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.

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. 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. 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.

2. Theoretical Model

2.1. 1disk rotor model with gyroscopic effect and RD coefficients caused by turbo elements

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

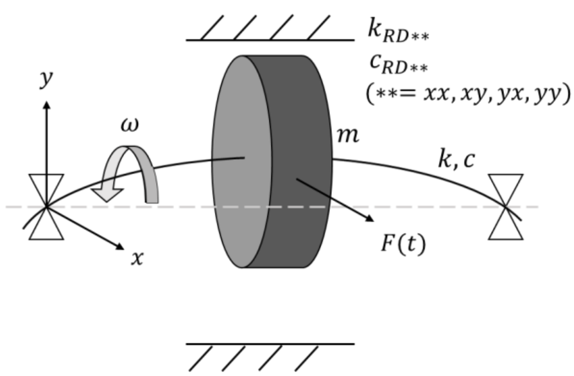


Figure 1. Theoretical model

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.

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

Table 1 shows the parameters used in the theoretical model.

Table 1. Theoretical Model Parameters

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]

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).

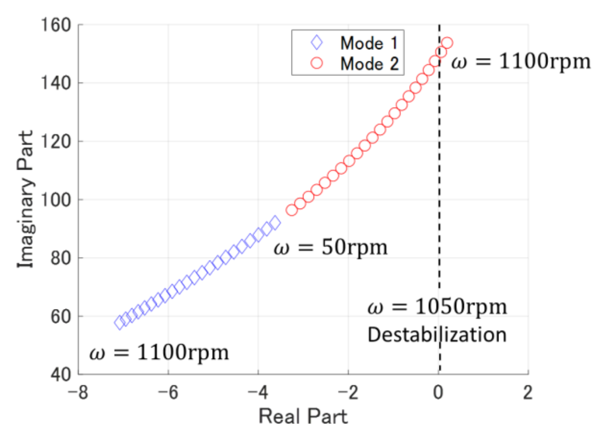


Figure 2. Variation of eigenvalues with rotational speed $\boldsymbol{\omega}$

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 ○, decreases 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). Figure 3 shows the FRF obtained from EMA using the external force data $f(t)$ and displacement data $x(t)$ obtained from the numerical simulations in Eqs. (1) and (2). Corresponding to Figure 2, two resonance peaks can be observed.

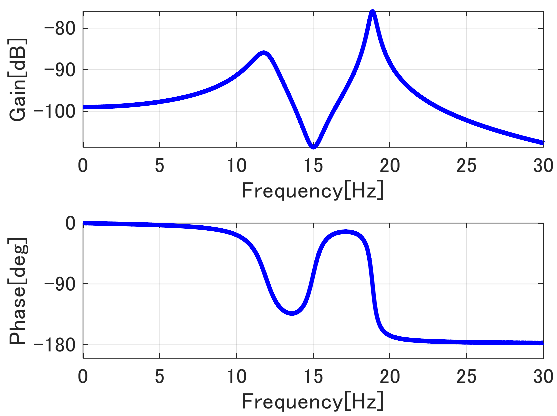


Figure 3. FRF by EMA（$\boldsymbol{\omega = 500}$**rpm**）

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 cross-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 FRF obtained from the EMA (Nagae et al. 2011) is shown in Figure 4.

