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Numerical study of intake flow field inside the high-speed refrigeration scroll compressor with spiral suction channels in the motor rotor

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Abstract: To control the temperature of the electric motor, the suction gas is introduced flowing through the motor in hermetic scroll compressors. For high-speed compressors, this flow resistance is high, which results in decreasing of the efficiency. In this paper, spiral flow channels were designed inside the motor rotor to decrease the suction flow resistance and increase the refrigerant flow rate inside the scroll compressor. Simulation of the suction flow through the motor was carried out. Two structure of straight and spiral intake channels were brought into the simulation model, under 3000 rpm, 5000 rpm, 7000 rpm and 9000 rpm. The results showed that the spiral channels could increase the refrigerant flow rate inside the motor rotor and the suction pressure of the compressor, and with the increase of the motor speed, the effect of the intake supercharging improved. At 9000 rpm, the suction pressure at the suction port of the compressor increased by about 4% by using spiral channels, while the mass flow rate increased by about 6%. The increases of total refrigerant flow rate in spiral channels are mainly due to the increase in unblocked channels. The size and the layout of the balance weight have a great effect on intake flow.

1. Introduction
Recently, the development of high-speed compressor has gained increased attention because of smaller size and lighter weight. The high-speed scroll compressors have been widely used in mobile platforms, such as electric vehicles [1]. High-speed scroll compressors are usually hermetic. In the hermetic compressors, it is impossible to observe the internal temperature, pressure and flow directly. Therefore, the cooling of motor is always a problem. Many scholars have reported the heat dissipation of the motor.

For compressor motor, suction cooling, a kind of air cooling, is the most widely used cooling method. The refrigerant enters the compressor, cools the motor first, and then flows into the compressor to operate. The inlet temperature of the refrigerant is lower than the motor temperature, and the refrigerant flows through the internal flow channels of the motor to reduce the motor temperature [2].

At the same time, in order to increase the heat exchange of the motor, the channels are usually set inside the motor. Huang et al. studied the axial cooling channels at the edges of housing or/and stator back, and carried out thermal analysis with rectangular, oval and elliptical flow channels [3]. He et al. developed a comprehensive model of the flow and thermal characteristics of a semi hermetic twin-screw refrigeration compressor. Straight cooling channels were set inside the stator and the rotor [4]. Tang et al. used CFD to analyze the influence of increasing rotor ventilation channel and optimizing stator cutting on motor cooling performance of a scroll compressor [5]. Yang et al. introduced a structure design of an external rotor permanent magnet synchronous torque motor, which adopted spiral magnetic pole and set cooling channels on the surface of the outer rotor and inside the stator core [6]. K. Rönnberg et al. compared different modelling approaches, and mainly focused on the stator side, where the straight cooling channels are arranged [7].

So far, the research about motor cooling only focuses on the heat dissipation effect of motor, and the research object is only the motor. However, for the motor used in the compressor, there is no report concern about the cooling structure and its impact on compressor performance. With the increase of heat transfer, the flow channels in the motor will bring resistance loss. Thanks to the high rotation speed, structures similar to turbine could be efficiently introduced to enhance the suction flow of the high-speed scroll compressor. At present, most cooling channels in the motor rotor or stator are straight. Therefore, in this paper, for high-speed scroll compressor, spiral inlet channels set in the motor rotor is proposed. In order to discuss the influence of refrigerant flow in the spiral channels on the performance of the compressor, the spiral channels with 240mm pitch and the straight channels are compared using CFD software. The simulation is carried out at 3000 rpm, 5000 rpm, 7000 rpm and 9000 rpm.

