Eliminating the Fluid-Induced Vibration and Improving the Stability of the Rotor/Seal System Using the Inerter-Based Dynamic Vibration Absorber

Qi Xu, Yuanqing Luo, Hongliang Yao, Lichao Zhao, and Bangchun Wen

1School of Mechanical Engineering, Shenyang University of Technology, Shenyang 110870, China
2School of Mechanical Engineering and Automation, Northeastern University, Shenyang 110819, China
3Installation Accessories Co. Ltd., Shenyang Blower Group Co. Ltd., Shenyang 110869, China

Correspondence should be addressed to Hongliang Yao; hlyao@mail.neu.edu.cn

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A theoretical research on eliminating the instability vibration and improving the stability of the rotor/seal system using the inerter-based dynamic vibration absorber (IDVA) is presented in this paper. The modified Jeffcott rotor and Muszynska nonlinear seal force models are employed. The proposed IDVA is the damping element of the traditional dynamic vibration absorber (DVA) replaced by one of the six configurations of the inerter. The instability threshold speed of the system is obtained by applying the Routh–Hurwitz stability criterion. The quantum particle swarm optimization (QPSO) method is utilized to optimize the parameters of the IDVA. The numerical method is applied to investigate the nonlinear dynamic responses and stability. The results show that the IDVA can effectively improve the stability and reduce the instability vibration of the rotor/seal system. Furthermore, the performance of the IDVA is more effective than that of the traditional DVA.

1. Introduction

The nonlinear seal force between the rotor and seal is prone to the instability vibration in the rotating machine. The instability vibration is one kind of the self-excited vibrations and adversely causes the amplitude to increase dramatically. Hence, the instability vibration must be controlled.

Great efforts for the instability vibration control had been made by Bently and Muszynska [1]. The nonlinear behavior and stability had been analysed via the Bently/Muszynska (B/M) nonlinear seal force model. The expression of the instability threshold was defined, and the “cross stiffness” component was the main reason for the instability. The fluid circumferential average velocity ratio is considered a key parameter for the instability vibration control. It can be realized using the various seals to disrupt the fluid flow [2–4]. The antiswirl flow in the seal was a typical method to eliminate the instability [5]. It was widely used in engineering. Zhang et al. [6] computed the dynamic and flow characteristics of the honeycomb seal and corresponding labyrinth seal in the rotor, and the results showed that the instability vibration could be reduced due to the additional aerodynamic force generated by the seal. Then, they presented two types of the labyrinth seals in the rotor system: the straight-through and corresponding interlaced labyrinth seal. The dynamic and flow characteristics of the rotor/seal system were analysed by the numerical method, and it showed that the straight-through labyrinth seal could reduce the rotor instability vibration [7]. Jia et al. [8] investigated the dynamic leakage rate of the rotor-labyrinth seal system with the variable rotating speed. The similar conclusion that the labyrinth seal could improve the rotor stability was obtained. Zhang et al. [9]
studied dynamic characteristics of the rotor/bearing/labyrinth-seal system. The research indicated that the shorter seal length could improve the stability of the rotor system, but the sealing effect was decreased.

Recently, the DVA has begun to attract attention to eliminate the rotor vibration. Hu and He [10] designed a rotor DVA to online control the rotor critical speed vibration and discussed the effect of the installation position on the suppression performance. Heidari and Monjezi [11] investigated the vibration control of the unbalanced rotor using the virtual passive DVA with the optimal parameters. The permanent magnet DVAs with the negative and tunable stiffness were presented to reduce the rotor unbalance vibration by Yao et al. [12, 13]. Besides, the nonlinear energy sink (NES, a kind of the nonlinear stiffness DVA) and centrifugal pendulum vibration absorber (CPVA, a kind of the torsion DVA) have been applied for the antiresonance and torsional vibration suppression of the rotor, respectively. For example, Taghipour et al. [14] studied the antiresonance for the nonlinear bearing of the rotor with the DVA and NES. Tehrani and Dardel [15] investigated the DVA and NES to prevent the contact between the rotor and stator. The NES was also employed for the vibration control of the unbalanced hollow rotor [16] and continuous rotor-blisk-journal bearing system [17]. Shi et al. [18] and Nishimura [19] et al. described the torsional vibration reduction in the rotor using the CPVA.

