Analysis on Leakage Characteristics of Labyrinth Piston Clearance in Reciprocating Labyrinth Compressor

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Abstract. In this paper, the piston clearance leakage model in reciprocating labyrinth compressor is established, and the leakage characteristics of labyrinth piston are analysed. The results show that in the labyrinth entrance section, the gas velocity decreases greatly, and the throttling effect is the most obvious. In the middle section of the labyrinth, the flow velocity descending gradient decreases. In the exit section, the flow rate begins to increase. When the labyrinth clearance is less than or equal to 0.1 mm, the clearance changes has little effect on the leakage. When the clearance is greater than 0.1 mm, the leakage increases rapidly with the clearance increasing. When the piston operates eccentrically, the leakage will increase by 1.5 ~ 2 times compared with the non-eccentric operation. Therefore, the eccentric operation of the piston should be avoided as much as possible and the clearance should be reduced.

Keywords. Labyrinth compressor, labyrinth piston, leakage, eccentric operation.

1. Introduction
Large reciprocating labyrinth compressor is widely used in petrochemical, pharmaceutical, military, bioengineering and other fields. Its structure diagram is shown in figure 1. In order to keep the gas clean, labyrinth seal without lubricating oil is usually used between the piston and the cylinder. The high-pressure gas is sealed by the labyrinth groove arranged on the piston. In order to avoid collision between the piston and the cylinder, sufficient radial clearance must be set. However, if the radial clearance is too large, it will inevitably increase the gas leakage.

Figure 1. Reciprocating labyrinth compressor structure.
The classical models for labyrinth leakage calculation include Martine, Kearton, Stodala, Egli, Vermes, etc. These algorithms calculate the leakage by modifying the "leakage coefficient". With the rapid development of computer, the calculation method of labyrinth flow field has been changed to CFD simulation. Wei [1] and Lin [2] simulated and studied the flow field of staggered labyrinth seal and straight labyrinth seal respectively. Zhang [3] used dynamic fluent grid technology to control the piston movement, and realized the real-time observation of the labyrinth flow field. Li [4] used the finite element method to analyse the coupling of flow field, temperature field and pressure field in the cylinder of low-temperature BOG reciprocating labyrinth seal compressor. Wang’s research team [5, 6] took the staggered labyrinth structure in labyrinth compressor as the research object, and established the fluid structure coupling model of labyrinth leakage flow and piston reciprocating motion. They proposed that the interaction between flow and solid structure has greatly influence on the flow field, and the results obtained from fluid structure coupling model is more close to the actual. At present, the research on labyrinth seal has involved thermal fluid structure coupling [7], tooth deformation [8], rotor dynamics characteristics [9] and two-phase flow leakage characteristics [10], and so on. The above researches enrich the flow theory in labyrinth seal and have a strong reference value for this paper. However, the influence of piston radial eccentric motion on labyrinth seal performance in reciprocating labyrinth compressor is not considered in the above research.

In this paper, a two-dimensional model of the labyrinth channel is established, and then the flow field is simulated to obtain the leakage under different inlet and outlet pressure ratios. Finally, the leakage of the piston under eccentric motion state is calculated and compared with that under non-eccentric motion state.

2. Methodology

2.1. Geometric Parameters of Labyrinth Seal
Taking a 4K-300MG reciprocating labyrinth compressor as an example, the compression medium is propyne, the flow rate is 4080 m$^3$/h, the rotating speed is 490 r·min$^{-1}$, and there are four rows of pistons in total. This paper takes the labyrinth piston of the first row as the research object. The suction pressure of the first row is 0.06 MPa, the exhaust pressure is 0.54 MPa, and the piston stroke is 250 mm. The main geometric parameters of the labyrinth seal ring groove are shown in figure 2. A total of 90 labyrinth seal ring grooves are distributed along the axial direction of the piston, the sealing section length is 180 mm.

