Preliminary Design and Performance Evaluation of Micro Scroll Compressor Used in Refrigeration Systems

Shuo Song, Yuanyang Zhao, Qichao Yang, Guangbin Liu, and Liansheng Li

Abstract Micro refrigeration systems can be used in many applications. Micro scroll compressors have the potential for application in these refrigeration systems. With its miniaturization, the clearance and geometry parameters have become more influential in the performance of scroll compressors. In this paper, several three-dimensional flow models with different pitches, thicknesses, and leakage clearances are established. These models are designed by the control volume method with R134a as the working fluid. By using the CFD method, the temperature distribution, pressure distribution, and flow characteristics in the working chamber are analyzed. By comparing compressors with different pitches and thicknesses, the effect of pitch and thickness on the volumetric efficiency and isentropic efficiency is obtained. By changing the radial and axial clearances, the change trends of the efficiency are obtained. The results provide a reference for the design of micro scroll compressors.

Keywords Micro scroll compressor · Design · Thickness · CFD · Efficiency

1 Introduction

In recent years, as fossil energy consumption increases and climate warming becomes more and more serious, energy conservation, emission reduction, and green development have become important issues for the sustainability of society [1]. As a necessary piece of modern technology for human existence, refrigeration has recently gained popularity as a research topic. Due to their compact size, lightweight, low noise, great volumetric efficiency, and other benefits, scroll compressors are frequently employed in refrigeration and air conditioning systems [2–4].

Compressors used in micro refrigeration systems have recently made some advancements as a result of the advancements in micro refrigeration system technology. In general, a compressor with a volume flow rate of less than 10 cm^3/

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rev is defined as a micro compressor. Yang et al. [5–7] developed different micro-refrigeration systems in which the compressor components were micro-volumetric compressors. Sathe et al. [8, 9] tested the volumetric efficiency of a mini rotor compressor at varying pressure ratios to lay the foundation for the application of miniature compressors. As a volumetric compressor, scroll compressors have the potential to be used in micro refrigeration systems.

With the development of Computational Fluid Dynamics (CFD) techniques, it has been used to study the working process and performance of scroll compressors. Zheng et al. [10–13] studied the internal flow of compressors and investigated the effects of parameters such as scroll wall roughness, speed, evaporation temperature, and pressure on compressor performance using CFD methods. Song et al. [14] analyzed the internal flow at different suction port positions based on the CFD method and analyzed the effect of suction ports on the mass flow rate of compressors. Wang et al. [15] investigated the variation of base circle radius and spreading angle of variable thickness scroll compressor on the length of leakage line, but the effect on leakage volume and performance parameters was not analyzed. Pereira et al. [16] investigated the leakage of compressors with different working fluids and geometrical parameters and concluded that the inlet loss of the axial clearance and the curvature of the radial clearance have a significant effect on the leakage. Zheng et al. [17] analyzed the effect of the number of slots and slot depth on the scroll teeth on the radial leakage and obtained that both the volumetric efficiency and isentropic efficiency of the compressor increased and then decreased with the increase of the slot depth. Bell et al. [18] researched the empirical friction factor of the isentropic nozzle model, calculated the leakage mass flow rate of the model, and concluded that the compressor leakage rate is related to the Reynolds number and the geometry of the leakage clearance.

The above studies provide references for studying the flow and performance effects of scroll compressor leakage. But most of them do not quantify the effects of clearance and scroll parameters on the performance of scroll compressors systematically. When scroll compressors are used in miniature refrigeration systems, the miniaturization of the compressor makes its performance more sensitive to the change of structural parameters. Therefore, an in-depth study of the effect of geometric parameters on the performance of miniature scroll compressors is needed. In this paper, a numerical analysis model with radial clearance and axial clearance is established and transient flow simulations are finished by the CFD method. The flow field characteristics and the influence of geometric parameters on the compressor leakage line length and volumetric efficiency are studied.

Table 1 Main performance parameters of the system under air-conditioning conditions

Parameters	Number
COP	2.93
Power/W	104
Volume ratio	3.63

Table 2 Micro scroll compressor structure parameters

Structural parameters	Number
Suction volume (cm^3/r)	2.0
Thickness (mm)	1.2/1.3 /1.4/1.5
Pitch (mm)	5.6/5.8/6.0/6.2
Involute final spreading angle ($^\circ$)	1980
Radial clearance (mm)	0.010
Axial clearance (mm)	0.007

2 Preliminary Design of Micro Scroll Compressor

2.1 Basic Parameters

The working fluid of the micro refrigeration system is R134a, and the working conditions of the system are standard air conditioning conditions, the design conditions for the scroll compressor are shown in Table 1. The suction volume of the micro scroll compressor is 2.0 cm^3/r, and the rotation speed is 4500 r/min. The scroll profile is a circle involute. The detailed profile parameters are shown in Table 2.

