Influences of Floating-ring Seal Parameters on Clearance Leakage

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Abstract. For a type of floating-ring seal widely used in the fields of aeronautics and astronautics, power and energy, and petrochemical engineering, the influences of various parameters on clearance leakage behavior of floating-ring seal are analyzed using numerical method and by means of MATLAB programming. Based on Bulk-flow model, the governing equations of the flow field in the clearance of floating-ring seal are established. The finite difference method and the staggered grid are used to discretize the governing equations. The SIMPLE algorithm is used to solve the discretized equations. The self-locking position of the floating-ring is calculated by analyzing the equilibrium condition of the floating ring, and accordingly the leakage flowrate of the flow field in the clearance of the floating-ring seal are obtained. Finally, the influences of fluid viscosity, inlet swirl, width of floating ring, and relative roughness of floating-ring and rotor surfaces on the leakage flowrate are discussed.

Keywords: Floating-ring seal; Leakage flowrate; Bulk-flow model; Seal parameter.

1. Introduction
Floating-ring seal is widely used in the pumps of petrochemical engineering, energy and power engineering and aeronautic and aerospace engineering systems. It is a type of semi-contacting seal with simple structure. The ring is able to move in radial direction and locks its position automatically by the dynamic fluid force produced by the annular flow, so that the impacting and rubbing are avoided to make the operation of pumps safer, more stable and reliable. Therefore, researches on the sealing performances of floating-ring seals have attracted much attention from researchers worldwide in this field.

Kirk in 1979 proposed for the first time a floating-ring seal model and used it to analyze the effects of the seal to the dynamic properties of a rotor system[1]. Ha et al studied the leakage behaviors and the rotordynamics of a high pressure floating-ring seal used in the turbopumps of a liquid propellant rocket engine[2]. Fei Wang et al used the Finite Difference Method (FDM) combining with the Successive Over Relaxation (SOR) scheme to solve the Reynolds equation, obtained the pressure distribution of the annular flow in a shallow slot split floating-ring seal and discussed the influences of the clearance on the leakage flowrate[3]. Wenbo Duan et al improved the Ha’s method with the FDM substituting for the FFT, obtained the leakage flowrate and the dynamic coefficients and analyzed the influences of floating-ring parameters on the performance of the seal[4]. Chang-Ho Choi et al used an experiment and calculation combined method to study the influences of the floating-ring clearance on the dynamic properties of a turbopump[5]. Guoqing Li et al carried out a rotary test in the cold and hot states respectively to investigate the leakage behaviors of the floating-ring seal of an airplane engine and compared the leakage behaviors of the floating-ring seal and a labyrinth
seal[6]. Zhi Chen et al employed the FLUENT to a laminar flow model to simulate the annular flows in the seal under different parameters[7]. Rui Xu et al put forward a transient CFD approach for the prediction of the dynamic properties of a liquid annular seal and verified its reliability with experiments[8]. Using the FLUENT Lijun Ma et al calculated and analyzed the lifting force, eccentricity and leakage flow rate under different conditions[9]. Zhansheng Liu et al utilized the Finite Element Method (FEM) established the fluid flow model and calculated the dynamic property coefficients for a floating-ring seal using the perturbation method. Experiments were carried out to verify the validity of calculated results[10].

Based on the Bulk-flow model, a set of governing equations of the fluid flow in the floating-ring seal are established in the present paper. The SIMPLE algorithm is used to solve the equations. The locked eccentricity, leakage flow rate and the pressure angle are obtained. Accordingly, the influences are analyzed of fluid viscosity, inlet swirl, floating-ring width and the relative roughness of stator and rotor on the leakage flow rate. The conclusions are expected to be helpful to the improvements of the sealing performance of the floating-ring seal.

2. Motion Mechanism and Fluid Governing Equations

The schematic of the floating-ring seal is illustrated in Fig. 1. The inserted nut and the supporting ring serve to limit the axial motion of the floating-ring. When the pump does not work, the ring freely suspends on the rotating shaft with its inner surface contacting with the top surface of the shaft due to the gravity and therefore has the maximum eccentricity. Once the pump starts working, the ring begins to move from high pressure area to low pressure area under the pressure difference between the two ends and contacts with the supporting ring in the end. At the same time, the floating-ring moves toward the rotating center due to the lifting dynamic fluid force produced by the annular flow and yields frictional force between the two contacting surfaces of the floating-ring and the supporting ring, which resists the movement of the floating-ring. The eccentricity of the floating-ring continuously decreases and in turn the dynamic fluid force reduces with its movement toward the rotating center. When the dynamic fluid force and the frictional force are in a balance, the floating-ring automatically locks itself at a particular position.

