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ABSTRACT

The single screw refrigeration compressor (SSRC) is widely used in refrigeration and air conditioning systems due to the advantages of simple structure, balanced forces on the rotor, small vibration, low noise level, high volumetric efficiency and so on. In the SSRC, working medium is an important factor influencing the thermal dynamic performance of the compressor. Restricted by the application environment and the requirements of environmental protection, the types of the working medium used in the SSRC will change gradually in future, this to a certain extent, affect the compressor adaptability for medium and the operation stability and reliability of compressor. Thus the effects of the working medium on the geometric characteristics and dynamic performance of the SSRC must be studied. In this paper, the geometric model of the exhaust port and the fluid flow calculation model of the exhaust process were established for analyzing the influence of the working medium on the exhaust port opening position and the flow resistance loss during exhaust process in the SSRC. A dynamic characteristic analysis model was proposed to calculate the force act on the meshing pairs and the bearings under the working condition as well as the load conditions of the shaft system under the shutdown-starting condition. All analysis results obtained in this paper can be useful for optimum design of the SSRC to improve the operation stability and reliability and the energy efficiency of the compressor with different working medium. The first major section of the manuscript is an abstract. The abstract should describe the contents of the paper, discuss the contribution to the field as well as present the most important results. Authors are responsible for typing accuracy and proofreading the manuscript. If accepted, the manuscript must be submitted for reproduction without being edited or retyped by the staff before printing. The manuscript must look professional and be technically correct in order to be accepted.

1. INTRODUCTION

Single screw compressor was first invented by Zimmern (1963), and then it was introduced to the field of refrigeration soon in the mid of 1970s (Constant *et al*. 1981 and Zimmern 1984). The SSRC mainly consists of one screw rotor and two star-wheels which symmetrically located with the screw to double the swept volume and balance the thrusts. So it has advantages of simple structure, without valve, balanced forces on the rotor, small vibration, low noise level, high volumetric efficiency and so on. Currently, it has been widely used in refrigeration and air conditioning systems. In the SSRC, working medium is an important factor influencing the thermal dynamic performance of the compressor. But today, restricted by the application environment and the requirements of environmental protection, the new agreement about accelerating the phase-out of production and consumption of hydro-chlorofluorocarhons (HCFCs) have been established. So the types of the working medium used in the SSRC will change gradually, this to a certain extent, affect the compressor adaptability for medium and the operation stability and reliability of compressor.

Over the years, a series of studies have been carried out for the HCFCs replacement technologies, the properties of the alternative refrigerants and the current situation of refrigerants in different types of refrigeration compressors. Ma and Wang (2010) analyzed the refrigerant alternative technologies, summarized the advantages and disadvantages of various hydro-fluorocarbons (HFCs) and natural refrigerants for substitution. From their study, the HFCs alternative refrigerants such as R410A, R134a and R407 series were considered to be viable transition alternative refrigerants, and the natural refrigerants such as the R290, NH_3, CO_2 and H_2O were considered to be environmental friendly alternative refrigerants, but the R290 and NH_3 had the security problems, furthermore the requirement for the compressor and system of the CO_2 and H_2O became more strict. Li (2011) enumerated a kind of transition alternative refrigerant, such as R32, DR-2, HFO-1234yf in addition to natural refrigerants, and analyzed

the thermal physical properties of this transition alternative refrigerant in his paper. Huo (2015) analyzed the feasibility of using R1234ze(Z) and R152a mixture as alternative for R22. Chen *et al.* (2016) presented a review of the latest research and development of the refrigerants based on the new version of the Montreal Protocol 2014 Assessment report, the 11th HR Gustav Lorentzen Conference on Natural Refrigerants 2014 and the 25th international congress of refrigeration 2015. The current situation of refrigerants in different types of refrigeration units and the properties of the alternative refrigerants were discussed in their study. Through these studies, many kinds of alternative refrigerants had been proposed to replace HCFC, and been used in refrigeration systems.
But the properties of the alternative refrigerants are difference from that of the HCFC, which will affect the geometric characteristics and working characteristics of the compressor in the refrigeration system used these alternative refrigerants. So the scholars at home and abroad have carried out some analysis about the compressor operating characteristics with different alternative refrigerants. Jurgen, Mitsuhiro and Hubacher *et al.* (1998, 2000, and 2002) researched the performance of the reciprocating compressor, the twin rotary compressor and the vane compressor for carbon dioxide cycle respectively. Yang *et al.* (2008) developed the hermetic rotary CO_2 compressor and measured its performance. Li and Yang *et al.* (2008) analyzed the CO_2 expander used in refrigeration system. Weight *et al.* (2000) researched the efficiency of the water vapor compressor. Brandon *et al.* (2000) analyzed the commercial feasibility of the use of water vapor as a refrigerant. Sariibrahimoglu *et al.* (2009) studied the effect of R600a on tribological behavior of sintered steel in hermetic refrigeration compressor. Xu *et al.* (2016) compared the performance of the compressor used R22, R404A and R407F. Song *et al.* (2015) compared the performance of the open screw compressor using R717 and R22. These studies showed that the structure and performance of the compressor were greatly influenced by refrigerant. Although many scholars have done a lot of work, so far no work has been done about the effect of working medium on single screw refrigeration compressor performance. In the SSRC, working medium is an important factor influencing the thermal dynamic performance of the compressor. Thus the effects of the working medium on the geometric characteristics and dynamic performance of the SSRC must be studied.