![Figure 2. Geometric parameters of the labyrinth seal.](image)

2.2. Meshing Division
The axial flow velocity of the leakage gas in the labyrinth channel is much greater than the circumferential velocity along the labyrinth ring groove. Pervious researches show that the difference between the calculation model considering gas circumferential flow and that ignoring circumferential flow is less than 3% [1, 2, 11]. It shows that the influence of circumferential flow on gas leakage can be negligible and does not affect the accuracy of calculation results.

ICEM software is used to mesh the two-dimensional labyrinth channel, as shown in figure 3. In order to analyse the influence of piston eccentric motion on leakage, the dynamic grid technology will be used in Fluent software. The piston eccentric motion trajectory in literature [11] will be used as the
boundary condition in this paper. The clearance between cylinder wall and piston adopts 2d-4quad structural grid to ensure that the piston boundary always has good grid quality under eccentric movement state.

![Meshing division](image)

**Figure 3.** Meshing division.

### 2.3. Governing Equations and Software Settings

When the leakage gas flows through the labyrinth channel, it needs to meet the mass conservation equation, momentum conservation equation and energy conservation equation. In addition, because the labyrinth turbulent flow field is a complex nonlinear system, different vortices will occur in each labyrinth chamber, and a reasonable turbulence model needs to be selected. The RNG $k-\varepsilon$ turbulence model takes into account both turbulent vortex and viscous force in the medium, which is adopted in this paper.

During the movement of the piston, the pressure at the inlet of the labyrinth seal, that is, the gas pressure in the cylinder, changes periodically with time, and its calculation method can refer to reference [12]. In this paper, the inlet pressure of the sealing uses the Profile function to define. Its variation curve is shown in figure 4. The outlet pressure of the labyrinth seal is 0.06 MPa.

![Inlet pressure of the labyrinth](image)

**Figure 4.** The inlet pressure of the labyrinth.

### 2.4. Validation of the Simulation Model

In this paper, the Vermes calculation method is used to validate the simulation model. The solution programs of Vermes are written in Mathcad software. This paper set the pressure ratio as 2 and the inlet pressure as 0.16 MPa to calculate the leakage of the labyrinth seal, and the results are shown in table 1. It can be seen from the table that the relative error between the calculation results and the simulation results is less than 7%, which shows that the turbulence model selection and boundary condition setting in the simulation model are reasonable and the calculation results are reliable.
Table 1. Comparison of leakage calculation results.

| Clearance /mm | Vermes calculation method /kg·s⁻¹ | Simulation model /kg·s⁻¹ |
|---------------|----------------------------------|--------------------------|
| 0.1           | 0.0046                           | 0.0048                   |
| 0.2           | 0.0119                           | 0.0112                   |
| 0.3           | 0.0227                           | 0.0216                   |

3. Results and Discussion

In order to analyse the flow pattern and energy loss of leakage gas, the flow field of leakage channel without eccentric operation of piston is simulated. Figure 5 is the dynamic pressure of the leakage gas and figure 6 is the velocity vector. For the convenience of observation, the whole flow field is divided into three stages. And the inlet section, middle section and outlet section are locally enlarged respectively. The two figures demonstrate that the flow pattern of leakage gas in the labyrinth channel is mainly the vortex flow at grooves and the jet flow at the top of the throttling tooth.

When the leakage gas enters the top of the first throttling tooth, due to the contraction of the flow beam, the gas pressure is converted into kinetic energy to rush through the throttling tooth at high speed. However, after passing through the throttling teeth, the gas begins to split, and part of the gas enters the labyrinth groove. Due to the sudden expansion of the volume and the obstruction of the windward surface, the flow velocity decreases rapidly, and a strong vortex is formed in the groove, resulting in the dissipation of energy. As the subsequent leakage gas continues to enter the labyrinth chamber, the pressure in the chamber increases and the energy is converted again. Under the action of differential pressure, it continues to flow to the next labyrinth chamber, and then forms a high-speed jet flow at the top of the throttling tooth where near the cylinder wall and rush to the next labyrinth chamber. Part of the gas continues to flow downward along the cylinder wall and does not enter the labyrinth chamber.

