Thermodynamic Model and Experimental Study of Oil-free Scroll Compressor

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Abstract. In order to study the performance characteristics of oil-free scroll compressor, this paper is based on the basic equation of circle involute profile, and uses the differential geometry theory to calculate the variation law of pressure with volume. Based on the basic law of thermodynamics, the thermodynamic model of the oil-free scroll compressor is established by considering the heat transfer model and the gas leakage model, considering the mass, energy conservation equation and gas state equation. The change of the mass flow rate of the gas in each chamber is obtained by solving the established model by using the improved Euler method. The experiment results show that with the increase of frequency, the temperature, the displacement and the power show a clear upward trend. The thermodynamic model has some guidance and reference for the development and performance analysis of oil-free scroll compressors.

1. Introduction

With the increasing global climate problems and the growing shortage of non-renewable resources, energy conservation, emission reduction, new energy development and utilization has become one of the important subjects of research in the world today. Green development will become a new concept of development, while increasing the promotion of new energy vehicles plan. And as a new energy vehicle gas source of the oil-free scroll compressor will be rapid development. Oil-free scroll compressors are widely used in refrigeration equipment, air-conditioning systems and various gas compression applications due to the advantages of light weight, small size, high volume efficiency and simple structure [1].

Study on thermodynamics model for oil-free scroll compressor, there are many scholars of the relevant research work. Zhao is famed for a fuel cell vehicle spray oil-free scroll compressor for detailed performance analysis and research [2]. Fang Shiyi ignoring gas and scroll wrap heat exchange case, calculating thermodynamic process of a new oil-free scroll compressor [3]; Le Wang designed and developed for aerospace applications of refrigeration oil-free scroll compressor, established the thermodynamic model and study the performance of the prototype [4]; Yanagisawa studied the volumetric efficiency of oil-free scroll compressors in detail [5]; Brycer. Shaffer has designed a plastic and metal material of a new oil-free scroll compressor, and the comparative study of the performance, plastic material more sensitive to temperature, but demands for precision low while costs are also relatively low [6]; Sunder through method of test research heat transfer process about fixed and
orbiting scroll, obtained the physical contact is important form of heat transfer in compression process [7]; Poi and Zhu through Two-dimension model to research two scroll plate heat transfer model [8]; Yi et al simulation the compression process of a car air-conditioning scroll compressor and verify the model by experiment [9, 10];

Based on the geometric model of oil-free scroll compressor, the first law of thermodynamic, conservation of mass and gas equation of State, considering the working chamber leakage model and heat transfer model, established the thermodynamic model of the prototype, and the thermodynamic model was validated by the test platform. The calculated results provide a theoretical basis and reference value for the performance analysis and structural optimization of oil-free scroll compressor.

2. Geometric model

The geometrical model of the oil-free scroll compressor is the basis for the analysis and study of the thermodynamic model, so it is necessary to study the geometric model of the scroll compressor.

Figure 1 shows the development of a new type of oil-free scroll compressor prototype structure.

![Figure 1. Oil-free scroll compressor.](image)

1-Fixed scroll; 2-Orbiting scroll; 3-Frame; 4-Small crank; 5-Spindle; 6-Leather belt wheel and Small balance iron; 7-Large balance iron;

The circle involute equation is used to deduce the spiral equation of the scroll plate, the circle involute baseline equation:

\[
\begin{align*}
x &= r_\gamma \cos \varphi + r_\gamma \sin \varphi \\
y &= r_\gamma \sin \varphi - r_\gamma \cos \varphi 
\end{align*}
\]  

After the baseline is determined, the inner and outer profiles of the scroll are determined according to the characteristics of the normal equidistant curve.