Now that the inerter, which is proposed for the vibration control of the vehicle suspension [20], has already been applied in the DVA, called the IDVA. Hu and Chen [21] investigated the parameter optimization to improve the performance of the IDVA. The extended fixed-point theory was used for the parameter optimization of the IDVA [22, 23]. Lazarek et al. [24] designed a tuned mass damper with the variable inerter. The IDVA has been utilized to the vibration control of the cable [25, 26], the offshore wind turbine [27], tall building [28], and so on.

The traditional DVA could eliminate the instability vibration in the rotor/seal system [29, 30]. The fluid-induced instability elimination of the rotor with the IDVA is proposed in this paper. First, the model of the rotor/seal system with the IDVA is established. Second, the expression of the instability threshold speed is obtained. Third, the numerical calculation is employed for the nonlinear dynamic and stability analysis. Then, the performance comparison between the IDVA and traditional DVA is discussed. Finally, some conclusions are drawn to summarize the effects of the fluid-induced vibration elimination and stability region extension on the rotor/seal system by the IDVA.

2. Model of Rotor/Seal System with IDVA

Figure 1 shows the model of the rotor/seal system with the IDVA. The instability vibration usually happens in the turbomachinery, such as the centrifugal compressor. Normally, the running speed of the kind of the rotor system is between the first- and second-order critical speeds. Therefore, the modified Jeffcott rotor model and B/M nonlinear seal force model are employed for the rotor/seal system model [1, 29]. The rotor model is employed for considering the first-order vibration shape. The nonlinear dynamic behaviors and stability of the real rotor/seal system can be described by the models. The IDVA is added in the rotor in the perpendicular directions by the rolling bearing in order to avoid rotation with the rotor [10]. The DVA and IDVA are depicted in Figure 2. The damping element of the DVA is replaced by one of the six configurations of the inerter: $c_1$, $c_2$, $c_3$, $c_4$, $c_5$, and $c_6$ [21]. Taking $c_3$ as an example in the main part of this paper, the equations of motion for the rotor/seal system with the IDVA are as follows:

$$\begin{align*}
mx + dx + kx + f_{x\text{Seal}} + k_a(x - x_a) + f_{x\text{IDVA}} &= m_e\omega^2 r \cos\omega t, \\
m\ddot{y} + d\dot{y} + ky + f_{y\text{Seal}} + k_a(y - y_a) + f_{y\text{IDVA}} &= m_e\omega^2 r \sin\omega t, \\
m_x\ddot{x} + k_a(x - x) - f_{x\text{IDVA}} &= 0, \\
m_y\ddot{y} + k_a(y - y) - f_{y\text{IDVA}} &= 0,
\end{align*}$$

where $m$, $d$, and $k$ are the rotor generalized mass, lateral external damping, and lateral isotropic stiffness, respectively. $m_e$ and $r$ denote, respectively, the eccentric mass and eccentricity. $\omega$ is the rotating speed. $m_x$ and $k_a$ denote, respectively, the mass and stiffness coefficients of the IDVA. $x$, $y$, $x_a$, and $y_a$ are the horizontal and vertical displacement generalized coordinates of the rotor and IDVA, respectively. $f_{x\text{Seal}}$ and $f_{y\text{Seal}}$ are the components of the nonlinear seal force in the $x$ and $y$ directions, respectively. The expressions are

$$\begin{align*}
f_{x\text{Seal}} &= m_i\ddot{x} + d_i\dot{x} + 2m_i\omega^2\lambda y + (k_i - m_i\omega^2\lambda^2)x + \omega d_l\lambda y, \\
f_{y\text{Seal}} &= m_i\ddot{y} - 2m_i\omega^2\lambda x + d_i\dot{y} - \omega d_l\lambda x + (k_i - m_i\omega^2\lambda^2)y,
\end{align*}$$

where $m_i$, $d_i$, and $k_i$ denote, respectively, the seal fluid inertia, seal fluid film radial damping, and seal fluid stiffness coefficients. $\lambda$ is the seal fluid circumferential average velocity ratio. The expressions of the nonlinear terms are

