Application of CFD analysis for static and dynamic characteristics of hydrodynamic journal bearing

Kenichiro Takenaka[1]*, Tomoyuki Hayashi[2], Kazuyoshi Miyagawa[2]

ISROMAC 2017

International Symposium on Transport Phenomena and Dynamics of Rotating Machinery

Hawaii, Maui

December 16-21 2017

Abstract

Journal bearings are one of the most important mechanical elements for the stable operation of rotors. Rotational machinery is becoming larger and more complicated in recent years, so proper design of bearings is necessary to offer stable support. However, bearings with shapes and operating conditions that can not be predicted by the conventional Reynolds equation method exist. In this paper ,therefore, a prediction method of vibration characteristics of journal bearings using CFD was constructed. To calculate the dynamic characteristics, mesh morphing methods simulating two oscillating states were studied. In order to compare with the calculation results, an active oscillating rotor using magnetic bearings was constructed. As a result, the calculation results were roughly consistent with the experimental results, and a prediction method by CFD was constructed.

Keywordssss

Journal bearing — Dynamic characteristics — Active magnetic bearing

[1] *Department of Applied Mechanics, Graduate School of WASEDA University, Tokyo, Japan*
[2] *Department of Applied Mechanics and Aerospace Engineering, WASEDA University, Tokyo, Japan*
***Corresponding author**: wishiflatse0324@asagi.waseda.jp

INTRODUCTION

With recent increase in the size and speed of rotational machinery, journal bearings are becoming the bearing type of choice. The choice of journal bearings is explained by the higher loading capacity and damping than ball bearings. However, in rotors supported by journal bearings, a self-excited vibration mechanism called oil whip occurs due to the large XY coupling effect of the dynamic characteristics during high speed rotation. This mechanism cause large vibrations that can lead to shaft system damage and accidents. Furthermore, for rotational machinery in horizontal drives, unless an appropriate bearing was designed, the rotor cannot float. This causes wear and damage of the bearings and efficiency reduction. Therefore, studies to investigate the characteristics of journal bearings have been conducted actively.[1][2][3] A typical calculation method used to investigate the bearing characteristics is the Reynolds equation, but it is unsuitable when the flow is turbulent due to high-speed rotation of the journal or when the bearing shape is complicated. In recent years, therefore, analysis of the bearing characteristics using CFD is often performed. [4][5] However, it seems that methods such as how to create a mesh of the bearing clearance are not established yet. For example, typical size of bearing clearances are in the range of 10-5 to 10-4 [m], so the number of elements becomes enormous unless the mesh aspect ratio in the circumferential and axial direction with respect to the radial direction is increased to some extent. If it is too large, the convergence may be difficult to obtain.

In this research, the static and dynamic characteristics of water lubricated journal bearings were investigated by active control of a rotor using magnetic bearings. Oscillation is performed in the single direction of each of x and y direction to investigate the bearing dynamic characteristics. In addition, CFD analysis of the oscillation of the journal is performed using the morphing method, and it is compared with the experiment result. Two patterns of movement of the journal by morphing were carried out. One is to deform the mesh in a single direction in each of the x and y direction, and the other is to deform the mesh so as to simulate journal whirling. We compare and examine these two analyses with experimental results.

1. METHODS

1.1 Experiment

In order to investigate the vibration characteristics of the journal bearing, an element test apparatus simulating a journal bearing was constructed. When investigating the bearing characteristics, it is necessary to measure the fluid force acting on the bearing. Generally, in a measurement of fluid force, a method of bringing load cells into contact with the bearing housing is used. But the seal system provided in the bearing which stops fluid leakage absorbs the fluid force and prevents accurate measurements. In this test apparatus, therefore, a 2-axis force sensor that can measure the fluid force is attached to the rotor so that the fluid force acting on the rotor can be measured directly.