In this paper, the geometric model of the exhaust port and the fluid flow calculation model of the exhaust process are established for analyzing the influence of the working medium on the exhaust port opening position and the flow resistance loss during exhaust process in the SSRC. A dynamic characteristic analysis model is proposed to calculate the force act on the meshing pairs and the bearings under the working condition as well as the load conditions of the shaft system under the shutdown-starting condition. All analysis results obtained in this paper can be useful for optimum design of the SSRC to improve the operation stability and reliability and the energy efficiency of the compressor with different working medium.

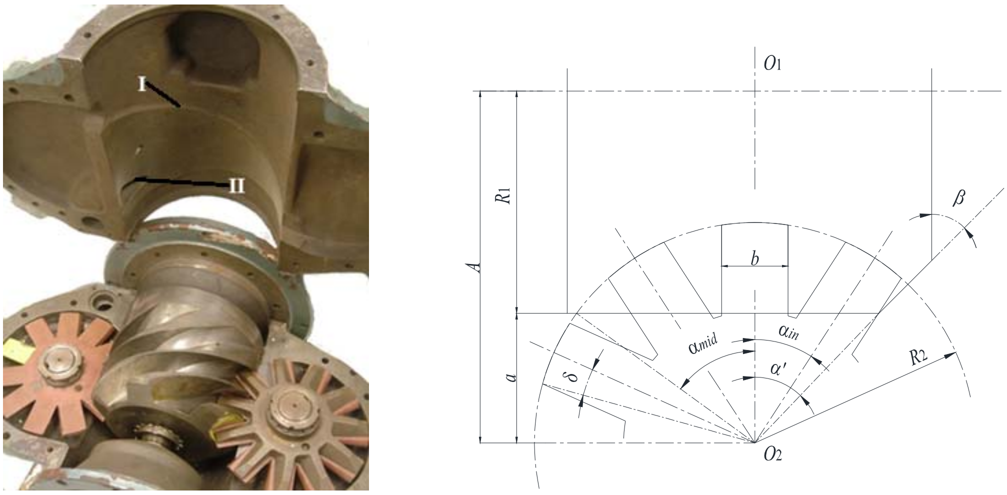
2. GEOMETRIC CHARATERISTICS ANANLYSIS

2.1 Geometric characteristics

In the SSRC, in order to increase the radial inlet area and reduce the friction area between the screw and the cylinder wall, the cylinder wall is usually designed to step as shown in Figure 1 (a), in which Ⅰ is the closed spiral and Ⅱ is the exhaust orifice. The spiral at the outside of the steps called close spiral. According to the geometric features of it, the closed helix refers to the spiral at the ventro-side of the tooth when the star wheel teeth form primitive volume. Because of that the properties of the refrigerants are difference from each other, the star wheel rotation Angle when the star wheel teeth form primitive volume is also different. So the location of the closed spiral will be affected by the refrigerants. In order to analysis the influence of refrigerants on the closed spiral, the closed spiral calculation model needs to be established.
Assuming that the star wheel Angle $\alpha = \alpha_{\text{in}}$ when the star-wheel tooth just forms a closed volume as shown in Figure 1 (b), then the equations of the closed spiral can be expressed as follows:

$$\begin{cases} Y_{\text{f}}\left(\alpha\right) = \left(A - R_{1\text{p}}\right)\tan\alpha - \dfrac{b}{2\cos\alpha} \\ S_{\text{f}}\left(\alpha\right) = PR_1(\alpha - \alpha_{\text{in}} - \dfrac{\alpha_\delta}{P}) \end{cases} \qquad \alpha_{\text{in}} \le \alpha \le \alpha_{\text{out}} \tag{1}$$

