Computer Simulation of Oil-free Scroll Compressor Compression Process

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Abstract

In this paper, the numerical simulation of the three-dimensional transient flow field in a scroll compressor is performed by the CFD dynamic mesh technique. During the procedure of the simulation, the air in the scroll compressor is treated as ideal gas, and meets the flow governing equations and gas state equation. In our approach, RNG k-ε model is used for turbulent model. The simulation results show that the flow field in the scroll compressor is time-periodic, in addition, the generation, motion and size change of the vortices inside the scroll compressor can be visualized. The opening and closing of the discharge port have significant influence on the flow loss. The simulation not only extremely vividly reveals the scroll compressor inherent laws but also provides theoretical references for the optimal design of scroll compressors.

Keywords: Scroll Compressor, Dynamic Mesh, Flow Field, Numerical Simulation

1. Introduction

Compared with piston compressors, scroll compressors currently have obviously competitive advantages in the air conditioning field with its high efficiency and low noise. The flow and heat transfer process of scroll compressors is very complex, and a lot of scholars have done multitude studies over the past years. The thermodynamic process were analyzed and the mathematical model of scroll compressors were established[1][2][3]. Nevertheless, it’s difficult to meet the general laws of scroll compressor flow field by these simplified models. With the rapid development of computer technology, computational fluid dynamics is used to study the flow field of compressors, which can get detailed flow field information and reveal potential physical process.

In the field of positive displacement compressors, many achievements have been achieved by using CFD technology. Yue Xiangji et al applied dynamic mesh technique in the analysis of rotary compressors transient flow and displayed visualized flow field in compression chamber[4]. Ma Yitai et al utilized dynamic mesh technique to simulate the transient flow field in rolling rotor expander[5][6]. Geng et al employed STAR-CD software to simulate the rolling rotor compressor internal flow field, and found a way to improve the compressor overall efficiency[7]. Kovacevic et al studied the screw compressor compression chamber mesh generation technology in detail, using CFD software to calculate the three-dimensional flow parameters in screw compressor compression chambers[8][9].

CFD rarely applied in the field of scroll compressors and the researchers are few in number. Most of conclusions have not been fully validated. S.Pietrowicz et al made a two-dimensional scroll compression chamber simulation using commercial mathematical software[10], Ooi simulated the internal flow field of a scroll compressor, and analyzed thermal conductivity in the scroll compressor [11].

In this dissertation, the thermodynamic process in scroll compressor is analyzed, and scroll compressor thermodynamic simulation models are established. Dynamic mesh technique is applied in the three-dimensional transient flow field numerical simulation of a scroll compressor. The method of dynamic mesh is described in detail. The flow field is shown in the case of different crank angle. At last, the simulation is verified by experiment.

2. Theoretical models

2.1. Physical models
Scroll compressors are the gas compression machinery by means of the volume change. After fixed scroll and orbiting scroll has been assembled, a number of closed crescent-shaped volumes can be formed. As shown in figure 1.

When the eccentric shaft promotes the orbiting scroll to round fixed scroll center, the volume of compression chambers expands or shrinks, thereby the suction, compression and exhaust of gas are realized.

2.2. Foundation of thermodynamic models

2.2.1. Assumptions

The study object of the thermodynamic system is the gas in compression chambers. The fixed scroll and orbiting scroll is the system boundary of the calculation domain. During the compression process, the gas in enclosed volume may leak because of the gap between the moving parts. Meanwhile, heat transfer phenomena occur between the compressed gas and scroll walls. The scroll compressor is oil-free and the analyzed object of the thermodynamic process is pure gas. Some assumptions are made: 1) There is identical temperature and pressure in each point of compression chambers, 2) The air in compressors is ideal gas and the internal energy is only related to temperature.

2.2.2. The mass equation

The mass conservation law is adopted. The quality difference between the mass into the control volume and the mass out of the control volume denotes as the increment mass of the control volume. As shown in the following formula:

\[ dm = dm_i - dm_o \]  

Where, \( dm \) is the quality differential in compression chambers and \( dm_i \) is the differential of the mass into compression chambers and \( dm_o \) is the differential of the mass out of compression chambers.