2.2 Structure Design

The design structure of the micro scroll refrigeration compressor is shown in Fig. 1. The compressor adopts the low-pressure chamber structure, and the refrigerant in the suction state enters the inside of the compressor through the suction port located on the side of the compressor shell. The refrigerant cools the motor first, and then it is sucked into the suction chamber composed of orbiting and fixed scroll. Then with the rotation of the orbiting scroll, the refrigerant is compressed to the high-pressure state in the space between the orbiting and fixed scrolls. Finally, it flows out from the compressor through the discharge tube located on the top of the compressor and flows into the condenser of the refrigeration system.

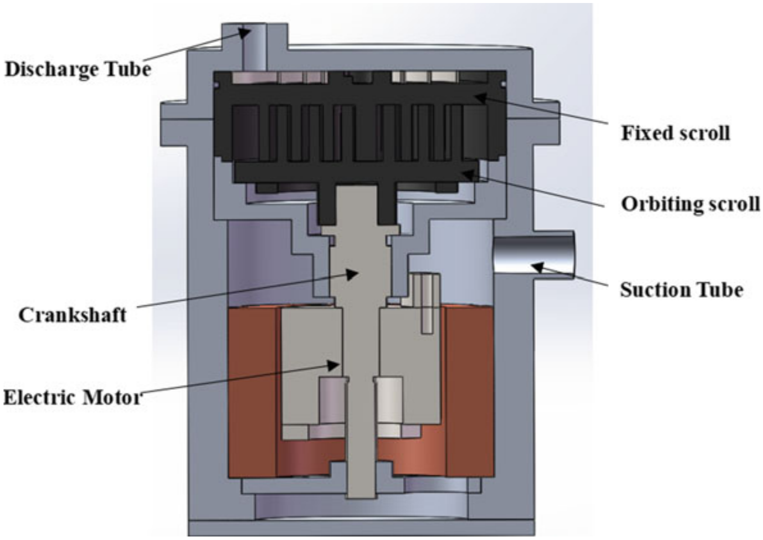


Fig. 1 Micro scroll compressor structure

3 Performance Analysis Model

3.1 Physical Model

A physical model was developed to estimate the performance of the micro scroll compressor, as seen in Fig. 2. The suction fluid domain, the discharge fluid domain, the working chamber fluid domain, and the axial clearance fluid domain were the four segments of the fluid domain in the physical model. Five monitoring sites were set up close to the wall of the fixed scroll, 90° apart from one another, to make it easier to analyze pressure change in the operating chamber of the scroll compressor (Fig. 3).

To study the influence trends of geometric parameters on the performance of micro scroll compressors, this paper adopts the control variates to ensure that the compressor displacement and the turn number of scrolls are constant. The pitch is fixed at 5.8 mm, and the compressor models with the thickness of 1.2, 1.3, 1.4, and 1.5 mm are established; the thickness is fixed at 1.5 mm, and the compressor models with pitches of 5.6, 5.8, 6.0, 6.2 mm are established. the specific structural parameters of the compressor are shown in Table 2.

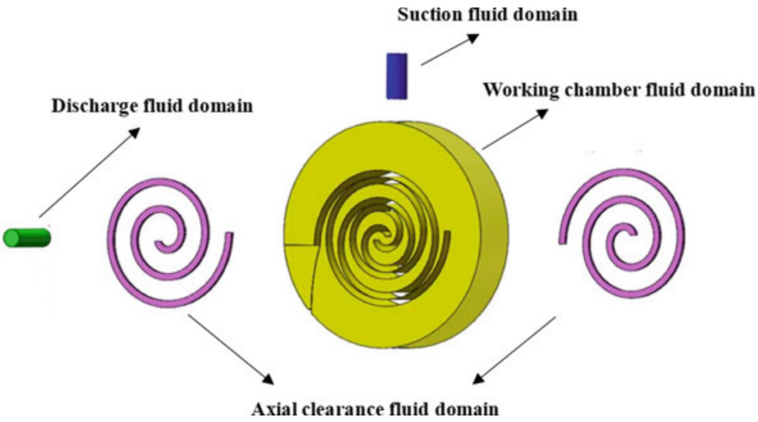
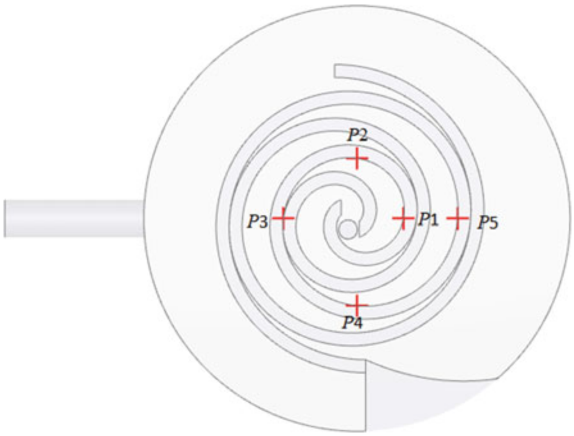


Fig. 2 Scroll compressor fluid domain model

Fig. 3 Monitoring points in the working chamber



3.2 *Meshing*

The moving region of the scroll compressor has meshed as the high-quality hexahedral structured grid. The inlet and outlet regions and the connection part are meshed by a common mesh template to generate a high-quality Cartesian mesh. The compressor mesh and the axial clearance mesh are shown in Fig. 4.

