Validation of fluid disturbance prediction method and control system design considering axial magnetic bearing frequency characteristics for centrifugal compressor

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Abstract
Magnetic bearings are a technology that can realize oil-free and maintenance-free rotating machinery and provide high added value to products. However, compared to rolling bearings and oil film bearings of the same size, their bearing stiffness is smaller, so they have the disadvantage of weakness for disturbance forces such as unbalance forces and fluid disturbances. Furthermore, because it is difficult to apply a laminated steel core to an axial magnetic bearing, it is necessary to consider the decrease in magnetic attraction force and phase lag due to eddy currents. Therefore, in this paper, with the purpose of ensuring product reliability for centrifugal compressors supported by magnetic bearings, a fluid disturbance prediction method and a practical modeling method for the frequency characteristics of axial magnetic bearings were developed. Regarding the fluid disturbance prediction method, a simple prediction method for disturbance forces was considered by assuming the change in pressure difference in the impeller when surge occurs. The frequency characteristics of the axial magnetic bearing were measured in a component level test of electromagnet, and the effectiveness of the proposed modeling method was confirmed. After that, the design reliability of a rotor system was confirmed in a system level test of a centrifugal compressor. Finally, as a product level test, actual operation was performed in a shop test. It was confirmed that normal operation was possible within the allowable vibration level even when a fluid disturbance occurred, and the fluid disturbance prediction method was validated.

Keywords : Magnetic bearing, Eddy current, Frequency characteristics, Centrifugal compressor, Surge, Rotating stall, Fluid disturbance

1. Introduction

Magnetic bearings for rotating machinery are a technology that uses the magnetic attraction of an electromagnet to support the rotor without contact. Examples of the application of magnetic bearings include compressor-driven high-speed rotating machinery such as turbomolecular pumps. In compressors, surges and rotating stalls can occur depending on the operating conditions, and unstable vibrations can become a problem when large disturbance forces act on the rotor (Knight and Corleto, 2011). When a compressor is operated in the low flow rate range, stall cells occur in the impeller and diffuser, and the stall cells rotate in the direction of rotation at a speed slower than the rotation speed. Rotating stall has various adverse effects such as noise, vibration, reduced efficiency, and fatigue damage to the impeller. Furthermore, when the flow rate is reduced and falls below the limit point, the discharge side fluid flows back, causing a surge in which the flow rate and pressure fluctuate violently over time. Sorokes provide actual measurement data of pressure fluctuations and shaft vibrations during impeller stall, diffuser stall, and surge, showing the impact of fluid disturbances on compressors (Sorokes, Marshall and Kuzdzal, 2014). Bianchini experimentally investigated the detailed behavior of fluid disturbances, including the transition from vaneless diffuser rotating stall to surge (Bianchini et al., 2015). It has been confirmed that the pressure fluctuation level increases rapidly and a large fluid disturbance occurs when entering surge from rotating stall. However, there are no reports on the proposal of prediction methods for fluid disturbances or their validation tests.

In axial magnetic bearings, because it is difficult to use a laminated steel core due to restrictions on structural strength, bulk materials are often used. It is known that the skin effect caused by eddy currents has a large effect on the magnetic flux passing through bulk materials, and the magnetic attraction force of axial magnetic bearings has frequency characteristics.

Many research groups have previously investigated the frequency characteristics of axial magnetic bearings. Fukata proposed a model that takes into account the influence of eddy currents in the analysis of the dynamic characteristics of axial bearings, and analyzed the time delay of the magnetic attraction force (Fukata, Kouya and Shimomachi, 1991). Kucera developed a generalized model of the magnetic flux equation by dividing an axial electromagnet into basic geometric shapes and approximating the magnetic flux distribution of each part using the theoretical solution of the magnetic field in a semi-infinite plate (Kucera and Ahrens, 1996). However, these modeling methods are nonlinear equations using Laplace operators, and are difficult to expand into the time domain. Meeker improved the model of Kucera to consider the frequency dependence of complex permeability using the Cauer expansion (continued fraction expression) of impedance (Meeker, Maslen and Noh, 1996). The modeling of Meeker can also be applied to the time domain, but it is for a simple shape, and the prediction accuracy may decrease for complex real shapes. On the other hand, Sawada applied a rational approximation model to the magnetic flux of an axial magnetic bearing and applied it to control design (Sawada et al., 2012). In the rational approximation model, the frequency response of the magnetic attraction force is estimated by FE analysis simulating the detailed shape, and the frequency response is modeled by curve fitting using the least squares method. However, unlike Kucera and Meeker et al., the modeling is not based on physical phenomena, and the prospects from a physical perspective are not good.

