The Characteristics of Rotordynamic Forces generated by Mechanical Seals

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Abstract. Evaluating rotodynamic reaction force caused by whirl excitation is very effective technique in order to investigate the dynamic properties of the mechanical seal. Especially, it is easy to understand the stability of the rotating component by investigating rotodynamic tangential force. In this paper, the stability of the mechanical seal was discussed by measuring the rotodynamic tangential force depending on the whirl frequency. Judging from the result, the mechanical seal used for this study has stable performance that the tangential force always acts against whirling direction at 1,000 rpm of rotational speed and up to 30 Hz of whirl frequency. Furthermore, whirl frequency ratio and effective damping were calculated using cross-coupled stiffness and direct damping which were obtained by solving quaternary linear equations after whirl excitation test. Whirl frequency ratio was always bellow 1.0, and effective damping was also always positive at all frequencies. Those results indicate that the mechanical seal definitely generate the force to stabilize this whirl vibration.

1. Introduction

In recent years, multidisciplinary optimization in rotor system design has been actively performed [1]. In order to find further improvements of stability, it will be necessary to design the rotor system considering with the rotodynamic forces generated by mechanical seals even if the contact forces occurred on the sliding surface are absolutely small. Especially, it is important for industrial products with high safety requirements such as rocket turbo pumps. Although there are many rotating machines that use mechanical seals as turbomachinery component, the rotodynamic forces generated by the mechanical seals have never been considered at the rotor system design stage. That is, there is no comprehensive data on the results of the rotodynamic forces related to the mechanical seals. The authors attempted to measure the rotodynamic force concerning the mechanical seal in the past [2]. It was very difficult to measure the rotodynamic force generated by mechanical seal accurately because the force was absolutely very small and was measured together with rotodynamic force due to oil seal which was installed to prevent fluid from entering the support bearing of the test rig. In this study, we developed a new test bench using active magnetic bearings (AMBs) and specially tuned PID controller with adaptive feedforward cancellation (AFC) [3] for the mechanical seal to experimentally obtain the rotodynamic force accurately. The control system can make a rotating shaft follow up an arbitrary orbit, and the rotodynamic forces can be measured in real-time during the operation.
As the result of the experiment, the quite small rotordynamic forces caused by the mechanical seals were obtained accurately. Tangential force grows as the whirl frequency increases. Tangential force is negative when the whirl frequency is positive range. On the other hand, positive tangential force is occurred when the whirl frequency is negative range.

Furthermore, whirl frequency ratio and effective damping were calculated using cross-coupled stiffness and direct damping which were obtained by solving quaternary linear equations after whirl excitation test in order to improve the data reliability. Whirl frequency ratio was always bellow 1.0, and effective damping was also always positive at all frequencies.

Those results indicate that this mechanical seal are used for this study definitely generate the force to stabilize this whirl vibration under the operating conditions.

2. Test bench

2.1. Hardware

The test bench overview is shown in figure 1. One radial AMB and one thrust AMB support a main shaft which is driven via an intermediate shaft and a flexible coupling by a 200 W rated output DC brushless motor made by Oriental Motor Co., Ltd. (Model BLM5200-A). On the opposite side of the main shaft, mechanical seal is mounted on an additional shaft which is attached to the end of the main shaft. Appearance of mechanical seal part is shown figure 2.

![Figure 1](image1.png)  
*Figure 1. Test bench overview.*

![Figure 2](image2.png)  
*Figure 2. Appearance of mechanical seal part.*

2.2. AMB specifications

Section view of AMBs is shown in figure 3. Shaft diameter is 60mm. The radial AMB is composed of heteropolar 8 poles. Magnetic flux density is 0.7 T. Control voltage is 90 VDC. PWM amplifiers supply power to AMB. Attractive radial force per two electromagnets is 260 N at 10 A. The thrust AMB is composed of homopolar 4 poles. Magnetic flux density is 0.7 T. Control voltage is 80 VDC. Thrust attractive force is 500 N at 10 A.

2.3. Mechanical seal specifications

Mechanical seals are mostly comprised of two rings. Rotary ring and stationary ring are installed on the shaft and on the machine housing. Type MBS100 (Eagle Industry Co., Ltd.) is used in this test. This type of mechanical seal is widely adopted in the oil refinery and petrochemical industries. Appearance of the mechanical seal used for this study is shown in figure 4. Material of Rotary ring and stationary ring are Carbon graphite and Silicon carbide, respectively. Metal bellows is made out of Inconel® 625. Secondary seals is made of FKM. The other metal parts are made out of Duplex. The mechanical seal is for 20 mm shaft diameter.
2.4. Feedback controller and digital signal processor

Feedback controller is built on MATLAB® /Simulink® (R2018b). C code is generated by Simulink® Coder™ using Simulink model. Building DSP program is completed after compiling the C code and linking object files by using Code Composer Studio™ (v6.2.0).