To verify the grid independence, the fluid domain model with a pitch of 5.8 mm and a thickness of 1.5 mm is selected as an example. The grid parameters and calculation results are shown in Table 3. When the grid number is 480,000, the mass flow rate of the compressor is 1.7198 g/s, and the mass flow rate changes very little when the grid number continues to increase. Therefore, the 480,000 grid model was used in the numerical simulation.

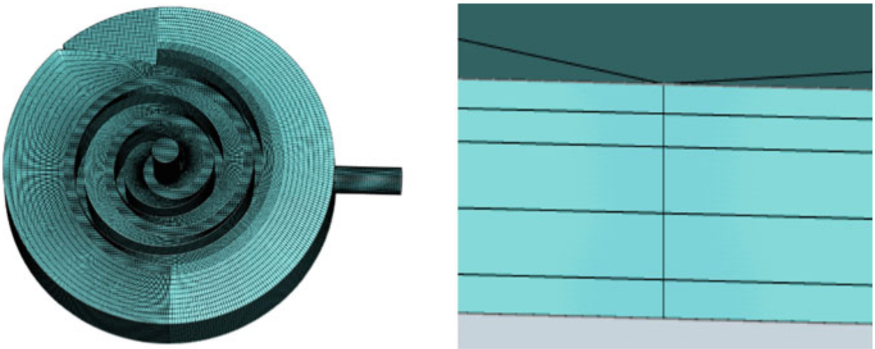


Fig. 4 Compressor calculation grid and axial clearance grid

Table 3 Grid independence and calculation time

Number of grids	Calculate the number of laps	Calculation time/h	Mass flow rate/g/s
280,000	5	11	1.5813
370,000	5	15	1.6768
480,000	5	18	1.7198
690,000	5	27	1.7203

3.3 Calculation Method and Boundary Conditions

R134a is used as the working fluid for the 3D transient simulation, and the thermophysical properties of R134a are called from the NIST database. The standard k-ε turbulence model is used to describe the refrigerant flow in the working chamber of the compressor. The compressor rotation speed is set to 4500 r/min. The pressure inlet and outlet conditions are chosen as the boundary conditions, where the inlet pressure is 0.3772 MPa, the inlet temperature is 291.45 K, and the outlet pressure is 1.4698 MPa. The first-order windward format is used, the pressure solver is chosen as the solver, and the pressure–velocity coupling equation is solved by the SIMPLES method. Since the compression process is very fast, the wall surface is regarded as an adiabatic wall surface.

4 Results and Discussion

4.1 Analysis of Internal Flow of Compressor

A micro scroll refrigeration compressor with a pitch of 5.8 mm and a thickness of 1.5 mm is selected in this part.

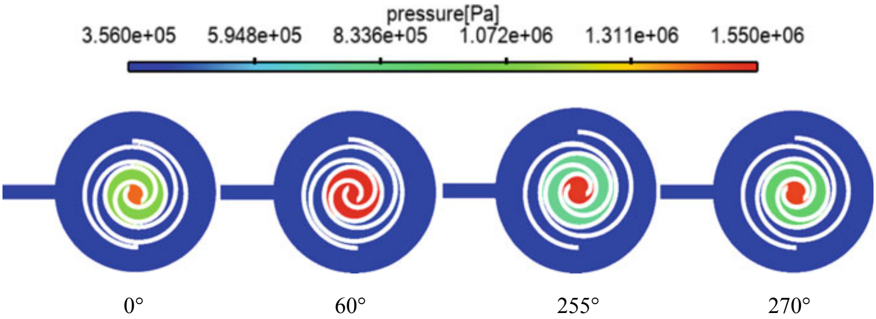


Fig. 5 Pressure distribution in working chambers

(1) **Pressure distribution**

Figure 5 shows the pressure distribution of the scroll compressor during one working cycle. Since the suction tube is not centrally symmetric, the pressure distribution in each pair of working chambers is not the same. When the crank angle is 0°, the pressure difference between the two suction chambers is about 5000 Pa. When the compressor rotates to 60°, there is pressure fluctuation in the discharge chambers because the volume of the discharge chambers changes more drastically and the connection area between the discharge chambers and the discharge channel is small, and the pressure rises steeply to 1.55 MPa. The region near the radial clearance is the same as a leakage channel with convergent and expansion parts. With the movement of the orbiting scroll, the pressure difference of the working chambers at two sides of the contact point is changing, the closer to the center, the greater the pressure difference between the two working chambers.

(2) **Temperature distribution**

Figure 6 shows the temperature distribution of the scroll compressor. The temperature distribution of each pair of working chambers is not uniform, and the temperature distribution in a working chamber is not uniform too.