In this paper, a prediction method of fluid disturbance that can be used at the design stage, and a practical modeling method for the frequency dependence of axial magnetic bearings were proposed. The validity of the proposed method was validated in actual measurement.

2. Nomenclature

Symbols

F: Force
K: Magnetic attraction force coefficient of magnetic bearings
I: Coil current of magnetic bearings
δ: Air gap (clearance) of magnetic bearings
R: Resistance of electromagnet
L: Inductance of electromagnet
V: Coil voltage of magnetic bearings

Subscripts

g: Air gap at magnetic bearing
f: Front side (suction side of impeller)
r: Rear side (discharge side of impeller)

3. Design and validation process of magnetic bearings

In order to ensure the product reliability as a rotating machine supported by magnetic bearings, it is important to perform appropriate disturbance estimation at the early design stage and design an electromagnet with sufficient load capacity. Figure 1 shows the design and validation flow for the magnetic bearing supported centrifugal compressor. In the magnetic bearing design flow, disturbance estimation is performed first. The rotor is affected by the rotor's own weight, unbalance force, and fluid disturbance in the impeller, but fluid disturbance is particularly difficult to estimate. Therefore, a reasonable method for predicting fluid disturbances at the design stage is required. In an axial magnetic bearing, the effect of frequency characteristics due to eddy currents is significant, so the frequency characteristics of the axial magnetic bearing must be taken into consideration when designing the rotor dynamics and control system. For the electromagnet, in order to properly evaluate the impact of a decrease in magnetic attraction force due to leakage flux and magnetic saturation, the shape was finalized by estimating the magnetic attraction force through FE analysis that simulated the detailed shape.

The validation test flow proceeded in the order of electromagnet component test, system level test of compressor, and actual operation test of the product. In the component test of the electromagnet, it was confirmed that the required magnetic attraction force could be realized with the designed electromagnet. Furthermore, after obtaining the frequency characteristics of the axial magnetic bearing, the validity of the modeling method for the frequency characteristics was also validated. In the system level test of the compressor, the validity of the rotor system and control system design was confirmed in a non-rotating condition. The behavior of the compressor rotor during magnetic levitation was actually measured, and it was confirmed that the frequency characteristics of the entire system were consistent with the prediction analysis. Finally, in the product level test, shop test was carried out under various conditions within the actual operating range in order to confirm the fluid disturbance level, which is particularly difficult to predict in advance. The behavior of the compressor during rotating stall and surge was measured, and the validity of the fluid disturbance prediction method was confirmed. As a result, the reliability of the magnetic bearing size and power amplifier capacity was also confirmed.

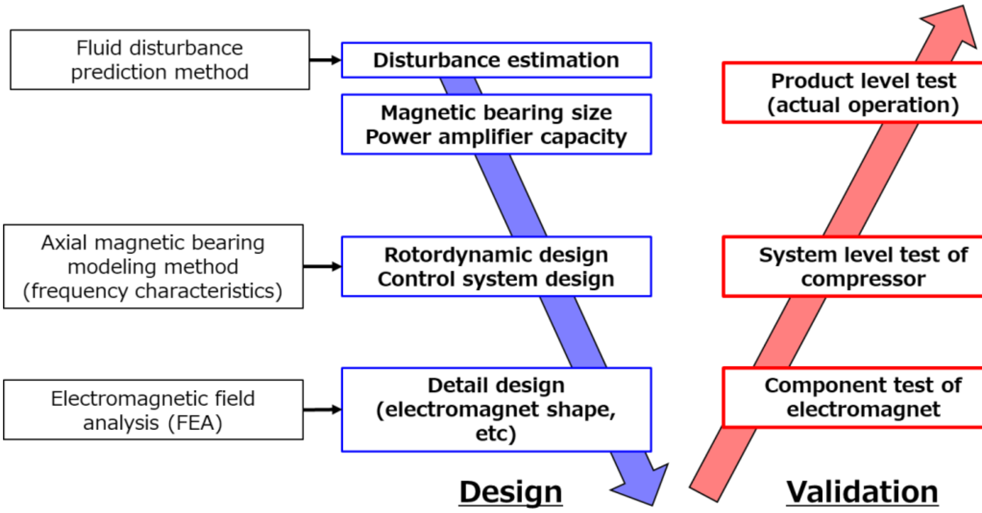


Fig.1 Design and validation test flow of compressor supported by magnetic bearings