$$\begin{align*}
d_i &= d_0\left(1 - z^2\right)^n, \\
k_i &= k_0\left(1 - z^2\right)^n, \\
\lambda &= \lambda_0\left(1 - z^2\right)^b, \\
z &= \frac{\sqrt{x^2 + y^2}}{r_l},
\end{align*}$$

where $d_0$, $k_0$, $\lambda_0$, $n$, and $b$ are the parameters of the nonlinear seal force. These parameters of the nonlinear force are described in detail by Bently and Muszynska [1, 31]. $r_l$ is the radial clearance of the seal.
$f_{\text{IDVA}}$ and $f_{\text{IDVA}}$ denote, respectively, the force components of the IDVA that acted on the rotor in the $x$ and $y$ directions. The expressions are as follows:

$$
\begin{align*}
  f_{\text{IDVA}} &= c (\dot{x} - \dot{x}_c) = k_3 (x_k - x_a) = b (\ddot{x} - \ddot{x}_k), \\
  f_{y\text{IDVA}} &= c (\dot{y} - \dot{y}_c) = k_2 (y_k - y_a) = b (\ddot{y} - \ddot{y}_k),
\end{align*}
$$

(4)

where $b$, $c$, and $k_2$ are the inertance, damping, and stiffness coefficients of the configuration of the inerter. $x_c$ and $y_c$ denote, respectively, the generalized displacements between the inertance and damping element in the $x$ and $y$ directions. $x_a$ and $y_a$ denote, respectively, the generalized displacements between the inertance and stiffness element in the $x$ and $y$ directions.

To define the dimensionless parameters:

$$
\begin{align*}
  \epsilon_c &= \frac{m_c}{m}, \\
  \epsilon_f &= \frac{m_f}{m}, \\
  \epsilon_a &= \frac{m_a}{m}, \\
  \epsilon_b &= \frac{b}{m_a} \\
  \zeta &= \frac{d}{2\sqrt{km}}, \\
  \zeta_a &= \frac{c}{2\sqrt{k_2m_a}}
\end{align*}
$$

(5)

and define the generalized coordinates (Taking the $x$ coordinate as an example):

$$
\begin{align*}
  X &= \frac{x}{r_f}, \\
  Y &= \frac{y}{r_f}, \\
  X_a &= \frac{x_a}{r_f}, \\
  Y_a &= \frac{y_a}{r_f}, \\
  X_c &= \frac{x_c}{r_f}, \\
  Y_c &= \frac{y_c}{r_f}, \\
  X_k &= \frac{x_k}{r_f}, \\
  Y_k &= \frac{y_k}{r_f}
\end{align*}
$$

(6)

$$
\tau = \omega t,
$$

(7)

The dimensionless transform is as following:
Conveniently, the dimensionless generalized coordinates are still expressed as \( \bar{x}, \bar{y}, x_a, y_a, \bar{\xi}, \bar{\eta}, \bar{x}_c, \bar{y}_c, \) and \( y_b, \) respectively. The original equations become the dimensionless equations:

\[
\begin{align*}
x'' + 2\bar{\omega}_x x' + \bar{\omega}^2 x + \bar{f}_{x\text{Seal}} + \epsilon_0 \bar{\omega}_b^2 (x - x_a) + \bar{f}_{x\text{IDVA}} &= \epsilon_0 \cos \tau, \\
y'' + 2\bar{\omega}_y y' + \bar{\omega}^2 y + \bar{f}_{y\text{Seal}} + \epsilon_0 \bar{\omega}_b^2 (y - y_a) + \bar{f}_{y\text{IDVA}} &= \epsilon_0 \sin \tau, \\
x''_a + \bar{\omega}_a^2 (x_a - x) - \bar{f}_{x\text{IDVA}} &= 0, \\
y''_a + \bar{\omega}_a^2 (y_a - y) - \bar{f}_{y\text{IDVA}} &= 0,
\end{align*}
\]

(8)

where

\[
\begin{align*}
\bar{\omega}_x &= \frac{\omega_x}{\omega}, \\
\bar{\omega}_y &= \sqrt{k/m}, \\
\bar{\omega}_a &= \frac{\omega_a}{\omega}, \\
\omega_a &= \sqrt{k_a/m_a}.
\end{align*}
\]

(9)