Comparing the distribution of pressure and temperature distributions, the leakage flow has a certain influence on the mass flow in the working chambers of the compressor. And the influence of leakage flow on the temperature distribution is much larger than that of the pressure distribution.

(3) **Velocity distribution**

Figure 7 shows the velocity distribution in the scroll compressor. Except for the radial clearance area, the fluid in the whole compression chamber is in a lower velocity motion. To better show the overall velocity distribution of the flow distribution, the upper-velocity limit is set to 15 m/s in the following section, while the actual maximum velocity in the leakage clearance is up to 204 m/s.

When the suction process is finished, the velocity distribution in the two suction chambers is not uniform. The suction flow channel is closed at this time and some

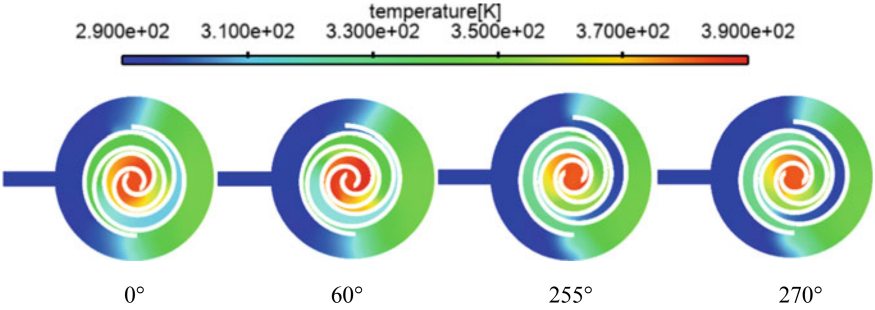


Fig. 6 Temperature distribution in working chambers

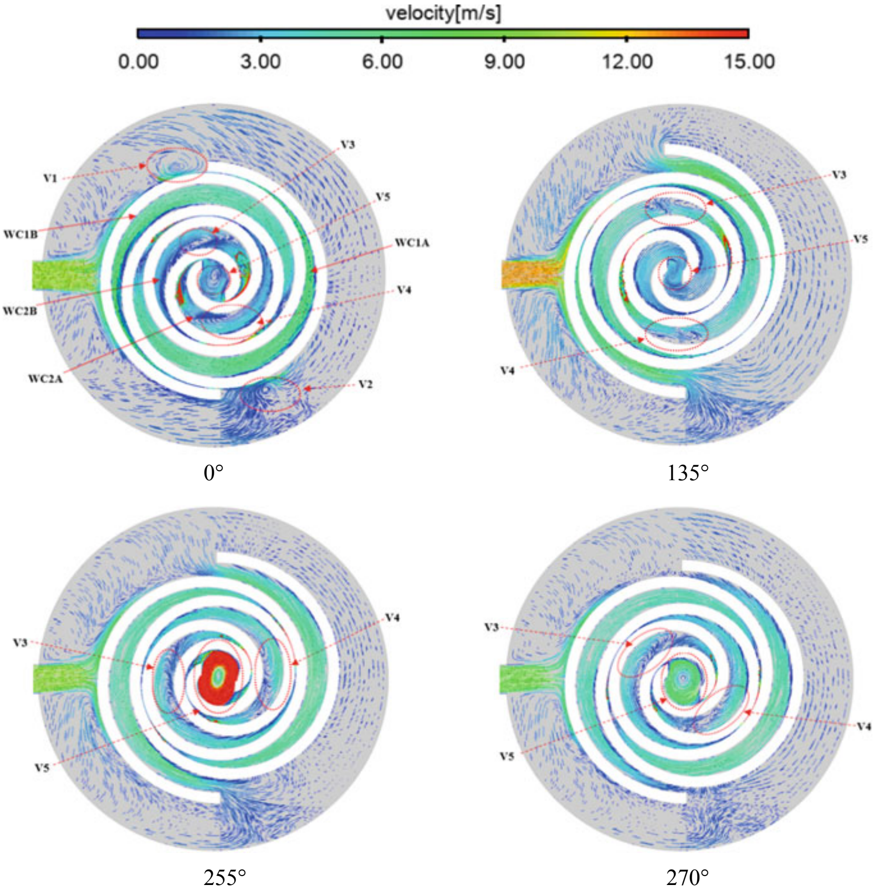


Fig. 7 Velocity distribution of working chamber

gas leaks through the clearance from the suction chamber. And the vortex V1, V2 is generated in the suction channel. At the radial clearance, the direction of leakage flow is from the chambers near the scroll center to the outside chambers.

With the movement of the moving scroll, the pressure difference between the two sides of the clearance increases, and tangential leakage flow gradually obvious. The leakage in the radial clearance leads to the vortex V3, V4 in the working chamber of compressors. It is found that the intensity of vortex V4 is always greater than that of V3. This is because the gas flow velocity of WC1A in the two suction chambers is always greater than that of WC1B. The gas in the WC2A chambers is more violent than that of WC2B. For the discharge chamber, the vortex V5 has always existed. As the moving scroll continues to move, the vortex V5 intensity gradually decreases.

