Optimization of Suction Chamber Structure in a Scroll Refrigeration Compressor

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Abstract. In order to improve the suction performance of the scroll compressors, five modification models are proposed by reducing or enlarging the area of the suction flow passage based on a scroll refrigeration compressor. The three-dimensional numerical simulation model is established using the commercial software PumpLinx to calculate the flow field and the performance of every modified model. The comparison and analyses has been done based on the simulation results. The results indicated that the enlargement of the suction passage only improve the volumetric efficiency a little while the shrink of the suction passage can improve it effectively. The mass flow rate pulsation when the modification angle is 270° is much higher than that of the original model, but it decreases with the modification angle. As the modification angle increases, the volume of inner suction passage and the back flow rate from the inner suction passage decrease and less fluid from suction chamber 2 flows back to the suction port, so the average mass flow rate increases and the mass flow rate pulsation decreases. The Model 5 has the maximum volumetric efficiency and lower mass flow pulsation, but the tangential and radial forces on the orbiting scroll are the highest.

1. Introduction

Scroll compressor is widely used in small refrigeration systems and air conditioners due to its simple structure, high efficiency, light weight, low energy consumption and high reliability, which greatly stimulates the enthusiasm of researchers for scroll compressor. How to develop a scroll compressor with higher performance has become the focus of current research.

The optimization of the scroll compressor through simulation method is an effective way to obtain the high performance compressor. Nieter [1] discussed the effect of geometry parameters such as the warp thickness, wrap height and volume ratio on the design, manufacture and energy loss of the scroll compressor and concluded that the wrap height and start angle were the two most important parameters. Ishii [2,3] studied the influence of the main geometry parameters on friction loss and leakage flow rate, so the mechanical efficiency and volumetric efficiency can be calculated out to obtain an optimum combination of the geometry parameters. The calculation procedures for the efficiency in the literatures [1-3] were mainly based on the fundamental equations with many assumptions. The chamber model provided a more accurate method to predict the performance of the scroll compressor [4-6]. Based on the chamber model, the influence of geometry parameters on the performance can be calculated out, so the optimum combination of the structure parameters can be obtained by combining the chamber model with the optimum procedures. Liu [7] integrates the mathematic model which included the
thermodynamic and dynamic model and the optimization solver to progress optimization research which aimed to reduce the friction loss of bearing components, and proposed some suggestions for designing bearings. Tseng [8] proposed a systematic design method for developing a family of scroll-type compressor using the general design optimization model, interactive session and discrete variable design optimization skills. Bell [9] developed an analytical model for the irreversibility during the compression process to optimize the scroll compressor for liquids-flooded Ericsson cycle application.

As the CFD (Computational Fluid Dynamic) simulation was applied in scroll compressors [10-13], it provided a method to calculate the flow fields in scroll compressor and helped to optimize the detailed structure of the suction and discharge ports directly. Feng [14] established a steady CFD model to investigate the flow field and flow loss in the discharge region of a scroll compressor, and indicated that the discharge loss mainly occurred at the onset of discharge and the easier-open discharge port led to the better discharge characteristic. Cui [10] conducted a CFD study of the working mechanism of the dummy port which was manufactured at the opposite to the discharge port to reduce the discharge flow loss, and pointed out that the dummy port made gas going through compressor easier and reduced the asymmetry of the gas pockets on the two sides of the discharge port. Song [15] investigated the influence of three arrangements of suction port on performance of a scroll expander, and the results showed asymmetry pressure distribution was caused by the blocking effect.

Although the suction process has significant impact on the volumetric efficiency and adiabatic efficiency, few literatures have been reported on the suction process. Cui [16-17] investigated the fundamental mechanism of suction process using CFD simulation and indicated that the dynamic features of the process showed expansion at the beginning and compression at a later stage of the suction process; the separations and vortices existed in the flow field during the gas intake process and the kinetic energy associated with these phenomena was dissipated into heat; and the suction pressure fluctuation reached 10% of the averaged inlet pressure and caused the performance loss, structure vibration and noise. Li [18] and Lv [19] pointed out that the inlet pressure pulsation would influence the inlet mass flow rate and make the suction process unsteady. Wu [20] studied the contribution of compressor noise to refrigerator overall noise and indicated that the suction and discharge pressure pulsation and vibration make great influence on refrigerator noise. Sun [21] established the three dimensional numerical simulation model of a scroll refrigeration compressor and analysed the flow field in the suction passage; it was found that there are flow phenomena such as ‘cut-off’ and ‘source’ in the flow passage which will limit the suction flow rate of the scroll compressor; but the paper did not discuss about the pulsation of the mass flow rate which will influence the vibration and noise of the compressor significantly.

