Observation and Prediction of Instability due to RD Fluid Force in Rotating Machinery by Operational Modal Analysis

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ABSTRACT

In recent years, as rotating machinery has become smaller and more efficient, various types of shaft vibration problems have arisen. Failure of rotating machinery may lead to major accidents and infrastructure shutdowns. Therefore, to prevent failures of rotating machinery, there is a growing need for the vibration analysis technology at the design stage and condition monitoring during operation stage. One of causes of the shaft vibration problems in rotating machinery is the rotordynamic (RD) fluid force acting on fluid elements such as journal bearings, seals, turbine blades, and so on. RD fluid force has a significant effect on the stability of rotating machinery and can destabilize the system. In recent years, operational modal analysis (OMA) methods, which identify modal parameters based on the measured data of a machine's operational condition, have been investigated in the condition monitoring. In this paper, the estimation of the modal parameters of rotating machinery using OMA from only the time history response of displacement data and, in particular, the prediction of the destabilization of rotating machinery caused by RD fluid force are investigated. As a result, the modal parameters are well estimated and, in particular, the destabilization of one mode due to RD fluid force is predicted and explained. The results are in good agreement with the results of the eigenvalue analysis of the original system, and the method is validated. Furthermore, the proposed method is applied to experimental data of the system destabilized by fluid force. The change in stability with rotational speed is observed, and the characteristics of the mode toward destabilization are confirmed. The results show the validity of OMA's predictions of destabilization in the experiments.

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1. INTRODUCTION

Rotating machinery such as pumps and gas turbines is used in various fields of industry. In recent years, various types of shaft vibrations have generated and become problems as rotating machinery become smaller and more efficient (Childs, 1978), (Childs & Moyer 1985). If these machines break down due to abnormal vibration, it can lead to infrastructure shutdowns and additional costs. Therefore, to prevent failures of rotating machinery, there is a growing need for the vibration analysis technology at both the design stage and condition monitoring during operation stage (Gecgel, Dias, Osire, Alves, Macado, Daniel, Castro & Cavalca, 2020).

One of the causes of shaft vibration in rotating machinery is the RD fluid force generated in fluid elements such as journal bearings, seals, turbine blades, and so on (Childs, 1983), (Nelson, 1984), (Nelson, 1985). The RD fluid force is caused by the interaction between the shaft vibration and the working fluid. It is also difficult to predict due to its nonlinearity. It is also known that RD fluid force has a significant effect on the stability of rotating machinery and can destabilize the system. (Yang, Iwatsubo & Kawai, 1984). Various methods have been studied to consider the effects of RD fluid force in shaft vibration analysis. The most common method is the expression using linear RD coefficients (Childs, 1993). Lund (1974) performed an eigenvalue analysis considering the RD coefficients to calculate the critical speed in a rotor system supported by journal bearings.

On the other hand, in the condition monitoring of a machine, it is desirable to determine the characteristics of the system from the condition of the actual machine, and to obtain its mathematical model and modal parameters. Traditionally, experimental modal analysis (EMA) has been widely used to identify modal parameters. For example, Chouksey, Dutt and Modak (2012) performed EMA considering RD coefficients in journal bearings and estimated modal damping ratios to predict destabilization phenomena in rotating machinery. However, to identify the modal parameters by EMA, an excitation test using a vibratory apparatus is required while the machine is stopped. This requires extra time and costs.

In recent years, operational modal analysis (OMA) methods, which identify modal parameters based on the measured data of a machine's operational condition, have been investigated. OMA does not require a vibratory apparatus, so it can perform condition monitoring without stopping the machine, which has the advantage of reducing costs. Santos and Svendsen (2017) applied OMA to experimental data of a rotor-bearing system with fluid film bearings to identify the modal parameters. And the validity was then confirmed by comparison with EMA. Castro and Zurita (2022) proposed a method to remove harmonic components, which is a challenge for OMA for rotating machinery, and verified it through experiments.