(4) **Working process of scroll compressor**

Figure 8 shows the pressure change curves of the scroll compressor with different scroll thicknesses. During the compression process, the pressure in the working chamber rises with the increase of the scroll thickness. During the discharge process, the pressure rises faster when the scroll thickness is small, as shown in Fig. 8a. When the rotation angle of about 600°, the pressure fluctuation appears firstly in the compressor chamber with a big scroll thickness, as shown in Fig. 8b.

Figure 9 shows the pressure change curves of the compressor at different scroll pitches. During the compression process, the gas pressure with a smaller pitch rises

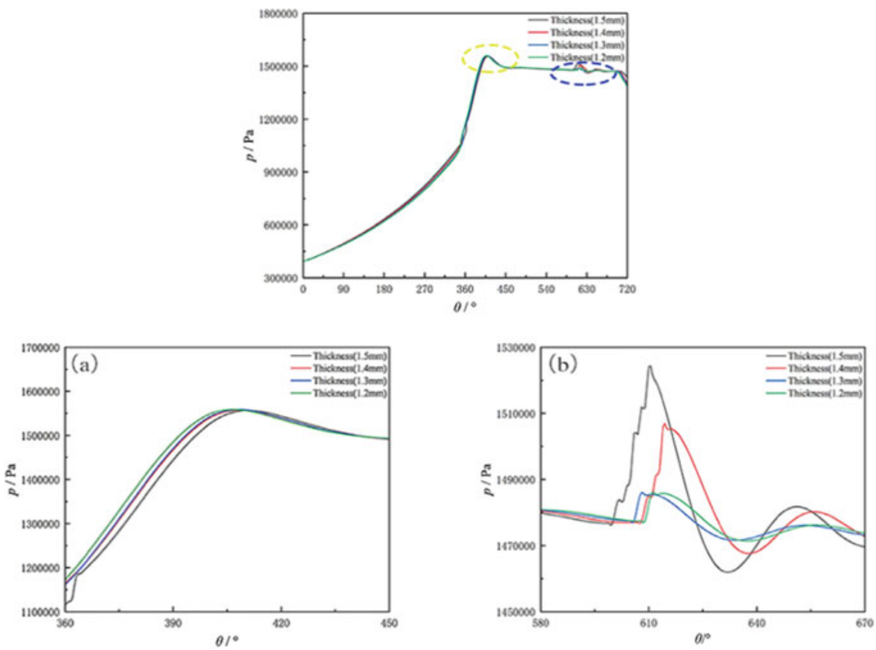


Fig. 8 Pressure variation during compressor operation with different thickness

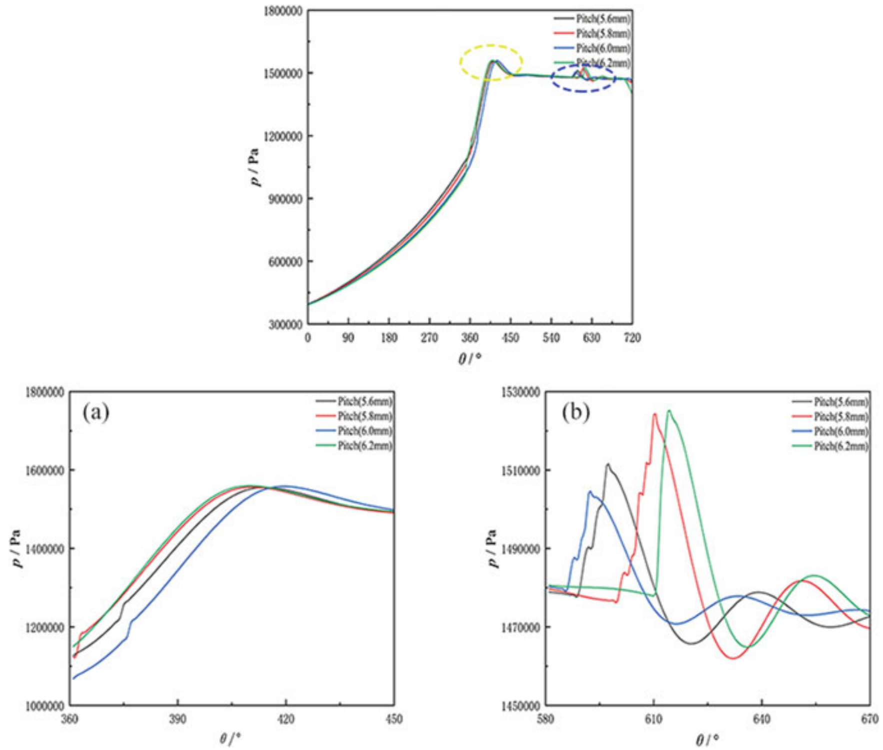


Fig. 9 Pressure variation during compressor operation at different pitches

faster, which means there is smaller leakage when the scroll pitch decreases. At the start of the discharge process, the pressure change trend changes. The larger the pitch, the faster the pressure rise, as shown in Fig. 9a. These may be caused by the change in the leakage flow during the discharge process. When the rotation angle is about 585°, the pressure fluctuation with a smaller pitch appears first, as shown in Fig. 9b. The effect of the scroll pitch on the working process is bigger than that of the scroll thickness.