The involute equation of inner circle:

\[
\begin{align*}
x &= r_\alpha \cos (\varphi + \alpha) + \varphi \sin (\varphi + \alpha) \\
y &= r_\alpha \sin (\varphi + \alpha) - \varphi \cos (\varphi + \alpha)
\end{align*}
\]  

The involute equation of outer circle:
\[
x = r_b \left[ \cos(\varphi - \alpha) + \varphi \sin(\varphi - \alpha) \right] \\
y = r_b \left[ \sin(\varphi - \alpha) - \varphi \cos(\varphi - \alpha) \right]
\]  
(3)

According to the above-mentioned scroll profile equation, combined with the geometric parameters listed in Table 1, the use of the baseline method to generate scroll inner and outer profile shown in Figure 2.

| Name                      | Symbol | Illustration               |
|---------------------------|--------|----------------------------|
| Type of spiral            |        | Circle involute            |
| Base circle radius        | \( r_b \) | 3.675mm                    |
| Offset distance           | \( R_o \) | 5.745mm                    |
| Height                    | \( h \)  | 36mm                       |
| Thickness                 | \( T \)  | 5.8mm                      |
| Ending angle              | \( \varphi_e \) | 8.49\( \pi \)              |
| Discharge angle           | \( \theta_d \) | 265°                      |
| Number of turns           | \( N \)  | 4.25                       |

3. Thermodynamics model
The thermodynamic model of oil-free scroll compressor is established to take full account of the effects of the suction heating and working chamber heat transfer as well as radial leakage and flank leakage on the thermodynamic model, the variety rule of the gas in actual working condition is obtained.

3.1. Basic equations
The change of the gas temperature in the working chamber with the rotation angle of the spindle [11]

\[
\frac{dT}{d\theta} = \frac{1}{mc_v} \left[ -T \left( \frac{\partial p}{\partial T} \right)_s \left( \frac{dV}{d\theta} \right) - \frac{V}{\omega} (m_{in} - m_{out}) \right] + \sum \frac{\dot{m}_{in}}{\omega} (h - h_f) + \frac{\dot{Q}}{\omega} 
\]

(4)

The mass of the gas in the working chamber varies with the spindle angle:

\[
\frac{dm}{d\theta} = \sum \frac{\dot{m}_{in}}{\omega} - \sum \frac{\dot{m}_{out}}{\omega}
\]

(5)

3.2. Heat transfer model.
There is a temperature difference in the environment will inevitably occur heat transfer phenomenon, the three ways of heat transfer are heat conduction, thermal convection, heat radiation. When the gas enters the suction chamber, the heat transfer process occurs in the chamber. In order to simplify the heat transfer model, neglecting the heat conduction and heat radiation, only consider the thermal convection phenomenon.

3.2.1. Suction heating. The suction process is calculated according to the convective heat transfer of the fluid in the tube, using the Dittus-Boelter equation, the surface heat transfer coefficient [12]
\[ h = 0.023 \frac{K}{d_p} Re^{0.8} Pr^{0.4} \]  \hspace{1cm} (6)

The temperature of the compressed gas entering the working chamber from the suction pocket can be calculated from equation (7):

\[ T_s = T_p - (T_p - T_{s,\infty}) \exp \left( \frac{\pi d_p L_p h}{mc_p} \right) \]  \hspace{1cm} (7)

The value of the heat transfer between the gas and the gas pipe is:

\[ \dot{Q}_m = mc_p (T_p - T_{s,\infty}) \left[ 1 - \exp \left( \frac{\pi d_p L_p h}{mc_p} \right) \right] \]  \hspace{1cm} (8)

3.2.2. Working chamber heat transfer. The average convection of the spiral plate heat exchanger is used to simulate the heat transfer process between the gas and scroll tooth, the surface heat transfer coefficient is \([13, 14]\)

\[ h = 0.023 \frac{K}{D_{ef}} Re^{0.8} Pr^{0.4} \left( 1 + 1.77 \frac{D_{ef}}{R_{\text{aver}}} \right) \times \left[ 1 + 8.48 \left[ 1 - \exp (-5.35 St) \right] \right] \]  \hspace{1cm} (9)

The equivalent diameter in this model is defined as the ratio of the working chamber volume to the scroll wrap:

\[ D_{ef} = \frac{4V}{A} \]  \hspace{1cm} (10)

The equivalent radius is defined as \([15]\):

\[ R_{\text{aver}} = R_s \left[ \frac{(\phi_k - \pi / 2) + (\phi_{k-1} - \pi / 2)}{2} \right] \]  \hspace{1cm} (11)

3.2.3. Area of heat transfer. The expression of the area of heat transfer between the scroll wrap and plate is in the range of increasing angle \(\phi_k\) to \(\phi_k - 2\pi\):

\[ dA = h r_s (\varphi - \phi_0) d\varphi \]  \hspace{1cm} (12)