The dimensionless components of the nonlinear seal force:

\[
\begin{align*}
\bar{f}_{x\text{Seal}} &= \epsilon_1 x'' + \mu_d x' + 2\epsilon_1 \lambda y' + (\mu_k - \epsilon_1 \lambda^2) x + \mu_d \lambda y, \\
\bar{f}_{y\text{Seal}} &= \epsilon_1 y'' - 2\epsilon_1 \lambda x' + \mu_d y' - \mu_d \lambda x + (\mu_k - \epsilon_1 \lambda^2) y,
\end{align*}
\]

(10)

where

\[
\begin{align*}
\mu_d &= \frac{d_f}{m \omega}, \\
\mu_k &= \frac{k_f}{m \omega^2}.
\end{align*}
\]

(11)

The dimensionless force components of the IDVA acted on the rotor:

\[
\begin{align*}
\bar{f}_{x\text{IDVA}} &= 2\epsilon_2 \bar{\omega}_b (\dot{x} - \dot{x}_c) = \epsilon_2 \bar{\omega}_b^2 (x_k - x_c) = \epsilon_2 (\bar{\omega}_c - \bar{\omega}_k), \\
\bar{f}_{y\text{IDVA}} &= 2\epsilon_2 \bar{\omega}_b (\dot{y} - \dot{y}_c) = \epsilon_2 \bar{\omega}_b^2 (y_k - y_c) = \epsilon_2 (\bar{\omega}_c - \bar{\omega}_k),
\end{align*}
\]

(12)

where

\[
\begin{align*}
\bar{\omega}_b &= \frac{\omega_b}{\omega}, \\
\omega_b &= \sqrt{k/b}.
\end{align*}
\]

(13)

\section*{3. Stability Analysis}

The unbalance force is assumed zero, the nonlinear terms and seal fluid inertia coefficient are neglected as they are minimal for the stability analysis [32]. The original dimensionless equations (8) can be rewritten:

\[
\begin{align*}
x'' + 2\bar{\omega}_x x' + \bar{\omega}^2 x + \mu_d x' + \mu_k x + \mu_d \lambda y + \epsilon_0 \bar{\omega}_b^2 (x - x_a) + \bar{f}_{x\text{IDVA}} &= 0, \\
y'' + 2\bar{\omega}_y y' + \bar{\omega}^2 y + \mu_d y' + \mu_d \lambda x + \epsilon_0 \bar{\omega}_b^2 (y - y_a) + \bar{f}_{y\text{IDVA}} &= 0, \\
x''_a + \bar{\omega}_a^2 (x_a - x) - \bar{f}_{x\text{IDVA}} &= 0, \\
y''_a + \bar{\omega}_a^2 (y_a - y) - \bar{f}_{y\text{IDVA}} &= 0.
\end{align*}
\]

(14)

The characteristic equation can be obtained:

\[
a_0 \mu^{2q} + a_1 \mu^{2q-1} + a_2 \mu^{2q-2} + \cdots + a_{2q-3} \mu^3 + a_{2q-2} \mu^2 + a_{2q-1} \mu + a_{2q} = 0,
\]

(15)

where \( \mu \) is the eigenvalue and \( q \) is the degree of freedom of equation (14). \( a_0, a_1, a_2, \ldots, a_{2q} \) are the coefficients of the characteristic equation.

According to the Routh-Hurwitz stability criterion [33], the instability threshold speed \( \omega_i \) of the rotor system can be obtained from the following expressions:

\[
H_1, H_2, \ldots, H_{2q-2} > 0, \\
H_{2q-1} = 0,
\]

(16)

where \( H \) is the Hurwitz determinant, and:

\[
\begin{pmatrix}
a_1 & a_3 & a_5 & \cdots & 0 \\
a_0 & a_2 & a_4 & \cdots & 0 \\
0 & a_1 & a_3 & \cdots & 0 \\
0 & a_0 & a_2 & \cdots & 0 \\
\vdots & \vdots & \vdots & \ddots & \vdots \\
0 & 0 & 0 & \cdots & a_{2q}
\end{pmatrix}
\]

(17)

The rotor/seal system is stable as \( \omega < \omega_i \) and unstable as \( \omega > \omega_i \).

