Numerical analysis of the transient flow in a scroll refrigeration compressor

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Abstract. In the present paper, the CFD technology is adopted to simulate the working process of a scroll refrigeration compressor with R22 as working fluid. The structural grids in the scroll compressor were updated continually during the solving process to cope with the movement boundaries of the fluid domain. The radial meshing clearance was 0.008 mm which was the same with that in the real prototype. The pressure, velocity and temperature distribution in chambers of compressor were computed. Also, the transient mass flux diagrams were calculated out. The results indicated that the pressure was asymmetrical in the two symmetrical suction chambers, because the suction port and passage were not absolutely symmetrical. The gradient of temperature was great in each working chamber due to leakage flow. Velocity vector distribution was asymmetrical in each pair of working chamber owing to the movement of orbiting scroll; the flow was complicated in the central working chamber. The movement of the orbiting scroll had different influence on the vortexes formation in each pair of compression chamber. The inlet and outlet mass flux fluctuated with the crank angle obviously. Because of the ‘cut-off’ of the refrigeration fluid in the suction chamber when the crank angle was larger than 220°, the inlet mass flux decreased remarkably. Finally, some useful advices were given to improve the performance of the scroll refrigeration compressor.

1. Introduction

Scroll compressors were widely used in small refrigeration systems and air conditioners, and it has become one of the most common used refrigeration compressors. The investigation of scroll compressors was essential to improve its performance.

In recent years, computational fluid dynamics (CFD) technology has gradually become a reliable means to analyze working process of the scroll refrigeration compressor. Cui [1] simulated the three dimensional transient flow in a scroll refrigeration compressor and found that the mass, velocity, pressure, and temperature were asymmetric in each pair of working chamber. Furthermore, the CFD method has been used to simulate the suction process [2] and the discharge process [3] to study the fundamental mechanism of them, and supposed the optimum proposal. Wang et al. [4] designed a multiphase scroll pump and analyzed the effects of gas volume fraction and rotation speed on the performance of scroll pump using the CFD method. Yue et al. [5] established the transient model of the scroll vacuum pump on the basis of dynamic mesh technology with the fluent code; three cases including under-compression, slight over-compression and over-compression were calculated and compared with the experimental results; the maximum error reached 30%. Hesse and Andres [6] used...
the Twinmesh and CFX to simulate the scroll vane pump with leakage transiently; the results indicated that the simulation can capture the working mechanism and flow conditions in the dry scroll vacuum pump. Ding et al. [7] had done a CFD simulation of an oil-flooded scroll compressor with VOF model and compared the simulation results with the experimental results, and proposed that the method is robust. Chang et al. [8] employed a CFD method for investigating the thermal-hydraulic behavior of the scroll type expanders; the unstructured grid was used and the unbalance pressure distribution had been found in the expanding process. Picavet and Angel [9] studied the scroll compressor with two intermediate discharge valves using CFD method. The flow field of the scroll chamber and the flow domain of the valve had been displayed and analyzed. Liu et al. [10] carried out a numerical simulation of the whole compression process in scroll compressor. The simulated mass flow rate and volumetric efficiency agreed well with the experimental ones. The forces were also calculated with the flow field data, and the non-uniform distribution of pressure caused the fluctuation of gas forces. Sun et al. [11, 12] simulated the unsteady flow of the scroll pump with dynamic mesh model. The results indicated that there was a high speed jet at the meshing clearance and the cavitation generated at negative pressure region in the downstream area.

Although many researchers have investigated the working process of the scroll compressor with the help of CFD methods and obtained some achievements, few researchers study the inner flow characteristics and mechanism in a scroll compressor thoroughly. In the present paper, the radial clearance is set to be 0.008 mm, which is same with that in the actual prototype. The inner flow characteristics and mechanism in a scroll refrigeration compressor were displayed and analyzed in detail. Eventually, some useful advices are given to instruct the optimization and improve the performance of scroll refrigeration compressors.

2. Methodology

2.1. Physical model and grids generation

| 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    |

Table 1 shows the main parameters of the scroll refrigeration compressor. As shown in Figure 1, flow domain includes inlet, working chambers and outlet. The axial clearance was neglected because of the application of the sealing strip the axial flexible adjusting mechanism in the actual scroll compressor prototype. The radial clearance was set to be 0.008 mm. The tangential leakage caused by the radial clearance had a great influence on the performance of scroll refrigeration compressor. Figure 2 shows the radial clearance and tangential leakage in the clearance.

![Figure 1: The flow domain of scroll compressor](image1)

![Figure 2: Radial clearance and tangential leakage](image2)
The hexahedral structured grids were generated in the compressing domain of scroll refrigeration compressor. The Cartesian grids were used in other domains [13]. Figure 3 and Figure 4 show the grids of the scroll refrigeration compressor. The unstructured grids were commonly used in the numerical analysis of the existing literatures. It was difficult to guarantee the orthogonality of unstructured grids in the radial clearance when the orbiting scroll moved. The convergence and accuracy of model were poor. In this paper, because the hexahedral structural grids were generated and updated as the orbiting scroll moved, there were many layers in the radial clearance and the number of layers remained unchanged during the solved process of the model.