2.Motor structure and computational model

2.1. Geometric model and spiral channels of the motor

In order to increase the suction pressure of the compressor and the refrigerant flow rate of the refrigeration system, spiral rotor channels are arranged inside the motor rotor. Figure 1 shows the cross section of the internal structure of the scroll compressor. The hermetic scroll compressor mainly divided into two parts: the motor and the compressor driven by the motor. As shown in Figure 1, all the refrigerant flow through the motor by the channels in the rotor and the gap between the rotor and the stator. After cooling the motor, the refrigerant then enters the suction port of the scroll compressor. Hence, we choose the flow passage of the motor part as the calculation domain, as shown in the Figure 1. The outlet pressure of the computational domain is the suction pressure of the compressor. The performance of the compressor will be improved with the increase of the outlet pressure and mass flow rate of the computational domain.
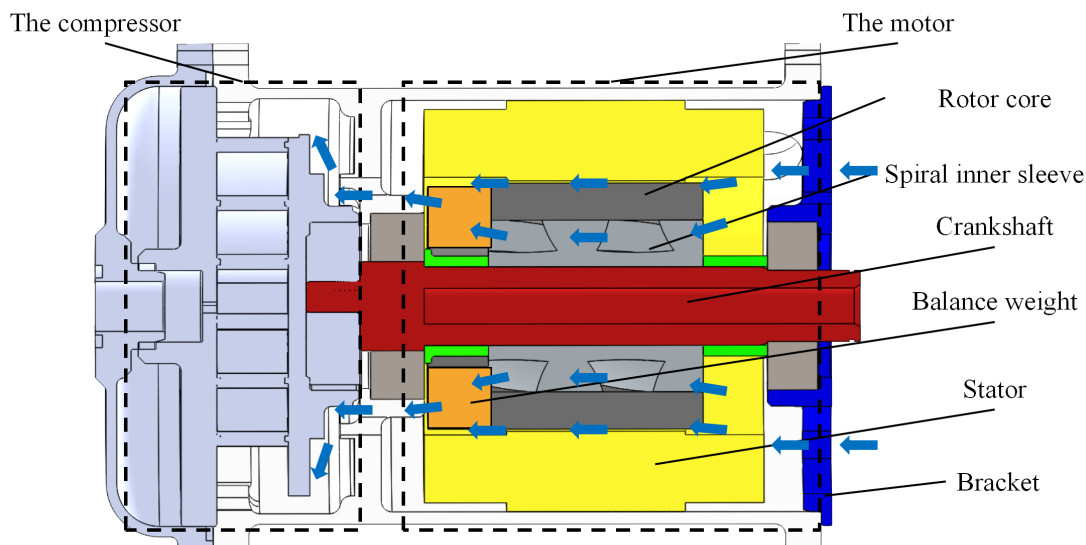
Figure 1. Cross section of internal structure of the scroll compressor.

The main structure of the motor shows in Figure 2. The motor is permanent magnet synchronous motor. The motor is composed of a crankshaft, a spiral inner sleeve, a rotor core, a balance weight and a stator. The inner rotor consists of rotor core and magnets. The balance weight is located on the rotor core. It is used to balance the rotating inertia force of the orbiting scroll and have an influence on the flow rate of the refrigerant. Accordingly, it's very important to design the size and shape of the balance weight. Axle plays an important part in the motor and supports the whole motor body, so its mechanical strength and rigidity must meet a high standard.
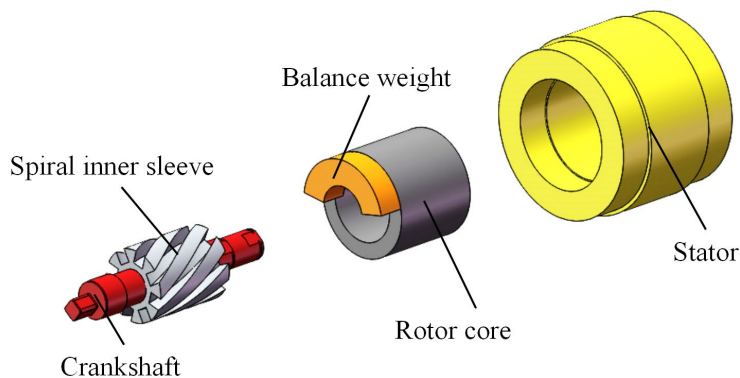
Figure 2. Main components of the motor.