A cross section view of this test apparatus is shown in Figure 1 and the details of the test section is shown in Figure 2. The test apparatus has a mechanism that water

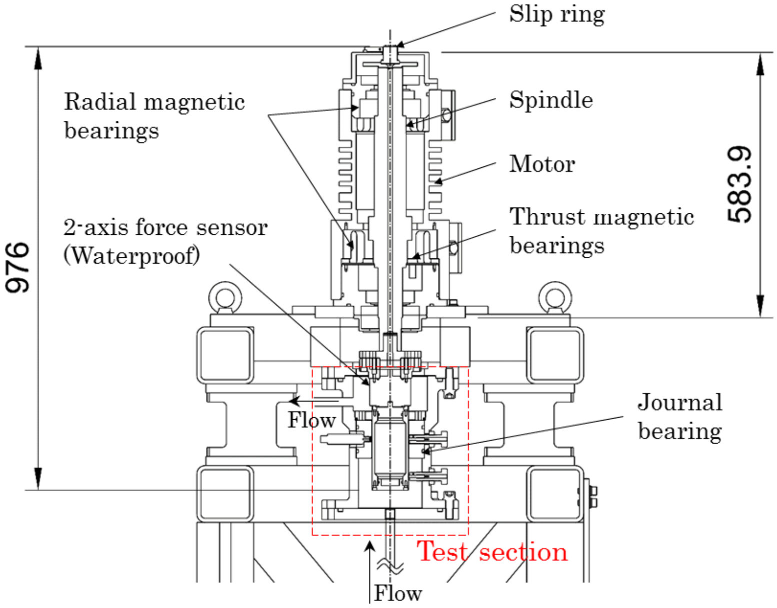

Figure 1 Journal bearing test apparatus cross-section

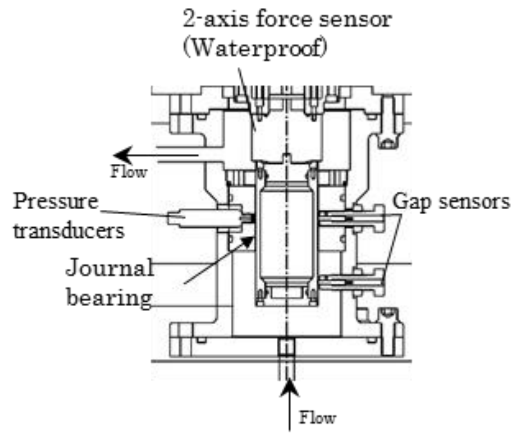

Figure 2 Detail of the test section

pressurized by the compressor from the tank to the lower part of the test bearing.

The flow passes through the bearing clearance and is drained from the upper part of the test section at atmospheric pressure.

The rotor consists of a hollow cylinder, a 2-axial force sensor, an axial force sensor mounting flange and a magnetic bearing spindle, and the rotor is rotated by a motor in the magnetic bearing system. The axial force sensor mounting flange and the spindle are hollow, so that the cable of the axial force sensor can pass through. The cable is connected to the slip ring via the adapter at the top of the spindle.

There are two radial magnetic bearings and one thrust magnetic bearing in the magnetic bearing system, which makes it possible to float the journal bearing. Moreover, by changing the input signal of the radial magnetic bearing, it is possible to eccentrically rotate the rotor while floating it, and to oscillate at arbitrary displacement / frequency in each of the x and y directions.

The test bearing's dimensions are $D = \varphi 80$, L = 80mm, Cr = 0.24mm, and its rotational speed is N = 1500rpm for all experiments. At this time, the fluid force F_x, F_y acting on the journal is measured by the axial force sensor and the influence eccentricity and oscillation was investigated. Also, the eccentricity of the journal was measured with the eddy current displacement sensors, and pressure fluctuation at the center of the bearing was measured with a pressure transducer.

1.2 CFD

CFD analysis was performed to compare the results of static and dynamic characteristics of the journal bearings with experiments. Commercial code SCRYU / Tetra V13 was used for the calculation. The analysis area is shown in Figure 3. The computational grid consists of 2.18 million hexa elements, the total number of nodes is 2.40 million points, the number of mesh layers in the thickness(radial) direction of the bearing clearance is 8 layers, and the average mesh aspect ratio in the circumferential direction and the axial direction is 1:20. As for the boundary condition, the static pressure at the inlet and the outlet was 0Pa, and non-slip wall was applied for the bearing wall surface.