2.2.3. The energy conservation equation

Based on the foregoing assumptions, at a particular time \( t \), the gas state in the control volume accord the first law of thermodynamics can be expressed as follow:

\[ dU = dQ + dH_i - dH_o - dW \]  

Where, \( dU \) is the increment of the gas internal energy and \( dQ \) is heat exchange amount and \( dH_i \) is the enthalpy of the gas into the control volume and \( dH_o \) is the enthalpy of the gas out of the control volume and \( dW \) is the work done by eccentric shaft.
On the base of thermodynamic theory, the following formula can be derived by combining the formula (1) and (2):

\[
\sum \sum \frac{dQ}{dt} = \frac{1}{mC_v} \frac{dT}{dt} + \frac{T}{m} \frac{dm}{dt} + \frac{1}{mC_v} \sum C_v \frac{dT}{dt} - \frac{1}{mC_v} \sum C_v \frac{dm}{dt} - \frac{R_C^2 \frac{dT}{dt}}{V C_v} \frac{dV}{dt}
\]  

(3)

The calculation formula of \(dQ\) and \(dm\), \(dm_o\) will be deduced behind.

2.2.4. The volume model of scroll compressors

Without consideration of the center chamber, the formula of the chamber volume can be shown as follow:

\[
V_i = ((2i - 1) - \frac{\theta}{\pi}) \pi p (p - 2b) h \quad (i \geq 2)
\]  

(4)

Where, \(V_i\) is the volume of each compression chamber and \(i\) is the number of compression chamber and \(\theta\) is crank angle and \(p\) is scroll pitch and \(b\) is the thickness of scroll wall and \(h\) is the height of scroll wall.

In order to gain the volume change rate, the formula (4) is differentiated:

\[
\frac{dV_i}{d\theta} = -p(p - 2b) h
\]  

(5)

The formula (5) is the law of the volume change rate. Seen from the above equation, the relationship of volume and angle is linear.

2.2.5. The leakage models

The leakage models of scroll compressors include tangential leakage models and radial leakage models. The tangential leakage models can be expressed as follow:

\[
\frac{dm_t}{d\theta} = \alpha_t \rho \frac{c_t h}{\omega} \sqrt{2(h_i - h_2)}
\]  

(6)

Where, \(m_{tq}\) is the tangential gas leakage mass and \(\omega\) is the angular velocity of orbiting scroll and \(\rho_t\) is high-pressure chamber gas density and \(c_t\) is the average radial clearance and \(\alpha_t\) is radial flow coefficient, related to the form of flow channel. \(h_i\) and \(h_2\) is the enthalpy of high-pressure gas and low-pressure gas.

The radial leakage models can be expressed as follow:

\[
\frac{dm_r}{d\theta} = \alpha_t \rho \frac{c_r L}{\omega} \sqrt{2(h_i - h_2)}
\]  

(7)

Where, \(m_{rj}\) is the radial gas leakage mass and \(c_r\) is the average axial clearance and \(\alpha_t\) is axial flow coefficient and \(L\) is the length of leakage line.

2.2.6. Heat transfer model

The heat transfer model between air and scrolls is generally considered as convective heat transfer, the model denotes as follow:

\[
Q = \alpha A (T_w - T)
\]  

(8)

Where, \(\alpha\) is the heat transfer area and \(T_w\) is the temperature of scroll wall and \(T\) is the temperature of air and \(\alpha\) is the convective heat transfer coefficient, which can be calculated by Dittus-Boelter equation for smooth tube turbulence.

2.3. Control equations

The summation of the time-averaged flow and the instantaneous pulsating flow may express turbulent flow, namely time average method:

\[
u = \bar{u} + u', \nu = \bar{v} + v', w = \bar{w} + w', p = \bar{p} + p'
\]  

(9)

The fluid control equation using formula (9) can be expressed as follow:

\[
\text{div} \bar{u} = 0
\]  

(10)
Formula (10) is mass conservation equation and formula (11)–(13) are momentum conservation equations. Where, \( \overline{\mathbf{u}} \) is time-averaged velocity vector. \( \overline{\mathbf{u}} \), \( \overline{\mathbf{v}} \), \( \overline{\mathbf{w}} \) is x, y, z direction component of \( \overline{\mathbf{u}} \). \( \overline{u} \), \( \overline{v} \), \( \overline{w} \) is x, y, z direction component of fluctuating velocity vector. \( \mu \) is dynamic viscosity and \( \rho \) is the density of ideal-gas and \( \overline{p} \) is the mean pressure and \( p' \) is fluctuating pressure.