As is shown in Figure 3, after the spiral inner sleeve added inside the rotor core, the refrigerant flow channels (F1-F8) are formed between the spiral inner sleeve and the rotor core. The spiral inner sleeve and the rotor core adopt interference fit to make rotation movement under the action of the crankshaft. The rotation direction of the spiral inner sleeve is set according to the rotation direction of the crankshaft. In this study the spiral inner sleeve rotates counterclockwise when viewed from the balance weight direction, as showed in Figure 3. However, due to the existence of the balance weight, some channels (F2-F4) are still blocked, so the design principle of the inner sleeve is to minimize the flow loss and ensure the maximum outlet area of channels.
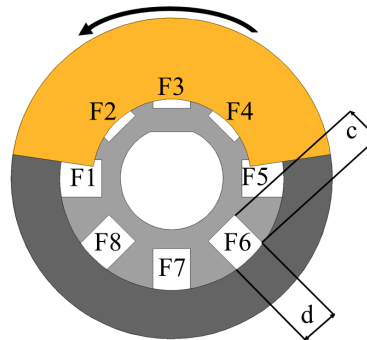
Figure 3. Rotor cooling channel
dimensions.

The radial dimension, the outside diameter and the inside diameter, and the axial length of the rotor core are decided by the structure of the motor. In this paper, the geometrical dimensions of the spiral inner sleeve we designed are shown in the Table 1 below. The outside diameter is 48 mm, the inside diameter is 22 mm, the axial length is 60 mm. Table 1 also show the d, c, the pitch and the total ventilation area, which are formed by the sleeve and the rotor core.

Table 1. The geometric dimensions of the spiral inner sleeve.

Geometric dimensions	Value
outside diameter	48 mm
inside diameter	22 mm
axial length	60 mm
d	8 mm
c	8.66 mm
pitch	240
total ventilation area	568.88 mm^2

2.2. Internal flow field model of the motor

In order to study the influence of the motor rotor channels on compressor performance, the internal runner of the motor was selected as the computational domain. The motor dashed box in Figure 1 shows the computational fluid domain. So, the calculation domain starts from the refrigerant entering the motor and ends at leaving the motor. It mainly includes the channels in the rotor core, the air gap and the gap between balance weight and crankshaft.

Figure 4 shows the computational model with straight channels, where (a) is the geometric model and (b) is the mesh of the geometric model. The computational model with spiral channels is show in Figure 5, where (a) shows the geometric model and (b) is the mesh of the geometric model. The motor rotor, the inner sleeve and the balance weight is rotating when the compressor is operating. Hence, the static domain is adopted for the gap between the rotor and the stator and the refrigerant inlet and outlet area, while the rotating domain is adopted for the internal flow channels of rotor and the gap between balance weight and crankshaft. In order to discuss the influence of refrigerant flow in the spiral channels on the performance of the compressor, the spiral channels with a pitch of 240mm and the straight channels are compared and analyzed. The number of the two types of channels are both 8. Besides, the cross-sectional area, shape and outlet location of the channels are all same.
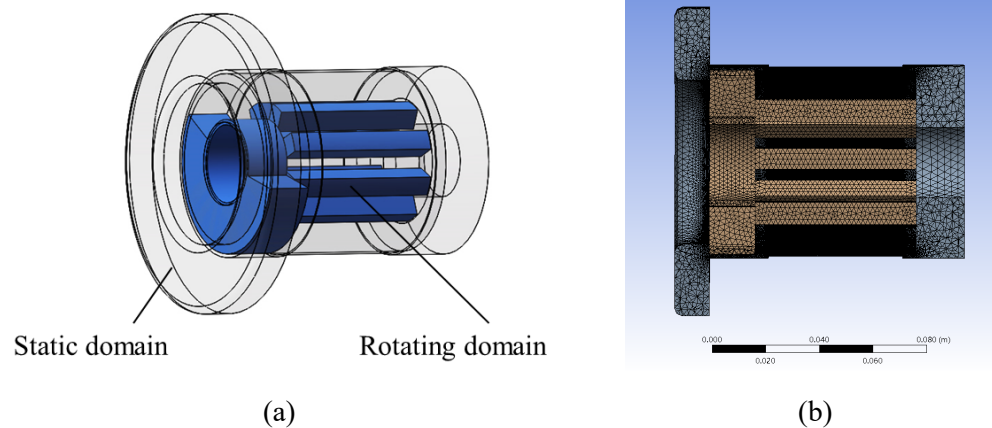
Figure 4. The geometric mode (a) and the section view of mesh (b) of the computational model with straight channels.
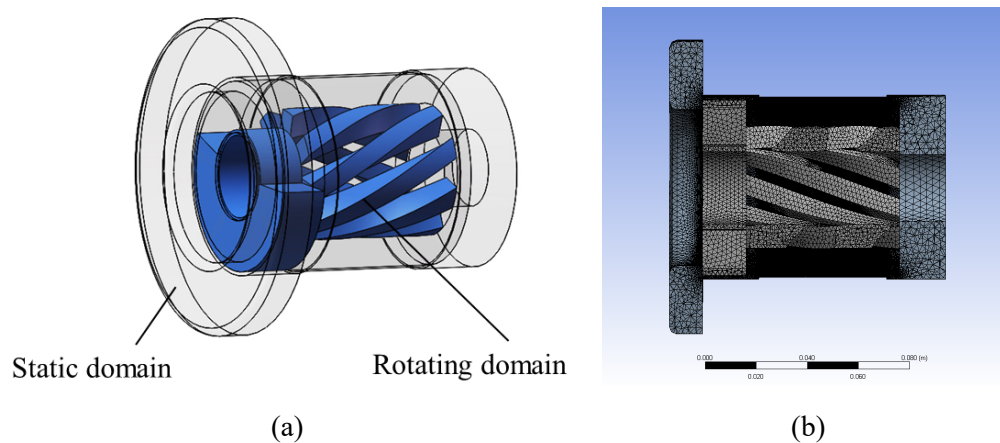