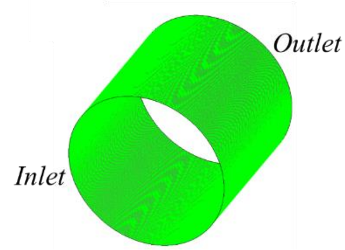

Figure 3 Computational domain

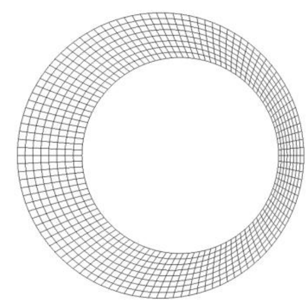

Figure 4 Moving deformation mesh

When static characteristics are calculated, the mesh with eccentricity = 0 is deformed by morphing:, the number of mesh layers in the bearing clearance does not change, so it can be analyzed with high accuracy in the case of high eccentricity. The schematic view of the moving deformation mesh is shown in Figure 4.

Next, the dynamic characteristics calculation method is described. Using an eccentric mesh, the shaft wall surface is oscillates with an amplitude of 10% of the bearing clearance. The journal movement was carried out in two patterns: unidirectional oscillations in the x and y direction

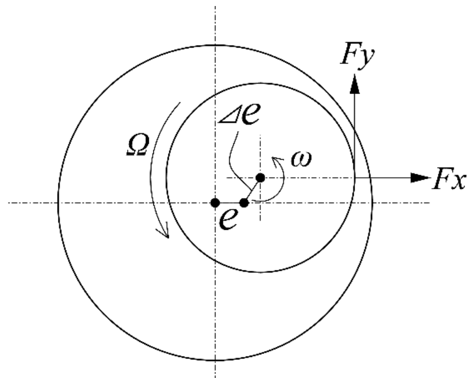

Figure 5 Whirl oscillate method

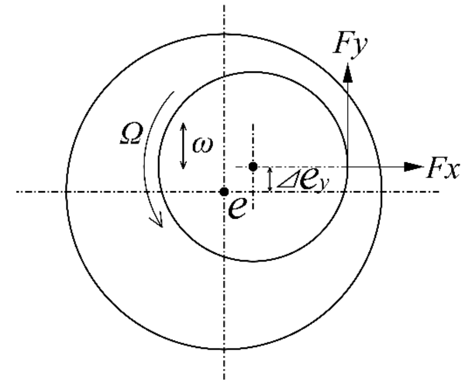

Figure 6 Unidirectional oscillate method

respectively and a whirling. Figure 5 and Figure 6 shows the outline of these movements.

In the CFD analysis, as shown in Figure 5 and 6, coordinates of the static equilibrium point are set to $(e, 0)$ in xy coordinates, and the journal rotates at a constant angular velocity ω with a small amplitude (whirl amplitude) $\Delta e(\Delta e/Cr = 0.1)$ around this point. At this time, the fluid force $F_x(t)$ and $F_y(t)$ are calculated. When the direction of eccentricity of the whirling journal coincides with the direction of the static equilibrium point, that is, when the journal center moves to (e + Δe, 0) and the clearance between the journal and the bearing is minimal, the values of $F_x(t)$ and $F_y(t)$ are substituted into the following expression :

$$\begin{Bmatrix} F_x(t) \\ F_y(t) \end{Bmatrix} = \begin{Bmatrix} F_{0x} \\ F_{0y} \end{Bmatrix} - \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix} \begin{Bmatrix} \ddot{x}(t) \\ \ddot{y}(t) \end{Bmatrix} - \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \begin{Bmatrix} \dot{x}(t) \\ \dot{y}(t) \end{Bmatrix} - \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{Bmatrix} x(t) \\ y(t) \end{Bmatrix} \tag{1}$$

Eqs.(1) holds only when the whirling amplitude is very small, and F_0 are the steady components of the fluid force.

Also, since the locus of the center of the journal is a circle, micro displacements x and y are given as follows :

$$\begin{cases} x = \Delta e\cos\omega t \\ y = \Delta e\sin\omega t \end{cases} \tag{2}$$

If the center coordinate of the journal is (e + Δe, 0) at $t = 0$,the clearance between the journal and the bearing is the smallest at $t = 2n\pi/\omega$, and at this time, Eqs.(1) can be transformed as follows :

$$\begin{cases} f_x|_{t=2n\pi/\omega}/\Delta e \\ f_y|_{t=2n\pi/\omega}/\Delta e \end{cases} = \begin{matrix} M_{xx}\,\omega^2 - C_{xy}\omega - K_{xx} \\ M_{yx}\,\omega^2 - C_{yy}\omega - K_{yx} \end{matrix} \tag{3}$$