Figure 1. Structure unit of floating-ring seal.

Here the Bulk-flow model is used to represent the actual flow so as to reduce the calculation. It is assumed that the fluid velocity and pressure are constant along the thickness of fluid film and that shearing force only exists on the contacting surfaces of the fluid and structure. The relationship between the shearing force and the fluid velocity is determined by experiments. By virtue of the very small annular clearance, the Bulk-flow model is reasonable in this case.

Basic governing equations, including the continuity equation and the momentum equations are listed as follows[11]:

Figure 2. Fluid control volume.
\[
\frac{\partial}{\partial t} \int_{C_1} \rho dV + \int_{C_2} \rho U_d \cdot dA = 0, \quad \frac{\partial}{\partial t} \int_{C_3} \rho U_\theta \cdot dA + \int_{C_3} \rho U_v \cdot dA = \sum F_i, \quad \frac{\partial}{\partial t} \int_{C_4} \rho U_\phi \cdot dA + \int_{C_5} \rho U_o \cdot dA = \sum F_0
\]  
(1)

The Moody friction factors are used in Eq. (1), having regard to control volume illustrated in Fig. 2, to get the following equations.

\[
\frac{\partial H}{\partial t} + \frac{1}{R} \left( \frac{\partial (HU_\theta)}{\partial \theta} + \frac{\partial (HU_v)}{\partial Z} \right) = 0, \quad -\frac{H \partial P}{\rho \partial \theta} = \frac{1}{2} U_z U_z f_S + \frac{1}{2} U_z U_z f_R + H \left( \frac{\partial U_\theta}{\partial t} + \frac{U_\theta U_v}{R} + \frac{\partial U_v}{\partial \theta} \right),
\]

\[
-\frac{H \rho R \partial P}{\partial \theta} = \frac{1}{2} U_z U_z f_S + \frac{1}{2} (U_\theta - R\omega) U_z f_R + H \left( \frac{\partial U_\theta}{\partial t} + \frac{U_\theta U_v}{R} + \frac{\partial U_v}{\partial \theta} \right)
\]

(2)

where, \(H\) is the thickness of the fluid film, \(P\) is the pressure of the fluid, \(R\) is the radius of the rotor, \(R_0\) is the inner radius of the floating-ring, \(U_z\) is the axial velocity of the fluid, \(U_\theta\) is the circumferential velocity of the fluid, \(U_S\) is the relative Bulk-flow velocity of the fluid to the floating-ring, taken as \(\xi_s / 2\), \(U_r\) is the relative Bulk-flow velocity of the fluid to the rotor, taken as \(\sqrt{U_z^2 + (U_\theta - R\omega)^2}\), \(\omega\) is the angular rotating velocity of the rotor, \(\rho\) is the density of fluid, \(a_1 = 1.375 \times 10^{-3}\), \(b_2 = 2 \times 10^4\), \(b_3 = 10^6\).

The zeroth governing equations are obtained through perturbation analysis and are nondimensionalized as

\[
\frac{b}{L} \frac{\partial}{\partial \theta} \left( h_0 u_{\theta 0} \right) + \frac{\partial}{\partial z} \left( h_0 u_{z 0} \right) = 0, \quad -h_0 \frac{\partial P}{\partial z} = \frac{L}{2C} u_{z 0} u_{s 0} f_S + \frac{L}{2C} u_{z 0} u_{r 0} f_R + h_0 \left( bL \frac{u_{\theta 0}}{R} \frac{\partial u_{z 0}}{\partial \theta} + u_{z 0} \frac{\partial u_{z 0}}{\partial z} \right),
\]

\[
-h_0 \frac{\partial P}{\partial \theta} = \frac{L}{2C} u_{\theta 0} u_{s 0} f_S + \frac{L}{2C} (u_{\theta 0} - 1) u_{r 0} f_R + h_0 \left( bL \frac{u_{\theta 0}}{R} \frac{\partial u_{\theta 0}}{\partial \theta} + u_{z 0} \frac{\partial u_{\theta 0}}{\partial z} \right)
\]

(3)

where \(z = Z / L, h_0 = H / \bar{C}, b = R\omega / \bar{V}, u_{z 0} = U_z / \bar{V}, u_{\theta 0} = U_\theta / R\omega, p_0 = P / \bar{V}^2, \bar{V} = Q / 2\pi r \bar{C}, f_S = a_1 \left[ 1 + \left( \frac{b_2 \xi_s}{h_0} + \frac{a_3}{h_0 u_s} \right) \right]^{1/3}, f_R = a_1 \left[ 1 + \left( \frac{b_2 \xi_r}{h_0} + \frac{a_3}{h_0 u_r} \right) \right]^{1/3}, \xi_s = \frac{b_3 \mu}{2\bar{V} \bar{C}}.