4.2 Performance Analysis Under Variable Working Conditions

The volumetric efficiency is a key parameter to express the performance of refrigeration compressors. The volumetric efficiency is defined as:

$$\eta_v = \frac{m_{\mathrm{ac}}}{m_{\mathrm{id}}} \tag{1}$$

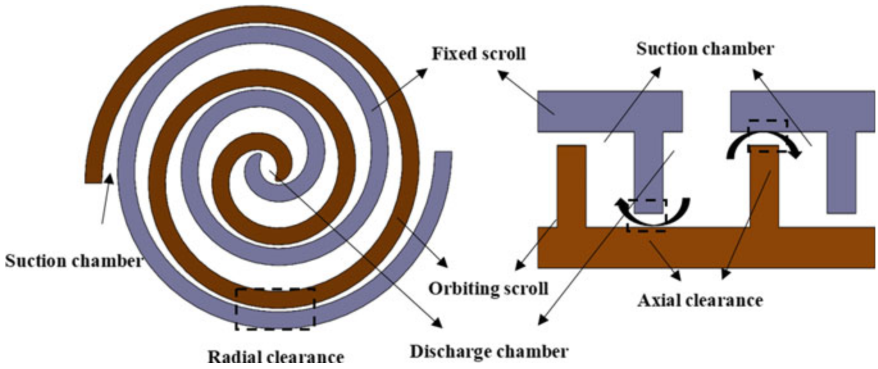


Fig. 10 Schematic diagram of internal leakage of scroll compressor

where m_{ac} is the actual mass flow rate, m_{id} is the design mass flow rate.

The isentropic efficiency is defined as:

$$\eta_s = \frac{P_{ad}}{P_{is}} \quad (2)$$

where P_{is} is the actual compression work, and P_{ad} is the isentropic process work.

The leakages of scroll compressors mainly are composed of tangential leakage through the radial clearance and radial leakage from the axial clearance (Fig. 10).

The length of the tangential leakage line is the height of the scroll, which can be defined as:

$$L_f = 2h \quad (3)$$

The length of the radial leakage line is related to the length of the scroll profile and the rotation angle, which can be defined as:

$$L_r = 2\int_{\varphi_i}^{\varphi_{i+1}} \sqrt{x^2 + y^2} d\varphi \quad (4)$$

where φ_i, φ_{i+1} are the starting and ending angles of the scroll profile.

(1) **Effect of scroll thickness**

Figures 11 and 12 show the volume efficiency with different thicknesses and axial clearances when the scroll pitch is 5.8 mm. In the actual design, scroll compressors can use an axial sealing mechanism such as a sealing strip to reduce the influence of axial clearance. Therefore, in the numerical simulation, this case can be simplified to the axial clearance of 0 μm. From Table 4 it can be obtained that the radial leakage line length of compressors with different thicknesses is the same, and the tangential leakage line length increases with the increase of thickness. When the axial clearance is 0 µm and the scroll thickness increases from 1.2 to 1.5 mm, the compressor volumetric efficiency decreases from 86.66 to 82.06%.

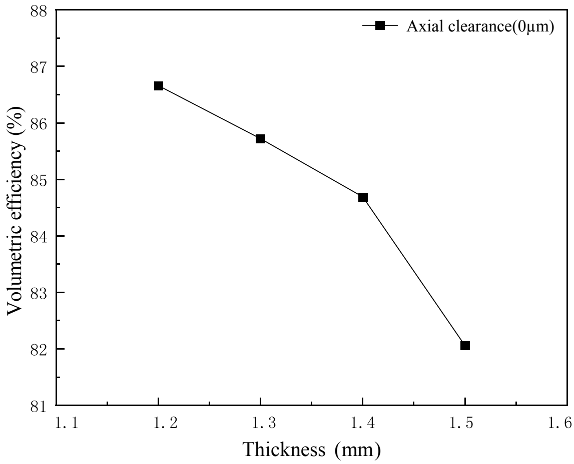


Fig. 11 Volumetric efficiency at different thicknesses without axial clearance

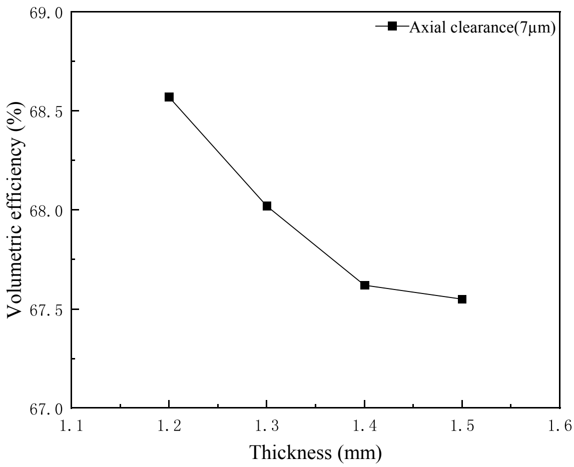


Fig. 12 Volumetric efficiency at different thicknesses for an axial clearance of 7 µm

