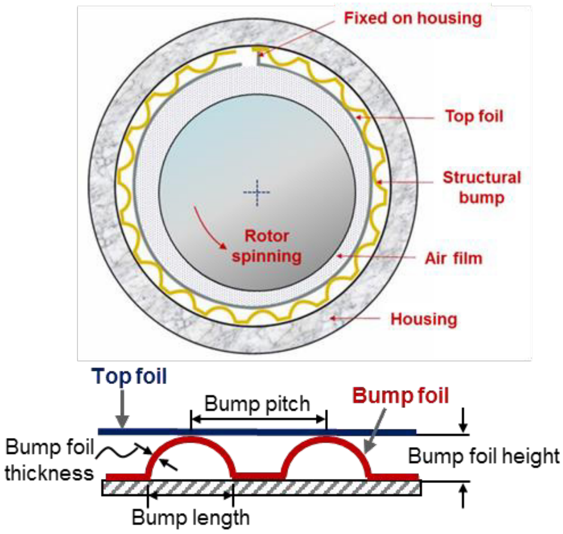

Fig. 4 Structure of AFB and bump foil geometry.

Table 2 Geometry and dimensions of AFB.

Parameters	Values
Bearing width	45 mm
Bearing housing diameter	36.4 mm
Rotor diameter	35 mm
Top foil thickness	0.147 mm
Bump foil thickness	0.127 mm
Bump pitch	3.0 mm
Bump half length	1.5 mm
Bump height	0.4 mm
Static stiffness	2.0 MN/m

However, an AFB suffers from initial rubbing and small damping coefficients for crossing the critical speeds of a rotor-bearing system. The bearing stiffness and load bearing capacity were analyzed by in-house codes. Rubio and San (2002) were established an analysis method of the bump foil structural stiffness. The bump foil was found to have a static stiffness of 2.0 MN/m.

2.2. Eigenvalue analysis of rotor bearing system

The rotor is made of SUS630 and is rigid; it has a length of 484 mm, weight of 4.4 kg, and diameter of 35 mm. It has a permanent magnet at its center. The critical mode is predicted by using the commercially available analysis software RAPP. The translatory mode and conical mode were obtained at 147 Hz (8,846 rpm) and 175 Hz (10,502 rpm), respectively, when the AFB's stiffness was 2 MN/m. In addition, the experiment results obtained between 120 and 160 Hz showed good consistency with the predicted results. The undamped critical speed analysis results and mode shapes are shown in Figure 5. Based on these analysis results, the AMB is working in rigid mode region by the AFB. It can be expected that the rigid mode vibration remove.

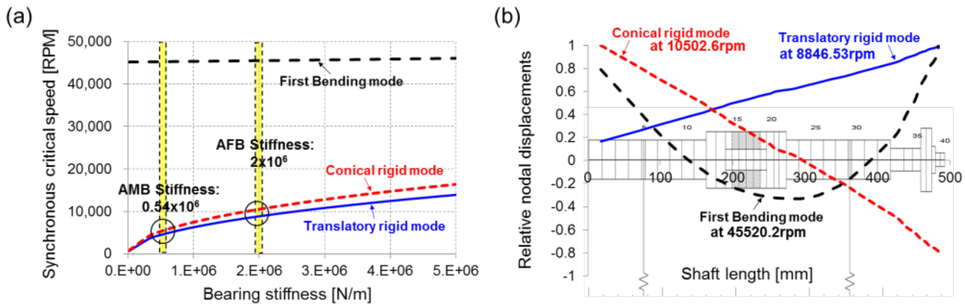

Fig. 5 Eigenvalue analysis result: (a) Undamped critical speed map, (b) Mode shapes.

3. Driving process and estimation of AMB control

3.1. HFMB driving algorithm

AMB operates on the closed loop driving method. First, the required current value for magnetic force control is calculated by the displacement controller using the position value received from the displacement sensor. Second, the current controller is applied to the AMB's coil. The rotor's position is constantly maintained by changing the magnetic force of the AMB, and the displacement value at this moment is fed back again. Thus, real-time displacement control is carried out in a closed loop mode. On the other hand, this study provides a magnetic effect by turning on the AMB's controller in the critical mode determination. This procedure is shown in Figure 6.