\section*{4. Numerical Simulations and Discussion}

The numerical calculation is employed due to the complicated nonlinear terms of equation (8) and implicit equation (16). The parameters of the rotor/seal system are obtained from reference [29] in order to compare with the previous research. The rotor \( m = 1 \text{ kg} \), \( \zeta = 0.025 \), \( \omega_i = 200 \text{ rad/s} \), \( r_i = 10^{-3} \text{ m} \) and \( \epsilon_1 = 0.01 \). The nonlinear seal force \( \epsilon_2 = 10^{-5}, \)
\( d_0 = 100 \text{ N/m} \), \( k_0 = 0.48 \), \( n = 2 \) and \( b = 0.5 \). Assuming that the dimensionless rotating speed \( s = \omega/\omega_i \), the frequency ratio \( \sigma_s = \omega_s/\omega_i \) and \( \sigma_b = \omega_b/\omega_i \). The QPSO algorithm is utilized to obtain the optimal IDVA parameters. The target function is the rotor instability threshold speed \( \omega_i \). The mass ratio \( \epsilon_m = 0.01 \) and \( \epsilon_b \) is a constant. Moreover, the damping ratio \( \zeta_a \), dimensionless frequency ratio \( \sigma_a \) and \( \sigma_b \).
are the optimization parameter. Correspondingly, the parameter optimization ranges are $0 < \omega_b < 1$, $0 < \sigma_a \leq 2$ and $0 < \sigma_b \leq 2$, respectively. The flow chart of the QPSO method is shown in Figure 3.

The optimization results are shown in Table 1. It can be seen that the maximum of the instability threshold speed is $\omega_1 = 2.759$ as $\omega_b = 0.03$, $\xi_a = 0.05$, $\sigma_a = 0.98$, and $\sigma_b = 0.95$. The convergence diagram of the optimization as $\omega_b = 0.03$ is in Figure 4. It shows that the optimization is convergent as the number of the iterations is greater than 15.

Therefore, the parameters of the IDVA are selected as follows: $\omega_b = 0.03$, $\xi_a = 0.05$, $\sigma_a = 0.98$, and $\sigma_b = 0.95$. The changes of the coefficients in equation (15) and Hurwitz determinants to estimate the stability are presented in Figure 5. It can be seen that $a_1$, $a_2$, $a_3$, $a_7$, $a_{11}$ are positive (Figure 5(a)); $H_1$, $H_2$, $H_3$, $H_6$ are also positive (Figure 5(b)), $H_9$ and $H_{10}$ are also positive at $s \leq 2.76$, and $H_{11}$ is equal to zero when $s$ is between 2.755 and 2.76 (Figure 5(c)). The coefficients and Hurwitz determinants satisfy the Routh–Hurwitz stability criterion. The bifurcation diagram, amplitude-frequency response curve, frequency spectrum, orbit, and Poincaré map are used to investigate the nonlinear behavior by the Newmark time integration method [34]. The algorithm is unconditionally convergent as the values of parameters $\beta \geq 1/4$ and $\delta \geq 1/2$ [35].

The bifurcation diagram and amplitude-frequency response curve of the rotor/seal system in the $x$ coordinate are shown in Figure 6. The blue and red solid dots and lines are the bifurcation and amplitude-frequency response curve of the rotor/seal system without and with the IDVA, respectively. It can be observed in Figure 6(a) that the vibration of the rotor/seal system is synchronous to one period at $s = 2.31$. Only one point is correspondingly shown in the bifurcation diagram for every rotating speed. Then, the vibration changes and bifurcates at $s = 2.31$. The vibration is double-periodic, multi-periodic, or quasi-periodic at $s \geq 2.31$ and can be determined by using the Poincaré map. $s = 2.31$ is the instability threshold speed of the rotor/seal system. The variances of the vibration in the rotor/seal system as adding the IDVA are as follows: the vibration of the system is synchronous to one period at $s < 2.805$; then the vibration changes and bifurcates at $s = 2.805$; the vibration is double-periodic, multi-periodic, or quasi-periodic at $s \geq 2.805$ and can be determined by using the Poincaré map; and $s = 2.805$ is the instability threshold speed of the rotor/seal system. Comparing with the rotor/seal system without the IDVA, the instability threshold speed increases by about 21.43%. The instability vibration between $s = 2.31$ and $s = 2.805$ becomes the synchronous vibration. The instability vibration is eliminated completely in this range. Simultaneously, the instability threshold speed increases by about 1.67% comparing with Table 1. The result illustrates that the instability threshold obtained by the linear equation (16) is less than that obtained by the nonlinear equation (8). Whether or not to add the IDVA, the amplitude of the system increases rapidly when the rotating speed exceeds the instability threshold speed as shown in Figure 6(b). But the instability amplitude can be reduced by the IDVA at $s \geq 2.805$. The instability vibration is eliminated partially in this range.