Table 4 Leakage line length at different thickness

Thickness/mm	Radial leakage line length/mm	Tangential leakage line length/mm
1.2	139.6812	8.074
1.3	139.6812	8.579
1.4	139.6812	9.147
1.5	139.6812	9.800

When the axial clearance is 7 μm and the thickness increases from 1.2 to 1.5 mm, the compressor volumetric efficiency decreases from 68.57 to 67.55%. It means that the leakage through the axial clearance has a greater impact on the volumetric efficiency. And the leakage caused by changing the thickness and the leakage through the radial clearance is small.

(2) **Effect of scroll pitch**

The scroll pitch is one of the important structural parameters and is directly related to the outer diameter of the compressor motor scroll. In the initial design of the compressor, the height-pitch ratio is recommended in the range of 1.0–2.5 to ensure the stability of the compressor. In this paper, the length of the leakage line is calculated for different pitch models which are shown in Table 5. The larger the pitch, the longer the radial leakage line and the shorter the tangential leakage line with the same thickness and number of scroll turns.

Figures 13 and 14 show the change in volumetric efficiency at different pitches and axial clearances when the scroll thickness is 1.5 mm. The compressor volume efficiency shows an increasing trend under different axial clearances. When the axial clearance is 0 μm and the pitch increases from 5.6 to 6.2 mm, the volume efficiency increases from 78.48 to 88.43%. When the axial clearance is 7 μm and the thickness increases from 5.6 to 6.2 mm, the volume efficiency increases from 65.69 to 70.74%. The effect of leakage through the axial clearance on the volumetric efficiency is much greater than that of the radial clearance, which leads to an increase in volumetric efficiency.

The above simulation results show that the effect of leakage through the axial clearance on the volumetric efficiency of the compressor is much greater than that of the radial clearance leakage. When designing a micro scroll compressor, a suitable thickness should be selected and the pitch should be increased as much as possible under the condition that the capacity and clearance remain unchanged.

(3) **Effect of clearance**

The leakage clearance has an important impact on the performance of micro scroll refrigeration compressors. A compressor model with a pitch of 6 mm and a thickness of 1.5 mm was established, and CFD simulations were conducted for different axial clearance and radial clearance, and the results are shown in Figs. 15 and 16.

When the axial clearance is 5 μm and the radial clearance is gradually increased from 5 to 11 μm, the volumetric efficiency of the compressor decreases from 86.38%

Table 5 Leakage line length at different pitches

Pitch/mm	Radial leakage line length/mm	Tangential leakage line length/mm
5.6	134.8648	10.931
5.8	139.6812	9.800
6.0	144.4977	8.842
6.2	149.3141	8.022

Fig. 13 Volumetric efficiency at different pitches without axial clearance

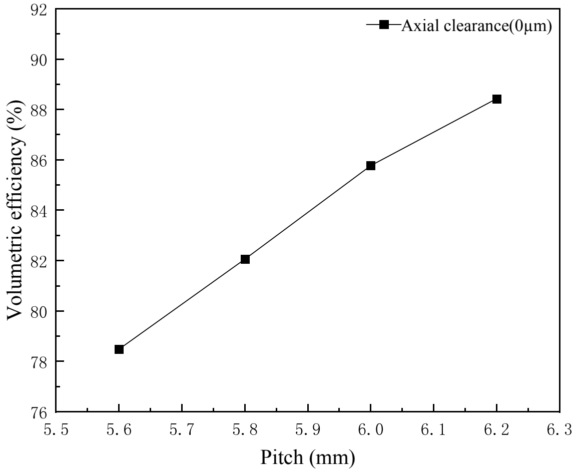
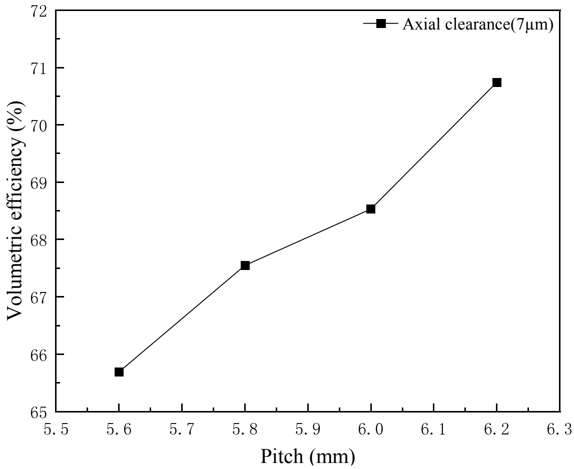


Fig. 14 Volumetric efficiency at different pitches for an axial clearance of 7 μm



to 75.11% and the isentropic efficiency decreases from 67.78 to 56.18%. When the axial clearance is 0 μm, the volumetric efficiency of the compressor decreases from 92.06 to 84.67% and the isentropic efficiency decreases from 78.35 to 64.19% when the radial clearance increases from 5 μm to 11 μm, and the changing trend is the same as that of the axial clearance.