![Overall grids](image)

![Grids at radial clearance](image)

2.2. Boundary conditions and calculated case

| Table 2. Boundary conditions |
|-------------------------------|
| Parameter                     | Option                      |
| Working fluid                 | R22                         |
| Rotational speed/(r/min)      | 2880                        |
| Conditions of sidewall        | Adiabatic/ No-slip          |
| Rotational direction          | Clockwise                   |
| Suction pressure/MPa          | 0.627                       |
| Suction temperature/K         | 307.7                       |
| Discharge pressure/MPa        | 2.146                       |

Table 2 is the operational conditions and model setting. The turbulence model was standard k-ε model and the standard wall function method was adopted in the simulation. The time step was $5.787 \times 10^{-5}$ s when the crank shaft rotated 1°, and the crank angle was defined as θ. Furthermore, Pressure inlet and pressure outlet were adopted for the pressure boundary conditions. Inlet specified temperature and outlet fully developed were adopted for the heat inlet and outlet conditions.

According to table 2, eight groups of grids were adopted as the grid independence check, from 1000,000 to 660,000, respectively. Based on the consideration of calculation, performance of the computer and the computing time, the total number of cells was 300,000 with 11 layers in the radial direction and 20 layers in the direction of height.

3. Simulation and discussion

In this section, on the basics of the simulated results, the inner flow characteristics and mechanism in the scroll refrigeration scroll would be studied in detail. To analyze the result of numerical simulation, the crank angle θ was defined as 0° when suction process began.

3.1. Pressure distribution
Figure 5. Pressure distribution with crank angle on a cross section area perpendicular to the z axis. The asymmetric phenomena were obvious when the crank angle was 90°. The pressure in the working chamber was almost homogeneous except at the meshing clearance. In flow domains of the radial clearance, the gradient of pressure was great, because the refrigerant fluid leaked from high pressure chamber to the low pressure chamber through the radial clearance.

3.2. Temperature distribution
Figure 6 shows the temperature distribution with crank angle on a cross section area perpendicular to the z axis. In every pair working chamber, the temperature distribution was inhomogeneous. The gradient of temperature was great in a working chamber. The temperature of fluid near the radial clearance was higher than that in other flow domain in a working chamber. The reason for that was the leakage flow brought a large amount of heat from the high pressure chamber.

Comparing the Figure 5 with 6, it was found that the pressure in the working chamber was almost homogeneous while the temperature in the working chamber was inhomogeneous. The reason was that the velocity of heat conduction was lower than that of pressure transmission.

Figure 6. Temperature distribution with crank angle on a cross section area perpendicular to the z axis
3.3. Velocity vector distribution

Figure 7. Velocity vector distribution with crank angle on a cross section area perpendicular to the z axis.

Figure 7 shows velocity vector distribution with crank angle on a cross section area perpendicular to the z axis. To display the velocity vector distribution in different flow domains preferably, the velocity was set as the variable and the upper limit of velocity was set to be 40 m/s. However, the maximum velocity was close to 200 m/s at the radial clearance in the simulation. In the section, the velocity vector distribution during the suction process, compression process and discharge process were analysed in detail, respectively.

3.3.1. Suction process
As Figure 7(a) shows, a small amount of fluid leaked out from the suction chambers because of the pre-compression of the refrigerant fluid. The leakage flow mixed with the suction flow, so two vortexes formed at the exit of the suction chamber.

As Figure 7(c) and (d) shows, the refrigerant fluid in suction chamber was cut off owing to the movement of orbiting scroll. As a result, a part of refrigerant fluid flowed back, and the other part of refrigerant fluid flowed into the inner suction chamber. Predictably, the inlet mass flux of the scroll compressor would decrease when the crank angle was larger than 220° and the volumetric efficiency would decrease owing to the ‘cut-off’ of the refrigerant fluid.

3.3.2. Compression process
As Figure 7(f) shows, the compression chamber 1 and the compression chamber 2 were a pair of working chamber. Meanwhile, the vortex 1 formed in the compression chamber 1 was larger than the vortex 2 formed in the compression chamber 2. It was because the movement of the orbiting scroll contributed to the vortex formation in the compression chamber 1 and restricted to the vortex formation in compression chamber 2.

3.3.3. Discharge process
The flow regime in the central working chamber was complicated all the time. Figure 7(a) and (b) show that there were three vortexes in the central working chamber from 0° to 150°. With the movement of orbiting scroll, the refrigerant fluid in the central working chamber was squeezed, so the vortex at the tip of the orbiting scroll moved to the outward and mixed with another vortex. As a result, as Figure 7(c)-(e) show, there were two vortexes in the central working chamber from 220° to 320°.