2.4.1. Structure of Feedback controller. In order to measure the rotordynamic force, the feedback controller is designed for controlling current sent to the AMBs. The feedback controller consists of the PID controller and the AFC. A block diagram of the control system is shown in Figure 5. In this system, the rotating shaft is controlled in two directions to achieve the arbitral whirling orbit. The directions are defined as x or y direction. \( P_i(s), P_y(s) \) are transfer functions of the plant in \( x \), \( y \) directions, respectively. \( s \) is Laplace operator. \( C_i(z), C_y(z) \) are the feedback controllers. \( z^1 \) is one-sample delay operator. \( d_i(t), d_y(t) \) are the disturbances. \( F_i(t), F_y(t) \) are the rotor dynamic forces. \( t \) is time. \( e_i(k), e_y(k) \) are the position error signal. \( r_x(k), r_y(k) \) are the reference signals which determine the whirling orbit of the rotating shaft. These are defined in equations (1).

\[
\begin{align*}
  r_x(k) &= A \sin(\Omega t) \\
  r_y(k) &= A \cos(\Omega t)
\end{align*}
\]

where, \( k \) is a sample number, \( A \) is an amplitude and \( \Omega \) is a whirl frequency. In this control system, the PID controller \( C_i(z), C_y(z) \) compensate for the disturbance \( d_i(t), d_y(t) \). Figure 6 indicates the amplitude spectra of \( d_i(t) \) and \( e_i(k) \). A criterion is set to the spectrum of \( e_i(k) \) as shown in Figure 6. In this paper, only the results in \( x \) direction are explained, because the results in \( y \) direction are almost same. The criterion indicates cumulative sum of the amplitude of \( e_i(t) \) is less than 1 \( \mu \)m (resolution of the gap sensor).

The AFC can compensate and estimate for \( F_i(t), F_y(t) \) simultaneously at \( \Omega \). The algorithm of the AFC in \( x \) direction can be described as equations (2).

\[
\begin{align*}
  u_i(k) &= p_x(k-1) \sin(\Omega Tk) + q_x(k-1) \cos(\Omega Tk) \\
  p_x(k) &= p_x(k-1) + \lambda_x e_x(k) \sin(\Omega Tk + \theta_x) \\
  q_x(k) &= q_x(k-1) + \lambda_x e_x(k) \cos(\Omega Tk + \theta_x)
\end{align*}
\]

where \( p_x(k) \) and \( q_x(k) \) are the adaptive parameters, \( \lambda_x \) is the step size parameter, \( \theta_x \) is the phase. The adaptive algorithm updates \( p_x(k) \) and \( q_x(k) \) using \( p_x(k-1), q_x(k-1) \) and \( e_x(k) \). In the adaptive algorithm, \( \lambda_x \) and \( \theta_x \) are the design parameters. The design method is proposed in the previous studies [4][5]. To realize the arbitral whirling motion, the adaptive parameters are updating to compensate for the influence due to the rotor dynamic force. In the feedback control system, the influence can be described as term (3).

\[
\frac{F_x}{1 + p_x C_x}
\]
When the adaptive parameters are convergence, \(u_i\) is equal to \(-F_x/(1 + P_x C_x)\). Therefore, the amplitude of the external force can be described as equations (4) and (5).

\[
\begin{align*}
\left| -\frac{F_x}{1 + P_x C_x} \right| &= \sqrt{p_x^2(k) + q_x^2(k)} \\
\arg \left( -\frac{F_x}{1 + P_x C_x} \right) &= \arctan \left( \frac{p_x(k)}{q_x(k)} \right)
\end{align*}
\]

The characteristic of \(P_x(s), C_x(z)\) can be identified in advance. Therefore, the amplitude and phase can be estimated by the adaptive parameters in real-time.

2.4.2 Digital signal processor. Digital signal processor (DSP) is 66AK2H06DAAW24 (Texas Instruments Inc.) which can be inserted Compact PCI bus. A/D and D/A conversion board are RPT1320-00A and RPT2320-00A (MIS Corporation), respectively. Resolution of each conversion board is 16 bit. In the DSP, the sampling time \(T_s\) is 0.1 ms.