However, to the best of the author's knowledge, there are no studies that use OMA to estimate the modal parameters of the system considering the RD coefficients and observe or predict the process of destabilization of rotating machinery. Therefore, in this paper, the estimation of the modal parameters using OMA and the observation and prediction of the destabilization of rotating machinery caused by RD fluid force are investigated. First, a theoretical model with RD coefficients is shown and numerical simulations are performed with changing the rotational speed statically. Next, a pseudo-frequency response function (FRF) is obtained using OMA from the obtained time history data of the vibration response. The pseudo-FRF is fitted to the quadratic system to estimate the modal parameters, and the degree of destabilization is predicted for each rotational speed. As a result, the modal parameters of the rotating shaft system are well estimated, and in particular, the destabilization of one of the modes due to the RD fluid force is predicted and explained. The results are in good agreement with the eigenvalue analysis of the original system, and the validity of the method is demonstrated. Furthermore, the proposed method of predicting destabilization by OMA is applied to experimental data of a system destabilized by fluid forces. Pseudo FRFs are created by acquiring data at each rotational speed of the stable state and the state just prior to destabilization. The change in stability with rotational speed is observed by fitting. As a result, the damping ratio of one of the modes decreased with rotational speed and moved toward destabilization. The results show the validity of the prediction of destabilization by OMA in the experiments.

2. THEORETICAL MODEL

2.1. Disk rotor model with gyroscopic effect and RD coefficients caused by turbo elements

The theoretical model used in this paper is shown in Figure 1.

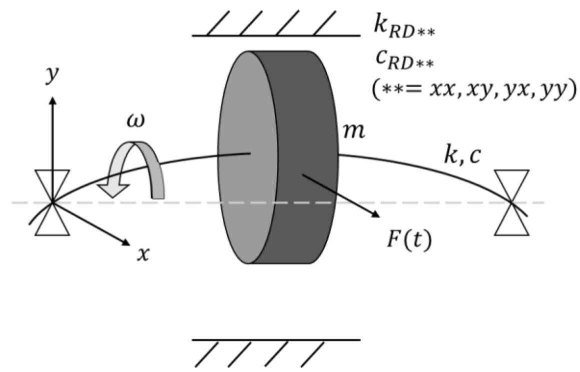


Figure 1. Theoretical model

The equations of motion for the theoretical model in Figure 1 are shown in Eq.(1).

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} c + c_{RDxx} & i_p\omega + c_{RDxy} \\ -i_p\omega + c_{RDyx} & c + c_{RDyy} \end{bmatrix}\begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} + \begin{bmatrix} k + k_{RDxx} & k_{RDxy} \\ k_{RDyx} & k + k_{RDyy} \end{bmatrix}\begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} F \\ 0 \end{bmatrix} \tag{1}$$

Where $i_p\omega\dot{y}, -i_p\omega\dot{x}$ are gyroscopic terms which account for gyroscopic moment acting in rotating machinery. The action of the fluid force from the fluid element is expressed as the RD coefficients. In this paper, as an example of the RD coefficients, an expression such as Eq.(2), which appears in vertical shaft system supported by a journal bearing, is used.

$$\begin{aligned} &k_{RDxy} = a\omega, \\ &k_{RDyx} = -a\omega, c_{RDxx} = c_{RDyy} = c_{RD}, \\ &k_{RDxx} = k_{RDyy} = c_{RDxy} = c_{RDyx} = 0 \end{aligned} \tag{2}$$

Table 1 shows the parameters used in the theoretical model.

Mass	m	10	[kg]
Damping coefficient	c	18.8	[Ns/m]
Stiffness coefficient	k	8.9×10^4	[N/m]
Gyroscopic coefficient	i_p	8.3	[kg/rad]
Rotational speed	ω	50~1100	[rpm]
Gradient of RD stiffness coefficient	a	10	[Ns/rad/m]
RD damping coefficient	C_{RD}	50	[Ns/m]

Table 1. Theoretical Model Parameters

2.2. Eigenvalue analysis

Figure 2 shows the variation of the real and imaginary parts of the eigenvalues with rotational speed ω obtained by eigenvalue analysis for the systems Eqs.(1) and (2).