In order to reduce the flow loss during the suction process, the present paper proposed a series of designations for the suction passage and simulated their flow fields using CFD methodology. The performance of the scroll compressors with these suction passages was predicted. The schemes will be compared and discussed to obtain a better one. And the flow fields in the optimum suction passage will be analyzed to acquire the reason for the better performance.

2. Mathematical Model

2.1. Physical Model

A scroll refrigeration compressor is taken as the research object in this paper. The main parameters of the scroll compressor are shown in Table 1. In order to optimize the suction process of the scroll refrigeration compressor, the suction flow passage of the scroll refrigeration compressor is modified using five different models, respectively. In the model 1 and model 2, the suction passage is magnified to reduce the flow loss from the suction port to the suction chamber 2 to impair the phenomenon ‘cut-off’ [21]. From the model 3 to model 5, the suction passage is shrunk to prevent the fluid from flowing back from the suction chamber 2 to the suction port. The modification angles which are in accordance with the crank angle from model 3 to 5 are 270°, 300° and 320° respectively.
Table 1. The main parameters of the original scroll refrigeration compressor

| Parameters                  | Value |
|-----------------------------|-------|
| Thickness of scroll vane/mm | 3.6   |
| Height of scroll vane/mm    | 40    |
| Numbers of circle           | 2.75  |
| Theoretical suction volume/mL| 82.4  |
| Radial clearance/mm         | 0.03  |

Figure 1. Schematic diagram of suction chamber modification

2.2. Grid generation
Figure 2 shows fluid domain and the grids of the original model. In this paper, the hexahedral grids were updated with the motion of the orbiting scroll during the solution process. The number of radial clearance grid layers remained unchanged in the calculation process, which ensured the convergence and accuracy of the calculation. Since the sealing strip was used in the axial clearance of the scroll compressor prototype, the axial clearance was neglected in this study.

Figure 2. Fluid domain and overall grids of the original model.

2.3. Boundary conditions
The pressures of inlet and outlet were set to be 0.627 MPa and 2.146 MPa, respectively. The inlet temperature was 307 K. R22 was used as the working fluid and its real parameters were obtained from
The parameters of R22 mainly including density, entropy, thermal conductivity and viscosity were added into PumpLinx [23]. The rotational speed was set to be 2880 r/min. It was assumed that there was no heat transfer with the outside of the compressor due to the high rotational speed, so the adiabatic model was used in the calculation of the sidewall surface.

2.4. Governing equation
Dynamic mesh technology was used in the process of numerical simulation in this paper. The basic of the governing equation could be written as follows [24]:

\[ \frac{\partial}{\partial t} \int_{\Omega(t)} \rho d\Omega + \int_{\sigma} \rho (v - v_\sigma) \cdot n d\sigma = 0 \]  

(1)

\[ \frac{\partial}{\partial t} \int_{\Omega(t)} pv d\Omega + \int_{\sigma} \rho ((v - v_\sigma) \cdot n) v d\sigma = \int_{\sigma} \nu n d\sigma - \int_{\sigma} p \cdot n d\sigma + \int f d\Omega \]  

(2)

\[ \frac{\partial}{\partial t} \int_{\Omega(t)} \rho C_p T d\Omega + \int_{\sigma} \rho C_p T ((v - v_\sigma) \cdot n) n d\sigma = \int_{\sigma} \left( \frac{\mu}{Pr} \frac{\partial C_p T}{\partial x_j} \right) \cdot n d\sigma + \int S d\Omega \]  

(3)

\[ \tau_{ij} = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu \frac{\partial u_k}{\partial x_k} \delta_{ij} \]  

(4)

Where: \( \Omega(t) \) is control volume, \( \rho \) is fluid density, \( \sigma \) is the surface of \( \Omega(t) \), \( n \) is surface normal of \( \sigma \), \( P \) is pressure, \( v \) is fluid velocity, \( v_\sigma \) is velocity of the \( \sigma \), \( f \) is body force, \( C_p \) is heat capacity, \( T \) is temperature, \( \mu \) is dynamic viscosity coefficient, \( Pr \) is Prandtl Number, \( S \) is source item.