L is the width of the floating-ring, \(\bar{C}\) is the average clearance, \(e\) is the eccentricity of the floating-ring, \(e_s\) is the relative roughness of the floating-ring surface, taken as \(\xi_s / 2\bar{C}\), \(e_r\) is the relative roughness of the rotor surface, taken as \(\xi_r / 2\bar{C}\), \(\bar{V}\) is the axial average velocity of the fluid, \(\xi_s\) is the relative roughness of the floating-ring surface, \(\xi_r\) is the relative roughness of the rotor surface, \(\mu\) is the viscosity of the fluid.

\[
\text{Figure. 3. Schematic of relations.}
\]
According to the geometrical relations indicated in Fig. 3, the fluid film thickness in the annular clearance can be defined as
\[
H = \sqrt{(R + C)^2 - e^2 \sin^2 \theta - e \cos \theta - R}
\] (4)

The boundary conditions are described as follows:
Inlet pressure
\[
p_0(0, \theta) = p_{in} - 0.5(1 + \zeta_{in})u_{s0}^2(0, \theta)
\] (5)

Outlet pressure
\[
p_0(1, \theta) = p_{ex} - 0.5(1 - \zeta_{ex})u_{s0}^2(1, \theta)
\] (6)

Inlet circumferential velocity
\[
u_{\theta 0}(0, \theta) = u_{sw}
\]
where, \( \zeta_{in} \) is the inlet pressure drop coefficient, taken as 0.5, \( \zeta_{ex} \) is the outlet pressure recover coefficient, taken as 0, \( p_{in} \) is the inlet pressure, \( p_{ex} \) is the outlet pressure, \( u_{sw} \) is the inlet swirl coefficient, taken as 0.25.

3. Analysis for the Self-locking Conditions of the Floating-ring

\[\text{Figure. 4. Forces analysis of floating-ring.}\]

The forces acting on the floating-ring are illustrated in Fig. 4. It can be known from the forces analysis that the locked floating-ring position depends on the balance of gravitational force, frictional force and fluid dynamic force. The gravitational force can be neglected because of the mass of the floating-ring being very small and thus resulting in a very small gravitational force compared with the other two forces. The frictional force can be determined by the pressure difference between the two sides of the floating-ring, the contacting area of the floating-ring and the supporting ring and the friction factor between the two contacting surfaces. The dynamic fluid forces can be defined by
\[
F_{c,x} = \int_0^{2\pi} \int_0^x p_0 \rho V^2 \cos \theta R d\theta dz, \quad F_{c,y} = \int_0^{2\pi} \int_0^x p_0 \rho V^2 \sin \theta R d\theta dz, \quad F_c = \sqrt{F_{c,x}^2 + F_{c,y}^2}
\] (7)

Evidently, the dynamic fluid forces are related to the steady pressure distribution of the annular flow. In turn, the steady pressure distribution depends on the locked floating-ring eccentricity that can be determined through iterations. To start with, an initial value of the eccentricity has to be supposed. The dynamic fluid forces can be obtained accordingly. Equilibrium conditions for the floating-ring are established to terminate the iterations. MATLAB codes are made to accomplish the iteration process. The formulas needed for the iterations are listed as follows.

The convergence function is defined by
\[
I = |F_\mu - F_c|
\] (8)

According to steepest descent algorithm[6], the iteration formula is defined by
Termination condition is given by

\[ l \leq 0.003 F'_{c} \]

4. Numerical Calculation and Results Analysis

SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm is widely used for numerical calculations in the field of fluid mechanics. The procedure starts with a guessed initial pressure distribution. In a sequence, a predicted velocity distribution is obtained by solving the momentum equations. The pressure correction equation is developed from the continuum equation accordingly so as to get a pressure correction value. The pressure and velocity distributions are updated with the pressure correction value. If the updated values do not satisfy the continuum equation, then the updated pressure values become the new guessed pressure distribution. The above steps are repeated until a convergent solution is achieved which satisfies the continuum equation.