Figure 5. The geometric mode (a) and the section view of mesh (b) of the computational model with spiral channels.

2.3. Computational Model

The simulation method and calculation conditions are presented in Table 2. The actual refrigerant R134a gas model is used for calculation. And the first-order upwind scheme is used to solve the momentum and energy equations. The convergence results are obtained by monitoring the residual of mass, energy and heat transfer equation. When the residual converges to 10^{-4}, the calculation stops.

Table 2. Simulation method and calculation condition.

Computation method		Operating conditions			
software	turbulent model	inlet temperature	inlet pressure	rotational speed (rpm)	outlet flow velocity (m/s)
				3000	4.3067
CFX17.2	k-Epsilon	288.15K	3.4966bar	5000	7.4789
				7000	10.1106
				9000	12.4621

In order to discuss the influence of refrigerant flow in the spiral channels on the performance of the compressor, the spiral channels and the straight channels are compared and analyzed. The simulation is

carried out at 3000 rpm, 5000 rpm, 7000 rpm and 9000 rpm. In addition, the grid independence verification is show in Table 3. After grid independence verification, the grid quantity of case 3 meets the quality requirement.

Table 3. Grid independence verification.

Type	Case	Mesh number	Outlet pressure (bar)
the computational domain with straight channels	1	3801056	336.085
	2	5391963	338.917
	3	6806930	338.638
	4	8189592	338.857
the computational domain with spiral channels	1	3384108	321.054
	2	4834654	324.767
	3	6297658	324.053
	4	7836120	324.671

The heat transfer between the surface of the motor and the refrigerant gas is convection heat transfer. In the setting of the wall boundary conditions, the outside temperature is assumed at 40℃, and the calculation of the heat transfer coefficient is relatively complex, and empirical formula is usually used to calculate it.