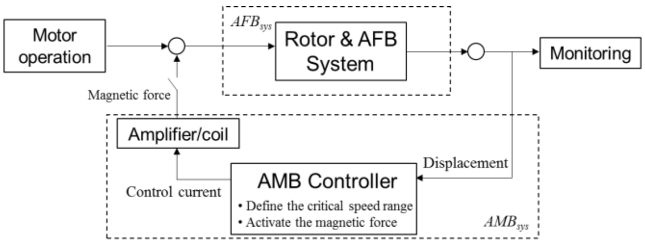

Fig. 6 Block diagram for HFMB system driving procedure.

3.2. Optimization of HFMB control gain

The AMB controller is using by a general proportional-derivative (PD) control method (Lewis, 1992). To experimentally determine the optimum region for the load distribution of both bearings, the stiffness of the HFMB was changed by varying the proportional control gain (*kp*). Jeong (2010) was studied the dynamic behavior and performance of the flexible rotor supported by hybrid bearing. Pham and Ahn (2014) were studied for optimization of a hybrid foil-magnetic bearing to support a flexible rotor. Because the experimentally determined control parameters are most important and strongly in HFMB. The load carrying capacity of each bearing can be adjusted by changing the proportional gain of the magnetic bearing, as shown in Figure 7. It shows the vibration orbit results for *kp* values of 2–7 A/mm at 135 Hz (8,100 rpm) of rigid mode rotating range.

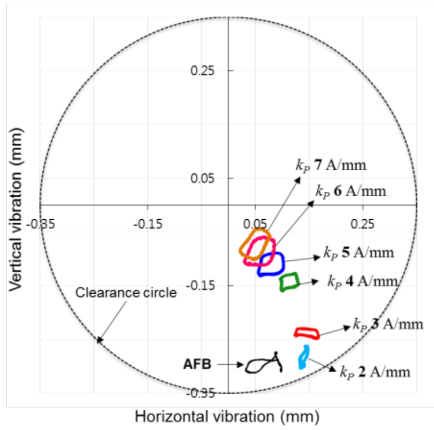

Fig. 7 Orbital vibration and rotor center comparison with various proportional control gain at 8,100 rpm.

As the proportional gain increases, the stiffness and the load of the magnetic bearing increase. Moreover, the rotor center position of the AMB changes the load sharing of the HFMB. As the rotor center approaches the journal center, the stiffness and load of the AFB decrease. The variable proportional control gain has the greatest impact on the vibration amplitude and controls the stiffness of the HFMB system. It is possible to move the center of the rotor by controlling *kp*, and this serves to help control the response when the AMB controller is turned ON/OFF. For these control gains, the load carrying capacity of the HFMB is determined to be insufficient. Based on these results, a proper *kp* value is obtained experimentally. Accordingly, it is confirmed that values of 4–7 A/m are appropriate for enhancing the HFMB performance.

4. Experiment and unbalance responses measurement
4.1 Test rig configuration and condition

Based on the critical speed modes of the rigid rotor, the control performance and dynamic characteristics of the HFMB were verified experimentally. In addition, an experimental setup featuring a rigid rotor supported by an HFMB was built. The HFMB test rig was constructed, as shown in Figures 8. The balance of the rigid rotor was maintained by the HFMBs at both ends of the rotor, which were controlled based on the position of the rotor as measured with sensors. The rigid rotor position was measured with position sensors. And the measured results were compared with those of the vibration measurements to evaluate the dynamic behavior.

The switching speeds were predetermined in controller. The strategy is that uses the HFMB effectively through switching control. First, the AFB support the load alone. And the HFMB is possible to support the rotor when the vibration occurs too large while the motor is running. If the rotor speed reaches the near mode speed, before a large vibration, the AMB was turned on. Therefore, the rotor speed is between 120-160Hz (the predetermined mode speed) AMB controller is turned on and off.