2.5. Measuring Instruments

Dedicated gap sensors for magnetic bearings made by SHINKAWA Electric Co., Ltd. are used. Those gap sensors are eddy current type (Model 1094-003). This sensor has a resolution of less than 1 \(\mu\)m. Differential amplifiers (Model 1195-510/01 and 1192/02) are used for good output linearity and temperature drift invariability. Data logger is combination of DL7000 and multiple GL7-HSVs (GRAPHTEC Corporation). Regarding measurement of temperature of water for the mechanical seal lubricant, K type thermocouples are used. FFT (Brüel & Kjær Type 3050-A-060) is used for real time monitoring of the whirl excitation as additional measurement.

3. Definition of rotordynamic forces

The equation of motion for the rotor is expressed as equation (6).

\[
\begin{bmatrix}
F_x \\
F_y
\end{bmatrix} =
\begin{bmatrix}
M & 0 \\
0 & M
\end{bmatrix}
\begin{bmatrix}
x'' \\
y''
\end{bmatrix} +
\begin{bmatrix}
C_{xx} & C_{xy} \\
C_{yx} & C_{yy}
\end{bmatrix}
\begin{bmatrix}
x' \\
y'
\end{bmatrix} +
\begin{bmatrix}
K_{xx} & K_{xy} \\
K_{yx} & K_{yy}
\end{bmatrix}
\begin{bmatrix}
x \\
y
\end{bmatrix}
\]

where \(M\) is the isotropic mass of the rotor including added mass. \(F_x\) and \(F_y\) are reaction forces from the rotor in \(x, y\) directions, respectively. These forces correspond to radial attractive forces of AMB. \(C_{xx}\),
$C_{yy}$, $K_{xx}$ and $K_{yy}$ are direct damping and spring coefficients. $C_{xy}$, $C_{yx}$, $K_{xy}$ and $K_{yx}$ are cross-coupled damping and spring coefficients. When the rotor whirs with a small amplitude and concentrically around the center of the radial AMB, equation (6) can be simply rewritten as equation (7).

$$-\begin{bmatrix} F_x \\ F_y \end{bmatrix} = \begin{bmatrix} M & 0 \\ 0 & M \end{bmatrix} \begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} C & c \\ C & -c \end{bmatrix} \begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} + \begin{bmatrix} K & k \\ -k & K \end{bmatrix} \begin{bmatrix} x \\ y \end{bmatrix}$$  \( \text{(7)} \)

Displacement, velocity and acceleration of rotor are expressed as sine and cosine functions, respectively. Substituting those sine and cosine functions into equation (7), and converting equation (7) to the tangential direction and vertical direction, equations (8) are obtained.

$$F_r = e(M\Omega^2 - c\Omega - K)$$

$$F_t = e(-c\Omega + k)$$  \( \text{(8)} \)

where $e$ is whirl amplitude. $\Omega$ is whirl frequency. $C$ and $K$ are direct damping and spring coefficients. $c$ and $k$ are cross-coupled damping and spring coefficients. $F_r$ and $F_t$ are radial and tangential components of rotordynamic force $F$, respectively. Positive $F_r$ is defined with outward direction, and positive $F_t$ is defined with the direction of the rotation. These are depicted together in figure 7.

**Figure 7.** Definition of rotordynamic forces.

4. **Whirl excitation test conditions and procedure**

The test conditions are shown in table 1. The whirl excitation test was performed when the two rings of the mechanical seal was in non-contact and in contact in order to purely measure the $F_r$ and $F_t$ caused by mechanical seal. $F_{rms}^r$ and $F_{rms}^t$ of rotordynamic forces generated by mechanical seal are calculated by subtracting the results of non-contact test from the results of contact test.

| Table 1. Test conditions. |
|---------------------------|
| Rotational speed ($\omega$) | 1,000 rpm |
| Whirl frequency ($\Omega$)  | 1 Hz, 2~30 Hz (even number) |
| Whirl amplitude ($e$)       | 30 $\mu$m |
| Water temperature ($T_w$)   | 24 $^\circ$C |

5. **Results and discussion**

All test data is recorded every 1 ms on a data logger. Figure 8 and figure 9 show the orbits and time histories of the displacement of the rotor during whirl excitation test at 30 Hz of whirl frequency,
respectively. At this time, the two rings of the mechanical seal contacted each other. It can be seen that the trajectory of the rotor is almost ideal circle and sine (cosine) waveform. Motion of rotor is well controlled with about 1 µm runout accuracy by using the state of the art PID controller.