(1) heat transfer coefficient of inner gap between stator and rotor

There is an annular gap between the rotating rotor, and the stationary stator. Nerg et al. [8] in their study pointed out heat transfer characteristics in such inner gap can be derived as:

$$Nu_g = 2, T_a^* < 1700$$
$$Nu_g = 0.128(Ta^*)^{0.367}, 1700 < Ta^* < 10^4 \tag{1}$$
$$Nu_g = 0.409(Ta^*)^{0.241}, 10^4 < Ta < 10^7$$

$$Ta^* = \frac{16997(0.0056 + 0.0571\zeta^2)(1 - \frac{\delta}{2r_o})Ta}{\pi^4 \zeta} \tag{2}$$

$$Ta = \frac{\rho^2 \omega^2 r_o \delta^3}{\mu^2} \tag{3}$$

$$\zeta = \frac{2r_o - 2.304\delta}{2r_o - \delta} \tag{4}$$

Where: Ta^* is the modified Taylor number, δ is the inner gap length (m), r_o is the radius of the rotor outer surface, ω is the rotation angular velocity (rad/s), μ is the dynamic viscosity (Pa·s) and ρ is the density(kg/m³).

(2) heat transfer coefficient of the rotor inner channels

The rotor inner channels circumferential distribution around the rotor axis, rotating along with the rotor and spiral sleeve. The heat transfer characteristics can be calculated as follow [4]:

$$N_{uc} = \frac{0.043 P_r}{(P_r)^{0.2} - 0.050} \frac{R_e}{\Gamma^{0.1}} \left[1 + \frac{0.061}{(\frac{R_e}{\Gamma^2})^{0.2}} \right],$$

(5)

$$\Gamma = \frac{R_e^{\frac{18}{11}}}{(G_r P_r^{\frac{2}{3}})^{\frac{10}{11}}},$$

(6)

$$G_r = \frac{H \omega^2 \beta (\tau \cdot \frac{d_c}{2}) \rho^2 d_c^3}{\mu^2},$$

(7)

Where: H is the distance from cooling hole axis to rotor axis (m), β is the thermal expansion coefficient and τ is the temperature gradient along the cooling hole axis, d_c is the equivalent diameter (m).

(3) heat transfer coefficient of end-winding space [9]:

$$\alpha_w = \frac{(1 + 0.04 v)}{0.045},$$

(8)

Where: v is the outer diameter straight velocity of the motor rotor(m/s).

(4) heat transfer coefficient of the inner and outer surface of the stator domain contacting with frame

The inner and outer of the stator domain contacted with the frame and the bearing at the area close to the outlet. Therefore, the refrigerant is heated by the compressor components. The heat transfer on the inner and outer surface can be treated as natural convection around a horizontal cylinder.

3. Results and discussion

3.1. Discussion on the compressor inlet mass flow rate and pressure

According to Figure 6, the outlet pressures of computational domain, the suction pressures of the compressor, with two type rotor channels are calculated. It is found that with the increase of rotation speed, the pressure loss increases due to the increase of refrigerant flow rate. However, under the same compressor operating condition, the outlet pressure of the spiral channel is higher than that of the straight channel, and the pressure loss is smaller. The pressure increases by 0.63% at 3000 rpm, 1.79% at 5000 rpm, 2.26% at 7000 rpm, and 4.35% at 9000 rpm. Therefore, the spiral channel can reduce the intake resistance of the compressor and the suction pressure is increased. And with the increase of rotation speed, the difference of outlet pressure is larger. Accordingly, in the high-speed compressor the spiral inner sleeve of the motor rotor is effective.

Under different rotational speeds of the compressor, the simulation results of the total refrigerant mass flow rate, the compressor suction mass flow rate, are shown in Figure 7. It can be seen from the figure that the flow rate of spiral channels is greater than that of straight channels at the same rotation speed. With the increase of the rotation speed, the difference of mass flow rate between the straight channels and spiral channels increases. The outlet mass flow rate increases by 3.94% at 3000 rpm, 4.19%