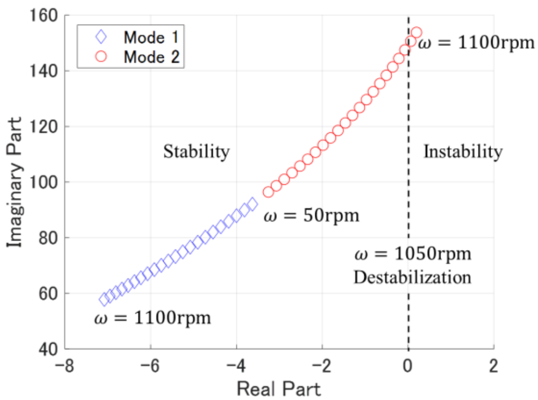


Figure 2. Variation of eigenvalues with rotational speed $\boldsymbol{\omega}$

Figure 2 shows that there are two modes in the system of Eqs. (1) and (2), and Mode 1 and Mode 2 are numbered according to the magnitude of their imaginary parts. In particular, the real part of Mode 2, indicated by the symbol ○, increases with increasing rotational speed ω, and becomes destabilized around at $\omega = 1050$ rpm.

2.3. Experimental Modal Analysis (EMA)

For the comparison to OMA, EMA is also used. EMA is a method to obtain the frequency response function (FRF) by taking the Fourier transform of the time response of the external force data $f(t)$ and displacement data $x(t)$ obtained from experiments (Chouksey, Dutt and Modak, 2012). In this paper, numerical simulations of Eqs. (1) and (2) are performed with the rotational speed varied quasi-statically. Figure 3 shows the example of FRF at $\omega = 500$ rpm. This FRF is obtained from EMA using the external force data $f(t)$ and displacement data $x(t)$. The rotational speed $\omega = 500$ rpm is chosen because the imaginary parts of two modes are significantly separated in Figure 2. Figure 3 indicates the occurrence of two resonance peaks as expected.

In EMA, the vibration characteristics can be obtained by curve-fitting this FRF. A comparison of the damping ratio obtained from EMA and the theoretical value obtained from eigenvalue analysis for the $\omega = 500$ rpm case is shown in Table 2.

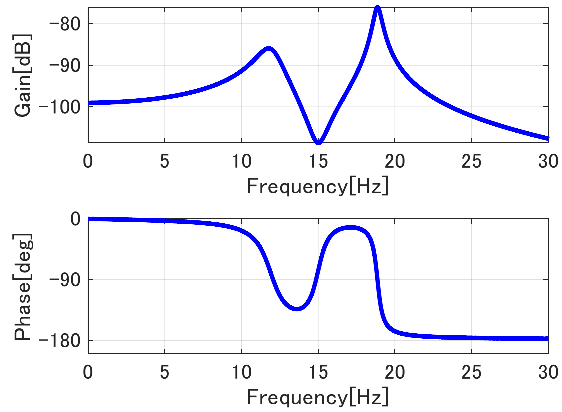


Figure 3. FRF by EMA（$\boldsymbol{\omega} = \mathbf{500rpm}$）

	Mode 1	Mode 2
Theoretical value [-]	0.070	0.014
Estimation value [-]	0.085	0.014
Error	21%	1%

Table 2. Comparison between the estimated damping ratio using EMA and theoretical solutions.

Table 2 shows that EMA can generally estimate damping ratios well from FRFs.