4. Prediction method of fluid disturbance

In this paper, prediction method of fluid disturbances in the axial direction has been proposed. When stall or surge occurs, the impeller cannot properly increase pressure, so the pressure difference from the suction part to the discharge part of the impeller decreases significantly. As a result, the backward axial force generated in the impeller main flow section is reduced, and the forward axial force acting on the rotor increases transiently. Prediction method of fluid disturbance was investigated by assuming such a transient change in pressure distribution during a surge. Therefore, as shown in Figure 2, in order to estimate the maximum fluid disturbance force, the most conservative pressure distribution during a surge was considered. In other words, assuming that the impeller main flow was constant at the suction pressure, the maximum forward axial load F_{ax_max} was defined by considering the minimum backward axial force F_{f_min} in the impeller main flow as shown in following equations.

During steady operation: $F_{ax} = F_f - F_r$ (1)

During surge: $F_{ax\_max} = F_{f\_min} - F_r$ (2)

This assumption may be too conservative for the actual pressure distribution, but it can be a very effective prediction method in the early design stage when there is no validation data from actual measurements. The safety factor of this method needs to be confirmed by a validation test in the actual environment.

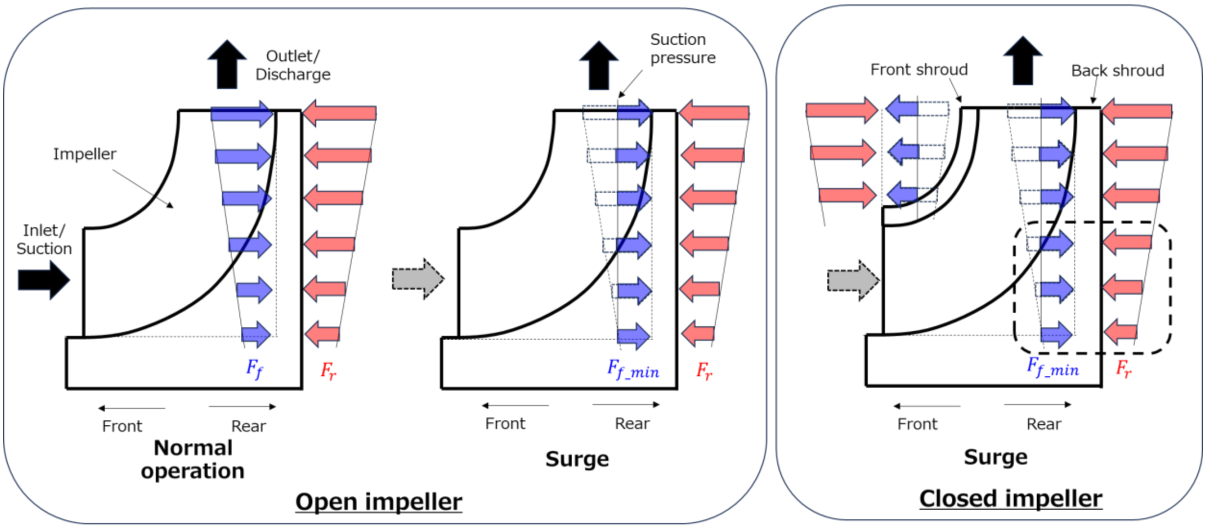


Fig.2 Assumption of pressure distribution of centrifugal compressor impeller in axial direction during surge

5. Component level test of electromagnet

In validating the design of a rotating machine supported by a magnetic bearing, component level tests were first performed on the electromagnet. The test apparatus is shown in Figure 3. The test equipment is composed of an axial disk and an axial bearing (only one side), and the magnetic attraction force can be measured by supporting the axial bearing via a load cell. 3 bearings were used as test specimens, as shown in Table 1. Maximum surface pressure was same in 3 bearings.