![Figure 8. Whirl orbit of the rotor.](image1)

![Figure 9. Time history of displacement.](image2)

5.1. Stability analysis by rotodynamic forces
Tangential force $F_t$ directly has a big influence on the vibration stability of the rotor. So first, tangential force was discussed. $F_t$ is positive and has destabilizing influence when the force is same as the direction of forward whirling motion and $\Omega$ is larger than 0 Hz. On the other hand, $F_t$ is negative and has stabilizing effect when the force is same as the direction of forward whirling motion and $\Omega$ is larger than 0 Hz. When $\Omega$ is negative, positive $F_t$ shows the stabilizing effect.

![Figure 10. Relationship between $\Omega$ and $F_{tr}^{\text{ms}}$.](image3)

![Figure 11. Relationship between $\Omega$ and $F_{tr}^{\text{ms}}$.](image4)

$F_{tr}^{\text{ms}}$ always acts in the opposite whirl direction at all whirl frequencies. This result indicates that the mechanical seal has clearly stabilizing effect against the shaft vibration under the test conditions at this time. And basically, $|F_{tr}^{\text{ms}}|$ rises as the $|\Omega|$ increases proportionally. It can be seen that $|F_{tr}^{\text{ms}}|$ asymptotically approach 0.02 N when the $\Omega$ is 0 Hz. It may have another meaning that $|F_{tr}^{\text{ms}}|$ become pure $|F_{tr}^{\text{ms}}|$ under radial misalignment of 30 µm without whirl. It might suggest that rotor system can be stabilized by giving parallel misalignment since this tangential force of 0.02 N works effectively on the system. Tadokoro also mentioned about stabilizing effect under parallel misalignment in circular sliding contact [6]. It is interesting that mechanical seal has a potential to stabilizing by giving parallel misalignment in addition to rotor whirling. Another interesting point is that $F_{tr}^{\text{ms}}$ becomes extremely small around 10 Hz. This phenomenon occurs on a couple of tries reproducibly. It may be due to a big change in the tribological properties of mechanical seal around 10 Hz. Although it also may be affected by the eigenvalues of the components of mechanical seal, the detail is unknown at the moment. Further research will be carried out by using a 6-axis torque sensor and additional measuring devices.
Regarding radial $F_r$, negative $F_r$ gives the effect of reducing the whirl amplitude of the rotor. Positive $F_r$ means inertial force, and increases the whirl amplitude of the rotor. Figure 11 shows the relationship between whirl frequency $\Omega$ and the mechanical seal radial force $F_{rms}$. The magnitude of $F_{rms}$ is much smaller than that of $F_r$. $F_{rms}$ is mostly negative when $\Omega$ is positive. On the other hand, $F_r$ is mostly positive when $\Omega$ is negative.

5.2. Stability analysis by whirl frequency ratio and effective damping coefficient
The previous section revealed that the mechanical seal generated stability force, but its level is unknown. Whirl frequency ratio ($WFR$) and effective damping coefficient ($C_{eff}$) are employed for evaluating seal stability [7]. $WFR$ is expressed as equation (9).

$$WFR = \frac{k}{C\Omega} \quad (9)$$

$C_{eff}$ is defined by equation (10) which is obtained from the second equation of equation (8).

$$C_{eff} = C \left(1 - \frac{k}{C\Omega}\right) \quad (10)$$

If $WFR$ is larger than 1.0, the rotor system is surely unstable because destabilizing force exceeds stabilizing force. Regarding $C_{eff}$, when $C_{eff}$ is larger, it means that stability of the rotor system is higher. $k$ and $C$ for calculating $WFR$ and $C_{eff}$ are able to generally approximated based on figure 10 and equation (8) for annular seal. However, it is very difficult to obtain those dynamic properties by approximate calculation because there is a singularity of the data at 10 Hz in this case. Therefore, spring coefficients and damping coefficients have to be obtained by directly solving equation (6) by substituting recorded experimental $F_r$, $F_c$, $x$ and $y$ and computed $\dot{x}$, $\ddot{x}$, $\dot{y}$ and $\ddot{y}$ to equation (6). Table 2 shows spring coefficients, damping coefficients, $WFR$ and $C_{eff}$ of the mechanical seal at four frequencies (4, 10, 20 and 30 Hz). $K_{ms}$, $k_{ms}$, $c_{ms}$ and $\epsilon_{ms}$ mean dynamic properties of mechanical seal. They are calculated by subtracting the dynamic properties when the mechanical seal is in non-contacting condition from the dynamic properties when the mechanical seal is in contacting condition as shown above. Superscript (contact) of $K_{contact}$ designates contact state, and superscript (non-contact) of $K_{non-contact}$ designates non-contact state. The same meaning of superscript is applied to $k$, $C$ and $c$.