Also, values of fluid forces $F_x(t)$ and $F_y(t)$ when $t = (4n+1)\pi/2\omega$ can be calculated at the same time, considering the signs and orientations of $F_x(t)$ and $F_y(t)$, Eqs.(1) can be transformed as follows :

$$\begin{cases} f_x|_{t=\frac{4n+1}{2}\pi/\omega}/\Delta e \\ f_y|_{t=\frac{4n+1}{2}\pi/\omega}/\Delta e \end{cases} = \begin{matrix} M_{xy}\,\omega^2 + C_{xx}\omega - K_{xy} \\ M_{yy}\,\omega^2 + C_{yx}\omega - K_{yy} \end{matrix} \tag{4}$$

Thus, when computing the characteristic value, 12 dynamic coefficients can be derived by finding $F_x(t)$ and $F_y(t)$ when the whirling frequency ω is changed while keeping Δe constant.

In the case of unidirectional oscillation, the displacement of the journal center can be described as follows :

$$\begin{cases} x = \Delta e\cos\omega t \\ y = 0 \end{cases} \text{ or } \begin{cases} x = 0 \\ y = \Delta e\sin\omega t \end{cases} \tag{5}$$

Considering t that $\cos\omega t = 0$ or $\sin\omega t = 0$, the following equations are obtained by substituting x and y from Eqs.(5) into Equ.(1) :

$$\begin{cases} f_x|_{t=2n\pi/\omega}/\Delta e \\ f_y|_{t=2n\pi/\omega}/\Delta e \end{cases} = \begin{matrix} M_{xx}\,\omega^2 - K_{xx} \\ M_{yx}\,\omega^2 - K_{yx} \end{matrix} \tag{6-a}$$

$$\begin{cases} f_x|_{t=\frac{4n+1}{2}\pi/\omega}/\Delta e \\ f_y|_{t=\frac{4n+1}{2}\pi/\omega}/\Delta e \end{cases} = \begin{matrix} C_{xx}\omega \\ C_{yx}\omega \end{matrix} \tag{6-b}$$

$$\begin{cases} f_x|_{t=2n\pi/\omega}/\Delta e \\ f_y|_{t=2n\pi/\omega}/\Delta e \end{cases} = \begin{matrix} -C_{xy}\omega \\ -C_{yy}\omega \end{matrix} \tag{7-a}$$

$$\begin{cases} f_x|_{t=\frac{4n+1}{2}\pi/\omega}/\Delta e \\ f_y|_{t=\frac{4n+1}{2}\pi/\omega}/\Delta e \end{cases} = \begin{matrix} M_{xy}\,\omega^2 - K_{xy} \\ M_{yy}\,\omega^2 - K_{yy} \end{matrix} \tag{7-b}$$

Therefore, by analyzing the cases of arbitrary oscillation frequency ω, it is possible to calculate the dynamic coefficients with the whirling oscillation.

In order to calculate the 12 dynamic coefficients (M_{ij}, C_{ij}, K_{ij}) by the whirl oscillation method, it is necessary to perform transient analyses on at least three different oscillation frequencies ω. On the other hand, in the single-directional oscillation method, it is necessary to perform analyses in at least two different excitation frequencies in the excitation directions in x and y directions, respectively, a total of 4 times. However, if the anisotropy of the bearing characteristics are small, it can be expressed as :
$(M, C, K)_{xx} = (M, C, K)_{yy}$, $(M, C, K)_{xy} = -(M, C, K)_{yx}$,
the unidirectional oscillation method can calculate

6 × 2 dynamic coefficients with only two analyses, which is advantageous in terms of calculation cost.

2. RESULTS AND DISCUSSION

Test results and analysis results of static and dynamic characteristics are shown below. Since the rotational speed of the journal in the experiment was set at 1500 [rpm], and the Reynolds number of the flow in the bearing was about 1700 in the transition area, the analysis was carried out by using the laminar flow model and the SST k-ω turbulence model.

2.1 Static Characteristics

Figure 7-a shows the relation between the Sommerfeld number and eccentricity, and Figure 7-b shows the relation between Sommerfeld number and attitude angle.