Table 1 Specifications of test bearings

Item	Unit	Brg Type 1	Brg Type 2	Brg Type 3
Bearing outer diameter	mm	212	212	302
Max surface pressure	MPa	0.46	0.46	0.46
Nominal Gap	mm	0.5	0.5	0.6
Max coil current (I_{max})	A	8	10	12
Coil turns	Turn	96	120	72

DC current was excited in the range up to the maximum coil current (I_{max}), and the magnetic attraction force was measured with a load cell. All measured forces were normalized using the maximum design value of the Brg Type 3. Figure 4 shows the measurement results of the magnetic attraction force. By using the simple equation shown below, it was found that sufficient accuracy was achieved in the low current region used in normal vibration control.

$$F = K \times I^2 \div \delta^2 \tag{3}$$

Compared to the simple equation, the FE analysis shows good agreement with actual measurement results even in high current ranges, and allows for highly accurate predictions even in conditions where large forces such as surges are required. With FE analysis, the effects of magnetic saturation can be estimated by analyzing the magnetic field of the actual shape, and prediction accuracy is high. The above results cinfirm the validity of the magnetic attraction force prediction method.

The same test apparatus was used to apply the desired DC + AC current with a power amplifier and measure the frequency characteristics of the magnetic attraction force. Figure 5 shows the results of measuring the frequency characteristics of the magnetic attractive force. As the excitation frequency increases, the fluctuating force level (gain) decreases and the phase lag also increases, and the measured data confirmed that the effect of frequency dependence on the magnetic attractive force is significant. Fig. 5 also shows the results of estimating the fluctuating force using FE analysis. The FE analysis showed a similar trend of the actual measurements, and it was confirmed that the predicted results of the frequency characteristics were close to the actual measurements. However, since the phase delay is larger in the actual measurements than in the FE analysis, it is considered necessary to improve the accuracy of the FE analysis method.

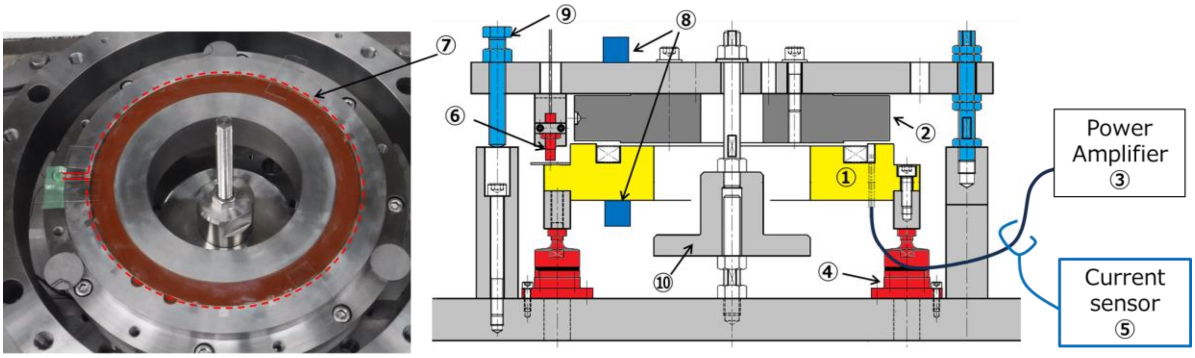


Fig.3 Test apparatus for component level test of magnetic bearing
1: Axial magnetic bearing, 2: Axial disk, 3: Power amplifier, 4: Load cell, 5: Current sensor,
6 Gap sensor, 7: Search coil, 8: Accelerometer, 9: Gap control bolt, 10: Centering jig