Where Y_{f} is the axial coordination of the closed spiral, A is the central distance between the star wheel and the screw rotor, $R_{1\text{p}}$ is the equivalent radius in the meshing point of the star wheel, α is the rotation position angle of the star wheel, b is the tooth width of star-wheel, S_{f} is the circumferential coordination of the closed spiral, P is the tooth number ratio of the star-wheel to the screw rotor, R_1 is the radius of the screw rotor, α_{in} is the flow inlet angle, α_{out} is the angle when the star-wheel tooth meshing out of the spiral groove, and α_{δ} is the position angle of the star-wheel meshing line.



(a) The internal surface of the cylinder wall (b) Geometric parameters relationship

Figure 1: The schematic of the single-screw compressor

Different from reciprocating compressor, the exhaust process of the SSRC is the forced exhaust process. The location and shape of the exhaust orifice is the necessary condition to guarantee the gas realizing the designed internal compression ratio in the primitive volume, and is the important factor to influence the compressor performance. As a result of the forced exhaust process, the exhaust process will begin when the front screw groove flank of the closed working chamber connected to the exhaust orifice. At this time, the front screw groove flank spiral is set as the exhaust orifice spiral (As shown in Figure 2). Similarly, the refrigerants will also affect the location and shape of the exhaust orifice spiral, so the exhaust orifice spiral calculation model needs to be established to analysis the influence of refrigerants on the exhaust orifice spiral.

Set the star wheel angle as the α_{d} when the front screw groove flank of the closed working chamber connected to the exhaust orifice, and set the star wheel angle as the α_{mid} when the front star-wheel tooth meshing out of the spiral groove. Used the star wheel rotation Angle as the characterized angle, the exhaust orifice spiral equation is as follows:

$$\begin{cases} Y_{\mathrm{p}}(\alpha) = \left(A - R_{1\mathrm{p}}\right)\tan\alpha + \dfrac{b}{2\cos\alpha} \\ S_{\mathrm{p}}(\alpha) = PR_1(\alpha - \alpha_{\mathrm{d}} - \dfrac{\alpha_{\delta}}{P}) \end{cases} \qquad \alpha_{\mathrm{d}} \le \alpha \le \alpha_{\mathrm{mid}} \tag{2}$$

Where Y_{p} is the axial coordination of the exhaust orifice spiral, and S_{p} is the circumferential coordination of the exhaust orifice spiral.

2.2 Flow resistance loss

In the exhaust process of the SSRC, the high pressure gas flow through the exhaust orifice and be discharged to the exhaust pipe. There will be a big flow resistance loss when the gas flows through the exhaust orifice which will affect the performance of the SSRC. The exhaust orifice position and area is different in the SSRC using different refrigerants. If the exhaust orifice was designed improperly, the flow resistance loss will be increased and the performance of compressor will be reduced inevitably. In order to calculate the flow resistance loss, the area of the exhaust orifice needs to be gained. The exhaust orifice area can be calculated through the integration of the exhaust orifice spiral equation and exhaust end line, the calculation formula is:

$$S = \int_{\alpha}^{\alpha_{\text{mid}}} \left(y_{\text{p}}\left(\alpha_{\text{out}}\right) - y_{\text{p}}\left(\alpha\right) \right) PR_1 \text{d}\alpha \tag{3}$$

After calculating the exhaust orifice area, the flow velocity which used to measure the flow resistance loss can be obtained as following:

$$v = \dot{V} / S \tag{4}$$

Where $\dot{V}$ is the exhaust gas volume from the exhaust orifice per second.