It was confirmed that the difference between the analysis results using SST $\boldsymbol{k}-\boldsymbol{\omega}$ analysis and laminar flow analysis was slight as indicated in Figure 7-a. It is assumed that this is because the Reynolds number in the bearing was in the transition region. However, it is shown that SST k-ω calculates the bearing load somewhat smaller than laminar flow analysis. Although the two analysis results are in good agreement with the experiment, a slight deviation is confirmed at the low eccentricity, and it is found that SST k-ω is relatively consistent with the experiment.

On the other hand, looking at the relationship of the eccentricity relationship with the attitude angle, although the simulations and the experimental results generally agree, the difference between the analysis result and the experiment result is slightly larger in the high Sommerfeld region. This is consistent with the region where the analytical value and the experimental value in Figure7-a are somewhat different, and it is probably due to the low accuracy of the experiment. The lower the eccentricity ratio, the smaller the bearing load $\boldsymbol{W}$ becomes, so the measurement accuracy decreased in all likelihood. In addition, it is confirmed that SST k-ω calculates a larger attitude angle than laminar flow analysis, but the difference is within the variation of the experimental result.

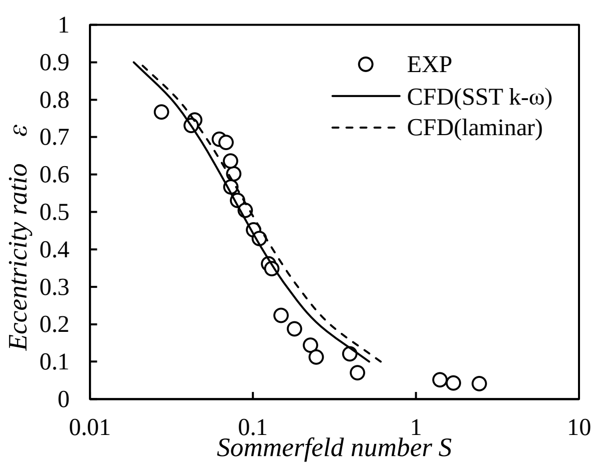

Figure 7-a Bearing load

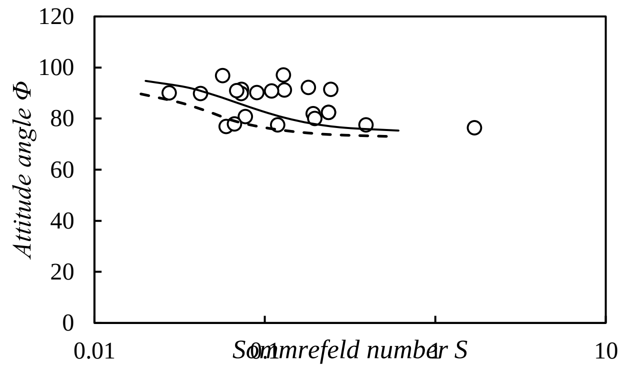

Figure 7-b Attitude angle

2.2 Dynamic Characteristics

The analysis results of dynamic characteristics are shown. As mentioned above, two methods, the unidirectional oscillation method and the whirl oscillation method, were analyzed, and the difference between them was investigated.

For both oscillation methods, analyses were performed for three points of oscillation frequencies $\boldsymbol{\omega}/\boldsymbol{\Omega}$ =0.5, 1.0, and 1.5, and the values of $\boldsymbol{f_x}$and $\boldsymbol{f_y}$ obtained by these analyseis are expressed by equations (3) and (4), (6) and (7). The dynamic coefficients $\mathbf{M}_{\mathrm{ij}}, \boldsymbol{C_{ij}}, \boldsymbol{K_{ij}}$were obtained by fitting using the least squares method as a function of $\boldsymbol{\omega}$. In addition, the eccentricity ratio of the journal was varied from 0.1 to 0.8, and the effects on the dynamic coefficient was examined. In the unidirectional oscillation method, it is possible to calculate all dynamic coefficients by analyzing only two frequencies $\boldsymbol{\omega}$, but those calculated using the analysis results at $\boldsymbol{\omega}$ = 0.5, 1.5 were almost the same as that calculated at 3 points, therefore those are omitted. Figures 8 to 10 show the results of laminar flow analysis, and Figures 11 to 13 show the result of turbulent flow analysis. Each dynamic coefficient is normalized by the oscillation frequency $\boldsymbol{\omega}$, the oscillation amplitude $\Delta\boldsymbol{e}$, and the static load $\boldsymbol{W}$.