According to the comparison of the data without axial clearance and with axial clearance of 5 μm, the difference between the two volumetric efficiencies gradually increases with the increase of radial clearance, and the maximum difference can reach 9.56%. The difference between the isentropic efficiencies gradually decreases, and the maximum difference is up to 10.57% when the radial clearance is 5 μm.

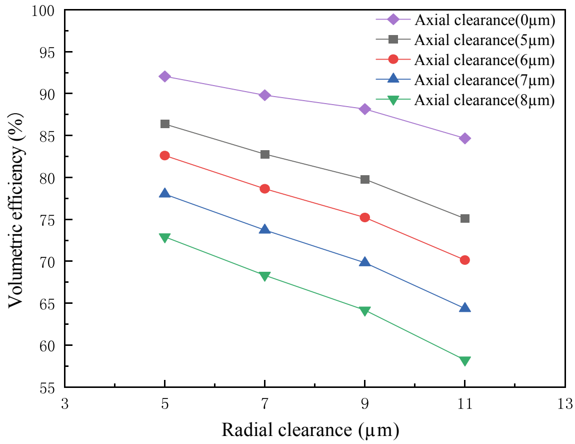


Fig. 15 Effect of clearance on volumetric efficiency

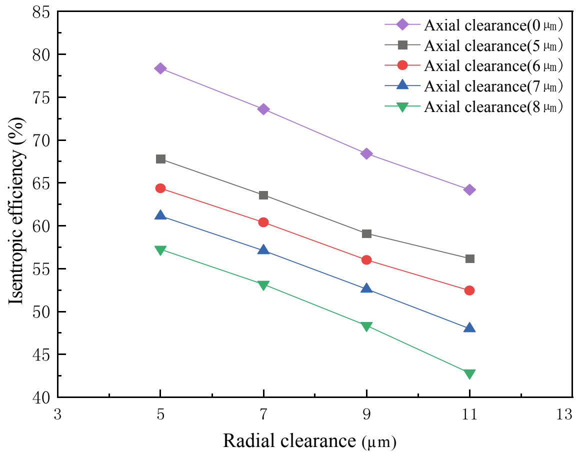


Fig. 16 Effect of clearance on isentropic efficiency

Compared with the radial clearance, the axial clearance has a greater impact on the volumetric efficiency and isentropic efficiency of the micro scroll compressor because the leakage channel of the axial clearance of the compressor is longer than that of the radial clearance.

During the operation of the scroll compressor, due to the machining accuracy, it is impossible to ensure that the two scroll plates are completely tight, so there must be radial clearance and axial clearance between the scroll plates, which will lead to radial leakage and tangential leakage. According to the above results, it can be seen that the axial clearance has more influence on the compressor performance, so in the actual design, other sealing methods are usually adopted to reduce the influence of the clearance, such as sealing strips and labyrinth seals. Since the size of the micro compressor is much smaller than the normal compressor, it is more difficult to take suitable axial sealing measures for the micro compressor, which is an important direction development for the future [19]. In this paper, to make the compressor volume efficiency reach 80% or more, the axial clearance should be less than 6 μm and the radial clearance less than 9 μm based on the calculation results.

5 Conclusions

In this study, the R134a micro scroll refrigeration compressor was modeled and analyzed using CFD numerical methods, and the influence trends of the scroll thickness, clearance, and scroll pitch on the compressor performance were analyzed.

There is always a vortex phenomenon inside the compressor, the existence of vortex and clearance leakage cause the flow change in the working chamber. Comparing the distributions of pressure and temperature, the influence of leakage flow on the temperature distribution is much larger than the pressure distribution.

In the ideal case of no axial clearance, the volumetric efficiency decreases slightly with the increase in the scroll thickness. After adding 7 μm of axial clearance, the compressor volumetric efficiency decreases from 68.57% to 67.55%. Ensuring the same thickness and number of scroll turns, the increase of the scroll pitch has a significant effect on the improvement of the volumetric efficiency. When the axial clearance is 7 μm and the scroll pitch increases from 5.6 to 6.2 mm, the volume efficiency increases from 65.69 to 70.74%.

With the increase in clearance, the volume and isentropic efficiencies show a decreasing trend. When the radial clearance changes from 5 to 11 μm and the axial clearance is 5 μm, the volume efficiency decreases from 86.38 to 75.11% and the isentropic efficiency decreases from 67.78 to 56.18%. For the compressor prototype in this paper, when we wish to increase the volumetric efficiency up to 80%, the axial clearance should be less than 6 mm and the radial clearance should be less than 9 mm.

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