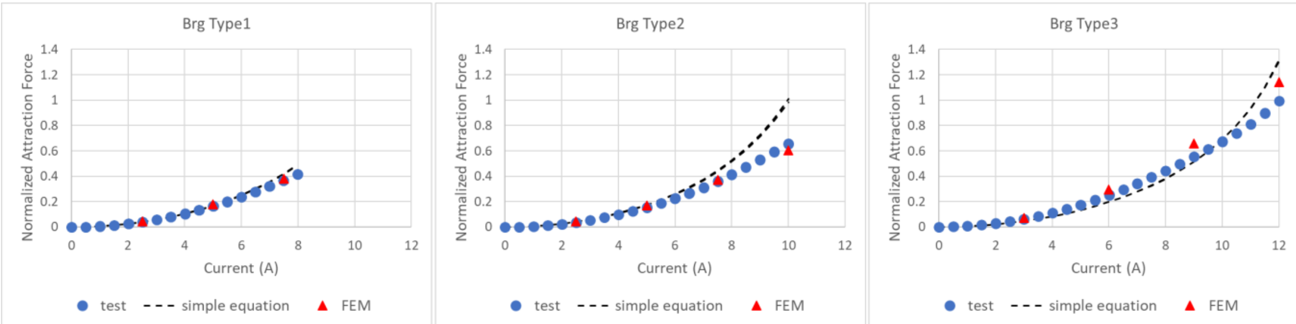


Fig.4 Test result of magnetic attraction force vs coil current
(Force is normalized by max design value of Brg type 3)

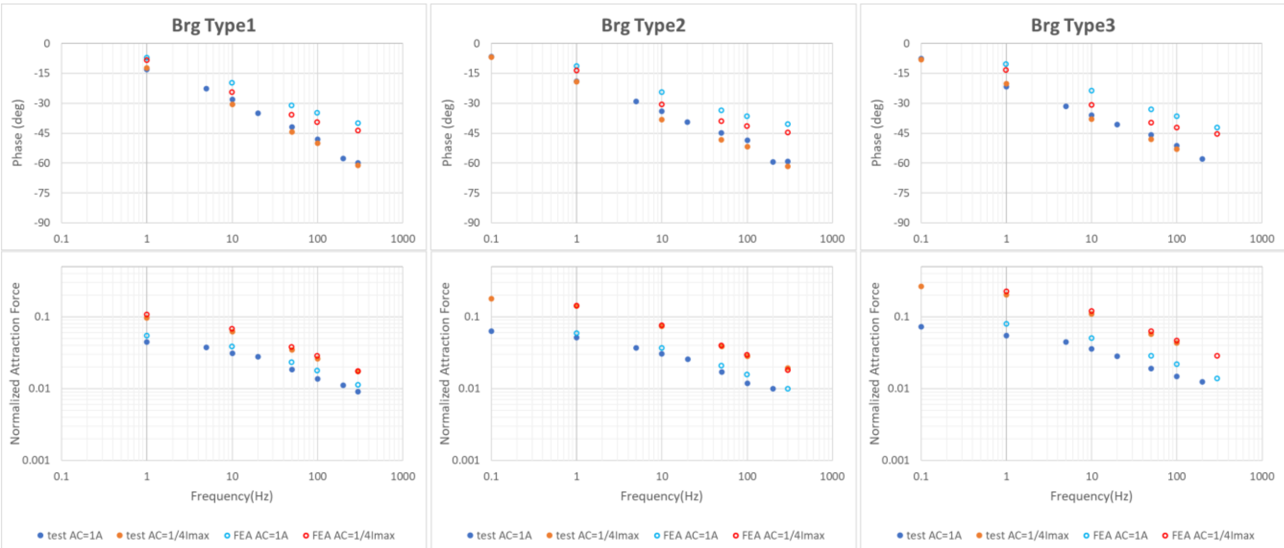


Fig.5 Test result of frequency characteristics of magnetic attraction force relative to the coil current
(AC indicates AC component of coil current at various current amplitude conditions. DC current is 1/2 I_{max}.)

Next, the frequency characteristics of impedance (transfer characteristics of coil current with respect to coil voltage) are shown in Figure 6. From the measurement results of the impedance characteristics, it can be confirmed that the gain reduction rate is 10 dB/dec and the phase lag in the high frequency range is around 45 deg. In the case of a laminated core, the frequency characteristics of an electromagnet are generally modeled as a first-order lag system based on coil resistance and inductance, in which case the gain reduction rate is 20 dB/dec and the phase lag in the high frequency range is 90 deg. Therefore, it was found that it is difficult to apply a first-order lag system to an axial magnetic bearing, which is greatly affected by eddy currents.

