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Article

Analysis on Dynamic Tracking Characteristics of Dry Gas Seals During Start-Up Process

Qiangguo Deng ^{1,*}, Yong Zhou ¹, Pingyang Yu ¹, Hengjie Xu ^{2,*}, Xuejian Sun ²
and Wenyuan Mao ²

¹ Faculty of Mechanical and Electrical Engineering, Kunming University of Science and Technology, Kunming, 650500, China

² Faculty of Chemical Engineering, Kunming University of Science and Technology, Kunming, 650500, China

* Correspondence: dengqiangguo@kust.edu.cn(Q.D.); why_551@kmust.edu.cn(H.X.)

Abstract: Based on the small perturbation method, the transient pressure control equation considering real gas effects was solved, and the fitting expression for the dynamic characteristic parameters of the gas film during the start-up process was obtained. Subsequently, the influence of structural parameters of spiral groove dry gas seals on the dynamic tracking of the stationary ring's motion during the non-steady-state start-up process under three degree of freedom perturbations was analyzed. The results show that when the stationary and rotating rings initially separate, the stationary ring exhibits good tracking performance for both axial and angular motions of the rotating ring, although the tracking capability varies significantly. As time and film thickness increase, the tracking capability gradually weakens, and at the working film thickness, the tracking parameters tend to stabilize when the working film thickness is reached. Larger spiral angles and deeper dynamic pressure grooves, the poorer the axial and angular tracking performance of the sealing ring. The number of grooves has a minimal impact on the axial and angular tracking performance of the stationary ring. A higher balance coefficient improves the axial and angular tracking performance of the stationary ring.

Keywords: dry gas seals; structural parameters; tracking performance; axial displacement; angular tilt

1. Introduction

Dry gas seals are a non-contact mechanical sealing technology, and widely used in high-speed rotating equipment such as centrifugal compressors and turbines due to its excellent performance, including zero wear, long lifespan, and high sealing efficiency [1,2]. During the operation of dry gas seals, the stationary ring must continuously track the motion of the rotating ring to maintain a stable sealing gap and prevent leakage. The sealing gap remains constant during stable operation. When subjected to external disturbances, the axial and angular vibration behavior of the dry gas seals, as well as the dynamic tracking characteristics of the stationary ring, directly affect the operational state of the dry gas seal. Ruan [3], Miller [4], Yelma [5], Zhang [6], and Chen [7] and others have conducted systematic research on the dynamic tracking performance of dry gas seals. Recently, Lee [8] used direct dynamic numerical methods to analyze the dynamic tracking performance of the stationary ring under external disturbances, concluding that while the rotating ring jumping and seal ring tilt have little impact on leakage, but can affect the probability of surface contact and the tracking performance of the stationary ring. Blasiak [9] suggested that surface texture depth and angular velocity are key factors influencing the amplitude of the stationary ring, and that appropriate operational and geometric parameters can reduce the angular vibration amplitude of the stationary ring and the leakage rate, thereby improving operational stability. Yang [10] analyzed the dynamic tracking performance of S-CO₂ dry gas seals using direct analytical methods and finite difference

methods, highlighting the significant impact of turbulence effects. Increasing the pump inlet pressure and reducing the rotational speed, stationary ring mass, spring stiffness, and O-ring damping were found to enhance the dynamic tracking performance of dry gas seals. Chen [11] provided a range of structural parameters for spiral groove dry gas seals that exhibit good dynamic tracking performance, identifying dynamic stiffness as the primary factor influencing tracking performance. He [12] also compared the dynamic tracking performance of dry gas seals under different flexible installation methods when subjected to external disturbances, concluding that dry gas seals with both stationary and rotating rings flexibly installed exhibit the best dynamic tracking performance. Teng [13] noted that excessive spring stiffness and O-ring damping are detrimental to the tracking performance of dry gas seals, but a higher axial gas film stiffness can improve seal tracking performance.

The start-up process of dry gas seals is a non-steady-state process where the sealing gap gradually increases. When subjected to external disturbances, the start-up characteristics of the gas film can easily be disrupted, resulting reduce or even eliminate of track ability of the stationary ring follow motion of the rotating ring. This presents a challenge to the opening capability and stability of dry gas seals. There is limited research on the dynamic tracking performance of dry gas seals during the start-up phase under external disturbances. This paper focuses on the dry gas seals device of a synthetic ammonia compressor. It analyzes the influence of spiral groove structural parameters on the dynamic tracking performance of the stationary ring under axial vibration and angular tilt disturbances, considering the real effects of carbon dioxide. The findings provide theoretical references for the design and selection of dry gas seals.

2. Computational Model

2.1. Physical Model

Figure 1 shows the dynamic physical model of a dry gas seals under three-degree-of-freedom perturbations. a_r represents the angular tilt of the rotating ring relative to the vertical plane of the main shaft; a_s represents the angular tilt of the stationary ring relative to the vertical plane of the main shaft; k_s is the spring stiffness; c_o is the O-ring damping. After the stationary and rotating rings separate, the seal end faces are filled with pressurized gas. As the rotational speed increases, the sealing gap (gas film thickness) gradually increases to the working film thickness of stable operation phrase. External disturbances caused by installation deviations, motor vibrations, are transmitted from the rotating ring to the flexibly installed stationary ring through the gas film, causing the stationary ring to experience corresponding axial vibrations and angular tilts. Therefore, the dynamic behavior of dry gas seals during the start-up process is more complex than during stable operation.

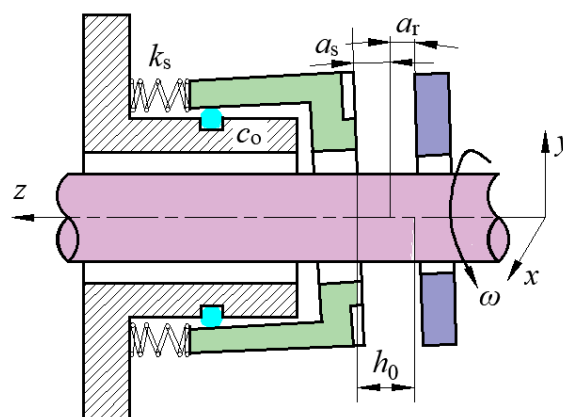


Figure 1. Dynamic model of dry gas seals under three degree of freedom perturbation.

2.2. Theoretical Model

2.2.1. Perturbed Pressure Control Equation

The gas film on the seal end faces is subjected to external three degree of freedom perturbations, which are generally small and can be considered as micro-perturbations. The perturbation process is similar to a spring-damper structure, often characterized by gas film stiffness and damping. The dynamic stiffness and damping of the gas film are referred to as gas film dynamic characteristic coefficients, which reflect the force state of the gas film and can be used to evaluate the stability of the gas film during the start-up process. The small perturbation method was used to solve the modified transient pressure control equation (1), considering slip flow effects, real gas effects, and dynamic viscosity, to analyze the gas film pressure distribution and its ability to resist external disturbances under three degree of freedom perturbations.

$$\frac{\partial}{\partial r} \left(\phi_q \frac{rh^3}{Z\eta} \frac{\partial p^2}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left(\phi_q \frac{h^3}{Z\eta} \frac{\partial p^2}{\partial \theta} \right) = 12\omega r \frac{\partial}{\partial \theta} \left(\frac{ph}{Z} \right) + 24r \frac{\partial}{\partial t} \left(\frac{ph}{Z} \right) \quad (1)$$