![Figure 3: The flow chart of the QPSO algorithm.](image)

**Table 1: The parameters optimization results of the IDVA**

<table>
<thead>
<tr>
<th>$\omega_b$</th>
<th>Optimized $\xi_a$</th>
<th>Optimized $\sigma_a$</th>
<th>Optimized $\sigma_b$</th>
<th>$\omega_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.01</td>
<td>0.05</td>
<td>0.98</td>
<td>1</td>
<td>2.743</td>
</tr>
<tr>
<td>0.02</td>
<td>0.04</td>
<td>0.99</td>
<td>0.98</td>
<td>2.756</td>
</tr>
<tr>
<td><strong>0.03</strong></td>
<td><strong>0.05</strong></td>
<td><strong>0.98</strong></td>
<td><strong>0.95</strong></td>
<td><strong>2.759</strong></td>
</tr>
<tr>
<td>0.04</td>
<td>0.05</td>
<td>0.98</td>
<td>0.94</td>
<td>2.750</td>
</tr>
<tr>
<td>0.05</td>
<td>0.05</td>
<td>0.98</td>
<td>0.92</td>
<td>2.753</td>
</tr>
<tr>
<td>0.06</td>
<td>0.05</td>
<td>0.98</td>
<td>0.91</td>
<td>2.745</td>
</tr>
<tr>
<td>0.07</td>
<td>0.05</td>
<td>0.98</td>
<td>0.90</td>
<td>2.739</td>
</tr>
<tr>
<td>0.08</td>
<td>0.06</td>
<td>0.97</td>
<td>0.85</td>
<td>2.735</td>
</tr>
<tr>
<td>0.09</td>
<td>0.05</td>
<td>0.98</td>
<td>0.88</td>
<td>2.731</td>
</tr>
<tr>
<td>0.10</td>
<td>0.05</td>
<td>0.98</td>
<td>0.87</td>
<td>2.728</td>
</tr>
</tbody>
</table>

![Figure 4: The optimization convergence result of the IDVA as $\omega_b = 0.03$.](image)

The speed $s = 2$, $s = 2.5$, and $s = 3$ are picked, respectively, for the dynamic behavior analysis in the range of $s < 2.31$, $2.31 < s < 2.805$, and $s > 2.805$ as shown in Figures 7–9. The blue and red solid dots and lines are the response of the
rotor/seal system without and with the IDVA, respectively. Figure 7 presents the response is synchronous to one period, and there is less change after adding the IDVA as the speed \( s = 2 \) is away from the resonance region. The response is five-periodic when \( s = 2.5 \) as shown in Figure 8. The instability vibration frequency is close to the first-order natural frequency of the rotor. The amplitude of the instability frequency is much higher than that of the rotating frequency. Then the response becomes one period, and the instability frequency disappears after adding the IDVA. As the speed continues to increase, Figure 9 represents the response is three-periodic when \( s = 3 \). The response becomes quasi-period, and the instability frequency is eliminated partially after adding the IDVA.

### 5. Comparison with the Previous Work

The traditional DVA has been applied to eliminate the instability vibration in the rotor/seal system in reference [29]. The comparison results of the instability threshold speed are shown in Table 2. It can be seen that the instability threshold speed of the rotor with the IDVA is larger than that of the rotor with the traditional DVA except the configuration of the inerter \( c_2 \). The maximum instability threshold speed is obtained from the configuration of the inerter \( c_3 \).

The comparison results of the bifurcation diagram and amplitude-frequency response curve are shown in Figure 10. The black and red solid dots and lines are the response of the rotor/seal system with the traditional DVA and IDVA, respectively. It can be seen that the instability threshold speed of the rotor with the IDVA \( (c_3) \) is larger than that of the rotor with the traditional DVA, and the instability amplitude of the rotor with the IDVA \( (c_3) \) is less than that of the rotor with the traditional DVA. The result still illustrates that the linear model appears more conservative in evaluating instability.