The expression of the heat transfer area at the bottom of the scroll is \([16]\):

\[ dA = \frac{1}{2} r_s^2 \left[ (\varphi - \phi_b)^2 - (\varphi - \pi - \phi_0)^2 \right] d\varphi \]  \hspace{1cm} (13)

Gas and scroll tooth, the bottom of the scroll heat exchange capacity in any working chamber can be calculated by the following formula \([17]\)
\[ \dot{Q} = h \int_A [T_{\text{scroll}}(\phi) - T(k, \theta)] \, dA \]  

(14)

### 3.3. Leakage model

There are two different leakage in the scroll compressor, namely, radial leakage through the axial gap and flank leakage through the radial gap. Leakage model shown in Figure 3.

For two internal leakage of the compressor, the gap can be expressed as a high and low side pressure function, resulting in a radial leakage area [18]

\[ A_r = \delta_r L_r \]  

(15)

For the radial leakage line length, the axial clearance \( \delta_r \) can be expressed as a linear function with respect to the pressure ratio:

\[ \delta_r = 1.1 \times 10^{-6} \left( \frac{P_h}{P_l} \right) + 10^{-6} \]  

(16)

Flank leakage area:

\[ A_f = h \delta_f \]  

(17)

Similarly, the radial gap \( \delta_f \) can also be expressed as a linear function of the pressure ratio:

\[ \delta_f = -9.615 \times 10^{-5} \left( \frac{P_h}{P_l} - 1.67 \right) + 20 \times 10^{-6} \]  

(18)

The leakage of the scroll compressor can be abstracted into a nozzle model in which the mass flow of the gas in the chamber is [19]

\[ \dot{m} = \varphi A_r \sqrt{\frac{2 \rho_h}{\gamma - 1} \left[ \left( \frac{P_l}{P_h} \right)^{\frac{\gamma}{\gamma - 1}} - \left( \frac{P_l}{P_k} \right)^{\frac{\gamma + 1}{\gamma - 1}} \right]} \]  

(19)
The flow rate is limited by the critical pressure ratio of the blocked flow conditions, the pressure ratio is:

$$\frac{p_l}{p_h} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$$  \hfill (20)

According to the character of the ideal gas compressible, the mass flow of the radial leakage can be determined by the following formula [20]:

$$\frac{p - p_m}{\rho g} = \lambda \frac{Lu_m^2}{4m2g}$$  \hfill (21)

Similarly, the mass flow of flank leakage can be represented by the following formula:

$$\frac{p - p_m}{\rho g} = \int_{-\phi}^{\phi} \frac{Rd\phi u_m^2}{4m2g}$$  \hfill (22)

4. Simulation analysis
In order to describe the thermodynamic model of oil-free scroll compressor in detail, this paper establishes the thermodynamic model of oil-free scroll compression based on the basic laws of thermodynamics, the mass and energy conservation equations and the gas state equation, considering the heat transfer model and the leakage model. The leakage of the oil-free scroll compressor is analyze-
End and studied deeply. The pressure-volume curve and the mass flow of leakage between the chambers are obtained by solving established thermodynamic model. As shown in the figure 4 is the flowchart of thermodynamic model of oil-free scroll compressor.

As shown in the figure 5, the curve of pressure with the volume. In the suction process, the volume continues increase to maximum of 213918 mm$^3$, while the pressure is always maintained at a fixed value of 101.3 kPa. When the gas enters the compression chamber, the volume of the gas is further compressed so that the volume inside the chamber is gradually reduced and the pressure continues to increase. When the spindle angle reaches the exhaust angle, the gas enters the exhaust chamber. At this time, the volume of the chamber is 44885 mm$^3$, and the gas is gradually discharged from the exhaust port. As shown in the figure 6, the mass flow curve of the fourth compression chamber. The radial mass flow of the spindle in the process of one cycle is increasing, indicating that the leakage of the third compression chamber to the chamber is lesser than the chamber to the next working chamber. And the flank mass flow rate tends to rise slightly, the leakage of the upper chamber to the chamber is slightly lesser than the chamber to the next chamber.

As shown in the figure 7, the third compression chamber mass flow curve. With the continuous rotation of the spindle, the mass flow of the radial leakage is declining, indicating that the leakage of the third compression chamber to the fourth compression chamber is lesser than the second compression chamber to the third compression chamber. While the mass flow of flank leakage remain essentially unchanged. As shown in the figure 8, the second compression chamber of the mass flow
Figure 7. The mass flow in third chamber.