From the analysis results, it was confirmed that the dynamic coefficient to the eccentricity using unidirectional oscillation method is consistent to as a function of the same tendency. This tendency was confirmed in both case laminar and turbulent.

First, concerning the inertia coefficients $\boldsymbol{M_{ij}}$, it was confirmed that almost no difference in value caused by the oscillation method was observed. The fluid inertia force decreases as the Sommerfeld number decreases (eccentricity increases) in both of them.

Next, looking at the damping coefficients $\boldsymbol{C_{ij}}$, it can be seen that the value of the direct terms in the x direction($\boldsymbol{C_{xx}}$) for the turbulent flow condition using the whirl method is larger than using the unidirectional method for all Sommerfeld numbers.

On the other hand, with the stiffness coefficients $\boldsymbol{K_{ij}}$, as the Sommerfeld number decreases, the value of the xy coupling terms are different depending on the oscillation exists. Therefore, it was confirmed that the evaluation of the bearing stability is different depending on the oscillation method.

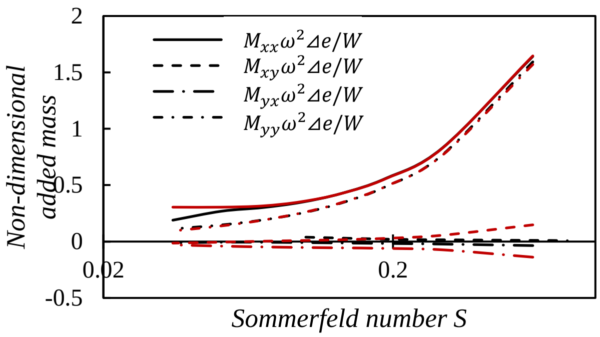

Figure 8 Added Mass Coefficient(— : uni-axial, —: whirl)

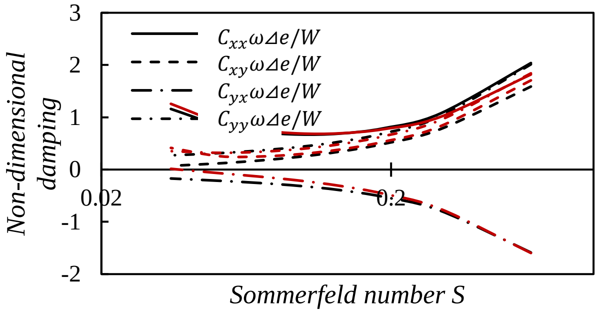

Figure 9 Damping Coefficient(— : uni-axial, —: whirl)

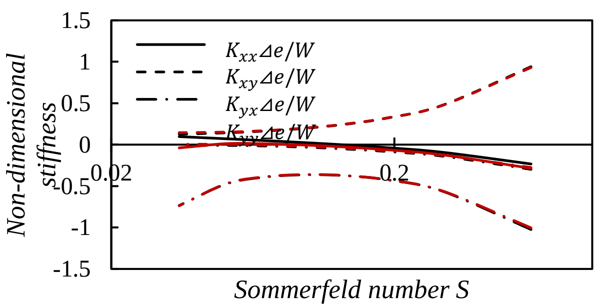

Figure 10 Stiffness Coefficient(— : uni-axial, —: whirl)

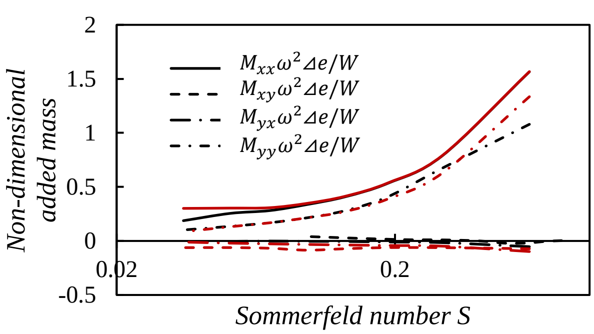

Figure 11 Added Mass Coefficient(— : uni-axial, —:whirl)