Because there is internal swirling flow in scroll compression chambers during the rotation of the orbiting scroll, RNG k-\( \varepsilon \) turbulence model is selected. Equation k and equation \( \varepsilon \) can be expressed as follows:

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho u_k)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \alpha_i \mu_{ij} \frac{\partial k}{\partial x_j} \right] + G_k + \rho e
\]

(14)

\[
\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho u_\varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \alpha_i \mu_{ij} \frac{\varepsilon}{k} \right] + \frac{C_{\varepsilon}}{k} G_k + \frac{\varepsilon^2}{k}
\]

(15)

Where, \( k \) is turbulent kinetic energy and \( \varepsilon \) is kinetic energy dissipation rate and \( \mu \) is turbulent viscosity and \( \mu_{ij} \) is the correction value to turbulent viscosity in RNG k-\( \varepsilon \) turbulence model.

In the low Reynolds number flow region near the wall, wall function method can solve the flow problems of the near-wall region, which can be expressed by the following equation:

\[
\mu^* = \frac{1}{k} \ln (E y^+), \quad y^+ = \frac{\Delta y_p C_{\varepsilon}^{1/4} k_p^{1/2}}{\mu}
\]

(16)

Where, \( y^+ \) and \( \mu^* \) is the dimensionless parameter which denotes distance and velocity. \( \Delta y_p \) is the distance between the node and wall. \( k_p \) is the turbulent kinetic energy of the node p and E is the constant about wall roughness.

3. Solution method

3.1. Thermodynamic calculation

As shown in the flowchart, compressor dimension parameters and initial state parameters are input. Because future value is used in the iteration of Runge-Kutta method, the temperature of entire compression process is calculated using adiabatic process models for initial calculation. In simulation crank angle \( \theta = 0 \) corresponds to the compression chamber just completed suction, and crank angle \( \theta = \theta_{out} \) corresponds to the compression chamber exhausting gas upcoming.
The calculated models are composed of the equation (1), (3) and (5). Formula (6) and (7) are substituted into equation (3). Owing to the unknown of temperature, outlet temperature must be iterated. That the outlet temperature difference between two iterations is very small can be considered the end of iteration.

3.2. Meshing

The computational domain is the fluid domain in scroll compressor. The shape of compressor and exhaust pipe is simplified in order to facilitate calculation. Meshing is completed in commercial software as shown in figure 3(a). There are about 280,000 grid cells. The computational domains consist of the compressor domain and the exhaust pipe domain. The computational domain of the compressor is made up of prism grid cell. The computational domain of the exhaust pipe is composed of hexahedral grid cell. Two computational domains are related by interface which ensures the correction of calculation when orbiting scroll reach exhaust port.

![Meshes of Scroll Compressor](image)

As shown in figure 3, triangular mesh can be found in front view, which may change to prismatic grid after longitudinal tensile. The prismatic grid guarantees the correct application about the dynamic method of spring smoothing and local remeshing.

3.3. CFD calculation

Dynamic method is needed due to the shape of compression chambers changes following the change of crank angle. The orbiting scroll can receive the move law through the macro DEFINE_CG_MOTION provided by FLUENT. The z coordinate remain unchanged when meshes move. When the x and y coordinate of grid changes, the dynamic mesh method combined spring smoothing and local remeshing is used in order to improve the quality of grid after deform.