Where ω is the rotational speed, and η is the dynamic viscosity, calculated using the temperature and pressure-dependent viscosity fitting method from Deng [14]. The slip rate factor is modeled using the F-K slip flow approach [15]:

$$\phi_q = \begin{cases} 1 + 6.0972/D + 6.3918/D^2 - 12.8124/D^3 & D \geq 5 \\ 0.83112 + 7.50522/D + 0.93918/D^2 - 0.05814/D^3 & 0.15 \leq D < 5 \\ -13.37514 + 12.64038/D + 0.09918/D^2 - 0.0004164/D^3 & 0.01 \leq D < 0.15 \end{cases} \quad (2)$$

Where Z is the real gas compressibility factor, modeled using the virial equation to describe the real gas behavior of carbon dioxide [15]:

$$Z = \frac{pV_m}{R_i T} \approx 1 + \frac{pB_v}{R_i T} + (C_v - B_v^2) \left(\frac{p}{R_i T} \right)^2 + \dots \quad (3)$$

Where B_v and C_v are called the second virial coefficient and the third virial coefficient, and their values are compared with the temperature T_r and eccentricity factor ϵ .

The three degree of freedom perturbations of the stationary ring in the axial and two angular directions are analyzed via normal mode analysis, as follows:

$$\begin{cases} \Delta z_s(t) = \Delta z_0 e^{if_1 t}, \Delta \dot{z}_s(t) = if_1 \Delta z_0 e^{if_1 t} \\ \Delta \alpha_s(t) = \Delta \alpha_0 e^{if_2 t}, \Delta \dot{\alpha}_s(t) = if_2 \Delta \alpha_0 e^{if_2 t} \\ \Delta \beta_s(t) = \Delta \beta_0 e^{if_2 t}, \Delta \dot{\beta}_s(t) = if_2 \Delta \beta_0 e^{if_2 t} \end{cases} \quad (4)$$

Where Δz_0 , $\Delta \alpha_0$, $\Delta \beta_0$ are the initial axial and angular perturbation magnitudes of the stationary ring, f_1 and f_2 are the equal perturbation frequencies in the axial and angular directions, and t is time.

The gas film thickness perturbation is expressed as:

$$\Delta h(r, \theta, t) = \Delta z_s(t) + \Delta \alpha_s(t)r \sin \theta - \Delta \beta_s(t)r \cos \theta \quad (5)$$

The gas film thickness at the equilibrium position at a given time post-perturbation is:

$$\begin{aligned} h(r, \theta, t) &= h_0 + \Delta h \\ &= h_0 + \Delta z_s(t) + \Delta \alpha_s(t)r \sin \theta - \Delta \beta_s(t)r \cos \theta \end{aligned} \quad (6)$$

Angular tilt causes uneven gas film thickness distribution in the sealing gap at any moment. Thus, this paper examines the gas film thickness response over time during start-up process under axial perturbation only.

The transient pressure is:

$$\begin{aligned}
 p(r, \theta, t) &= p_0 + \Delta p \\
 &= p_0 + p_z \Delta z_s(t) + p_\alpha \Delta \alpha_s(t) + p_\beta \Delta \beta_s(t) \\
 &= p_0 + p_1 + p_2 + p_3
 \end{aligned} \tag{7}$$

Equations (6) and (7) are substituted into Equation (1), and higher-order perturbation terms are neglected, simplifying the nonlinear transient pressure control equation into a steady-state equation and a perturbed dynamic pressure equation. The three degree of freedom pressure terms in the latter are expressed as real and imaginary parts.

$$p_j(r, \theta, t) = p_{jr} + ip_{ji} \quad (j = z, \alpha, \beta) \tag{8}$$

The parameters in the dynamic pressure control equation are nondimensionalized as follows:

$$\begin{aligned}
 P_0 &= p_0 / p_{re}, \quad H = h / h_g, \quad R = r / r_o, \quad \Lambda = 6\omega r_o^2 / (p_{re} h_g^2), \quad \Gamma = \nu / \omega, \quad t^* = \nu t \\
 P_{zl} &= p_{zl} h_g / p_{re}, \quad P_{je} = p_{je} h_g / (p_{re} r_o) \quad (j = \alpha, \beta; l = r, i)
 \end{aligned} \tag{9}$$

Where p_{re} , h_{re} are the reference pressure and film thickness, p_{jr} , p_{ji} ($j=z, \alpha, \beta$) are the real and imaginary parts of the axial and angular perturbation pressures, and Γ is the frequency ratio.

The dimensionless steady-state pressure control equation is:

$$\frac{\partial}{\partial R} \left(\frac{\phi_q R H_0^3}{Z \eta} \frac{\partial P_0^2}{\partial R} \right) + \frac{\partial}{\partial \theta} \left(\frac{\phi_q H_0^3}{Z \eta} \frac{\partial P_0^2}{\partial \theta} \right) = 2\Lambda R \frac{\partial}{\partial \theta} \left(\frac{P_0 H_0}{Z} \right) \tag{10}$$

The real part of the dimensionless dynamic pressure equation is:

$$\begin{aligned}
 &\left\{ \begin{aligned} &\frac{\partial}{\partial R} \left(\frac{\phi_q R H_0^3}{Z \eta} \frac{\partial P_0 P_{zr}}{\partial R} + \frac{3\phi_q R H_0^2}{2Z \eta} \frac{\partial P_0^2}{\partial R} \right) \\ &+ \frac{\partial}{\partial \theta} \left(\frac{\phi_q H_0^3}{Z \eta} \frac{\partial P_0 P_{zr}}{\partial \theta} + \frac{3\phi_q H_0^2}{2Z \eta} \frac{\partial P_0^2}{\partial \theta} \right) = \Lambda \frac{\partial}{\partial \theta} \left(\frac{P_0 + P_{zr} H_0}{Z} \right) - 2\Lambda \Gamma \frac{P_{zi} H_0}{Z} \\ &\frac{\partial}{\partial R} \left(\frac{\phi_q R H_0^3}{Z \eta} \frac{\partial P_0 P_{\alpha r}}{\partial R} + \frac{3\phi_q R^2 H_0^2 \sin \theta}{2Z \eta} \frac{\partial P_0^2}{\partial R} \right) \\ &+ \frac{\partial}{\partial \theta} \left(\frac{\phi_q H_0^3}{Z \eta} \frac{\partial P_0 P_{\alpha r}}{\partial \theta} + \frac{3\phi_q R H_0^2 \sin \theta}{2Z \eta} \frac{\partial P_0^2}{\partial \theta} \right) = \Lambda \frac{\partial}{\partial \theta} \left(\frac{P_0 R \sin \theta + P_{\alpha r} H_0}{Z} \right) - 2\Lambda \Gamma \frac{P_{\alpha i} H_0}{Z} \\ &\frac{\partial}{\partial R} \left(\frac{\phi_q R H_0^3}{Z \eta} \frac{\partial P_0 P_{\beta r}}{\partial R} - \frac{3\phi_q R^2 H_0^2 \cos \theta}{2Z \eta} \frac{\partial P_0^2}{\partial R} \right) \\ &+ \frac{\partial}{\partial \theta} \left(\frac{\phi_q H_0^3}{Z \eta} \frac{\partial P_0 P_{\beta r}}{\partial \theta} - \frac{3\phi_q R H_0^2 \cos \theta}{2Z \eta} \frac{\partial P_0^2}{\partial \theta} \right) = \Lambda \frac{\partial}{\partial \theta} \left(\frac{P_{\beta r} H_0 - P_0 R \cos \theta}{Z} \right) - 2\Lambda \Gamma \frac{P_{\beta i} H_0}{Z} \end{aligned} \right. \tag{11}
 \end{aligned}$$