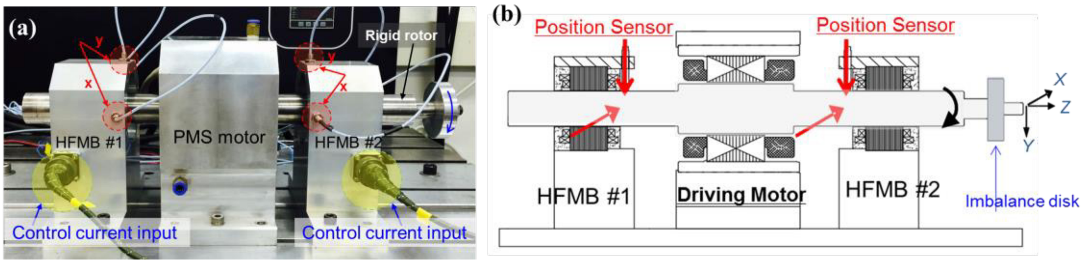

Fig. 8 Test rig configuration for HFMB: (a) Photo of test rig, (b) Schematic view of test rig.

4.2 Vibration measurement of AFB and HFMB

Rotor vibrations of run-up tests from 0 to 200 Hz are shown in Figure 9. The results in Figure 9(a) indicated that the AFB's rigid mode vibration is ~150 Hz. When the magnetic effect is applied entirely, as shown in figure 9(b), the HFMB's rigid mode vibration is ~90 Hz. The unbalance response in Figure 9(c) is much smaller than that in the other two settings. In particular, the rigid mode vibrations of the AFB are well suppressed by switching the HFMB's controller ON/OFF. In these experiments, a test that produced rigid mode vibration control in the rotational machine was conducted successfully, and the vulnerability of the AFB owing to its low damping force was verified. Furthermore, when a large unstable vibration component occurs owing to the rigid mode or rotor imbalance in a real turbo-machine, it can cause fatal damage to an AFB-supported system. Thus, a study to improve the AFB's damping force will be needed. The third result verifies the robustness of the AMB's controllability and stability. It is observed that by replacing the AFB's weak damping force, the AMB can suppress the critical mode vibration of a rotational machine. Accordingly, not only an improvement in the vibration performance of rotational machinery but also improvements in the durability and stability of the whole system can be expected by selectively using the AMB's dynamics in the rigid mode or at the point of critical speed in the AFB's operation. It was further shown that this prevents damage to turbo-machinery by sensing and monitoring the critical vibration caused by the occurrence of rigid mode vibration. On the other hand, when the AMB controller is turned on and off, the rotor is moving to center position. Therefore, sudden peak data in Figure 9(c) means that rotor position moved.