3. OPERATIONAL MODAL ANALYSIS (OMA)

In this paper, the OMA analysis is carried out using the method of Nagae et al. (Nagae, Watase and Tamaki, 2011). This method calculates the auto-correlation function shown in Eq. (3) from the time history data of the vibration response obtained experimentally and obtains a pseudo-FRF by taking its Fourier transform.

$$\{corr(x,x)(\tau)\} = \begin{cases} \dfrac{1}{n}\displaystyle\sum_{l=1}^{n-\tau} x(lT_s)\cdot\{x(lT_s+\tau T_s)\} & \tau \geq 0 \\ \dfrac{1}{n}\displaystyle\sum_{l=1}^{n-\tau} x(lT_s)\cdot\{x(lT_s-\tau T_s)\} & \tau < 0 \end{cases} \tag{3}$$

where n is the number of data points, T_s is the time step of the data, and τT_s is the time delay. A comparison of the pseudo-FRF obtained by OMA and the example of $\omega = 500$ rpm within the FRF obtained from the EMA (Nagae et al. 2011) is shown in Figure 4.

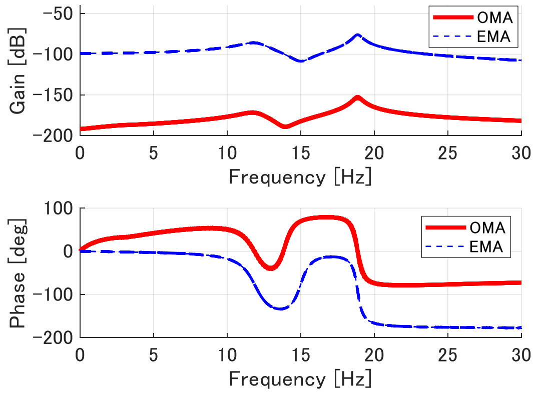


Figure 4. Comparison of FRF by EMA and OMA（ω = 500rpm）

In Figure 4, the gain and phase differ by about 100 dB and 90 degrees, between OMA and EMA, respectively. However, these differences do not affect the estimation of vibration characteristics.

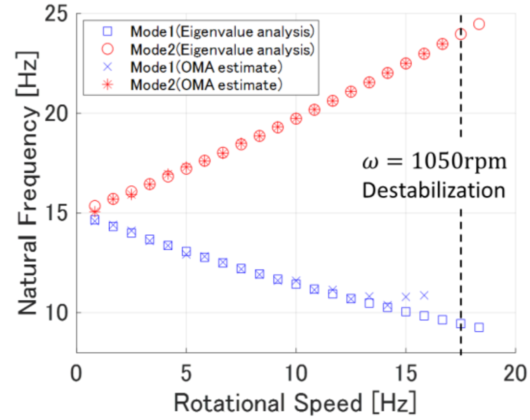
The modal parameters are obtained by fitting the pseudo FRF shown in Figure 4 (red lines) which is obtained by OMA with a quadratic system. To evaluate the estimated degree of destabilization of the system due to the influence of the gyroscopic moment and RD coefficients, the eigenvalues of the system are obtained and evaluated for each rotational speed ω.

4. ESTIMATION OF DESTABILIZATION USING OMA

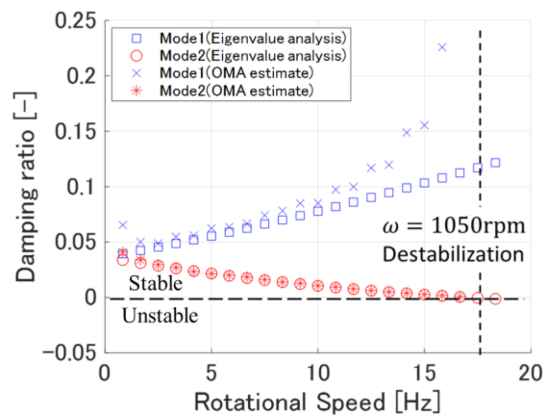
4.1. Estimation of Eigen-Characteristics Using OMA

For each rotational speed ω, the pseudo FRFs shown in Figure 4 are calculated and fitted to estimate the natural frequency and damping ratio. Figure 5 show a comparison of estimated result by OMA and theoretically obtained eigenvalue for the systems Eqs.(1) and (2), respectively. Figure 5(a) shows natural frequencies, and Figure 5(b) shows damping ratios. An enlarged view of the area near the destabilization speed of the Mode 2 of Figure 5(b) is shown in Figure 5(c).