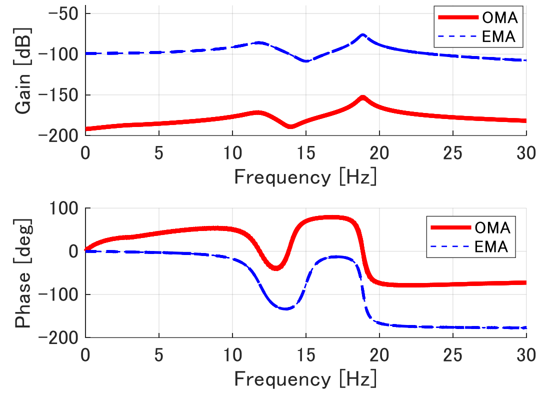


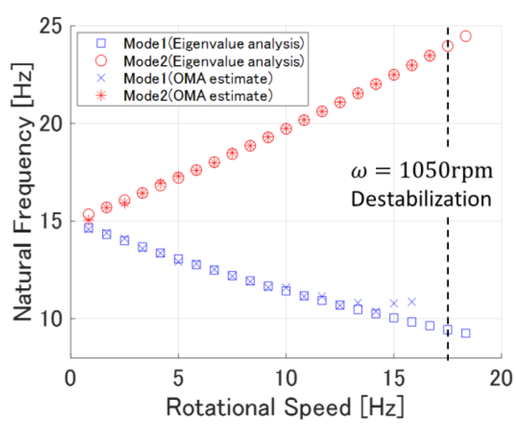
Figure 4. Comparison of FRF by EMA and OMA（$\omega =$ 500rpm）

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.

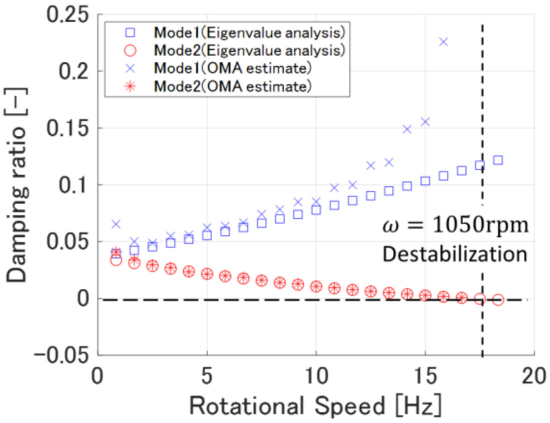
4. Estimation of Destabilization Using OMA

4.1. Estimation of Eigen-Characteristics Using OMA

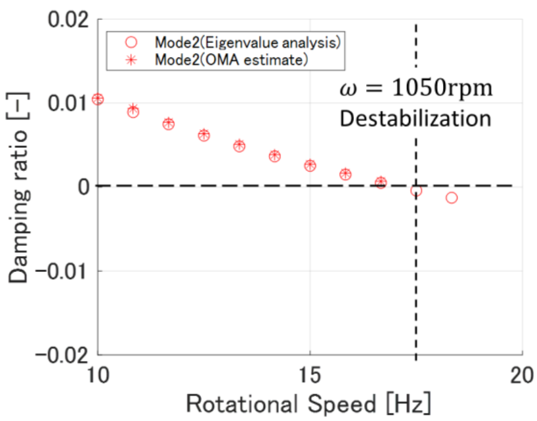
For each rotational speed ω, the pseudo FRFs shown in Figure 4 are calculated and fitted to estimate the natural frequency and damping ratio. Figure 5 show a comparison of estimated result by OMA and theoretically obtained eigenvalue for the systems Eqs.(1) and (2), respectively. Both the natural frequencies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



(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

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. However, this error does not affect the estimation of the destabilization speed of Mode 2, so the error is not discussed in this paper.

4.2. Evaluation of estimation of system stability

Table 2 shows the comparison of the damping ratio of Mode 2 near the destabilization, estimated by OMA, shown in Figure 5 and the theoretical values obtained from the eigenvalue analysis.

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

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 2 indicates that estimation results of the damping ratio of mode 2 from OMA are almost accurate up to ω = 1000 rpm, which is just before the destabilization speed. Therefore, the degree of destabilization can be predicted by estimating the modal parameters using OMA and observing their variation with rotational speed.

5. Discussion

In OMA of actual rotating machinery, there is always a rotational synchronous component due to unbalance. The rotational synchronous signal usually accounts for a large proportion of the total vibration signal, so OMA method cannot be applied without modification. Therefore, it is necessary to perform signal processing of experimental data from actual rotating machinery so that the OMA method used in this paper can be applied. First, signals containing rotational synchronous components and harmonic components caused by unbalance were removed from spectrum in advance by signal processing. OMA was performed using the time history data of the vibration response regenerated from the spectrum with the rotational synchronous components removed. The results are shown in Figure 5. Moving average using a Gaussian filter was performed to remove noise. The gain and phase changes are clearly captured, and each mode characteristics can be adequately estimated. Thus, it is confirmed that the OMA method in this paper is effective for vibration signals including rotational synchronous components caused by unbalance in actual rotating machinery.