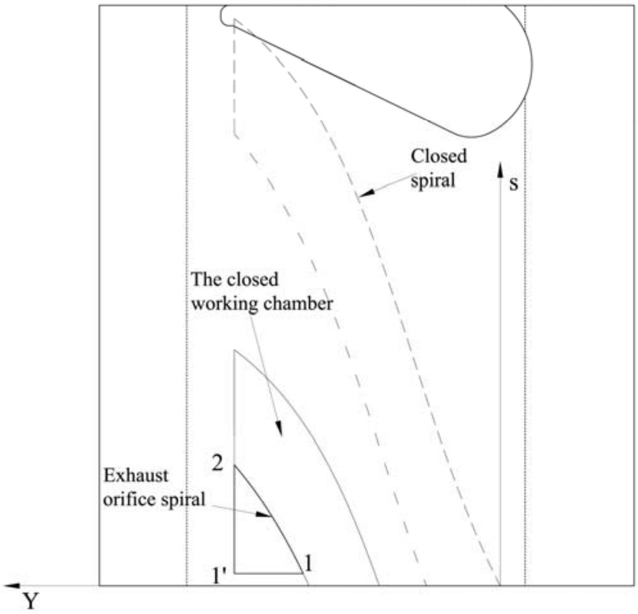


Figure 2: The expanded view of the internal surface of the cylinder wall

3. DYNAMIC CHARACTERISTICS ANALYSIS

3.1 Thermal process analysis

In the SSRC, the gas pressure in the compression chamber under different star wheel rotation angle can be expressed as:

$$p\left(\alpha\right) = \begin{cases} p_{\text{s}} & \alpha < \alpha_{\text{in}} \\ \dfrac{V_0{}^n}{V\left(\alpha\right)^n} p_{\text{s}} & \alpha_{\text{in}} \le \alpha \le \alpha_{\text{d}} \\ p_{\text{d}} & \alpha \ge \alpha_{\text{d}} \end{cases} \tag{5}$$

Where p_{s} is the suction pressure, V_0 is the groove volume at suction closure of the SSRC, $V\left(\alpha\right)$ the volume of the compression chamber under different star wheel rotation angle, which can be calculated by the equation listed in the Wang's study(2014), p_{d} is the exhaust pressure of the SSRC.

By using this equation, the gas temperature in the compression chamber under different star wheel rotation angle can be expressed as:

$$T\left(\alpha\right) = T_{\text{in}} \left(\frac{p\left(\alpha\right)}{p_{\text{s}}} \right)^{\frac{n-1}{n}} \qquad \alpha_{\text{in}} \le \alpha \le \alpha_{\text{d}} \tag{6}$$

3.2 Dynamic characteristics

In the SSRC, the symmetrical arranged star wheels make the radial gas force of the screw which caused by the compressed gas in the screw groove in balance with each other. On the other hand, the axial force of the screw also can be kept balance due to the compressed gas filled in the screw groove and the axial gas channel opened in the screw. Based on the above characteristics, the dynamic characteristics of the screw are very good. But the gas forces act on the star wheel can not be kept balance. The refrigerants will affect the dynamic characteristics of the star wheel, and then the operation reliability of the compressor will be affected. So in this study the dynamic characteristics of the star wheel must be analyzed.

The forces act on the star wheel and shaft system are shown in Figure 3, F_g is the resultant gas force acting on the multiple star wheel teeth at the same time. F_n is the normal force acting on the star wheel tooth flank. M_{fA} and M_{fB} are the torque of the star wheel bearing. The value of F_n is much smaller than that of the gas force under quasi-static equilibrium state, therefore its influence in the dynamic characteristics analysis of the star wheel shaft can be ignored.

For a standard gear ratio SSRC, there are three teeth engage in screw. Set the last tooth as the basis tooth to calculate resultant forces F_g. The gas force applied on the each star wheel tooth is expressed as follows:

$$F_g(\alpha_i) = \begin{cases} \int_{-\frac{b}{2}}^{\frac{b}{2}} \left(p(\alpha_i) - p_s \right) \cdot dS_{s,i} & \alpha_{in} \le \alpha_i \le \alpha_{mid} \\ \int_{-\frac{b}{2}}^{R_2 \cdot \sin\gamma} \left(p(\alpha_i) - p_s \right) \cdot dS_{s,i} & \alpha_{mid} \le \alpha_i \le \alpha_{out} \end{cases} \tag{7}$$

$$\gamma_0 = arc\cos\left(\frac{A - R_1}{R_2}\right) - \delta \tag{8}$$

So the resultant gas forces act on the multiple star wheel teeth at the same time can be calculated as following:

$$F_g(\alpha) = \sum F_{g,i}(\alpha_i) = \sum_{i=0}^{n} F_{g,i}(\alpha + i\lambda) \tag{9}$$

Where $p(\alpha_i)$ is the gas pressure of the working chamber at Arbitrary star wheel rotation Angle, p_s is the suction pressure, $S_{s,i}$ is the tooth area engage in screw, R_2 is the radius of the star wheel, δ is the half width tooth angle, λ is the intersection angle of the center line of the adjacent two star wheel.