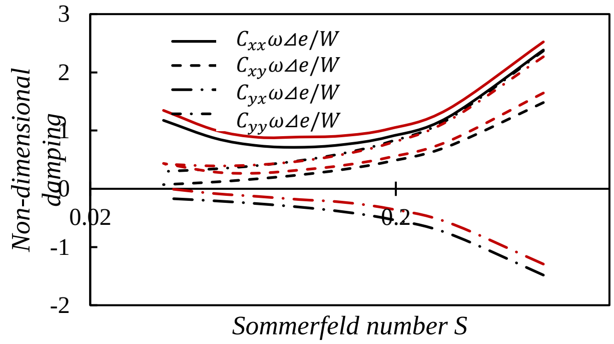

Figure 12 Damping Coefficient(— : uni-axial, —: whirl)

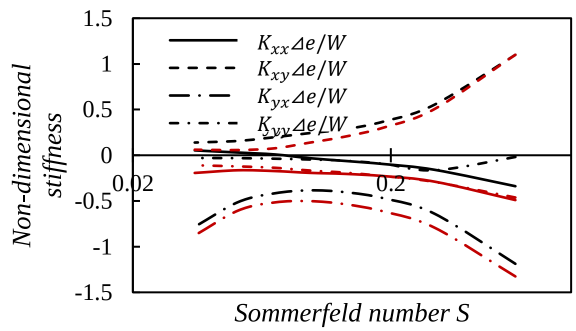

Figure 13 Stiffness Coefficient(— : uni-axial, —: whirl)

In order to investigate the difference in stability in more detail, the energy loss E of the bearing was introduced and organized. $\boldsymbol{E}$ is expressed by the equation (8), which is the energy lost by the fluid while the journal vibrates one period. A larger $\boldsymbol{E}$ implies higher stability.

$$E = \pi \Delta e^2 \{\omega^2 (C_{xx} + C_{yy}) - (K_{xy} - K_{yx})\} \quad (8)$$

The results obtained by organizing using $\boldsymbol{E}$ are shown in Fig. 14 and Fig. 15. From these graphs, the higher the eccentricity ratio is, the larger the value of $\boldsymbol{E}$ obtained by the whirl method compared to unidirectional, and it can be confirmed that it tends to evaluated more stably.

When $\boldsymbol{C_{ii}}$were calculated using the Eqs. (3) and (4) from the result obtained by the whirl method, it is influenced by $\boldsymbol{M_{ij}}(\boldsymbol{i} \neq \boldsymbol{j})$ during the curve fitting of the analysis results. As can be seen from Figure8 and 10, the value of $\boldsymbol{M_{yx}}$ due to the Sommerfeld number is slightly larger than the value of $\boldsymbol{M_{xy}}$. Therefore, it is considered that the value of $\boldsymbol{C_{xx}}$ is influenced by the value of $\boldsymbol{M_{yx}}$.

The above results suggest that, it is better to use the unidirectional oscillation method when evaluating the bearing stability because the direct damping coefficients $\boldsymbol{C_{ii}}$ are not affected by other factors.

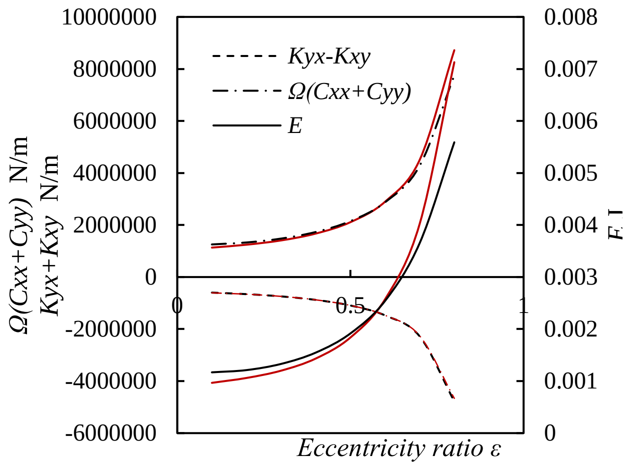

Figure 14 Energy Loss of bearing (Laminar)

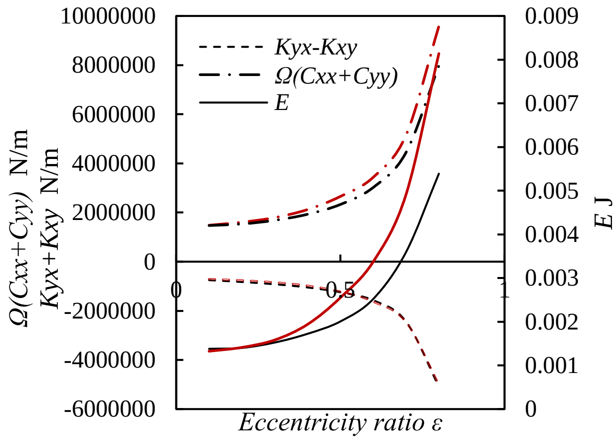

Figure 15 Energy Loss of bearing (Turbulent)

3. Conclusion

In order to investigate a water lublicated journal bearing using experiments and CFD, the following contents were clarified.