The imaginary part of the dimensionless dynamic pressure control equation is:

$$\begin{aligned}
 &\left\{ \begin{aligned} &\frac{\partial}{\partial R} \left(\frac{\phi_q R H_0^3}{Z \eta} \frac{\partial P_0 P_{zi}}{\partial R} \right) + \frac{\partial}{\partial \theta} \left(\frac{\phi_q H_0^3}{Z \eta} \frac{\partial P_0 P_{zi}}{\partial \theta} \right) = \Lambda \frac{\partial}{\partial \theta} \left(\frac{P_{zi} H_0}{Z} \right) + 2\Lambda \Gamma \left(\frac{P_{zr} H_0 + P_0}{Z} \right) \\ &\frac{\partial}{\partial R} \left(\frac{\phi_q R H_0^3}{Z \eta} \frac{\partial P_0 P_{\alpha i}}{\partial R} \right) + \frac{\partial}{\partial \theta} \left(\frac{\phi_q H_0^3}{Z \eta} \frac{\partial P_0 P_{\alpha i}}{\partial \theta} \right) = \Lambda \frac{\partial}{\partial \theta} \left(\frac{P_{\alpha i} H_0}{Z} \right) + 2\Lambda \Gamma \left(\frac{P_{\alpha r} H_0 + P_0 R \sin \theta}{Z} \right) \\ &\frac{\partial}{\partial R} \left(\frac{\phi_q R H_0^3}{Z \eta} \frac{\partial P_0 P_{\beta i}}{\partial R} \right) + \frac{\partial}{\partial \theta} \left(\frac{\phi_q H_0^3}{Z \eta} \frac{\partial P_0 P_{\beta i}}{\partial \theta} \right) = \Lambda \frac{\partial}{\partial \theta} \left(\frac{P_{\beta i} H_0}{Z} \right) + 2\Lambda \Gamma \left(\frac{P_{\beta r} H_0 - P_0 R \cos \theta}{Z} \right) \end{aligned} \right. \tag{12}
 \end{aligned}$$

The opening force of sealing ring end face is obtained:

$$F_o = \int_0^{2\pi} \int_{r_i}^{r_o} p(r) r dr d\theta \quad (13)$$

The closing force of the sealing ring is composed of back pressure acting force on the static ring and spring force:

$$F_c = p_i \pi (r_b^2 - r_i^2) + p_o \pi (r_o^2 - r_b^2) + p_{sp} \pi (r_o^2 - r_i^2) \quad (14)$$

Where $r_b = \sqrt{r_o^2 - B(r_o^2 - r_i^2)}$ is the balance radius, B is the balance ratio and $p_{sp} = 0.03$ MPa is the spring pressure.

2.2.2. Dynamic Mathematical Model

According to the three degree of freedom perturbation model of the dry gas seal system in Figure 1, external disturbances create forces in three directions on the springs and O-rings of the rotating ring, stationary ring, and stationary ring seat. Define the Z-direction as the axial direction of the dry gas seal, and the Z-direction force of the static ring F_z ; The angular deflection moments M_x and M_y relative to the X and Y directions. Based on component stiffness and damping, the stationary ring force balance dynamic mathematical model:

$$\begin{cases} m_s \ddot{z}_s + (c_{zz} + c_{oz}) \dot{z}_s + (k_{zz} + k_{sz}) z_s = k_{zz} z_r + c_{zz} \dot{z}_r \\ I_s \ddot{\alpha}_s + (c_{\alpha\alpha} + c_{oa}) \dot{\alpha}_s + c_{\alpha\beta} \dot{\beta}_s + (k_{\alpha\alpha} + k_{sa}) \alpha_s + k_{\alpha\beta} \beta_s = c_{\alpha\alpha} \dot{\alpha}_r + c_{\alpha\beta} \dot{\beta}_r + k_{\alpha\alpha} \alpha_r + k_{\alpha\beta} \beta_r \\ I_s \ddot{\beta}_s + (c_{\beta\beta} + c_{ob}) \dot{\beta}_s + c_{\beta\alpha} \dot{\alpha}_s + (k_{\beta\beta} + k_{sb}) \beta_s + k_{\beta\alpha} \alpha_s = c_{\beta\beta} \dot{\beta}_r + c_{\beta\alpha} \dot{\alpha}_r + k_{\beta\beta} \beta_r + k_{\beta\alpha} \alpha_r \end{cases} \quad (15)$$

Where m_s and I_s are the mass and moment of inertia of the stationary ring. z_s , α_s , β_s represent the axial displacement (Z-direction) and angular tilt displacements (X and Y directions) of the stationary ring, z_r , α_r , β_r correspond to the axial displacement and angular tilt displacements (X and Y directions) of the rotating ring, respectively. k_{zz} , $k_{\alpha\alpha}$, $k_{\beta\beta}$, $k_{\alpha\beta}$, $k_{\beta\alpha}$ denote the axial and cross-coupled angular tilt stiffnesses (X and Y directions) of the gas film, while c_{zz} , $c_{\alpha\alpha}$, $c_{\beta\beta}$, $c_{\alpha\beta}$, $c_{\beta\alpha}$ represent the axial and cross-coupled angular tilt damping coefficients (X and Y directions) of the gas film. k_{sz} , k_{sa} , k_{sb} are the axial and angular tilt stiffnesses (X and Y directions) of the springs, respectively. c_{oz} , c_{oa} , c_{ob} are the axial and angular tilt damping coefficients (X and Y directions) of the O-rings, respectively. They can be obtained from the calculation formula [16]:

$$k_{sz} = k_s, \quad k_{sa} = k_{sb} = 0.5k_s r_s^2, \quad c_{oz} = c_o, \quad c_{oa} = c_{ob} = 0.5c_o r_b^2 \quad (16)$$

Where k_s and c_o are the stiffness and damping of the spring and O-ring, respectively; r_b is the equilibrium radius of the sealing ring; r_s is the installation radius of the spring, taken as $r_s = 0.5(r_i + r_o)$.

The external excitation of the rotating ring with three degrees of freedom is considered to follow a trigonometric function variation, and the amplitude and phase are equal. The motion form of the rotating ring is:

$$z_r = Z_{rz} \sin(ft), \quad \alpha_r = Z_{rr} \cos(ft), \quad \beta_r = Z_{rr} \sin(ft) \quad (17)$$

Where Z_{rz} , Z_{rr} are the rotating ring axial and angular excitation amplitudes, and f is the disturbance frequency.

Solution Methodology

3.1. Boundary Conditions

Boundary conditions for solving the perturbed pressure control equation:

$$\begin{cases} P_{jr} = P_{ji} = 0 & (j = z, \alpha, \beta) \\ P_0 = P_{in} / P_{re} & (R = 1) \\ P_0 = P_{out} / P_{re} & (R = r_i / r_o) \\ P_0(R, \theta) = P_0(R, \theta + 2\pi/N_g) \end{cases} \quad (18)$$

Initial and boundary conditions for the motion equations:

$$\begin{cases} z_s(0) = \beta_s(0) = \dot{\alpha}_s(0) = 0 \\ \dot{z}_s(0) = Z_{rz} \\ \alpha_s(0) = \dot{\beta}_s(0) = Z_{rr} \end{cases} \quad (19)$$

3.2. Computational Procedure

From the time response of the dry gas seals during the speed-up process, a fitting expression for the equilibrium film thickness as a function of time is derived. Subsequently, the transient pressure control equation of the gas film, accounting for real gas effects, is solved using the small perturbation method. This provides the dynamic characteristic parameters of the gas film (axial and angular dynamic stiffness and damping) at each time instance. These parameters are mathematically fitted, and the resulting expressions are substituted into the three degree of freedom perturbation dynamic model. By solving this model, the axial displacement difference and the relative angular tilt of the stationary ring as functions of time are obtained, illustrating the motion law of the stationary ring tracking ability the rotating ring.