By using a search coil to measure the fluctuation of the gap magnetic flux density, it was confirmed that there was almost no gain reduction or phase delay in the transmission from the magnetic flux density to the magnetic attraction force. Therefore it was found that the frequency characteristics of the magnetic attraction force occur in the transmission from the coil current to the gap magnetic flux density.

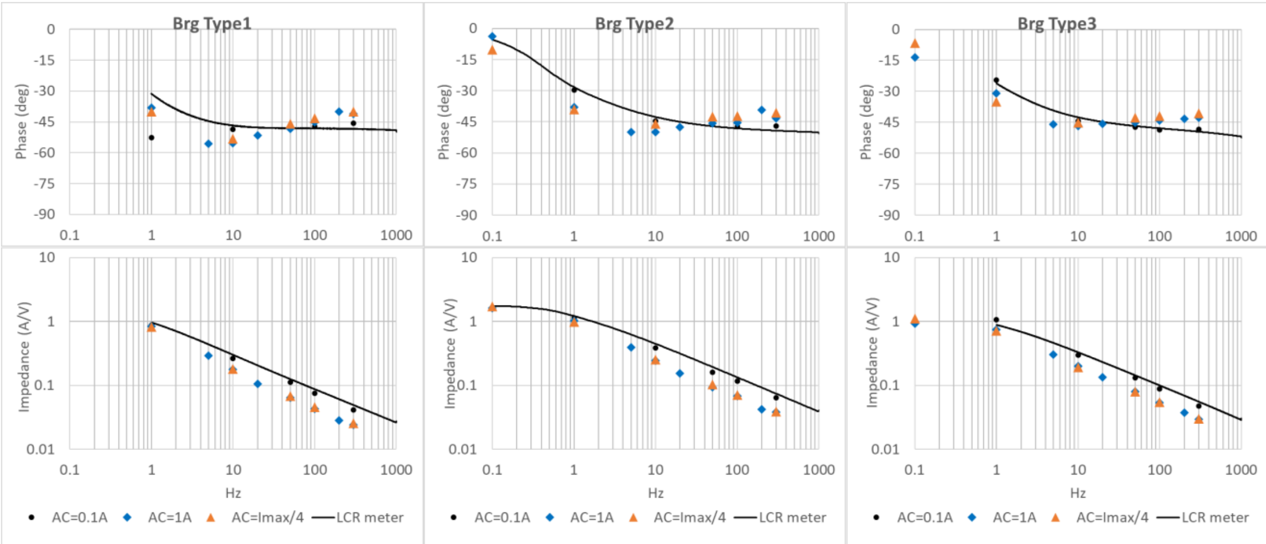


Fig.6 Test result of frequency characteristics of impedance
(Plot data indicates AC component of coil current of dynamic magnetic attraction force test (Fig.5) at various current amplitude conditions. Black line indicates LCR meter results.)

6. Practical modeling method for axial magnetic bearings

In this paper, based on the modeling method of Meeker, a new modeling method was proposed to curve-fit the frequency characteristics of an electromagnet of actual shape. Figure 7 shows an overview of the axial magnetic bearing modeling method proposed in this study. The overall impedance of the axial magnetic bearing is expressed as a series combination of the gap part, which is not affected by eddy currents, and the core part, which is greatly affected by eddy currents. In this study, the impedance of the core part is expressed with a Cauer expansion using resistance and inductance as in Meeker's model, and the frequency characteristics of the impedance in the detailed shape are curve fitted using the values of each resistance and inductance as fitting parameters. The overall impedance is modeled as follows using a series combination of the coil resistance R_0, the gap inductance L_1, and the eddy current term Z_{ed}.

$$\frac{I_0}{V} = \frac{1}{R_0 + \frac{1}{\frac{1}{L_1 s} + \frac{1}{Z_{ed}}}} \tag{4}$$

$$Z_{ed} = R_2 + \frac{1}{\frac{1}{L_3 s} + \frac{1}{R_4 + \frac{1}{\frac{1}{L_5 s} + \frac{1}{R_6 + \cdots}}}} \tag{5}$$