at 5000 rpm, 6.86% at 7000 rpm, and 6.87% at 9000 rpm. It is obvious that the spiral channels help to increase the refrigerant flow and the refrigeration capacity of the compressor system.
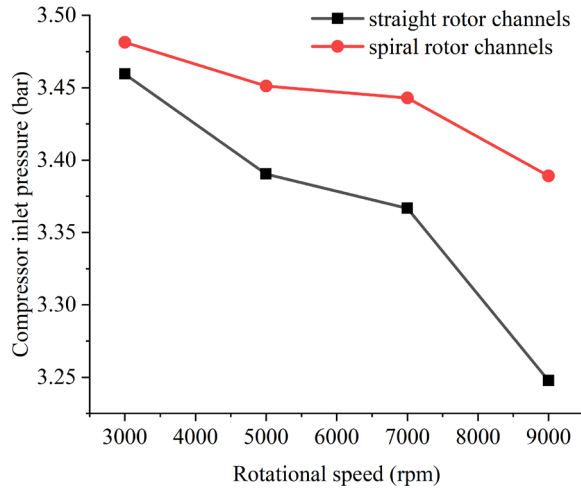

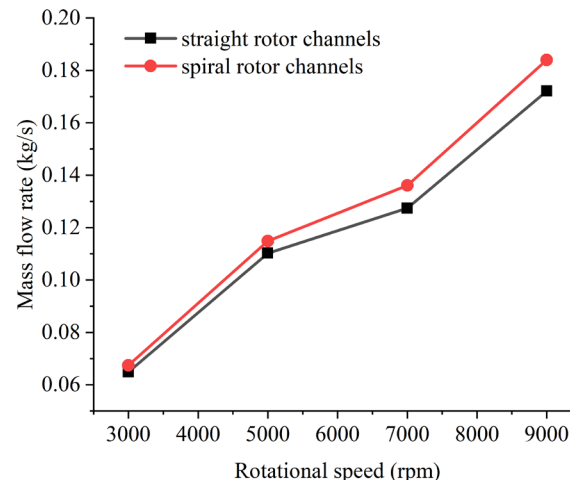**Figure 6.** Compressor inlet pressure of rotor channels at different rotation speeds.

Figure 7. Mass flow rate of rotor channels at different rotation speeds.

3.2. Discussion on refrigerant distribution

The outlet position of the rotor channels is shown in Figure 3. At 3000 rpm, 5000 rpm, 7000 rpm and 9000 rpm, the refrigerant mass flow distribution shows the same law. Take 9000 rpm as an example. Figure 8 (a) (b) shows the sectional drawing of velocity vector of the computational domain with straight and spiral channels. The refrigerant flows into the gap between balance weight and crankshaft through the air gap and the channels in the rotor, and finally reaches the outlet. It can be seen that the velocity of refrigerant in unblocked channels is larger. By comparing the straight channel with the spiral channel, it can be found that in the unblocked channel, the refrigerant flow rate is higher, which results in larger mass flow rate of the spiral channels.
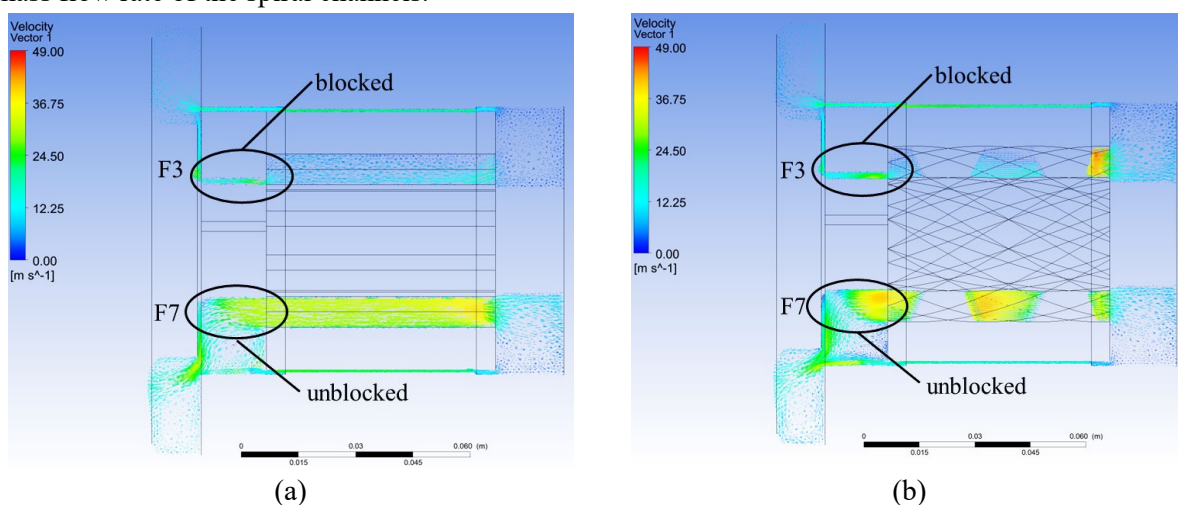