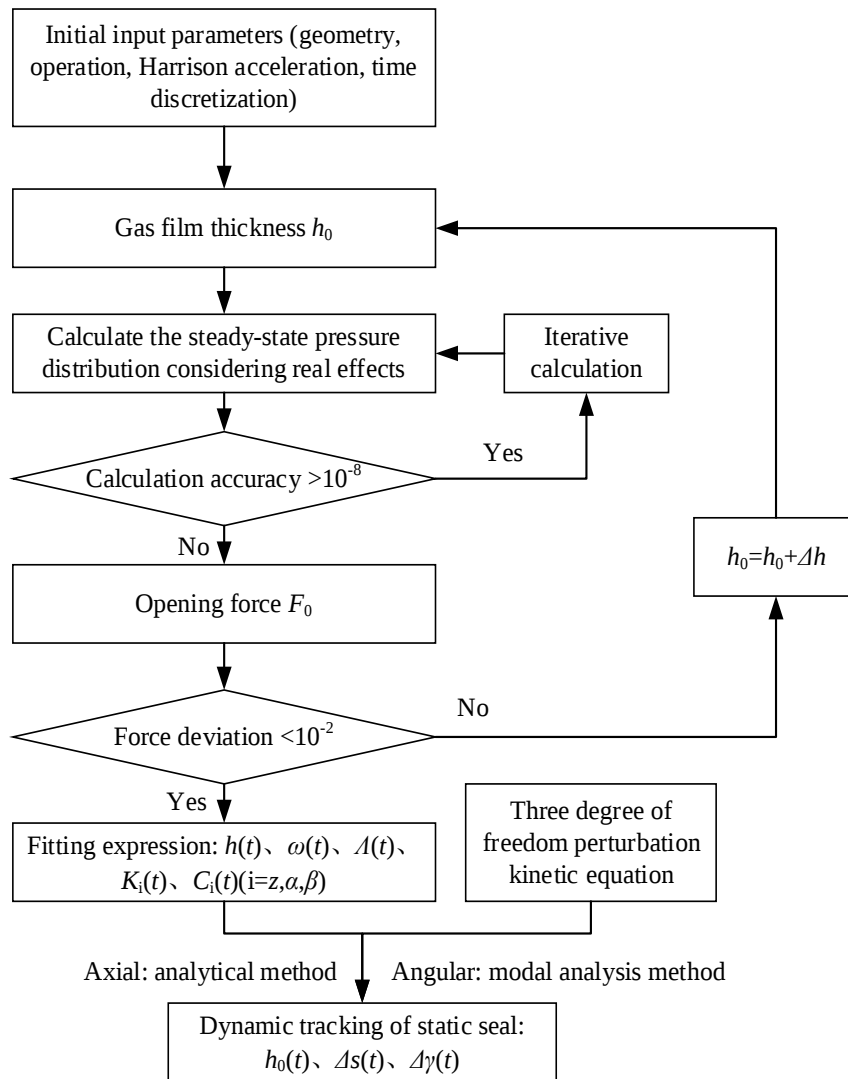


Figure 2. Flow chart for numerical calculation.

The Harrison acceleration method [17], valued for its exceptional start-up and acceleration performance, is widely adopted in dry gas seals. expressed as:

$$\omega(t) = \omega_{\text{end}}(1 - \cos(\pi t / t_a)) / 2 \quad (20)$$

Where ω_{end} is the rotational speed at the working film thickness, t_a is the acceleration time, and t represents any instant during the acceleration process.

Based on the quasi steady-state assumption [15] (where the stationary ring maintains force balance during start-up process, the opening force equals the closing force, $F_o = F_{\text{close}}$). the gas film thickness, angular velocity, and compressibility number at various moments during the dry gas seals start-up process are computed based on the Harrison acceleration method. Fitting expressions for these parameters as functions of time are subsequently established.

$$\begin{bmatrix} h \\ \omega \\ \Lambda \end{bmatrix} = \begin{bmatrix} f_h(t) \\ f_\omega(t) \\ f_\Lambda(t) \end{bmatrix} = \begin{bmatrix} c_{h1} & c_{h2} & c_{h3} & c_{h4} & c_{h5} & c_{h6} \\ c_{\omega1} & c_{\omega2} & c_{\omega3} & c_{\omega4} & c_{\omega5} & c_{\omega6} \\ c_{\Lambda1} & c_{\Lambda2} & c_{\Lambda3} & c_{\Lambda4} & c_{\Lambda5} & c_{\Lambda6} \end{bmatrix} \begin{bmatrix} t^6 \\ t^5 \\ t^4 \\ t^3 \\ t^2 \\ t \end{bmatrix} + \begin{bmatrix} c_{h7} \\ c_{\omega7} \\ c_{\Lambda7} \end{bmatrix} \quad (21)$$

Where the polynomial coefficients $c_{i1}, c_{i2}, c_{i3}, c_{i4}, c_{i5}, c_{i6}, c_{i7}$ ($i=h,\omega,\Lambda$) represent the fitting expression coefficients, as listed in Table 1.

Table 1. Coefficient of fitting expression for various parameters during speed increase.

polynomial coefficients	$i=h$	$i=\omega$	$i=\Lambda$
c_{i1}	-62.26	-42.98	-19.63
c_{i2}	148.64	2.87×10^3	1.31×10^3
c_{i3}	-142.58	-4.6×10^3	-2.1×10^3
c_{i4}	72.2	-2.92×10^3	-1.34×10^3
c_{i5}	-24.31	4.81×10^3	2.2×10^3
c_{i6}	8.31	1.87×10^3	8.54×10^3
c_{i7}	0.9	1.67×10^2	76.1431
SSE	3.79×10^{-5}	1.87×10^{-3}	3.89×10^{-4}
RMSE	1.59×10^{-3}	1.11×10^{-2}	5.1×10^{-3}

3.3. Dynamic Characteristics Coefficients of the Gas Film

The finite difference method is used to solve the discrete pressure control equation. According to literature studies [18], the axial and angular perturbations of the dry gas seals are independent motions. The dynamic characteristic coefficients for the two angular tilts are numerically identical, and the cross-coupled dynamic characteristic coefficients resulting from mutual angular perturbations are equal in magnitude but opposite in direction. The dimensionless dynamic characteristic coefficients of the gas film are derived as follows:

$$\begin{bmatrix} K_{zz} & C_{zz} \\ K_{\alpha\alpha} & C_{\alpha\alpha} \\ K_{\alpha\beta} & C_{\alpha\beta} \end{bmatrix} = -\iint \begin{bmatrix} P_{zr}R & \frac{1}{\Gamma}P_{zi}R \\ P_{\alpha r}R^2\sin\theta & \frac{1}{\Gamma}P_{\alpha i}R^2\sin\theta \\ P_{\beta r}R^2\sin\theta & \frac{1}{\Gamma}P_{\beta i}R^2\sin\theta \end{bmatrix} dRd\theta \tag{22}$$

The Gaussian function is used to fit the relationship between the aerodynamic coefficient and time, expressed as:

$$\begin{cases} K_j = \sum_{m=1}^7 a_{Kj,m} e^{\left(\frac{-(t-b_{Kj,m})}{d_{Kj,m}}\right)^2} \\ C_j = \sum_{m=1}^7 a_{Cj,m} e^{\left(\frac{-(t-b_{Cj,m})}{d_{Cj,m}}\right)^2} \end{cases} \quad (j = zz, \alpha\alpha, \alpha\beta) \tag{23}$$

The coefficients of the fitting expression in the equation are provided in Table 2.