4. Simulation results and analysis

4.1. Results of thermodynamic simulation

An oil-free air scroll compressor produced by a compressor company in Jiangxi province is adopted. The parameters of this scroll compressor are as follows: the radius of basic circle is 3.1 mm, and the thickness of scroll wall is 4.4 mm, and the speed of orbiting scroll is 2800 r/min. The simulation parameters are as follows: Constant volume specific heat is 0.716, and the constant pressure specific heat is 1, and the adiabatic index is 1.4. The distance of simulation step is π/180, and the iteration number is 50. The fixed-order Runge-Kutta method is used for solving differential equations. The results of volume, pressure and temperature changes are shown in Figure 4 (a), (b), (c) below.
Figure 4. Results of Thermodynamics Simulation

Figure 4(a) is the compression chamber volume change excluding the central compression chamber, which is linear. Figure 4(b) signifies the pressure change in compression chambers. With the orbiting scroll rotating, the pressure in compression chambers is rising continuously. Figure 4(c) indicates the temperature change in the compression chambers. Since it is an adiabatic process, with the orbiting scroll rotating, the temperature in compression chambers increases significantly.

4.2. CFD results

The finite volume method is employed in the whole process. The PRESTO! method is adopted to calculate pressure term and the SIMPLC algorithm is utilized to solve the pressure-velocity coupling equation, respectively. CFD results can be obtained as follows.

Figure 5. Distribution of Pressure

The pressure in compression chambers during suction, compression, exhaust are shown as figure 5. With narrowing of compression chambers, gas pressure is gradually increased. Maximum average pressure appeared in the center compression chamber, which meets the theoretical laws of pressure change. The maximum pressure appeared in place of scrolls engagement for the greatest speed change.

Figure 6. Speed Field

Figure 7. Flow Process during Exhausting

Figure 6 demonstrates the velocity field of the scroll compressor. The maximum velocity appeared in the position of scrolls engagement. The air flow from the chambers near the center to the chambers faraway the center. Figure 7 displays the flow process in the exhaust port. As the volume decrease in the center of the compression chamber, the gas flows from center chamber to exhaust pipe. The area of
exhaust port changes with the movement of orbiting scroll. As a result, the flow loss is affected significantly by the movement.

![Velocity Field in Center Compression Chamber](image)

**Figure 8. Velocity Field in Center Compression Chamber**

The velocity field in center compression chamber is shown in figure 8(a). With the movement of the orbiting scroll, vortex flow may be observed in center chamber, meanwhile, the phenomenon that the large vortices is broken up to small vortices may be observed too. The gas in center compression chamber flows rapidly into the exhaust port when the orbiting scroll reaches the position in figure 8(b).

### 4.3. Experimental verification

The scroll compressors are tested on the experiment equipment as displayed in figure 9. The compare between actual value and simulation value is shown in table 1. The simulation value meets the actual value well and the results of simulation are reasonable.

![Laboratory Equipment](image)

**Figure 9. Laboratory Equipment**

<table>
<thead>
<tr>
<th>Units</th>
<th>Flow rate</th>
<th>Pressure</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>simulation value</td>
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<td>0.75</td>
<td>195</td>
</tr>
<tr>
<td>actual value 1</td>
<td>4.33</td>
<td>0.66</td>
<td>187</td>
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<tr>
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<td>0.77</td>
<td>194</td>
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<td>0.65</td>
<td>183</td>
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<tr>
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<td>4.49</td>
<td>0.78</td>
<td>198</td>
</tr>
<tr>
<td>actual value 6</td>
<td>4.12</td>
<td>0.66</td>
<td>184</td>
</tr>
<tr>
<td>actual value 7</td>
<td>4.18</td>
<td>0.70</td>
<td>192</td>
</tr>
</tbody>
</table>

**Table 1. Actual Value and Simulation Value**

### 5. Conclusion

In the paper, the mathematical models suitable for oil-free scroll air compressors are established in the light of thermodynamic theory. The simulation is carried out on the comprehensive consideration about the heat transfer and leaks. The results demonstrate that the value from simulation coincides with actual value.

The three-dimensional flow field simulation model of scroll compressors is established reasonably. The problem of orbiting scroll movement in CFD is solved. The simple and reasonable boundary conditions are identified. The simulation of three-dimensional flow field is realized using RNG k-ε turbulence model. The simulation results include intuitive and rich flow field information. Vortex generation, movement, rupture process can be observed obviously. Those are much better than the theoretical analysis and experimental researches.
The simulation can be used to predict the thermal performance of oil-free scroll compressor, and improve the scroll compressor design. It can be taken as an important theoretical foundation for the further optimization of oil-free scroll compressors.

6. References