The comparison results of the relationship between the instability threshold speed and frequency ratio \( \sigma_a \) and the instability threshold speed and damping ratio \( \zeta_a \) are shown in Figure 11. The black dashed line represents the instability threshold speed of the rotor. Comparing the rotor with the traditional DVA, Figure 11(a) shows that the range of increase in instability threshold speed is almost constant as the frequency ratio \( \sigma_a \) changes, while Figure 11(b) shows that the range of increase in instability threshold speed is decreased as the damping ratio \( \zeta_a \) changes. And the maximum of the instability threshold speed increases by about 3.13%.
Figure 7: The response of the rotor/seal system at $s = 2$. (a) The frequency spectrum of the $x$ coordinate. (b) The orbit of the rotor. (c) The Poincaré map of the $x$ coordinate.

Figure 8: The response of the rotor/seal system at $s = 2.5$. (a) The frequency spectrum of the $x$ coordinate. (b) The orbit of the rotor. (c) The Poincaré map of the $x$ coordinate.

Figure 9: The response of the rotor/seal system at $s = 3$. (a) The frequency spectrum of the $x$ coordinate. (b) The orbit of the rotor. (c) The Poincaré map of the $x$ coordinate.

Table 2: The comparison results of the instability threshold speed.

<table>
<thead>
<tr>
<th></th>
<th>$\epsilon_b$</th>
<th>Optimized $\zeta_a$</th>
<th>Optimized $\sigma_a$</th>
<th>Optimized $\sigma_b$</th>
<th>$\omega_i$</th>
<th>Increment</th>
</tr>
</thead>
<tbody>
<tr>
<td>DVA</td>
<td>—</td>
<td>0.05</td>
<td>0.99</td>
<td>—</td>
<td>2.687</td>
<td>—</td>
</tr>
<tr>
<td>IDVAc1</td>
<td>0.01</td>
<td>0.05</td>
<td>1.00</td>
<td>—</td>
<td>2.696</td>
<td>+0.33%</td>
</tr>
<tr>
<td>IDVAc2</td>
<td>0.10</td>
<td>0.05</td>
<td>1.00</td>
<td>—</td>
<td>2.504</td>
<td>−6.81%</td>
</tr>
<tr>
<td>IDVAc3</td>
<td>0.03</td>
<td>0.05</td>
<td>0.98</td>
<td>0.95</td>
<td>2.759</td>
<td>+2.68%</td>
</tr>
<tr>
<td>IDVAc4</td>
<td>0.09</td>
<td>0.03</td>
<td>1.01</td>
<td>0.76</td>
<td>2.728</td>
<td>+1.53%</td>
</tr>
<tr>
<td>IDVAc5</td>
<td>0.09</td>
<td>0.14</td>
<td>0.93</td>
<td>1.79</td>
<td>2.730</td>
<td>+1.60%</td>
</tr>
<tr>
<td>IDVAc6</td>
<td>0.04</td>
<td>0.01</td>
<td>0.98</td>
<td>1.03</td>
<td>2.755</td>
<td>+2.53%</td>
</tr>
</tbody>
</table>
6. Conclusions

The fluid-induced vibration elimination and stability margin extension of the rotor/seal system using the IDVA is proposed in this paper. The rotor/seal system dynamic model is established, and the six configurations of the IDVA are added, respectively, in the system model. The instability threshold speed of the system is obtained by applying the Routh–Hurwitz stability criterion. The QPSO algorithm is utilized to obtain the optimal IDVA parameters. The numerical method is applied to investigate the nonlinear dynamic responses and stability of the rotor/seal system without and with the IDVA. The results are compared with the previous work. The conclusions are drawn as follows.

1. The IDVA can eliminate the fluid-induced instability vibration and improve the instability threshold speed of the rotor/seal system.

2. Comparing with the traditional DVA, the effect of the IDVA on the instability vibration elimination and stability improvement is better, but the range of increase in the instability threshold speed is decreased as the damping ratio $\zeta_a$ changes.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

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