Table 2. Coefficient fitting expression of stiffness and damping.

Coefficients	$i=K, j=zz$	$i=C, j=zz$	$i=K, j=\alpha\alpha$	$i=C, j=\alpha\alpha$	$i=K, j=\alpha\beta$	$i=C, j=\alpha\beta$
a_{ij1}	14.5681	6.83×10^{11}	2.972	48.0122	0.4906	-1.06×10^{14}
a_{ij2}	25.8175	1.54×10^9	9.910^3	1.0582	15.0778	0
a_{ij3}	41.0531	-0.1847	27.2903	0.9252	0.0517	-2.489
a_{ij4}	0	16.0941	0.2264	2.49×10^{14}	0.6853	0
a_{ij5}	0	0	0.6595	-1.5237	1.3948	0
a_{ij6}	0	0	0	0	3.3057	0
a_{ij7}	0	0	0	0	0	0
b_{ij1}	-0.0113	-1.1545	0.0785	-0.1142	-0.0063	-2.7463

b_{ij2}	0.1667	-3.3627	0.1266	0.0381	-0.1518	-0.2855
b_{ij3}	0.8107	0.4562	0.7473	0.0225	0.079	0.7463
b_{ij4}	0	-7.563	0.0342	-117.4525	0.0218	0
b_{ij5}	0	0	0.0426	0.2475	0.0619	0
b_{ij6}	0	0	0	0	0.7102	0
b_{ij7}	0	0	0	0	0.7606	0
d_{ij1}	0.2134	0.2288	0.1426	0.0819	0.0153	0.4856
d_{ij2}	0.3703	0.7747	0.3419	0.0601	0.1158	0.009
d_{ij3}	0.6641	0.1968	0.9848	0.1519	0.0538	0.9844
d_{ij}	0	11.0114	0.0053	21.1259	0.1596	0
d_{ij5}	0	0	0.062	0.4267	0.3808	0
d_{ij6}	0	0	0	0	1.4309	0
d_{ij7}	0	0	0	0	0.0022	0
SSE	0.060871	0.0063894	0.017086	0.0079032	0.00014378	0.041877
RMSE	0.068428	0.025277	0.049405	0.033601	0.011991	0.056757
R-square	9.99×10^{-1}	0.99999	0.99954	0.99996	0.99999	0.97681
Adjusted R-square	0.99875	0.99997	0.99861	0.99987	0.011991	0.96254

3.4. Tracking Parameters of the Stationary Ring

The tracking performance of the stationary ring relative to the rotating ring is commonly characterized by the axial displacement difference between the stationary and rotating rings and the relative angular tilt angle.

The axial displacement difference between the stationary and rotating rings:

$$\Delta S = Z_s - Z_r$$

(24)

The relative angular tilt angle between the stationary and rotating rings:

$$\Delta \gamma = \sqrt{(\alpha_s - \alpha_r)^2 + (\beta_s - \beta_r)^2}$$

(25)

Due to the angular tilt of the stationary and rotating rings, the film thickness at all nodes on the seal ring end face varies at any given time, preventing the direct use of the true gas film thickness value. Moreover, the film thickness increment induced by the tilt is negligible. Thus, the gas film thickness over time is defined as the sum of the sealing gap and the axial displacement difference between the stationary and rotating rings under perturbation, excluding the film thickness change caused by angular tilt.

Results and Discussion

Controlling the dynamic tracking performance of dry gas seals during the start-up acceleration process is considerably more difficult than during stable operation phase. External disturbances affecting the dynamic gas film thickness cause real-time variations in the seal ring tracking performance parameters, frequently leading to deviations from the intended trajectory and motion behavior. This presents substantial challenges to the opening capability and operational stability of dry gas seals.

Table 3. Parameters of dry gas seals.

Parameters	Value	Parameter	Value
Outer Radius r_o /mm	77.78	Spiral Groove Angle $\alpha/^\circ$	15
Inner Radius r_i /mm	58.42	Number of Grooves $N_g/\mu\text{m}$	12
Groove Root Radius r_g /mm	69	Groove Depth $h_g/\mu\text{m}$	5
Balance Radius r_b /mm	61.3	Outlet Pressure p_i /MPa	0.101
Groove-to-Land Ratio γ	1	Inlet Pressure p_o /MPa	4.5852

Moment of Inertia $I_s/\text{kg}\cdot\text{m}^2$	1.33×10^{-4}	Gas Constant $R_r/\text{J}\cdot(\text{mol}\cdot\text{K})^{-1}$	8.3145
Stationary Ring Mass m_s/kg	0.17	Gas Viscosity $\eta/10^{-6}\text{pa}\cdot\text{s}$	20.275
Spring Specific Pressure p_{sp}/MPa	0.03	CO ₂ Molecular Mass $M/\text{g}\cdot\text{mol}^{-1}$	44
Axial Amplitude $a_{rz}/\mu\text{m}$	20	CO ₂ Temperature T/K	400
Angular Axial Amplitude $a_r/\mu\text{rad}$	50	Rotating Ring Poisson's Ratio ν_1	0.14
O-ring Damping $c_o/\text{N}\cdot\text{s}\cdot\text{m}^{-1}$	1000	Stationary Ring Poisson's Ratio ν_2	0.29
Spring Stiffness $k_s/\text{N}\cdot\text{m}^{-1}$	11.419×10^6	Disturbance Frequency f_z	2π

Table 3 presents the parameters of the dry gas seals. The operating conditions are defined as follows: according to reference [15], the opening film thickness at seal ring separation is $0.9\mu\text{m}$, and the stable working film thickness is $3\mu\text{m}$. The Harrison acceleration method is used during start-up process, with the rotating ring perturbation frequency matching the motor angular velocity and a frequency ratio of $\Gamma=1$. The axial and angular motions of the dry gas seals dynamic model are solved independently. The axial motion, governed by a second-order nonlinear homogeneous differential equation, is solved analytically. The angular motion, involving coupled X and Y direction equations, is solved using modal analysis [19]. Figure 3 illustrates the time-dependent variations in the gas film thickness, axial displacement difference, and relative angular tilt angle of the dry gas seals end face after perturbation.