In the present paper, the FDM with staggered grids is used to discrete the governing equations with the consideration of the needs of using the SIMPLE algorithm and eliminating the effects of nonlinearity and coupling. After the solutions of the governing equations are obtained, the pressure solution is taken into Eq. (7) to get the fluid dynamic forces. The velocity solution is taken into Eq. (10) to get the leakage flowrate. Upon all the solutions being obtained, the influences of fluid density, fluid viscosity, swirl coefficient of inlet velocity, width and thickness of floating ring, and surface relative roughness of floating-ring and rotor on the leakage flowrate are discussed.

\[ Q = \int_{0}^{2\pi} h_{0,0} \cdot C_{u}(0,0) \cdot \bar{T}_{d} \theta \]  

Parameters for the calculations: radius of the rotating shaft is 0.0265 m; radius of the supporting ring is 0.0275 m; thickness of the floating-ring is 4.5×10^{-3} m; length of the floating-ring is 8×10^{-3} m; average film thickness is 1×10^{-4} m; density of the fluid is 1110 kg/m^{3}; viscosity of the fluid is 1.86×10^{-4} N·s/m^{2}; inlet pressure is 6.83 MPa; outlet pressure is 0.42 MPa; rotating speed is 50000 rpm; relative roughness of the surfaces of the floating-ring and the rotor is 1×10^{-5}; friction factor between the contacting surfaces of the floating-ring and the supporting ring is 0.1. Results are displayed in the figures from Fig. 5 to Fig. 8.

**Influences of the Viscosity.** The viscosities are taken as 1×10^{-4} N·s/m^{2}, 2×10^{-4} N·s/m^{2}, 3×10^{-4} N·s/m^{2}, 4×10^{-4} N·s/m^{2}, 5×10^{-4} N·s/m^{2}. The influences of the viscosity on the leakage flowrate are illustrated in Fig. 5. It is can be seen that the leakage flowrate decreases in a lightly linear manner with the increase of the viscosity. The fluid viscosity is a critical factor that contributes to the inner frictional force produced by the relative motions between the fluid masses. The larger the viscosity, the larger the frictional force, resulting in a larger resistance to the leakage and thus a smaller leakage flowrate.

**Influences of the Inlet Swirl.** The inlet velocity swirl coefficients are taken as 0, 0.5, 1, 1.5, 2. The influences of the inlet swirl on the leakage flowrate are illustrated in fig. 6. It can be seen that the
leakage flowrate decreases with the increase of the swirl coefficient and that the effects become stronger as the swirl coefficient increases. The swirl coefficient represents the magnitude of the circumferential component of the flow velocity. The circumferential velocity undermines the axial velocity so that it makes the leakage flowrate decrease.

**Influences of the Floating-ring Width.** The floating-ring widths are taken as 7×10^{-3} m, 8×10^{-3} m, 9×10^{-3} m, 10×10^{-3} m, 11×10^{-3} m. The influences of the floating-ring width on the leakage flowrate are illustrated in Fig. 7. It can be seen that the leakage flowrate decreases in a lightly linear manner with the increase of the floating-ring width. Evidently, the larger the floating-ring width, the larger the resistance from the flow passage. This justifies the calculated results.

![Figure 7. Influence of L on leakage flowrate.](image)

![Figure 8. Influence of es and er on leakage flowrate.](image)

**Influences of the Relative Roughness of the Floating-ring and Rotor Surfaces.** The relative roughnesses of the floating-ring and rotor surfaces are taken as 2×10^{-4}, 4×10^{-4}, 6×10^{-4}, 8×10^{-4}, 10×10^{-4}. The influences of the relative roughness of the floating-ring and rotor surfaces on the leakage flowrate are illustrated in Fig. 8. It can be seen that the leakage flowrate decreases in a lightly linear manner with the increase of the relative roughness. Obviously, the larger the roughness, the larger the resistance to the fluid flow, leading to the decrease of the leakage flowrate.

**5. Conclusions**

Based on the Bulk-flow model, numerical simulations are carried out for a type of floating-ring seal with MATLAB. The influences of fluid viscosity, inlet swirl, width of floating-ring and surface relative roughness of floating-ring and rotor on the leakage flowrate are analyzed with the following conclusions, which not only contribute to sealing theory but also provide guide for improving the sealing performance of this type of floating-ring seal.

1. The leakage flowrate decreases with the increase of the viscosity, inlet swirl, floating-ring width and the roughness of the floating-ring and rotor surfaces by the path of changing the annular flow.
2. Among all the four parameters, the influences become strong as the inlet swirl intensifies.
3. The sealing performance of the floating-ring seal could be improved by means of increasing the roughness of the floating-ring and rotor surfaces or adopting special surface structures, increasing the floating-ring width, and installing inlet swirl devices.

**Acknowledgement**

This research was financially supported by the National Science Foundation of China (11172270).

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