3. Results and Discussion

3.1. Results Comparison
Table 2 shows the performances of the five models compared with the original model. The volumetric and adiabatic efficiency had the same definitions in literature [18]. The discharge temperature was obtained using time-integration algorithm, as formulated below:

\[ T_{\text{dis,ave}} = \frac{1}{2\pi} \int_0^{2\pi} T_{\text{dis}}(t) dt \]  

(5)

From the Table 2, the suction mass flow rate, volumetric efficiency and adiabatic efficiency of all modification models are higher than the original model. It is indicated that the suction passage has significant impacts on the performance of the scroll compressor. The detailed analyses of the performance of the modification models are shown in the following sections.

| Table 2. Time-averaged performances of the five geometric models |
|---------------------------------------------------------------|
| **Suction mass flow rate [kg/s]** | **Volumetric efficiency/%** | **Adiabatic efficiency/%** | **Indicated power [W]** | **Discharge temperature [K]** |
|----------------------------------|-----------------------------|-----------------------------|--------------------------|-------------------------------|
| Original Model                   | 0.07956                     | 86.69                       | -                        | 55.52                         |
| Model 1                          | 0.08131                     | 88.60                       | 1.91                      | 55.90                         |
| Model 2                          | 0.08105                     | 88.32                       | 1.63                      | 55.50                         |
| Model 3                          | 0.08401                     | 91.55                       | 4.86                      | 55.90                         |
| Model 4                          | 0.08771                     | 95.57                       | 8.88                      | 56.47                         |
| Model 5                          | 0.08978                     | 97.83                       | 11.14                     | 56.71                         |
From the Table 2, the volumetric efficiency of Model 1 is 1.91 larger than that of the original model while the volumetric efficiency of Model 5 is 11.14% larger. It indicates that the magnification of the suction passage will not improve the suction performance a lot. The shrink of the suction passage will improve the volumetric efficiency remarkably. The Model 5 has the highest suction mass flow rate and its volumetric efficiency reaches 97.83%.

The adiabatic efficiency of each model in a period is shown in Table 2. Basically, the adiabatic efficiency of the modified models changes a little, which is different from the volumetric efficiency. Compared with the adiabatic efficiency of the original model, the efficiency of Model 1 increases by 0.38% while that of Model 2 decreases by 0.02%. When the flow passage shrinks, the adiabatic efficiency increases slightly. The Model 5 has the highest adiabatic efficiency which is 1.19% higher than that of the original model. As the volumetric and adiabatic efficiency increase, the discharge temperature decrease. Nevertheless, it does not change a lot. The discharge temperature of the Model 5 is 388.47K, which is only 1.4K lower than that of the original model.

3.2. Pulsation of the suction mass flow rate
Figure 3 shows the variation of mass flow rates with crank angle. The different suction flow passage leads to different periodic variations of the suction flow rate.

The Model 1 and Model 2 have larger suction flow passages, so the flow passage wall and orbiting scroll will not mesh and the ‘cut-off’ phenomenon will not form during the suction process. It can be seen that suction mass flow rate of Model 1 and Model 2 do not falls abruptly after 220°, which indicates that the ‘cut-off’ phenomenon does not happen. But they still have the peak at about 45°, which is caused by the big volume of the suction passage which will accommodate an amount of gas temporarily at the beginning of the suction process.

The pulsation of the mass flow rate in Model 3 is the highest. As shown in Figure 3, the mass flow rate diagram of Model 3 can be divided into 6 processes by five special angles which were labelled A to E. The mass flow rate reaches a peak at crank angle A (30°) and then falls in to a trough position at crank angle of 95° at position B. Then the mass flow rate increases until the crank angle C (165°). At the crank angle of D (290°), the mass flow rate falls suddenly and becomes negative at about 320°. The minimum mass flow rate happens at crank angle of 338° which is marked by E. Then the mass flow rate increases rapidly to the maximum value at crank angle A. It can be seen that the pulsation of mass flow rate decreases with the modification angles from Model 3 to Model 5, but it is still larger than the pulsation of the original model. And the beginning crank angle where the mass flow rate decreases suddenly increases from 290° to 340°.