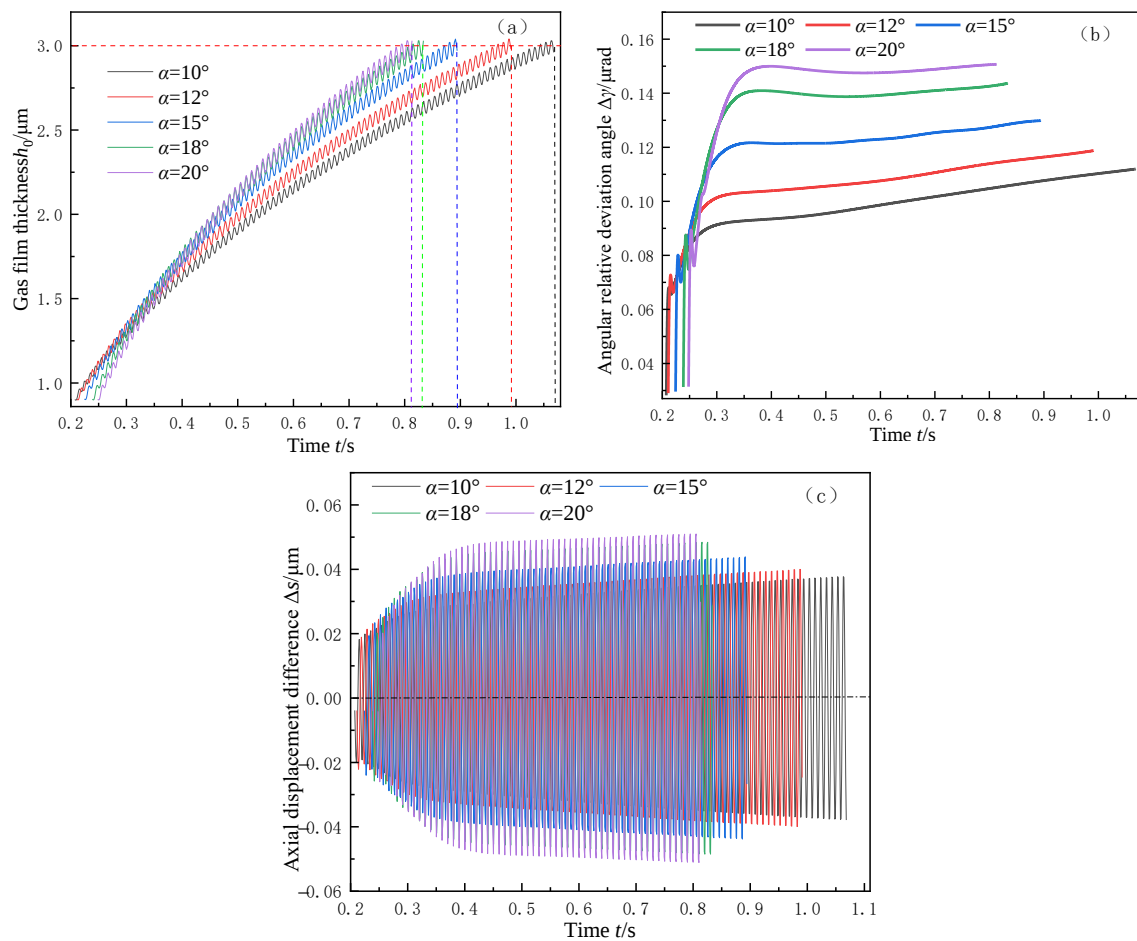


Figure 3. The influence of spiral groove angle on the tracking characteristics.

4.1. Effect of Spiral Angle on Tracking Performance

An optimized seal ring structure enhances the dynamic tracking performance and sealing efficiency of dry gas seals in stable operation [20]. When external disturbances occur during the start-up process, a dynamic analysis of how structural parameters affect the stationary ring tracking performance relative to the rotating ring not only advances the theoretical study of the start-up process but also offers valuable insights for optimizing seal ring design and selection.

Figure 3(a) illustrates the dynamic variation of gas film thickness over time for dry gas seals with different spiral groove angles (ranging from 10° to 20°). As shown in the figure, the gas film thickness increases oscillatory during the process of the initial opening thickness of $0.9\mu\text{m}$ to the working thickness of $3\mu\text{m}$. This oscillatory behavior occurs because, as the rotational speed increases, the gas film thickness grows gradually according to the Harrison speed-up mechanism. When subjected to disturbance excitation from the rotating ring, the micro-disturbance axial displacement difference between the rotating and stationary rings superimposes on the gas film, manifesting the disturbance characteristics in the gas film growth curve. From the gas film fluctuation patterns, it can be observed that the fluctuations are relatively small when the sealing rings initially separate. As time progresses and the gas film thickness increases, the fluctuations become more pronounced. Upon reaching the working film thickness, the fluctuations stabilize, and their amplitudes converge, indicating a steady tracking performance. This demonstrates that the intensity of gas film disturbances increases with film thickness, but once the gas film thickness stabilizes, the disturbance amplitude also tends to stabilize. This phenomenon is attributed to the reduction in both the stiffness and damping of the gas film as its thickness increases, which weakens the gas film ability to resist fluctuations and suppress oscillations.

The time required for dry gas seals with different spiral groove angles to reach the opening film thickness and the working film thickness varies significantly. For instance, in the case of dry gas seals with spiral groove angles of 10° and 20° , the separation times for the seal rings are 0.207 seconds and 0.248 seconds, respectively, while the times required to achieve the working film thickness are 1.068 seconds and 0.813 seconds, respectively. The durations for the gas film thickness to increase from the opening thickness to the working thickness are 0.861 seconds and 0.565 seconds, respectively. These results indicate that a larger spiral groove angle delays the separation of the seal rings but enables faster attainment of the working film thickness, resulting in a shorter total duration for the start-up phrase. This suggests that dry gas seals with larger spiral groove angles can more rapidly achieve the working film thickness after entering the fluid lubrication phase, effectively reducing the risk of contact and wear between the rotating and stationary rings and enhancing operational stability. On the other hand, dry gas seals with smaller spiral groove angles exhibit shorter times before separation, which reduces the risk of contact vibrations during the mixed lubrication phase and minimizes frictional wear and heat accumulation on the seal ring surfaces. Therefore, the influence of the spiral groove angle on the duration of the mixed lubrication phase (characterized by contact vibrations) and the fluid lubrication phase is inversely related: a shorter duration in the former phase corresponds to a longer duration in the latter, and vice versa. This necessitates a comprehensive trade-off by seal designers, considering the relative importance of stability in both lubrication phases under specific operating conditions. The selection of the spiral groove angle should not be based solely on the time required for gas film thickness changes but should instead account for the overall performance requirements of the seal.

Figure 3(b) illustrates the time-dependent dynamic changes in the relative angular tilt angle between the stationary and rotating rings for dry gas seals with varying spiral angles. During the fluid lubrication phase, the relative angular tilt angle increases sharply at the initial stage, experiences oscillatory fluctuations, and then the growth rate slows before eventually stabilizing. A smaller relative angular tilt angle suggests that the stationary ring can rapidly respond to the rotating ring angular tilt. Over time, the relative angular tilt angle, influenced by rotational speed and film thickness, reflects the stationary ring's angular tracking capability. A larger relative angular tilt angle indicates inferior angular tracking performance. When the relative angular tilt angle stabilizes, it signifies that the stationary ring tracking performance has become constant, with diminished angular