Figure 8. Cross section of velocity vector (a) with straight channels (b) with spiral channels at 9000 rpm.

Table 4 shows the percentage of mass flow in the rotor channel F1-F8 in the total flow at 9000 rpm. In different types of channels, the mass flow rate of refrigerant increases with the increase of the rotation speed. And the refrigerant distribution in each channel is uneven, among which, F1-F5 channels are blocked to varying degrees at the outlet, and F6-F8 are directly connected with the rotating domain of the gap between balance weight and crankshaft. Among them, the straight channels have higher refrigerant flow rate than the spiral at F1-F5 channel, while the spiral channels have higher refrigerant flow rate than the straight at F6-F8 channel. The increases of total refrigerant flow rate in spiral channels are mainly due to the increase of F6-F8 flow rate.

Table 4. The percentage of mass flow in the rotor channels (F1-F8) in the total flow at 9000rpm.

Rotor channels	Percentage of mass flow rate in rotor channels		Deviation
	Straight channels	Spiral channels	
F1	15.58%	14.54%	-1.04%
F2	4.27%	2.96%	-1.31%
F3	4.21%	3.31%	-0.90%
F4	4.71%	3.97%	-0.74%
F5	18.68%	18.65%	-0.03%
F6	18.52%	19.96%	1.44%
F7	17.28%	18.72%	1.44%
F8	16.74%	17.88%	1.14%

The main reason for the great difference of flow rate is the existence of balance weight of scroll compressor. Due to the existence of the balance weight, the refrigerant flow and flow rate in the F1-F5 channels are reduced. Therefore, in order to further increase the refrigerant flow in the motor of the scroll compressor, the location of the balance weight of the scroll compressor can be optimized. On the premise of ensuring that the balance weight meets the requirements of the compressor, the volume is reduced and the layout should be reasonable, so as to increase the refrigerant flow and reduce the pressure loss of the channels of the motor rotor.

4. Conclusion

In this paper, a kind of spiral channel in the motor rotor of the high-speed scroll compressor is proposed. The spiral inner sleeve is added inside the rotor core, the spiral refrigerant flow channels are formed between the spiral inner sleeve and the rotor core. In order to discuss the influence of the rotor channels structure on the performance of the compressor, the spiral channels with a pitch of 240 mm and the straight channels are compared and analyzed. The simulation is carried out at 3000 rpm, 5000 rpm, 7000 rpm and 9000 rpm.

It is found that under the same compressor operating condition, the outlet pressure of the spiral channels is higher than that of the straight channels and the refrigerant mass flow rate of spiral channels also is greater. The pressure increases by 0.63% at 3000 rpm, 1.79% at 5000 rpm, 2.26% at 7000 rpm, and 4.35% at 9000 rpm. And, at different rotational speeds, the outlet mass flow increases by 3.94% at 3000 rpm, 4.19% at 5000 rpm, 6.86% at 7000 rpm, and 6.87% at 9000 rpm. With the increase of the motor speed, the supercharging effect of spiral channels in rotor is more obvious.

Due to the existence of balance weight of scroll compressor, the refrigerant distribution in each channel is very uneven. The refrigerant flow velocity and mass flow rate are larger in the unblocked

channels than blocked channels. And in the unblocked channels, the spiral channels show better flow characters. The increases of total refrigerant flow rate in spiral channels are mainly due to the increase in unblocked channels. Therefore, in order to further increase the refrigerant mass flow in the motor of the scroll compressor, the size of balance weight is reduced and the layout should be reasonable.

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