1. Static characteristics of the bearing in the transition zone were investigated. There were almost no differences between the analysis using the laminar flow model and the turbulent model(SST k-ω). However, the bearing load was found to be somewhat closer to the experimental results in the analyses using SST k-ω. And it was found that the static characteristics could be calculated with relatively high accuracy.

2. Dynamic characteristics of the bearing were investigated using CFD. In this analysis, journal oscillations were simulated using two mesh morphing methods, and comparisions were made between them.

2-1 Non-dimentional damping coefficient $C_{xx}\omega\Delta e/W$ calculated by the whirl oscillation method larger in all eccentricity than the unidirectional oscillation method.

2-2 The dynamic stability of the bearing was evaluated from the viewpoint of energy loss. The result showed that the analysis results using the whirl oscillation method was tend to evaluate more stable than the unidirectional oscillation method. This tendency is particularly noticeable for high eccentricity ratio, it is considered that the value of M_{ij} at whirl oscillation could not be ignored and could not be calculated C_{ii}completely independently from the fluid force.

NOMENCLATURE

$R=D/2$	Bearing radius	[m]
L	Bearing width	[m]
Cr	Radial clearance	[m]
F	Fluid force	[N]
Ω	Angular velocity of journal	[min^{-1}]
ω	Oscillation frequency	[min^{-1}]
W	Bearing load	[N]
e	Journal eccentricity	[m]
φ	Attitude angle	[deg]
$\varepsilon=e/Cr$	Journal eccentricity ratio	
θ	Circumferential angle from maximum clearance point	[deg]
U	Journal surface velocity	[m/s]
ν	Kinematic viscosity of lubricant	[m^2/s]
K_{ij} (i, $j=x$, y)	Stiffness or spring coefficient	[N/m]
C_{ij} (i, $j=x$, y)	Damping coefficient	[N・s/m]
M_{ij} (i, $j=x$, y)	Mass coefficient	[N・s^2/m]
$Re=CrU/\nu$	Reynolds number	
$S=\mu NLD(R/Cr)^2/W$	Sommerfeld number	

ACKNOWLEDGMENTS

This work was done as part of the project financially supported by the NEDO (New Energy and Industrial Technology Development Organization). We would like to thank the organizers of this work.

REFERENCES

[1] Iwatsubo, T., Ono, M., and Tsuzimoto, Y., “Calculations of Dynamic Force of Parallel Annular Seal with Eccentricity”, Transactions of the Japan Society of Mechanical Engineers, Series C, Vol.60, No.93-1355, 4136-4141, 1994.

[2] Kanki, H., Ozawa, Y., Oda, T., and Kawakami, T., “Study on the Dynamic Characteristics of a Water-Lubricated Pump Bearing”, Transactions of the Japan Society of Mechanical Engineers, Series C, Vol.51, No.85-0107 B, 2650-2654, 1985.

[3] Childs, D. W., “Turbomachinery Rotordynamics, Phenomena, Modeling, and Analysis”, John Willey & Sons, Inc, New York, 145-162, 1993.

[4] Zenglin Guo, Hirano, T., R. Gordon Kirk, “Application of CFD Analysis for Rotating Machinery—Part I_Hydrodynamic, Hydrostatic Bearings and Squeeze Film Damper”, ASME, Journal of Engineering for Gas Turbines and Power, Vol. 127, 2005.

[5]K. M. Panday, P. L. Choudhury, and N. P. Kumar, Numerical Unsteady Analysis of Thin Film Lubricated Journal Bearing, International Journal of Engineering and Technology, Vol. 4, 2012

[6] Eguchi, M., “Development of measurement system for seal dynamics applied magnetic bearings”, Dynamics and Design Conference 2000, No.411, 2000.

[7] Someya, T., “Journal-Bearing Databook”, The Japan Society of Mechanical Engineers, 1989.