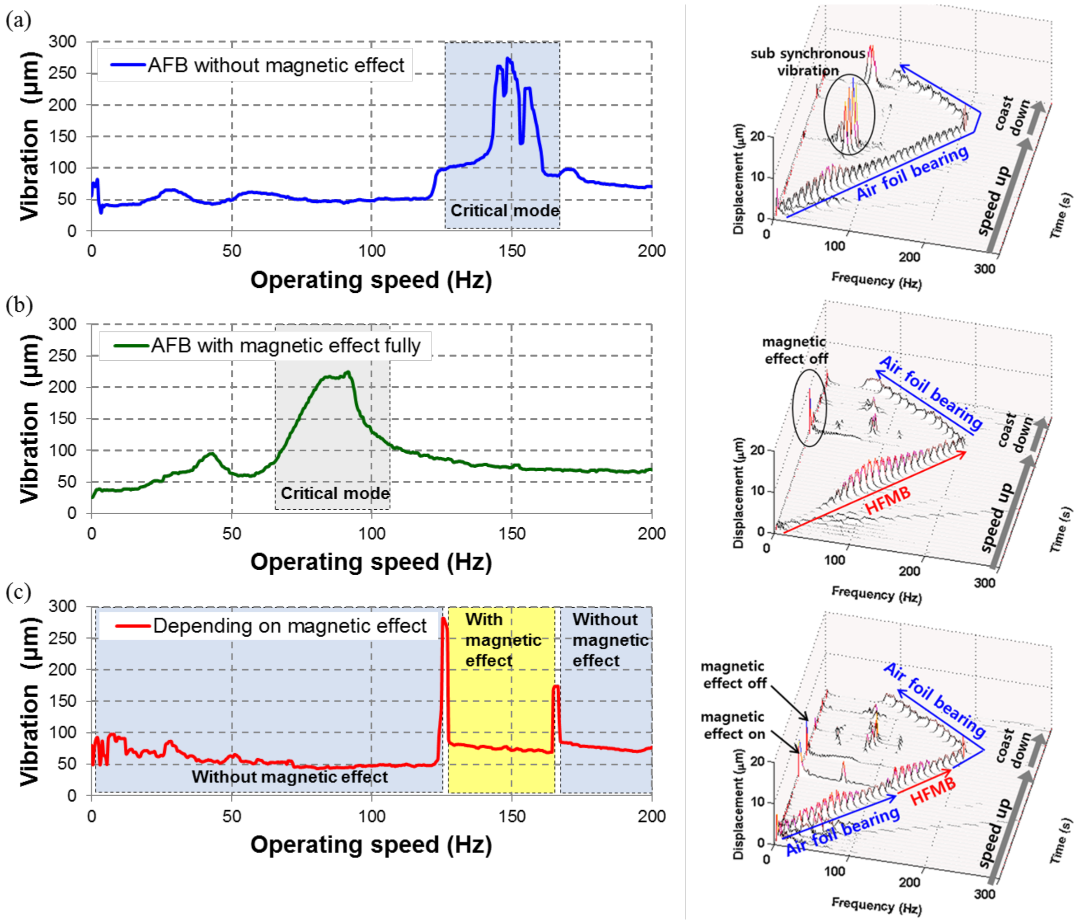

Figure 9. Test results for rigid mode vibration suppression: (a) AFB without magnetic effect, (b) AFB with full magnetic effect, (c) Depending on magnetic effect.

5. Discussions for rotordynamic behavior with HFMB

5.1 FFT data analysis at rigid mode vibration

Figure 10 shows the results of FFT in the rigid mode period for AFB and HFMB bearings. The AFB shows the rigid mode at 140 Hz (1x); at this time, the component of 0.5x vibration that appears is largely asynchronous. The HFMB shows the rigid mode at 100 Hz without any subsynchronous vibration; however, its 1x vibration amplitude is larger than that of the AFB. These results indicate the need to reduce the vibration of the entire system by properly and selectively using the magnetic effects of the HFMB.

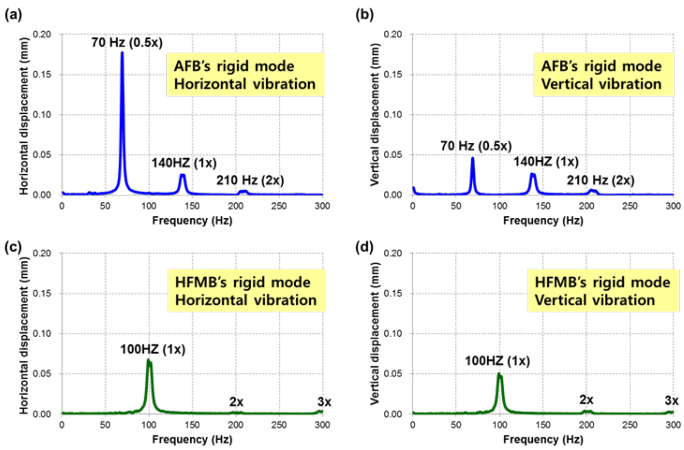

Figure 10. FFT data comparison of rigid mode frequency: (a) AFB's horizontal vibration, (b) AFB's vertical vibration, (c) HFMB's horizontal vibration, (d) HFMB's vertical vibration.

Figure 11(a) and (b) show the moments when the AMB controller is switched ON and Figure 11(c) and (d), the moments when it is switched OFF. The controller power was applied at 120 Hz and turned off at 160 Hz. Subsynchronous vibration of AFB was almost suppressed, and the 1x vibration amplitude did not increase further while the AMB controller was turned on. It is possible to evaluate the robust controllability of the HFMB.