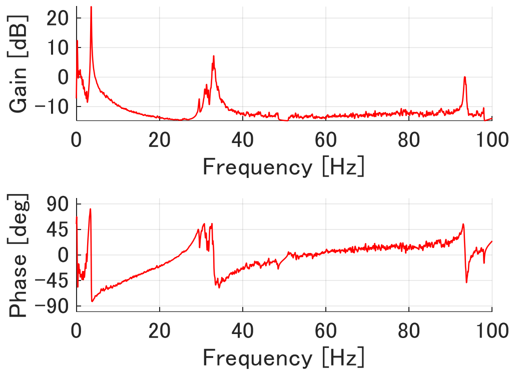


Figure 6. Pseudo FRF based on OMA for vibration signals obtained from the experimental apparatus

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.

References

Castro, G. & Zurita, G., (2022). Applications of Operational Modal Analysis in Gearbox and Induction Motor, Based on Random Decrement Technique and Enhanced Ibrahim Time Method. *Journal of applied sciences, 2022,12,5284*

Childs, D. W.(1978). The Space Shuttle Main Engine High-Pressure Fuel Turbopump Rotordynamic Instability Problem. *ASME J. Eng. Gas Turbines Power Vol.100 No.1: pp. 48-57. DOI: 10.1115/1.3446326.*

Childs, D. W. (1983). Finite Length Solutions for RD Coefficents of Turblent Annular Seals. *ASME Journal of Lubricatuion Technology Vol.105 pp437-445.*

Childs, D. W. & Moyer, D. S. (1985). Vibration Characteristics of the HPOTP (High-Pressure Oxygen Turbopump) of the SSME (Space Shuttle Main Engine). *ASME J. Eng. Gas Turbines Power Vol.107 No.1 : pp. 152-159. DOI: 10.1115/1.3239676.*

Childs, D. W. (1993). Turbomachinery rotordynamics: phenomena, modeling, and analysis. *John Wiley & Sons.*

Chouksey, M., Dutt, J. K. & Modak, S.V. (2012). Modal analysis of rotor-shaft system under the influence of rotor-shaft material damping and fluid film forces. *Mechanism and Machine Theory, 48:81-93*

Gecgel, O., Dias, J. P., Osire, S. E., Alves, D.S., Macado, T. H., Daniel, G. B., Castro, H. F. & Cavalca, K. L. (2020). Simulation-Driven Deep Learning Approach for Wear Diagnostics in Hydrodynamic Journal Bearings. *Journal of Tribology, August 2021, Vol. 143*

Lund, J. W. (1974). Stability and Damped Critical Speeds of a Flexible Rotor in Fluid-Film Bearings. *Journal of Engineering for Industry, 96(2): 509-517*

Nagae, N., Watase, M. & Tamaki, T., (2011). Operational modal analysis using cross correlation function. Japan society of civil engineers, *Journal of structural engineering,Vol.57A (in Japanese)*

Nelson, C. C. (1984). Analysis for Leakage and Rotordynamic Coefficients of Surface-Roughened Tapered Annular Gas Seals. *ASME Journal of Engineering for Gas Turbines and Power Vol.106(1984) pp927-934*

Nelson, C. C. (1985).RD coefficients for Copmressible Flow in Tapered Annular Seals. *ASME Journal of Tribology Vol.107 pp318-325*

Santos, I. F. & Svendsen, P. K.,(2017). Noninvasive Parameter Identification in Rotordynamics Via Fluid Film Bearings- Linking Active Lubrication and Operational Modal Analysis. *Journal of Engineering for Gas Turbines and Power, JUNE 2017, Vol. 139*

Yang, B. S., Iwatsubo, T. & Kawai, R. (1984). A Study on the Dynamic Characteristics of Pump Seal: 1st Report, In case of Annular Seal with Eccentricity. *Bulletin of the JSME Vol.27 No.227: pp. 1047-1057. DOI: 10.1299/jsme1958.27.1047*

Biographies

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