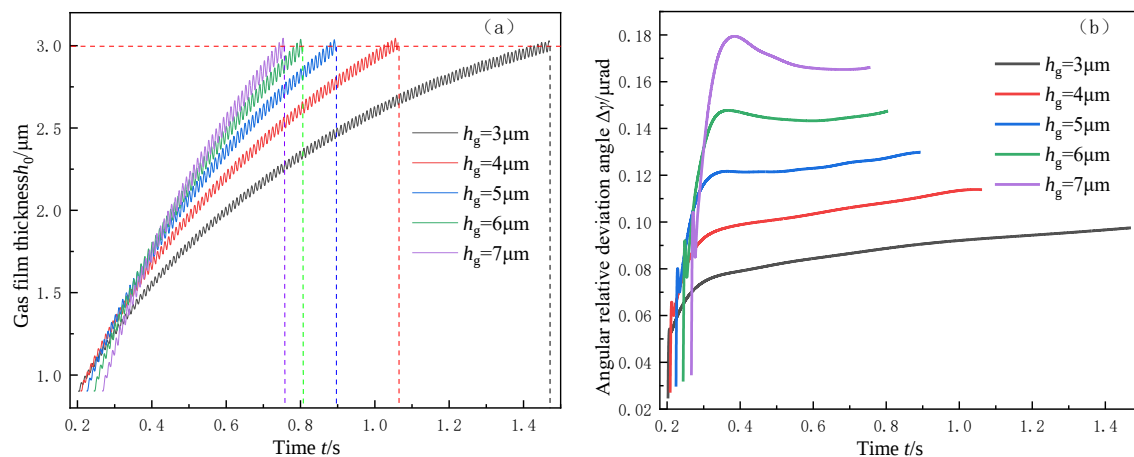
tilt fluctuations and gradual stabilization. Larger spiral angles lead to greater relative angular tilt angles and a delayed start-up time for the stationary ring to track the rotating ring angular tilt.

Figure 3(c) presents the axial displacement difference between the rotating and stationary rings of dry gas seals with varying spiral groove angles during the initial start-up phase. The experimental results demonstrate that following the separation of the rotating and stationary rings and their transition into the fluid lubrication phase, the relative axial displacement difference exhibits an initial rapidly increase and gradually stabilizes. This phenomenon suggests that the axial tracking capability of the stationary ring diminishes with increasing gas film thickness, and it stabilizes rapidly within a short duration. Furthermore, the analysis reveals that larger helix angles negatively impact the axial tracking performance, resulting in both reduced tracking accuracy and delayed initiation of axial tracking response.

4.2. The Influence of Groove Depth on Tracking Characteristics

Figure 4 illustrates the variation of tracking performance parameters over time for dry gas seals with different groove depths (ranging from 3 to 7 μm). As can be observed, the gas film thickness increases with the start-up time, and the deeper the hydrodynamic groove, the longer it takes for the seal rings to separate. However, once separated, the deeper grooves reach the working film thickness faster, resulting in a shorter total duration of the opening phase. As the speed increases, the relative angular deflection between the rotating and stationary rings becomes more pronounced with deeper grooves. This indicates that the angular tracking capability of the stationary ring decreases with increasing groove depth. This phenomenon is attributed to the enhanced hydrostatic pressure effect within the sealing gap caused by larger groove depths, which increases the risk of gas film self-excited instability, ultimately leading to a decline in the stationary ring angular tracking performance.

At the 0.753-second mark, the axial displacement differences between the rotating and stationary rings for dry gas seals with groove depths of 3 μm and 7 μm are 0.0286 μm and 0.0557 μm , respectively. This demonstrates that deeper grooves induce higher amplitude fluctuations in axial displacement difference, and their impact on the stationary ring axial tracking capability follows the same trend as the angular deflection behavior.



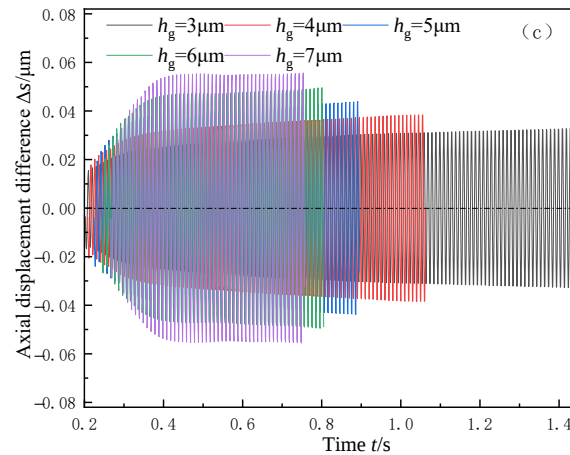
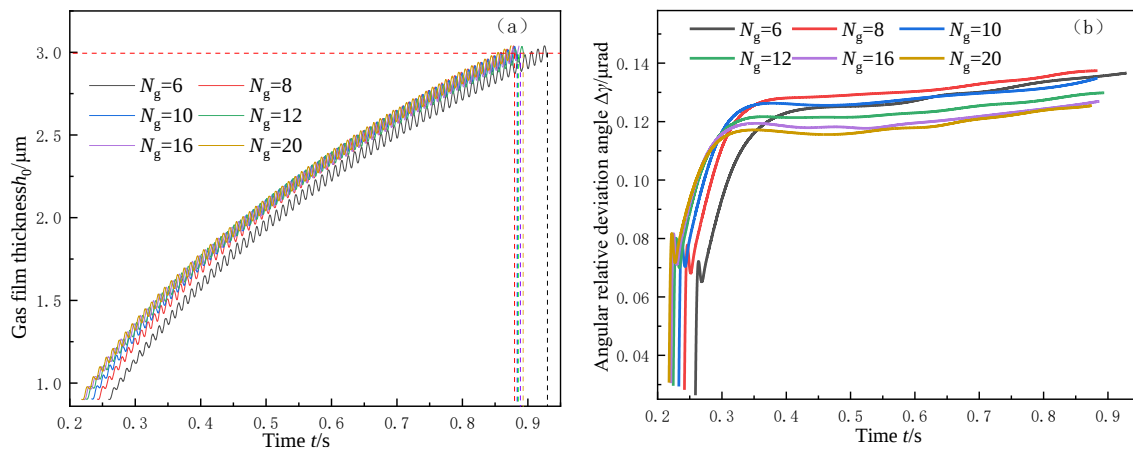


Figure 4. The influence of groove depth on the tracking characteristics.

4.3. The Influence of Groove Number on Tracking Characteristics

The tracking characteristics of dry gas seals with varying groove numbers are presented in Figure 5. As depicted in Figure 5(a), the separation times for seals with 6 and 20 grooves are measured at 0.263 seconds and 0.222 seconds, respectively, while the durations required to achieve the working film thickness are 0.935 seconds and 0.883 seconds, respectively. These results demonstrate that an increase in the number of grooves significantly reduces both the separation time and the duration of the opening phase, while enhancing the opening capability. This phenomenon can be attributed to the increased number of pumping channels for carbon dioxide gas, which improves the pressurization effect of the lubricating gas within the grooves. Consequently, a greater number of high-pressure zones are formed, leading to an elevated gas film force at the end faces. Furthermore, the relative angular deflection between the rotating and stationary rings exhibits a rapid increase with rotational speed before stabilizing at a constant deflection angle. Notably, the relative deflection angles for seals with different groove numbers remain closely aligned, with a maximum deviation of only 7.28% observed during the 0.4~0.87-second start-up interval. This suggests that the number of grooves has a negligible influence on the angular tracking motion of the seal rings. In terms of axial displacement, the difference between the rotating and stationary rings initially increases rapidly with rotational speed before gradually stabilizing. The deviation in axial displacement caused by variations in groove number decreases as the rotational speed increases, declining from 8.77% to 3.69% within the 0.4~0.87-second start-up interval. This indicates that the impact of groove number on the tracking performance of the seal rings diminishes at higher rotational speeds.