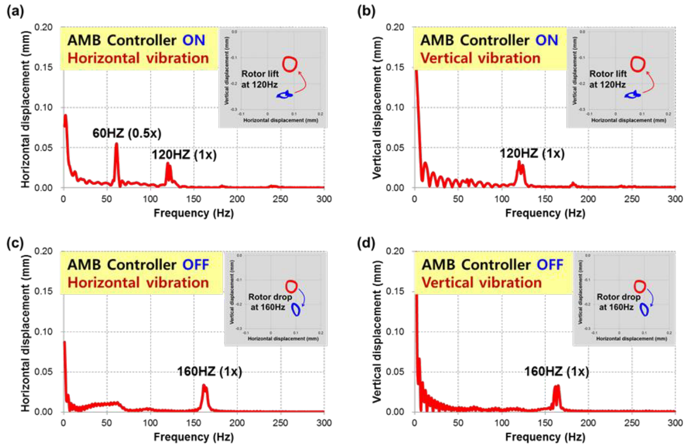

Figure 11. FFT data comparison at the moment the AMB controller is switched ON/OFF: (a) AMB ON Horizontal vibration, (b) AMB ON Vertical vibration, (c) AMB OFF Horizontal vibration, (d) AMB OFF Vertical vibration.

5.2 Rotor trajectory motion and vibration with AFB vs. HFMB

The applied the magnetic effect lifts the rotor to the center housing and can reduce the friction and vibration between the rotor and the bearing. Figure 12 show the rotor being moved to the center by applying the magnetic effect during rotor speed-up. This HFMB is characterized by the high load capacity at critical speeds of the AFB that prevent rubbing and that leverages the controllability of the magnetic bearing to suppress excessive vibration. The test results show that the vibration performance of the HFMB is better than that of the AFB at critical speed. At rigid mode speed, the HFMB plays a dominant role that makes the rotor lift up from the top foil of the AFB. This levitation reduces the friction between the rotor and the top foil of the AFB and efficiently reduces motor current consumption. At high speed, an air gap is produced between the rotor and the top foil of AFB. This air gap pushes the foil away from the rotor more than occurs by the effect of levitation by AMB. Therefore, the AFB has the advantage of saving energy at high speed compared with the HFMB.

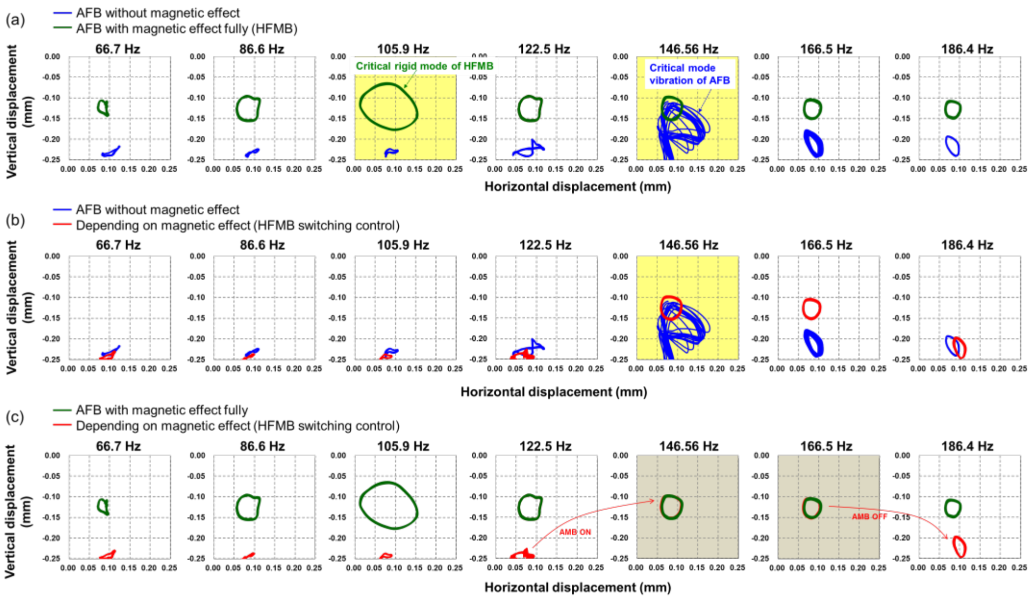

Figure 12. Vibration trajectory comparison according to the rotating speed: (a) AFB vs. HFMB, (b) Vibration suppression of AFB's rigid mode, (c) Vibration reduction of HFMB's rigid mode.