As shown in Figure 3, the resultant gas force act on the tip position of the star wheel tooth, so an overturning moment will be produced by the gas force. In order to balance this overturning moment, bearings on both ends of the star wheel bracket need to be installed. The torque of the star wheel bearing can be obtained by the following equation:

$$M_i(\alpha_i) = \begin{cases} \int_{-\frac{b}{2}}^{\frac{b}{2}} \int_0^{\mu} \left(p(\alpha_i) - p_s \right) \cdot r(\alpha_i) \, d\sigma d\eta & \alpha_{in} \le \alpha \le \alpha_{mid} \\ \int_{-\frac{b}{2}}^{R_2 \cdot \sin\gamma} \int_0^{\mu} \left(p(\alpha_i) - p_s \right) \cdot r(\alpha_i) \, d\sigma d\eta & \alpha_{mid} \le \alpha \le \alpha_{out} \end{cases} \tag{10}$$

$$W(\alpha) = \sum W_i(\alpha_i) = \sum_{i=0}^{n} W_i(\alpha + i\lambda) \tag{11}$$

Where $r(\alpha_i)$ is the distance between the gas force equivalent application point and the star wheel axis, which can be calculated by the equation listed in the Wang's study (2014).

Influence by the work environment, the SSRC alternate in downtime and normal work condition. When the compressor is in a state of shutdown, the refrigerant in the system will keep in an equilibrium state. In this case, the internal pressure of the system is much higher than suction pressure. So in the shutdown-starting condition, the suction gas is the refrigerant vapor in the equilibrium state. The pressure of the suction gas under the shutdown-starting condition is much higher than the normal suction pressure of the SSRC. So the forces act on the meshing pair will become higher than the normal working conditions. As the refrigerant used in the SSRC is different, the pressure of the refrigerant vapor under equilibrium state will be different too. So the refrigerants will affect the dynamic characteristics of the meshing pairs under shutdown-starting condition. In order to analyze the influence of the refrigerant on the dynamic characteristics under shutdown-starting condition, the forces acting on the star wheel under the shutdown-starting condition also have to be calculated in this study.

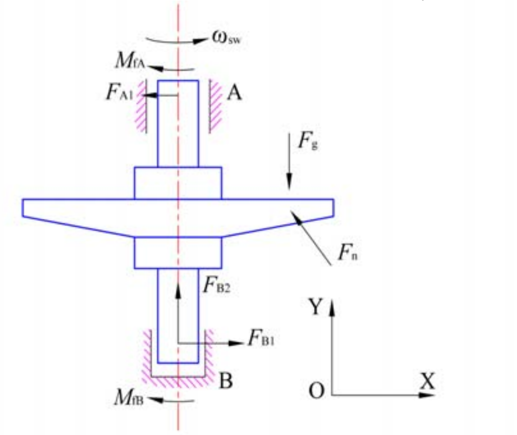


Figure 3: The forces act on the star wheel and shaft system

4. RESULTS AND DISCUSSION

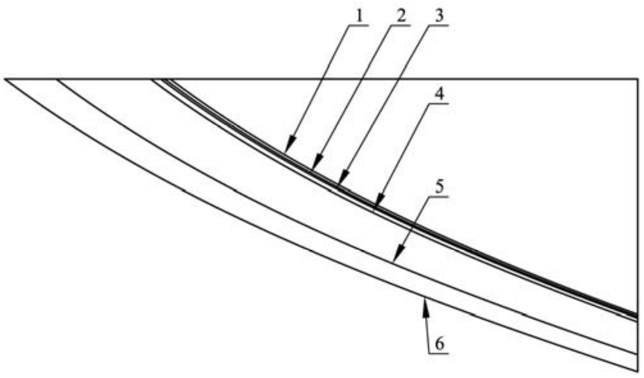
In this paper, a typical SSRC used in refrigeration and air conditioning field with the structure and operation parameters as shown in Table 1 was chosen for the calculate example. Several refrigerants such as R22, R134a, R410A, R290, R717 and R32 were used in this study to contrastive analysis the influence of the refrigerant on the geometric characteristics and the dynamic performance of the SSRC.