The flow of leakage gas in the labyrinth channel is the continuous repetition of the above process. In the entrance section, because the pressure in the cylinder is greater than that in the labyrinth channel, the gas continuously enters the labyrinth channel. Due to the labyrinth throttling effect, the speed of the leakage gas in the entrance section decreases step by step and the pressure decreases. In the middle section, when the labyrinth channel is filled with leakage gas, its flow rate and dynamic pressure drop to a certain extent, and the descending gradient decreases In the exit section, because the gas pressure in the labyrinth channel is greater than the outlet pressure, the pressure difference increases, and the gas flow rate in the labyrinth channel begins to increase again until it flows out of the labyrinth outlet. The static pressure of leakage gas in the whole labyrinth channel continuously decreases, as shown in figure 7.
Figure 6. Velocity vector of the leakage gas.

Figure 7. Static pressure of the leakage gas.

Figure 8 shows the turbulent kinetic energy distribution of leakage gas. It demonstrates that the turbulent kinetic energy changes strongly in the entrance section and exit section, and the energy dissipation is significant at both sections. In each labyrinth chamber, the maximum turbulent kinetic energy appears at the throttling tooth tip on the windward side, which is not only the diversion area of the leakage gas, but also the area where the leakage gas begins to shrink, boost, accelerate and generate jet flow, finally resulting in severe energy dissipation.

Figure 8. Turbulent kinetic energy distribution of leakage gas.

In order to obtain the influence of inlet pressure on leakage, this paper calculate the leakage when the outlet pressure is 0.16 MPa and the pressure ratio $\gamma$ is 2, 3 and 4 respectively. When the piston does not run eccentrically in the cylinder, the variation of leakage with labyrinth clearance width is shown in figure 9. The figure shows that with the clearance width increasing, the labyrinth throttling effect decreases and the leakage increases. However, when the clearance width is less than 0.1 mm (i.e. less than 1/3 of the design width), the leakage changes slowly with the gap width variation, which can effectively reduce the labyrinth leakage.

However, in the actual working process of the labyrinth compressor, the piston is always work under eccentric operation. This paper obtain piston eccentricity distance from piston eccentricity trajectory in reference [11] via coordinate transformation, as shown in figure 10. It should be noted that the design clearance between the labyrinth and the cylinder is 0.3 mm, but the actual clearance fluctuates with the eccentricity of the piston. In this paper, UDF file about the piston eccentric movement is compiled as the piston movement boundary condition. During the eccentric operation of the piston, there is a slight inclination between the piston and the cylinder wall. K. Graunke’s research shows that compared with the parallel movement state, the piston incline state makes the leakage increase slightly which can be ignored [13]. Therefore, when defining the eccentric motion of the piston, this paper considers that the piston is always parallel to the cylinder wall.

Figure 11 shows the changes of leakage when the piston moves eccentrically and non-eccentrically under different pressure ratios. When the piston is in eccentric operation, the leakage is 1.5 ~ 2 times greater than that without eccentric operation. Therefore, if the coaxial between the piston and the cylinder can be realized, that is, the piston has no eccentric state during operation, the leakage will be
reduced. When the piston has no eccentricity, the collision between the piston and the cylinder can be avoided, and the labyrinth clearance can also be reduced to 0.1 mm. At this time, the leakage will be reduced by more than 80%. Therefore, reducing the labyrinth clearance and realizing the coaxial between the piston and the cylinder wall are the most effective methods to reduce the leakage and improve the volumetric efficiency.

![Image of Figure 9](image-url)

**Figure 9.** Variation of leakage with labyrinth clearance width.

![Image of Figure 10](image-url)

![Image of Figure 11](image-url)

**Figure 10.** Eccentric distance of first row piston.  **Figure 11.** Changes of leakage when the piston moves eccentrically and non-eccentrically.