6. Conclusions

Base on the critical speed modes of the rigid rotor, the control performance and rotordynamic behavior of the HFMB were verified experimentally. In addition, an experimental setup featuring a rigid rotor supported by an HFMB was built. The vibration responses of the rigid rotor supported by three different bearings compared to verify the effectiveness of the HFMB. Accordingly, the HFMB can afford the high load capacity of the AFB at critical speed as well as the high damping force to prevent rubbing and the control capability of the AMB to suppress excessive vibration. Therefore, the HFMB improves the performance of rotating machinery significantly in terms of range of operation and reliability. In addition, the effects of load distribution between the AFB and the AMB are investigated by changing the proportional control gain of the AMB. The HFMB for a high-speed rotor is experimentally optimized by altering the load distribution of two bearings in terms of unbalance responses measurement. The results obtained are summarized as follows.

1. An HFMB has a short length compared to the hybrid bearing suggested by Heshmat. Therefore, it affords design advantages in high-speed turbo-machinery applications.

2. Load sharing between the AFB and the AMB are critical issues in an HFMB. It was investigated experimentally by varying the proportional control gain of the AMB. The variable proportional control gain has the greatest impact on the vibration amplitude and rotor position of HFMB. The optimum control parameters were improved the rotordynamic performance with HFMB.

3. An experiment that induced a magnetic effect in a rotor supported by an HFMB was implemented. A switching control algorithm was proposed that senses a rotor's rigid mode and allows the HFMB to improve the vibration performance and stability based on the magnetic force. The results were verified experimentally. Further, HFMB can expect to be helpful in the development of improved technology to turbo smart machinery.

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References

Agrawal, G. L., Foil air/gas bearing technology – an overview. International Gas Turbine & Aeroengine Congress & Exhibition, Orlando, Florida, 1997, ASME paper 97-GT-347.

Heshmat, H, Chen, H. M. and Walton, J. F., On the performance of hybrid foil-magnetic bearings. ASME J Eng Gas Turbines Power 2000; 122(1): 73–81.

Jeong, S., Ahn, H. J., Kim, S. J. and Lee, Y. B., Vibration control of a flexible shaft supported by a hybrid foil-magnetic bearing. In: IFToMM 8th International Conf. on Rotor Dynamics, Seoul, Republic of Korea, 12–15 September 2010.

Jeong, S., Choe, B. S. and Lee, Y. B., Rotordynamic Behavior and Performance of Controllable Hybrid Foil-Magnetic Bearing. Proceedings of the 9th IFToMM International Conf. on Rotor Dynamics, Mechanism and Machine Science, 2015; 21: 1465-1475.

Lee, Y. B. et al., Airfoil-magnetic hybrid bearing and a control system thereof. Patent US8772992 B2, USA, 2010.

Lewis, F. L., Applied Optimal Control & Estimation: Digital Design & Implementation. TI Series, 1992. p.266-268, Prentice Hall, Englewood Cliffs, NJ.

Pham, M. N. and Ahn, H. J., Experimental optimization of a hybrid foil-magnetic bearing to support a flexible rotor. Mechanical Systems and Signal Processing, 2014; 46(2): 361-372.

Rubio, D. and San, A., Bump-type foil bearing structural stiffness: Experiments and predictions. In: Proceedings ASME Turbo Expo, Vienna, Austria, 2004, ASME Paper No. GT 2004-53611.

Swanson, E., Heshmat, H. and Walton, J. F., Performance of a foil-magnetic hybrid bearing. ASME J Eng Gas Turbines Power 2002; 124(22): 375–382.