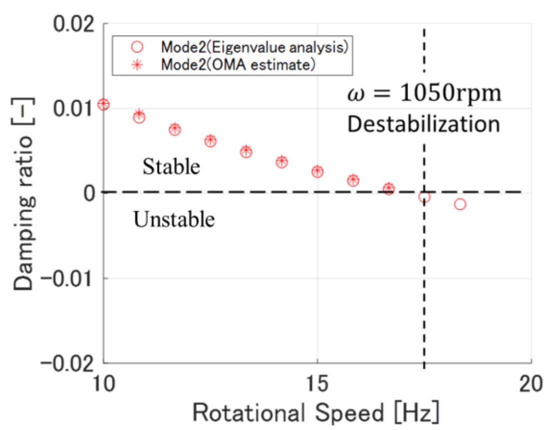
Figure 5 indicates that the modal parameters obtained from the pseudo-FRF by OMA correspond to the results of the eigenvalue analysis of the equations of motion shown in Eqs.(1) and (2). The estimation results of modal parameters of Mode 1 have errors in two regions: in the small rotational speed region and near the destabilization speed. The large errors in these two areas are discussed in the next section.



(a) Natural Frequency



(b) Damping ratio



(c) Expansion of Mode 2 damping ratio near destabilization

Figure 5. Comparison of OMA estimation of eigenvalues and eigenvalue analysis results

4.2. Cause of Error at low rotational speed range

This section discusses the large error between the predictions by OMA for Mode 1 and Mode 2 when the rotational speed is low in Figure 5. Figure 6 shows the pseudo FRF at a rotational speed of $\omega = 50$rpm. As shown in Figure 6, when the rotational speed is low, the effect of gyroscopic moments is small, and the two modes are close together. Estimating the modal parameters by OMA when the modes are in close proximity is known to be difficult (Dreher, Storti, and Machado, 2023).

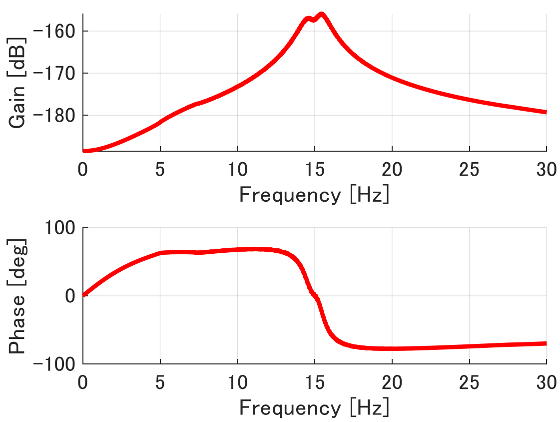


Figure 6. Pseudo-FRF by OMA ($\boldsymbol{\omega} = \mathbf{50}$ **rpm**)

4.3. Cause of Error at high rotational speed range

This section also discusses the large error in the Mode 1 predictions near destabilization, when the rotational speed increases in Figure 5. Figure 7 shows the pseudo FRF at a rotational speed of $\omega = 1000$rpm. As shown in Figure 7, the peak of Mode 1, which is stable, is extremely small relative to mode 2. This makes it difficult to estimate Mode 1 characteristics because Mode 2 characteristics are primarily evaluated during the fitting process. On the other hand, when focusing on destabilizing modes (Mode2), the estimation is accurate near destabilization.

4.4. Evaluation of estimation of system stability

Table 3 shows the comparison of the estimated and theoretically obtained damping ratio of Mode 2 near the destabilization. These values are corresponding to the data shown in Figure 5 around the destabilization speed.