4. Conclusion

In this paper, the labyrinth flow field and leakage analysis model are established, and the gas leakage characteristics in labyrinth are analysed. The following conclusions are drawn:

1) In the labyrinth entrance section, the labyrinth throttling effect is the most obvious, the velocity of leakage gas decreases step by step, and the pressure decreases significantly. In the middle section, both the descending gradient of velocity and dynamic pressure are weak. In the exit section, the gas flow velocity begins to increase again until it flows out of the outlet. The pressure of leakage gas in the whole labyrinth channel can always be in a decreasing state.

2) The clearance has a great influence on the leakage in labyrinth. When the clearance is less than or equal to 0.1 mm, the clearance has little effect on the leakage. However, because the piston is always under eccentric motion state, the clearance (0.1 mm) will cause the piston to collide with the
cylinder wall. When the clearance is greater than 0.1 mm, the leakage in the compressor increases rapidly with the increase of clearance.

3) When the piston operates eccentrically, the leakage will increase by 1.5 ~ 2 times than the design clearance. If the same axis can be realized between the piston and the cylinder, the gap between the piston and the cylinder can be reduced, and the leakage will be greatly reduced.

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References
[1] Wei W 2013 Numerical Simulation on Labyrinth Seal Flow Field Which Based on the Method of CFD (Daqing: Northeast Petroleum University).
[2] Lin M 2012 3D Flow Field Analysis and Piston Structure Optimization for Reciprocating Compressor Based on CFD (Shenyang: Shenyang University of Technology).
[3] Zhang S T 2015 Numerical Simulation for Labyrinth Seal Compressor Using Dynamic Mesh Technique (Shenyang: Shenyang Ligong University).
[4] Li H 2014 The Fem Analysis of Thermo-Mechanical Coupling and Flow Field of Labyrinth Sealing Reciprocating Compressor Cylinder System (Shenyang: Shenyang University of Technology).
[5] Tang H N, Lu J and Wang S J 2015 Fluid-structure interaction analysis of tooth profile angle effect on labyrinth seal performance Lubrication Engineering 40(10) 36-40.
[6] Tang H N, Wang S J and Zhao J 2015 Effect of fluid-structure interaction on sealed flow field and leakage rate based on computational fluid dynamics Journal of Shanghai Jiaotong University(Science) 20(3) 326-330.
[7] Zhou M, Sun D, Zhao H, Zhi Q and Zhang G C 2021 Leakage characteristics and formula establishment of nonmetallic labyrinth seal based on fluid-solid-thermal coupling Journal of Aerospace Power (Preprint)
[8] Sun D, Zhou M, Zhao H, Wang P and Du C Y 2021 Numerical study on fluid-structure interaction of labyrinth seal leakage characteristics considering tooth deformation Lubrication Engineering 46(3) 130-136.
[9] Wang T H, Li Z G and Li J 2021 Rotordynamic characteristics of the straight-through labyrinth seal based on the applicability analysis of leakage models unsing bulk-flow method J.Eng.Gas Turbines Power (Preprint gtp-21-1403).
[10] Yang J and Luis S A 2021 An Analytical Two-phase flow model for prediction of leakage in wet gas labyrinth seals and pocket damper seals.Is simplicity still desired J.Eng.Gas Turbines Power (Preprint gtp-21-1296).
[11] Cheng J M, Zeng X J, Liu Z, Yu X L and Feng Q K 2016 Research on dynamic modeling and electromagnetic force centering of piston/piston rod system for labyrinth piston compressor Proc IMechE Part I: J Systems and Control Engineering 230(8) 786–798.
[12] Qu Z C 2019 Principle of Reciprocating Compressor (Xi’an, Xi’an Jiaotong University Press).
[13] Graunke K and Ronnert J 1984 Dynamic behavior of labyrinth seals in oil-free labyrinth-piston compressors Proceeding of the 1984 International Compressor Engineering Conference Purdue University, W Lafayette, IN USA p 7-15.