In curve fitting of eddy current terms, the key point is to set constraints for curve fitting, such that the higher the order, the larger the resistance ($R_2 < R_4 < R_6 < \cdots$) and the smaller the inductance ($L_3 > L_5 > L_7 > \cdots$), referring to the modeling by Meeker. By adding such constraints, it is possible to obtain curve fitting estimates based on the frequency characteristics of the theoretical model, rather than simply least-squares curve fitting. The method proposed in this study is formally equivalent to the rational approximation model (polynomial of Laplace operator s) proposed by Sawada et al. However, the method proposed in this study is significantly different from the method proposed by Sawada et al. in that it is able to fit the curves based on physical phenomena. Figure 8 shows the fitting results of the impedance characteristics using the proposed method. Fig. 8 also shows the magnetic attraction force estimated using the fitted eddy current terms

Z_{ed}. Model order is 5th order. From this result, it was confirmed that the frequency characteristics of the magnetic attraction force can be estimated with high accuracy by curve fitting the impedance characteristics using the proposed method.

In axial magnetic bearings, magnetic flux feedback using Hall elements is sometimes used to compensate for the frequency dependence caused by eddy currents. However, by using the proposed method, it is possible to efficiently compensate for frequency dependence without using additional sensors such as Hall elements.

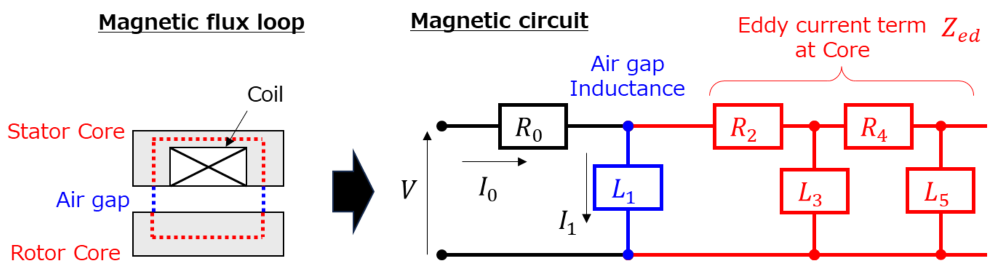


Fig.7 Overview of the axial magnetic bearing modeling method proposed in this study