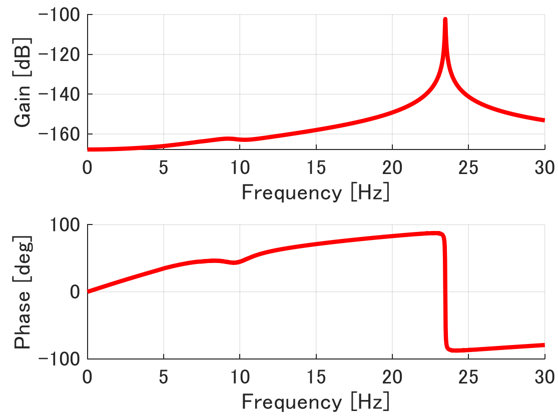


Figure 7. Pseudo-FRF by OMA ($\boldsymbol{\omega} = \mathbf{1000}$ **rpm**)

Rotational speed [rpm]	Estimated value (OMA) [-]	Theoretical value (Eigenvalue analysis) [-]
900	0.0027	0.0025
950	0.0017	0.0015
1000	6.84×10^{-4}	5.04×10^{-4}
1050	NaN	-4.1×10^{-4}

Table 3. Comparison of damping ratios of OMA and eigenvalue analysis of Mode 2

Table 3 indicates OMA estimation results of the damping ratio of mode 2 agree with the results of eigenvalue analysis of the original system. As a result, it indicates that OMA can predicts the destabilization of rotating machinery accurately by obtaining a pseudo-FRF and estimating the damping ratio of corresponding mode.

5. EXPERIMENTAL VERIFICATION

5.1. Test-rig and experimental conditions

Figure 8 shows a photograph of the test rig for the vertical rotor system used in this paper.

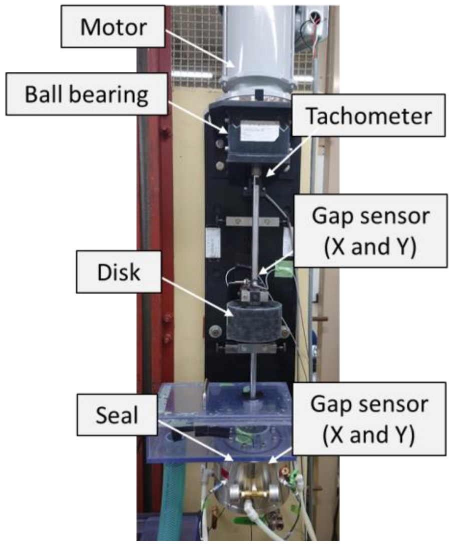


Figure 8. Photographs of test rig

This system is supported at the upper end by ball bearings and at the lower end by a seal (Miyake, Inoue, Watanabe, 2019), (Kunori, Inoue & Miyake, 2021), (Kimura, Inoue, Taura, Heya, 2023). The rotor is directly driven by a motor and rotates at an optional rotational speed ω. The working fluid is water. The gap sensor (PU-05) is used to measure the vibration data of the x and y displacements of the disk and seal. Table 4 shows the experimental conditions.

Seal clearance	75	[µm]
Onset speed of instability	2100	[rpm]
Rotational speed	1200, 1500,2040	[rpm]
Sampling rate	1	[kHz]
Measurement time	5	[min]

Table 4. Experimental conditions

The onset speed of instability during this experiment was approximately 2100 rpm (Miyake et al., 2019), (Kunori et al., 2021), (Kimura et al., 2023). Tests were conducted at three different rotational speeds: 1200rpm and 1500rpm, the stable state, and 2040rpm, just before destabilization. Measurements were taken for 5 minutes at each rotational speed. A pseudo-FRF was created using the time history response of the seal's x displacement measured at each rotational speed, and the vibration characteristics were estimated by fitting.

5.2. OMA procedures on actual machine data

Figure **9** shows the analysis procedure for the OMA method used in this paper.