From the Table 2 and Figure 3, it can be concluded that the magnification of the flow passage could reduce the flow pulsation but cannot improve the volumetric efficiency effectively. In addition, the diameter of the fixed scroll will increase when the flow passage is enlarged, so the benefit of magnification of flow passage seems not to overtake the disadvantage of the modification. For the Model3, Model4 and Model5, the diameter of fixed scroll decreases and the volumetric efficiency increase largely, but the flow pulsation is larger than the original model. As the volumetric efficiency increase with the modification angle, and the flow pulsation decrease with it, the Model 5 has the best performance. However, it is early to state that the 320 degree is the optimized angle. The cases whose modification angles are 340 and 360 degree are calculating. It can be deduced that the modification angle of 360 degree could be the optimal one.
In order to investigate the reason for the pulsation of mass flow rate in Model 3, four monitor points were arranged in the suction passage, as shown in Figure 4. The suction passage is divided into outer flow passage and inner flow passage. The outer flow passage is tight so the orbiting scroll will engage with its side wall. The inner flow passage has a volume which can contain a quantity of fluid.

The suction conditions of the refrigerant fluid are 0.627 MPa and 307.7 K and the discharged pressure is 2.146 MPa. The initial pressure at point 1 in the suction pipe is much lower than the inlet pressure. When the crank angle changes from 30° to 95° and from 280° to 338°, the pressure at Point 1 is higher than the suction pressure. So the suction mass flow rate shows a downward trend in both stages. The pressure at Points 3 and 4 reached a maximum value at the crank angle is about 280°. The pressure at Points 3 and 4 fluctuates greatly in one cycle. The intense pulsation of the pressure in the flow passage results in the pulsation of the suction mass flow rate.
Figure 5. Pressure variations of monitor points: (a). Model 3, (b). Model 5

The suction mass flow rate of model 3 in the stage from E to A has a quick increase, which is mainly caused by the higher pressure difference between suction inlet and suction pipe. As the pressure of Point 1 gets down to the minimum value at crank angle of 5°, the fluid from the inlet to the Point 1 continues accelerating until the crank angle of 30°. The flow velocity at the Point 1 in suction pipe increases quickly and reaches a maximum value due to the higher pressure difference between Point 1 and Point 3, which is shown in Figure 6(a) and Figure 6(b). Because the flow passage is fully opened in the stage from E to A, a large amount of working fluid were sucked into the flow passage, and the mass flow rate has a quick increase and reached a peak at position A.

In Figure 5(a), the pressure at Point 3 becomes higher than the Point 1 in the stage from A to B. The fluid flows into the suction passage and stops, which leads to the increase of the pressure of Point 3. The pressure of Point 1 also increases as the fluid flowing into the suction passage decelerates. Although the pressure of Point 1 is higher than the inlet pressure, the gas continues flowing into the suction passage under the effect of the inertia force of the fluid.

When the crank angle reaches 60°, the volume of the suction chambers increases and the pressure of Point 2 and Point 3 begin to decrease. The pressure of Point 1 also decreases as the fluid flows into the suction chambers. At the crank angle of 95°, the pressure of Point 1 is lower than the inlet pressure with the increase of suction volume, as Figure 6(c) shows. Therefore, the inlet velocity begins to accelerate and the mass flow rate increases. The pressure at Point 2, Point 3 and Point 4 in the stage from B to C is smaller than the inlet pressure, which is mainly caused by the volume enlargement of both suction chambers. Because the pressure of Point 2 and Point 3 decreases, the inlet mass flow rate increases and reaches a peak at the crank angle of 165°. After the crank angle of 135°, the refrigerant fluid in the suction flow passage is slightly compressed owing to the movement of the orbiting scroll, and the pressure at the Point 3 and Point 4 begins to increases.
In the stage from C to D, the mass flow rate shows a relatively stable process due to the slow increase of the volume of suction chamber 1. Since the orbiting scroll begins to engage with the suction passage wall, the pressure of Point 3 and Point 4 increases. The fluid only flows into the suction chamber 1, as shown in Figure 6(d). The pressure at Point 1 and Point 2 changes a little during this stage, so the mass flow rate almost keeps constant.