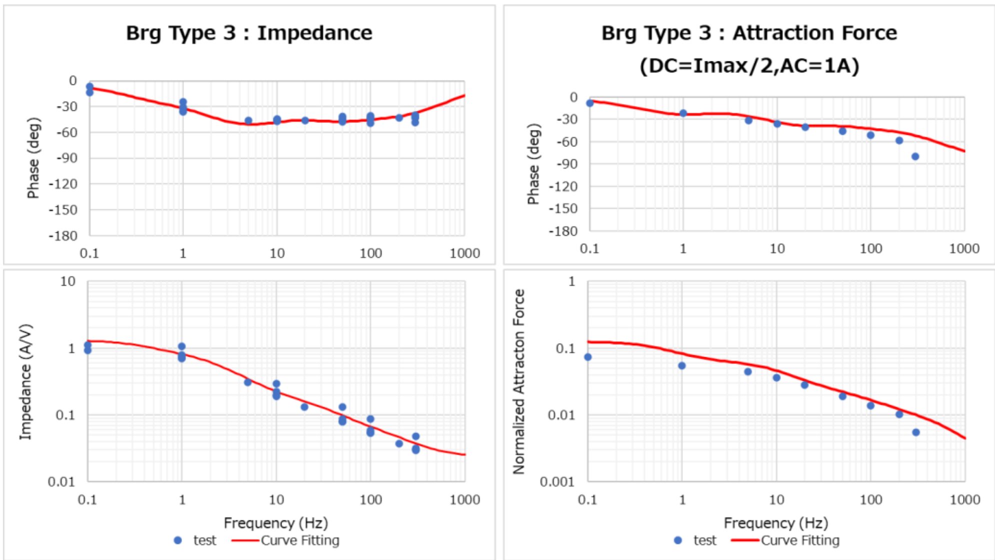


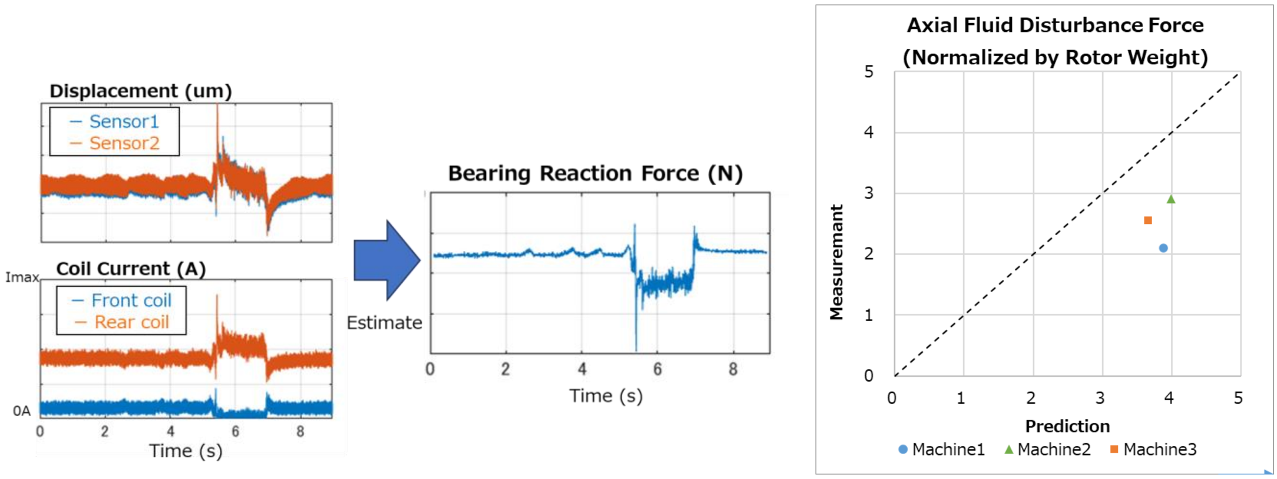
Fig.8 Fitting results of impedance characteristics using the proposed method and estimated frequency characteristics of magnetic attraction force (5th order)

7. Product level test by actual operation

Finally, in order to confirm the actual fluid disturbance level, which is particularly difficult to predict in advance, a shop test under actual load operation was conducted using a centrifugal compressor whose preliminary design was confirmed in the system level test. Product level tests were conducted on three 100kW class centrifugal compressor with different output levels. The product reliability was validated in the actual product environment by operating under various conditions within the product's operating range and checking the behavior of the compressor when rotating stall or surge occurs. Under all operating conditions including surge occurrence, it was confirmed that unstable vibration or saturation of coil current did not occur, and the centrifugal compressor could be operated without problems.

Next, the axial fluid disturbance force was estimated based on the measured data of rotor displacement and coil current at the time when the surge occurred as shown in Figure 9 (a). The axial fluid disturbance force was calculated by adding the bearing reaction force of the front and rear axial bearings and the rotor inertia force. The maximum force fluctuation range during the surge was estimated from the obtained time history waveform of the fluid disturbance force, and the validity of the fluid disturbance prediction method was confirmed by comparing it with the predicted value.

Figure 9 (b) shows the comparison result between the measured and predicted fluid disturbance forces. Measured force is normalized by rotor weight. The predicted results of the fluid disturbance force were greater than the actual measured results, and it was confirmed that an appropriate safety factor was ensured. From the above results, the validity of the fluid disturbance prediction method proposed in this study was also confirmed in the product level validation test.



(a) Estimation of fluid disturbance force during surge (b) Comparison between test and prediction

Fig.9 Comparison of measurements and prediction analysis results for fluid disturbance forces during surge

8. Conclusion

In this paper, with the purpose of ensuring product reliability for centrifugal compressors supported by magnetic bearings, a fluid disturbance prediction method and a practical modeling method for the frequency characteristics of axial magnetic bearings were developed.

Regarding the prediction method of fluid disturbance, a simple prediction method was considered by assuming the change in pressure difference in the impeller when surge or rotating stall occurs.

For the frequency characteristics of axial bearings, a practical modeling method that can be applied to complex real shapes was proposed by improving the method used in previous research. The frequency characteristics of the axial magnetic bearing were measured in a component level test of the electromagnet, and the effectiveness of the proposed modeling method was confirmed.

As a validation test at the product level, actual operation was performed in a shop test and it was confirmed that normal operation was possible within the allowable vibration level even when a surge occurred, and the prediction method of fluid disturbance was validated.

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