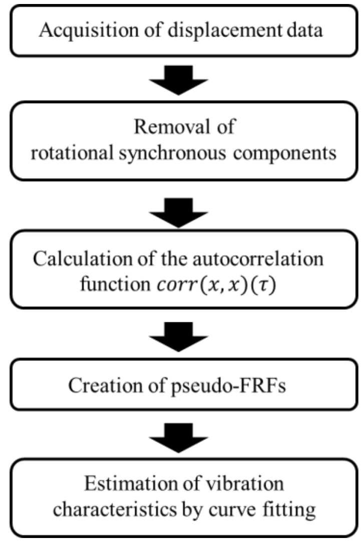
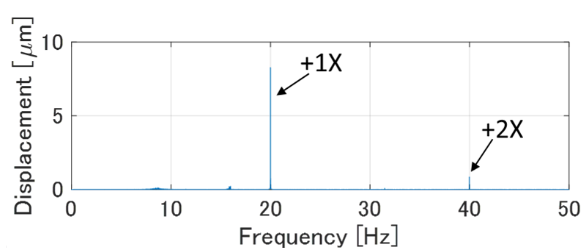


Figure 9. Analysis flow

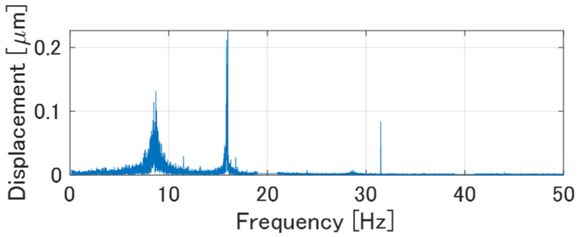
In OMA analysis of actual rotating machinery, there are always rotational synchronous components due to unbalance. The rotational synchronous component usually accounts for a large proportion of the total vibration signal, so pseudo-FRF cannot be created without modification. Therefore, removing the rotational synchronous component from the obtained vibration data is necessary. Then, the autocorrelation function is calculated using the data after removing the rotational synchronous components to obtain a pseudo-FRF.

5.3. Pre-processing in the creation of pseudo-FRFs

As discussed in Section 5.2 the rotational synchronization component must be removed in the real machine data. The procedure is described below, using data for rotor rotational speed of 1200rpm (20Hz). Figure **10** shows the FFT before and after removing the rotational synchronous component (20 Hz), respectively.

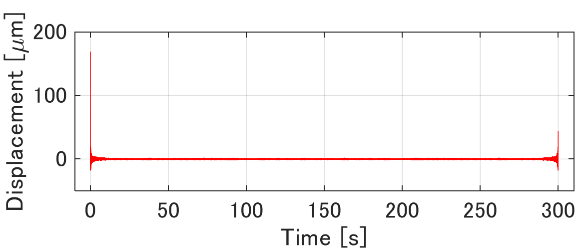


(a) Before removal of rotational synchronous components

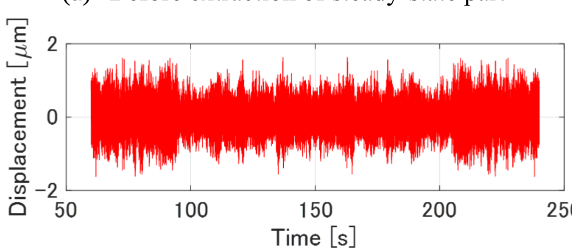


(b) After removal of rotational synchronous components
Figure 10. Removal of rotational synchronous components in FFT

In Figure **10** (a), the rotational synchronous component (20 Hz) and its double component (40 Hz) are dominant. In this paper, signals in the range of ±1 Hz are excluded from this rotation synchronous component and its integer multiple components. As a result, these components, which were dominant in Figure **10** (a), are removed in Figure **10** (b). Figure **11** (a) shows the result of computing the inverse FFT of the FFT after removing the rotational synchronous component and its integer multiplier components to create the pseudo-FRF. As can be seen from this figure, the effect of removing the rotational synchronous component and its integer multiplier components occurs at both ends. Figure **11** (b) shows the time history response for the intermediate steady-state part (40s~260s).