| $\Omega$ (Hz) | 4   | 10  | 20  | 30  |
|---------------|-----|-----|-----|-----|
| $K_{ms}$ (= $K_{contact}$ - $K_{non-contact}$, N/m) | 2961.89 | 25805.78 | 16566.20 | 36029.21 |
| $k_{ms}$ (= $k_{contact}$ - $k_{non-contact}$, N/m) | 2759.00 | -18465.60 | 16355.88 | 25520.88 |
| $c_{ms}$ (= $C_{contact}$ - $C_{non-contact}$, N·s/m) | 258.22 | 2602.06 | 553.03 | 837.19 |
| $\epsilon_{ms}$ (= $\epsilon_{contact}$ - $\epsilon_{non-contact}$, N·s/m) | 151.83 | -368.14 | 46.98 | 159.40 |
| $WFR$ (-) | 0.43 | -0.11 | 0.24 | 0.16 |
| $C_{eff}$ (N·s/m) | 148.45 | 2895.94 | 422.87 | 701.80 |

Relationship between whirl frequency $\Omega$ and spring coefficient $K_{ms}$ and damping coefficient $C_{ms}$ are shown in figure 12 and figure 13, respectively. It can be seen that $K_{ms}$ and $C_{ms}$ increase in quadratic function as whirl frequency increases except for 10 Hz (see red dot in figure 12). $k_{ms}$ also increases in quadratic function as whirl frequency increases, whereas a correlation between $\epsilon_{ms}$ and whirl frequency is almost constant throughout the whole whirl frequency judging from the data at 4 and 30 Hz though $\epsilon_{ms}$ at 20 Hz is very small value. Regarding $WFR$ and $C_{eff}$, relationship between whirl frequency $\Omega$ and $WFR$ and $C_{eff}$ are shown figure 14 and figure 15. $WFR$ decreases gradually with increasing whirl frequency. On the other hand, $C_{eff}$ increases in quadratic function as whirl frequency increases.
There was a singularity of stability at 10 Hz in the figure 10 as mentioned above. Experimental $C_{\text{eff}}$ at 10 Hz is about 12 times larger than $C_{\text{eff}}$ which obtained from fitting curve. It can be explained that the singularity is due to the extremely large $C_{\text{eff}}$.

As shown above, it can be seen that mechanical seal is always stable. The fact that mechanical seal generates stabilizing force against whirl vibration is good news for mechanical seal manufacturers.

6. Conclusion

In this study, the authors measured rotordynamic force, and additionally obtained dynamic properties of a mechanical seal by using new developed AMB facilities. Summarised results are as follows.

- The mechanical seal generates stabilising force against rotor whirl.
- Stabilizing tangential force $|F_{\text{ms}}|$ increases proportionally as the $|\Omega|$ increases.
- The mechanical seal has a possibility of generating effective tangential force to suppress the rotational movement due to parallel misalignment.
- Direct spring, cross-coupled spring and direct damping increase in quadratic function as whirl frequency increases, whereas a correlation between $c_{\text{ms}}$ and whirl frequency is nearly constant.

References

[1] Uchiumi M, Shimagaki M, Yoshida Y and Adachi K 2012 28th International Congress of the Aeronautical Sciences 52 ICAS2012-4.3.4
[2] Kuroki Y, Uchiumi M, Nagao N, Inoue H, Hiromatsu J and Sato K 2013 JSME Fluids Engineering Conference 0430 (in Japanese)
[3] Bodson M, Sacks A. and Khosla P 1994 IEEE Transactions on Automatic Control 39 (9) 1939
[4] Yabui S Kajiwara, I Nakamura S and Atsumi T 2013 Journal of Advanced Mechanical Design, Systems, and Manufacturing 7 (6) 903
[5] Yabui S and Inoue T 2019 Journal of Vibration and Control 25 (4) 793
[6] Tadokoro C, Nagamine T and Nakano K 2018 Tribology International 120 16
[7] Iwatsubo T and Ishimaru H 2010 Journal of System Design and Dynamics 4 (1) 177