In the stage from D to E, the suction mass flow rate drops suddenly as well as the pressure at Point 3 and Point 1. When the crank angle reaches the position D, the orbiting scroll leaves from the outer flow passage wall and the outer flow passage connects with the inner flow passage. The fluid in the suction chamber 2 and inner flow passage flows back and the pressure of Point 3 and Point 4 decreases sharply. This phenomenon can also be seen in Figure 6(e). The backflow results in the pressure increase at Point 1, which becomes bigger than the inlet pressure rapidly. The inlet mass flow rate falls down. At the crank angle of 320°, the inlet mass flow rate becomes negative, which means that the fluid flows out at the inlet. Figure 6(f) shows the flow field and pressure distribution at the crank angle of 338°. The backflow prevents the gas from flowing into the flow passage and the mass flow rate reaches the minimum value, but the suction passage begins to open to the suction pipe. After the crank angle of 338°, the pressure of Point 1 which decreases with the pressure of Point 3 becomes lower than the inlet pressure, so the negative mass flow rate decreases. After the crank angle of 355°, the inlet mass flow rate becomes positive and increases with the pressure difference between the inlet pressure and the pressure of Point 1. The mass flow pulsation principles of Model 4 and Model 5 are similar with the Model 3.

Figure 5(b) shows the variation rule of transient pressures at the monitor points in model 5 during a period. It can be seen that the pressure pulsation amplitudes of point 1 in model 5 is smaller than that in the model 3 in one cycle. The maximum pressure of point 1 is 0.65 MPa and the minimum pressure is...
0.58 MPa, while in Model 3 the maximum pressure of Point 1 is 0.64 MPa and the minimum pressure is 0.57 MPa. Therefore, the maximum of the suction mass flow rate of model 5 is smaller than the model 3 and the minimum suction flow rate is larger than that of Model 3. The volume of the suction flow passage decreases as the modification angle increases. After the flow passage is cut off, the working fluid in the flow passage is compressed for a longer time, so the pressure at the point 3 is higher. Because the modification angle of Model 5 is larger than that of Model 3, the crank angle when the inner suction passage connects with the outer suction passage is larger and is 330 degree in Figure 5(b). At this moment, since the suction chamber 2 has already closed, only the fluid in the inner suction passage flows back to the outer suction passage, which can be indicated by the pressure of point 4. Therefore, the backflow process is shorter and its influence on the suction flow is lower when the modification angle increases.

3.3. Gas forces on the orbiting scroll

![Fig. 7 Variations of the gas forces on the orbiting scroll of different geometric models](image)

The gas forces on the orbiting scroll of the scroll compressor consist of tangential force $F_t$ and radial force $F_r$ in this paper. These values of forces can be obtained from the CFD results. Fig. 7 shows the gas forces on the orbiting scroll for different geometric models of the optimization schemes. Compared with the time-averaged tangential force in original model, it increases by 1.41%, 4.56% and 10.16% in Model 1, Model 3 and Model 5, respectively. The radial force for these three models increases by 0.75%, 2.77% and 4.62% comparing with the original model. The profiles of the tangential force and radial force for these models show the same crank angle at the maximum value, as shown in Fig. 7 Suction Chamber Structure. The forces in geometric Model 1 are approximately same with the original one. The tangential force and radial force in Model 5 are the highest. One reason for the increase of the force is that the pressure in working chamber increases as the suction mass flow rate increases. Another reason is that the asymmetric pressure in a pair of working chambers causes extra force on the orbiting scroll when the two suction chambers inhales gas differently.

4. Conclusions

In order to improve the suction performance of scroll compressor, five modification models are proposed by reducing or enlarging the area of the suction flow passage based on a scroll refrigeration compressor. The CFD simulation model is established to calculate the flow field and the performance of every model. The results are compared and analysed, and the conclusions are summarized as follows:

1) The enlargement of suction passage will reduce the pulsation of the suction mass flow rate and improve the volumetric efficiency, but the improvement is limited.
(2) The shrink of suction passage will improve the mass flow rate effectively, but the mass flow rate pulsation also increases.

(3) When the modification angle is larger than 270°, the volumetric efficiency of the modified model increases with the angle and the mass flow rate pulsation decreases with the angle. And the Model 5 has the maximum volumetric efficiency and lower flow pulsation than that of the Model 3 and Model 4.

(4) As the modification angle increases, the volume of inner suction passage and the back flow rate from the inner suction passage decrease and less fluid from suction chamber 2 flows back to the suction port.

(5) The radial force and the tangential force in Model 5 are the highest because of the increase of the suction mass flow rate and the pressure difference between the symmetrical working chambers.

Acknowledgments

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