Figure 8. The mass flow in second chamber.

Curve. The mass flow of radial leakage is significantly higher than the fourth and third compression chambers. This is due to the continuous rotation with the spindle, the pressure of working chamber rise further, while the leakage to the third compression chamber is decreased. At this time, the mass flow of flank leakage is gradually increasing.

As shown in the figure 9, the central exhaust chamber mass flow curve. With the spindle angle increases, radial leakage to the second compression chamber leakage mass flow decreased. This is because the central exhaust chamber can only leakage downwards to the second compression chamber, resulting the mass is decreased. While the flank and radial leakage mass flow is declining until the spindle angle reaches the exhaust angle $1.472\pi$, the central chamber began to exhaust, the mass of the chamber is quickly discharged the outside. As shown in the figure 10, the total mass flow curve. The mass flow of the above-mentioned chamber in radial and flank leakage are superimposed on each other, resulting in a curve change as shown in the figure 10. The mass flow of radial leakage is much larger than the flank leakage, and the variation of mass flow in radial direction is larger than flank leakage.

Figure 9. The mass flow in center exhaust chamber.

Figure 10. The mass flow of total leakage.
5. Experimental Verification

In order to verify the correctness of the oil-free scroll compressor thermodynamic model and analyze the performance of the whole machine under different working conditions, a prototype of the new oil-free scroll compressor is developed. Figure 11(a) shows the test prototype and according to the test system schematic diagram of Figure 11(b), the test platform of oil-free scroll compressor is built and the simulation and test data are compared.

Figure 12 is in different pressure conditions, the temperature variety with the frequency. Under the same discharge pressure, with the frequency increases, the temperature gradually increased. Under the same frequency, with the discharge pressure increases, the discharge temperature also gradually increased. Figure 13 under the different pressure conditions, the displacement with the frequency. Under the same discharge pressure, with the frequency increases, the displacement gradually increased. Under the same frequency, the discharge volume is slightly reduced as the discharge pressure increases. The discharge pressure for the displacement of smaller impact, the change of displacement is little with different discharge pressure.

Figure 14 compares the simulated and tested shaft power. With the increase of rotational speed, the values of the simulated shaft power and the measured values are increasing, and the errors are small,
which verifies the correctness of the thermodynamic model. Figure 15 shows the variety in power with frequency. In the case of the same discharge pressure, with the increase of frequency, the power continues to increase. Under the same frequency, the power gradually increases with the increase of discharge pressure.

![Figure 14. Simulation and test shaft power.](image1)

![Figure 15. Power with frequency.](image2)

The figure 16 shows the variety of voltage with spindle speed under different pressure conditions. Under the same discharge pressure, with the increase of the spindle speed, the voltage increases. Under the same spindle speed, the discharge pressure increases, voltage is almost kept constant. The discharge pressure for voltage is smaller, and the voltage variation is slight with different discharge pressure. Figure 17 shows the current with frequency under different pressure conditions. At the same discharge pressure, the current decreases with increase of frequency. In the same frequency conditions, the current increases with the discharge pressure increases. The variation of the current at low frequency is large, and the variation range at high frequency is small.

![Figure 16. Voltage vs. spindle angle.](image3)

![Figure 17. Current vs. frequency.](image4)

6. Conclusion

According to the oil-free scroll compressor, the relevant thermodynamics model is established. Based on the developed prototype, the performance of the prototype is tested by constructing the test platform. The following conclusions can be drawn:
1) Based on the geometric parameters and structural dimensions of oil-free scroll compressor, the geometrical model of prototype was established and the variation curve of pressure-volume was calculated.

2) Based on the conservation of energy, the law of conservation of mass and the gas state equation, the thermodynamic model of oil-free scroll compressor is established, considering the heat transfer model and leakage model.

3) The leakage mass flow of each chamber is obtained by solving the established thermodynamic model by using the improved Euler method. Based on the developed prototype model, the thermodynamic model is tested and verified.

4) The test results show that under different discharge pressure, the temperature and power change range is larger, and variation of displacement is small. The experimental data for new oil-free scroll compressor research and performance analysis has a certain guidance and reference.

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