(a) Before extraction of steady-state part

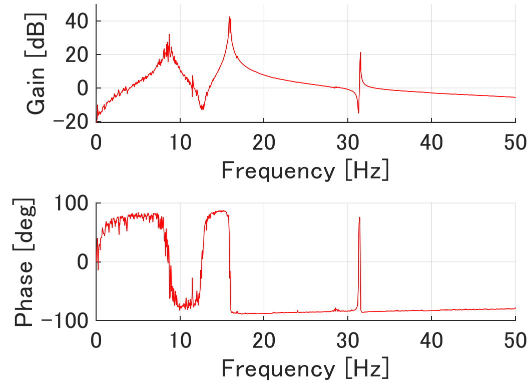


(b) After extraction of steady-state part
Figure 11. Removal of inverse FFT effects

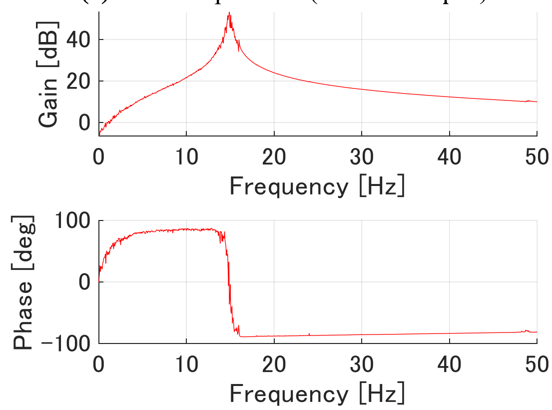
In this paper, a pseudo-FRF is created using the time history response of Figure **11** (b), from which only the steady-state part is extracted.

5.4. Prediction of destabilization

Figure **12** shows the results of creating pseudo-FRFs during stable operation and just before destabilization, respectively.



(a) Stable operation (ω = 1200rpm)



(b) Just before destabilization (ω = 2040rpm)
Figure 12. Pseudo-FRFs from experimental data

Table **5** shows the results of estimating vibration characteristics by curve fitting the natural frequency peak (around 15 Hz) of these pseudo-FRFs.

Rotational speed [rpm]	Natural Frequency [Hz]	Damping Ratio [-]
1200	15.9	0.0056
1500	15.9	0.0025
2040	16.0	0.0014

Table 5. Vibration characteristics estimated by fitting pseudo-FRFs

Table **5** shows that the damping ratio decreases just before destabilization (2040rpm) compared to the stable operation (1200rpm). This confirms that the system is moving toward destabilization. It gave an estimation of destabilization speed as 2250 rpm. As a result, this estimated value well agreed to the actual destabilization speed, 2100 rpm, shown in Table 4.

6. CONCLUSION

In this paper, a simple rotor system with the effect of gyroscopic moments and RD coefficients of fluid elements was considered. And the destabilization of the rotor system is predicted by using displacement data only and obtained the following conclusions.

The OMA analysis was performed using time history data of the vibration response of a rotating machine to identify the modal parameters. The variation of damping ratio with rotational speed among the modal parameters was observed, and the destabilization of rotating machinery was successfully predicted. The validity of the prediction was confirmed by comparing with the results of eigenvalue analysis of the original system.

Furthermore, the proposed method of predicting destabilization by OMA was applied to experimental data of a vertical rotor system. A pseudo-FRF was created from the acquired time history data, and the modal parameters were identified. Then, the destabilization speed is estimated, and it well agreed to the actual destabilization speed.

The method proposed in this paper only requires removing the rotationally synchronous component and calculating the autocorrelation function. Therefore, the data can be analyzed on-the-fly. However, in the case of a noisy system, noise removal by averaging is necessary. This may lead to a longer data acquisition time, which poses a problem. In addition, it is also desirable to measure the rotor signal directly with a sensor.

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BIOGRAPHIES

Daiki Goto was born in Nagoya, Japan, in 1998. He received his Bachelor of Engineering from Nagoya University, Japan, in 2022. He is currently a Master student in the Graduate Department of Mechanical Systems Engineering, Nagoya University (Japan). He is a member of Japan Society of Mechanical Engineers Society of Japan. His major research interests are rotor dynamics, fault diagnosis.

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