Table 1: The main parameters

Parameters	Value	Parameters	Value
P	11/6	Length of the screw (mm)	155.7
R_1 (mm)	181	Thickness of the star wheel (mm)	7.25
R_2 (mm)	181	Lubricating oil	Suniso 4GS
A (mm)	144.8	Evaporation temperature (K)	278.15
b (mm)	26.5	Condensation temperature (K)	313.15
n (r.min^{-1})	2880	Superheat temperature (K)	5

4.1 Geometric characteristics analysis results

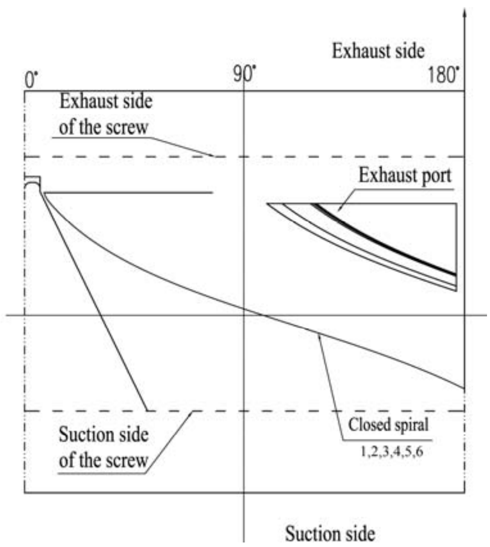
The location and shape of the exhaust orifice will be affected by the refrigerants. Figure 4 shows the opening location and the shape of the exhaust orifice in the SSRC with different refrigerant. The contrast of the exhaust orifice of the SSRC with different refrigerant shows that the opening location and the area of the exhaust orifice are greatly affected by the refrigerants because of that the pressure ratio and the adiabatic index of the refrigerants are very different from each other. The internal volume ratio increase by degress from curve 6 and curve 1.For the refrigerants used in this study, the exhaust orifice opening time of the SSRC with R410A and R32 which have a lower pressure ratio and higher adiabatic index is earlier than that of the SSRC with R22, R134a, R290 and R717 which have a higher pressure ratio and lower adiabatic index. And the exhaust orifice area of the SSRC with R410A and R32 is larger than that of the SSRC with R22, R134a, R290 and R717. So in the design phase, the exhaust orifice must be designed separately for each kind of the refrigerant. If the design is not changed, the over or under compression process will occur, then the additional energy losses will be caused.



1-R134a, 2-R290, 3-R717, 4-R22, 5-R410A, 6-R32

Figure 4: The location and shape of the exhaust orifice

Figure 5 shows the closed spiral location of the SSRC with different refrigerant. As can be seen from the contrast of these closed spirals, the difference of closed spiral location of the SSRC with different refrigerant is very small. So the conclusion of that the influence of refrigerants on the closed spiral is very small and is smaller than that on the location and shape of the exhaust orifice. So in the design phase, only one closed spiral needs to be designed for all kinds of the refrigerant, but the closed spiral must be designed to meet the requirements of the refrigerant with the minimum inlet pressure.



1-R134a, 2-R290, 3-R717, 4-R22, 5-R410A, 6-R32

Figure 5: The closed spiral location

Figure 6 shows the exhaust orifice flow speed ratio of the compressor using different refrigerants changing with the pressure ratio of the refrigerants. The variation tendency of the curves in Figure 6 shows that the exhaust orifice flow speed of the compressor will influence by the pressure ratio and the adiabatic index of the refrigerants rather than the evaporating pressure and condensing pressure of the refrigerants. For the refrigerants used in this study, the exhaust orifice flow speed of the compressor with R410A and R32 which have a lower pressure ratio and higher adiabatic index is smaller than that of the compressor with R22, R134a, R290 and R717 which have a higher pressure ratio and lower adiabatic index. That means the exhaust flow resistance loss of the compressor with R410A and R32 is lower than that of the compressor with R22, R134a, R290 and R717.