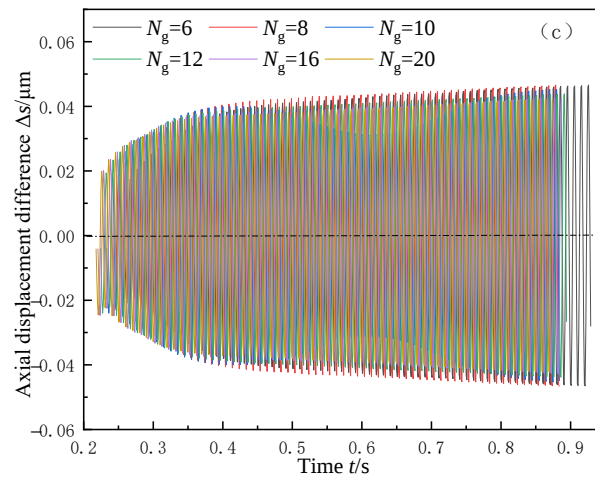
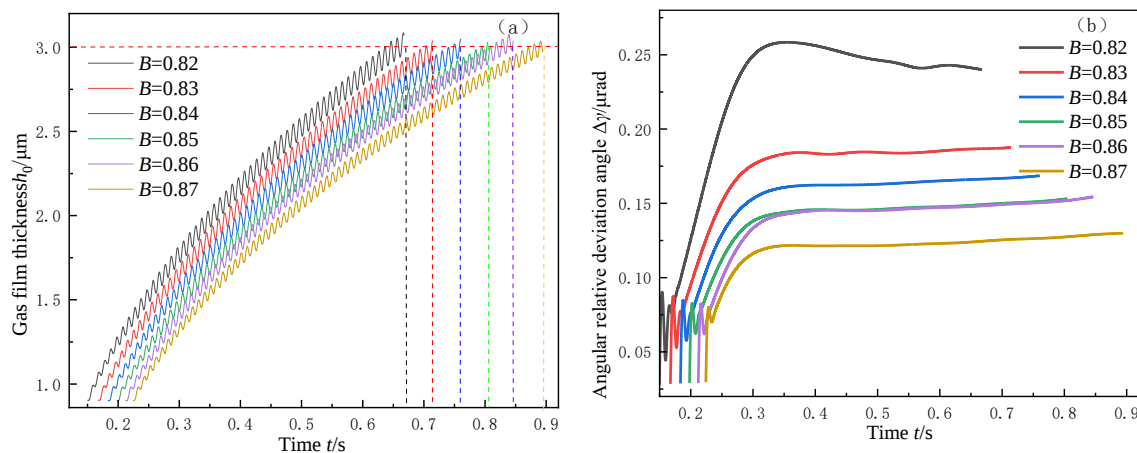


Figure 5. The influence of groove number on the tracking characteristics.

From the observed trends in axial displacement difference and relative angular deflection during the start-up phase, it is evident that the number of grooves has minimal influence on the ability of the stationary ring to track the axial and angular motions of the rotating ring. This is primarily because the number of dynamic pressure grooves only represents the number of channels on the sealing ring end face, which only affects the pressure distribution and opening ability, but cannot change the structure or geometric characteristics of the channels, and has little impact on the motion performance of the sealing ring. As a result, the motion performance of the seal rings is largely unaffected by the number of grooves, rendering its impact on tracking performance relatively insignificant.

4.4. The Influence of Equilibrium Coefficient on Tracking Characteristics

The influence of the equilibrium coefficient on the tracking performance of dry gas seals is illustrated in Figure 6. Observing the variation of gas film thickness with rotational speed, it can be seen that dry gas seals with lower equilibrium coefficient have shorter opening times and stronger opening capabilities. This is because the equilibrium coefficient, which characterizes the balance radius of the seal ring, directly affects the closing force of the seal ring. A lower equilibrium coefficient requires a smaller gas film force to balance the closing force, enabling the seal ring to separate and reach the working film thickness at a lower rotational speed.



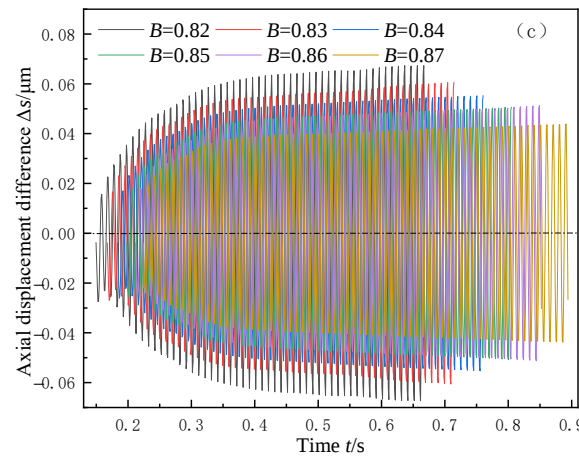


Figure 7. The effect of equilibrium coefficient on the tracking characteristics.

Within the equilibrium coefficient range of 0.82~0.87, both the axial displacement difference and the relative angular deflection between the rotating and stationary rings increase with rotational speed. Notably, seals with a lower equilibrium coefficient exhibit significantly larger relative angular deflections, with the maximum relative deflection deviation approximately doubling. This indicates that the equilibrium coefficient has a pronounced influence on the angular tracking characteristics of the seal rings, and a lower equilibrium coefficient is detrimental to the angular tracking motion. When the working film thickness is achieved, the axial displacement difference between the rotating and stationary rings is greater for seals with a lower equilibrium coefficient. The maximum deviation in axial displacement difference between seals with different equilibrium coefficient reaches up to 37%. This demonstrates that the equilibrium coefficient significantly affects the ability of the seal rings to track both axial and angular motions. This behavior can be attributed to the influence of the equilibrium coefficient on the closing force of the seal ring. A lower equilibrium coefficient reduces the closing force, resulting in larger fluctuations in the stationary ring's ability to track the axial and angular motions of the rotating ring, thereby weakening the overall dynamic response of the seal rings.

The study on the axial and angular tracking performance of stationary rings in spiral groove dry gas seals reveals that different structural parameters exhibit varying degrees and patterns of influence on tracking performance. Therefore, a reasonable seal ring selection strategy must take into account the operating conditions and comprehensively consider factors such as the seal ring opening capability, gas film stability, and the tracking performance of the stationary ring.

4. Conclusions

When the rotating and stationary rings initially separate, the stationary ring exhibits good tracking performance for both the axial and angular motions of the rotating ring. However, the tracking capability undergoes significant variations during this phase. As time progresses and the gas film thickness increases, the tracking ability gradually weakens. By the time the working film thickness is achieved, the tracking performance parameters stabilize and remain constant.

Micro-disturbances influence the gas film thickness and exhibits an oscillatory increase during the start-up acceleration process. Furthermore, the amplitude of these oscillations becomes more pronounced as the gas film thickness increases.

The influence of structural parameters on the tracking characteristics of the stationary ring can be summarized as follows: Larger spiral groove angles and deeper grooves delay the separation of the rotating and stationary rings, reduce the total duration of the start-up phase, and increase the relative angular deflection and axial displacement difference, collectively resulting in degraded tracking performance. A higher number of grooves shortens the opening time and enhances the opening capability of the dry gas seals, although it has a relatively minor influence on the axial and

angular tracking performance of the stationary ring. Conversely, a higher equilibrium coefficient increases the opening time and reduces the opening capability but improves both the axial and angular tracking performance of the stationary ring.

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