4. Mass flux curve and discussion

Figure 8 shows the variation of mass flux with the crank angle in one cycle. The inlet and outlet mass flux fluctuated with the crank angle remarkably. The inlet and outlet integral average mass flux were 0.0866 kg/s and 0.0876 kg/s, respectively. The relative error was less than 1.2%; hence, it could be regarded as mass conservation during the simulation.

For the inlet mass flux curve, the maximum value happens at 25°, and the minimum value exists at 290°. When the crank angle was 220°, the inlet mass flux began to decrease. As shown in Figure 8, the suction volume in the suction chamber usually increased with the crank angle from 0° to 310° and then decreased, and so the mass flux was expected to increase with the increment of the suction chamber volume and decrease with the reduction of the volume. However, the inlet mass flux curve showed different trends. The influence of the movement of the orbiting scroll and flow passage on the suction
process was the main reason for the difference. When the crank angle was 220°, the ‘cut-off’ caused by the movement of the orbiting scroll prevented the refrigerant fluid flowing into the suction chamber, so the inlet mass flux began to decrease and reached the minimum value at 290°.

As shown in Figure 8, the outlet mass flux decreased in general from 0° to 111.5°. When the crank angle was 111.5°, the central working chamber connected with the near working chamber and the discharge process began. However, at the moment, because the scroll refrigeration compressor was under-compression under the current operation condition, the backflow formed and the minus outlet mass flux reached a minimum at 135°. After 135°, the pressure in the central working chamber increased gradually with decrement of the discharge volume, and reached the discharge pressure when the crank angle was about 150°. Therefore the backflow phenomenon disappeared after this moment. Subsequently, the volume of the central working chamber decreased because of the squeezing of the orbiting scroll, so the pressure in the central working chamber increased. Therefore, the outlet mass flux increased from 150° to 240° and reached the maximum at 240°.

5. Efficiency
Volumetric efficiency and isentropic efficiency are two key performance parameters in a scroll refrigeration compressor. The volumetric efficiency is defined as:

\[ \eta_v = \frac{\dot{m}_{ac}}{\dot{m}_{id}} \]

in which: \( \dot{m}_{ac} \) is the actual mass flux, \( \dot{m}_{id} \) is the ideal mass flux. The isentropic efficiency is defined as:

\[ \eta_i = \frac{p_i}{p_{ad}} \]

in which: \( p_i \) is the isentropic theory compression work, \( p_{ad} \) is the adiabatic indicated work. \( \eta_v \) and \( \eta_i \) can be calculated by the simulated results, which were 94.35% and 80.57%, respectively. Because the axial clearance was neglected in the simulated model, the volumetric efficiency was slightly higher.

![Figure 9. Volumetric efficiency with radial clearance value](image)

To investigate the influence of the radial clearance value on the volumetric efficiency and the isentropic efficiency, other four groups of radial clearance value had also been simulated, which were from 0.012 mm to 0.030 mm. Figure 9 shows the volumetric efficiency and the isentropic efficiency with the radial clearance value. It could be found that the volumetric efficiency and the isentropic
efficiency decreased with the increasing of the radial clearance value. When the radial clearance value increased from 0.008 mm to 0.030 mm, the volumetric efficiency decreased from 94.35% to 86.69% and the isentropic efficiency decreased from 80.57% to 70.13%.

6. Conclusion
In present paper, on the basics of the simulated results, the inner flow characteristics and mechanism in a scroll refrigeration compressor were studied in detail. Some conclusions are summarized as follows:

1. Pressure distribution was asymmetrical in the two symmetrical suction chambers. The pressure in the working chamber was almost homogeneous except at the meshing clearance. In a working chamber, the temperature distribution was inhomogeneous because of the leakage.

2. The inlet and outlet mass flux fluctuated with the crank angle remarkably. Because of the “cut-off” in the suction flow passage, the inlet mass flux decreased remarkably when the crank angle was larger than 220°. Hence, it was essential to optimize the suction port and flow passage to improve the volumetric efficiency. When the discharge process began, the backflow formed and the outlet mass flux reached a minimum value at 135°. A check valve should be installed to prevent the backflow.

3. Velocity vector distribution was asymmetrical in each pair of working chamber owing to the movement of orbiting scroll, the flow conditions was complicated in the central working chamber all the time. In addition, the movement of the orbiting scroll contributed to the vortex formation in the compression chamber1 and restricted the vortex formation in compression chamber 2.

4. When the radial clearance value increased from 0.008 mm to 0.030 mm, the volumetric efficiency decreased from 94.35% to 86.69% and the isentropic efficiency decreased from 80.57% to 70.13%. It indicated that the value of the radial installation clearance had a great influence on the volumetric efficiency and the isentropic efficiency of the scroll refrigeration compressor.

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