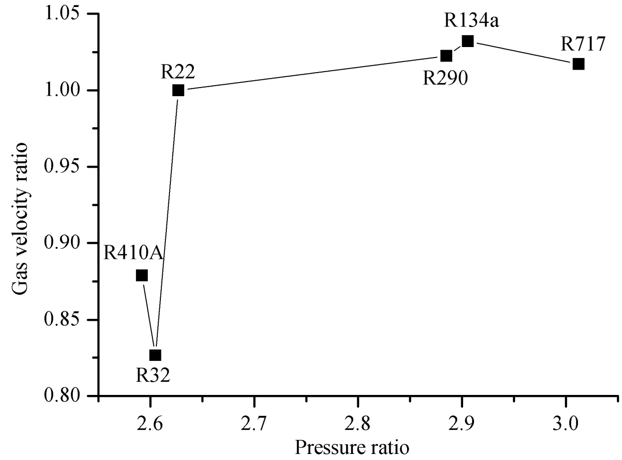


Figure 6: The exhaust orifice flow speed of the compressor with different refrigerants

4.2 Dynamic characteristics analysis results

The gas force acts on the star wheel and the torque acts on the shaft system are shown in Figure 7 and Figure 8 respectively. The contrasts of the gas force curves and the torque curves in Figure 7 and Figure 8 show that the gas force acts on the star wheel and the torque acts on the shaft system of the compressor will be affected by the evaporating pressure and condensing pressure of the refrigerant rather than the pressure ratio and the adiabatic index of the refrigerant. The gas force and the torque will increase with the increase of the evaporating pressure and condensing pressure of the refrigerant. So in the design phase, the dynamic characteristics of the meshing pair and the shaft system have to be checked if a refrigerant with high evaporation pressure and condensing pressure used in the SSRC. For each refrigerant, the gas force first increases and then decreases with the change of the star-wheel angle. Due to the change of work pressure ratio, the star wheel Angle is different to achieve the maximum gas force.

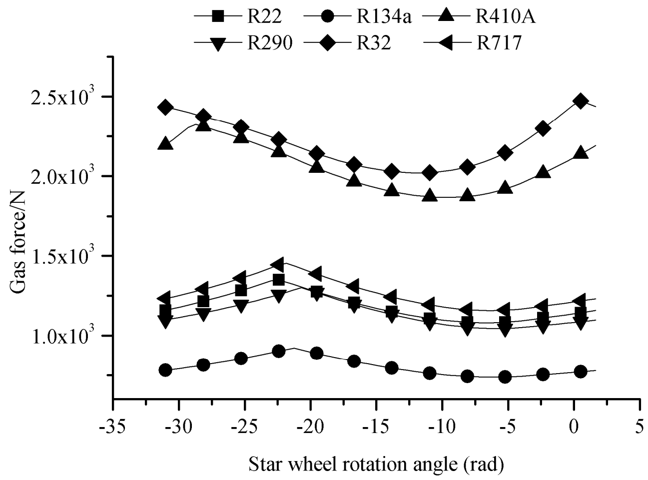


Figure 7: The gas force acts on the star wheel

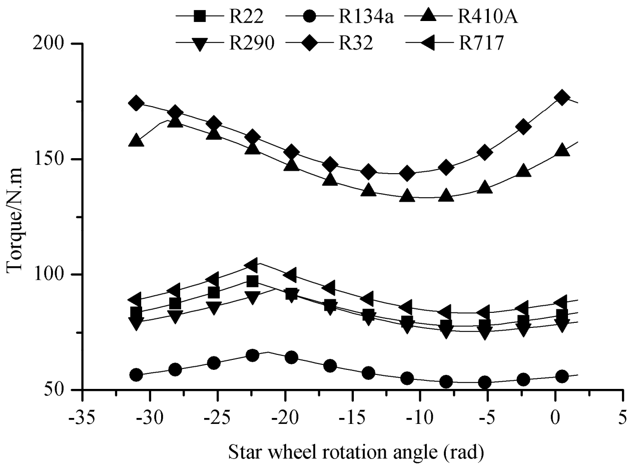
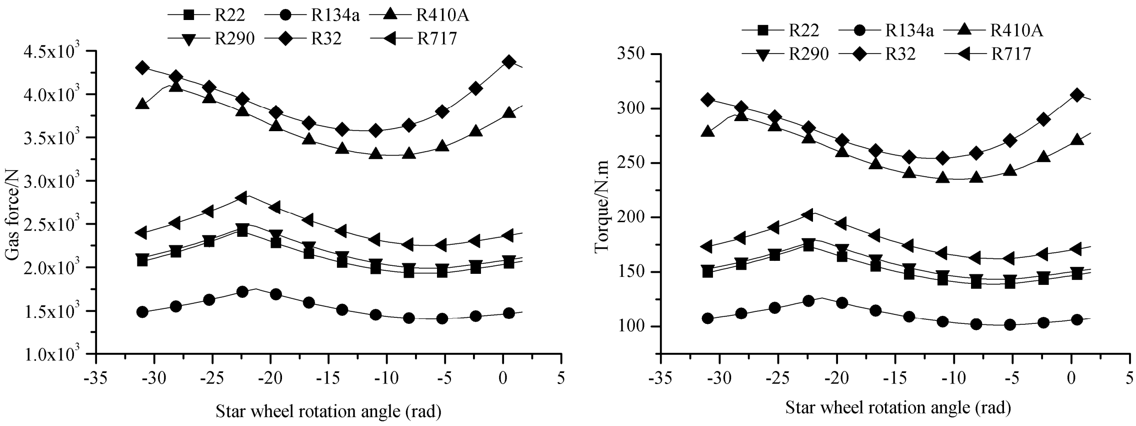


Figure 8: The torque acts on the shaft system

The gas force and the torque act on the shaft system under the shutdown-starting condition are shown in Figure 9. The contrasts of the gas force curves and the torque curves in Figure 7, Figure 8 and Figure 9 show that the influence law of the refrigerant on the gas force and the torque under the shutdown-starting condition are the same as that under the normal working condition. But the gas force and the torque act on the shaft system under the shutdown-starting condition are twice the gas force and the torque act on the shaft system under the normal working condition.



(a) Gas force (b) Torque

Figure 9: The gas forces and torque under the shutdown-starting conditon

5. CONCLUSIONS

(1) The location and shape of the exhaust orifice will be affected by the refrigerants. The exhaust orifice opening time of the SSRC with refrigerant which have a lower pressure ratio and higher adiabatic index is earlier than that of the SSRC with refrigerant which have a higher pressure ratio and lower adiabatic index. And the exhaust orifice area is larger when the refrigerant has a lower pressure ratio and higher adiabatic index.
(2) The influence of refrigerants on the closed spiral is very small and is smaller than that on the location and shape of the exhaust orifice.
(3) The exhaust orifice flow speed of the compressor will be affected by the pressure ratio and the adiabatic index of the refrigerants rather than the evaporating pressure and condensing pressure of the refrigerants. The exhaust orifice flow speed of the compressor with refrigerant which have a lower pressure ratio and higher adiabatic index is smaller than that of the SSRC with refrigerant which have a higher pressure ratio and lower adiabatic index. That

means the exhaust flow resistance loss is lower when the refrigerant has a lower pressure ratio and higher adiabatic index.
(4) The gas force acts on the star wheel and the torque acts on the shaft system of the compressor will be affected by the evaporating pressure and condensing pressure of the refrigerant rather than the pressure ratio and the adiabatic index of the refrigerant. The gas force and the torque will increase with the increase of the evaporating pressure and condensing pressure of the refrigerant.
(5) The influence law of the refrigerant on the gas force and the torque under the shutdown-starting condition is the same as that under the normal working condition. But the gas force and the torque act on the shaft system under the shutdown-starting condition are twice the gas force and the torque under the normal working condition.

NOMENCLATURE

The nomenclature should be located at the end of the text using the following format:

A	The central distance between the star wheel and the screw rotor	(m)
b	The tooth width of star-wheel total cost	(m)
F	Force	(N)
n	Rotate speed	(r/min)
P	Tooth number ratio of the star-wheel to the screw rotor	
p	Pressure	(Pa)
R	Radius	(m)
r	The gas force equivalent application distance	(m)
R_{1p}	Equivalent radius in the meshing point of the star wheel	(m)
S	The circumferential coordination of the closed spiral	
$S_{s,i}$	The tooth area engages in screw	(m^2)
V_0	The primitive volume of the SSRC	(m^3)
$\dot{V}$	The exhaust gas volume from the exhaust orifice per second	(m^3/s)
W	Torque	(N.m)
Y	The axial coordination of the closed spiral	
α	Star-wheel rotation angle	(rad)
α_δ	The position angle of the star-wheel meshing line	(rad)
δ	The half width tooth angle	(rad)
λ	The intersection angle of the center line of the adjacent two star-wheel	(rad)

Subscript

1	Screw rotor
2	Star-wheel
d	Exhaust
f	Closed spiral
g	Gas
in	Inlet
mid	The front star-wheel tooth meshing out of the spiral groove
out	The star-wheel tooth meshing out of the spiral groove
p	Exhaust orifice spiral
s	Suction

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