Experimental study of the axial force on the rotors in a twin-screw refrigeration compressor

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Abstract

The high operating pressure of a twin-screw compressor employed in high temperature heat pump or NH₃/CO₂ cascade refrigeration systems often leads to extremely large axial force on the compressor rotors. Accurate calculation of this axial force is very important in selection of bearing and design of balancing pistons. In this paper, an experimental study of the axial force on the twin-screw compressor rotors was conducted. The variations of the axial force under different operating conditions were investigated. Furthermore, the axial force was also calculated by employing the mean pressure model and sector pressure model, and was compared with that of the experimental results. It was concluded that the sector pressure model is more accurate in predicting the axial force on the rotors in comparison with that of the mean pressure model. However, the mean pressure model was more accurate under operating conditions with large built-in volume ratio. Moreover, the axial force on the rotor end face was found to have larger influence than that on the rotor helical surface.

Keywords: Twin-screw compressor; Refrigeration; Axial force; Balance piston

Nomenclature

F	axial force [N]
A	area [mm ²]
p	pressure [MPa]
N_i	indicated power [kW]
T	screw lead of the male rotor [mm]
Z_l	number of lobes of the male rotor
n	rotating speed of the male rotor [rpm]

Greek symbols

φ	rotating angle of the male rotor [°]
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Subscripts

g	gas
e	end face
ss	shaft shoulder
b	balance piston
n	negative component of force sensor
p	positive component of force sensor
s	suction
d	discharge
j	journal
r	rotor
k	number of the sector section

1. Introduction

Compared with those employed in air compression applications, the twin-screw refrigeration compressor has larger axial force on the rotors and relatively more complicated structure, such as balance piston and slide valve. With the development of the refrigeration industry, the twin-screw refrigeration compressor is operating under higher and higher pressure levels. In recent years, the extremely large axial force on the rotors becomes a big issue regarding to the reliability of the compressor, especially for applications in high temperature heat pump system [1-4] and NH₃/CO₂ cascade refrigeration system [5-8].

In a twin-screw compressor, high gas pressure acts on the rotor helical surface and hence generates the axial force acting on the male and female rotors. There are also axial forces acts on the rotor discharge and suction end faces. It is very important for the selection of bearings and design of balance piston in the twin-screw refrigeration compressor. Therefore, accurate calculation of the axial force on the rotors is very important in the design and operation of the

twin-screw refrigeration compressor.

Researchers have put a lot of efforts to investigate the axial force in the twin-screw compressors. Zhou et al. [9] simplified calculated the axial forces on the rotors of a screw compressor by simplifying the complicated helical surface of the rotor. G.P. Adams [10-12] proposed an integral method to compute forces on the rotor in a screw compressor and the calculation accuracy was improved. Xing [13] put forward a classical method of the rotor force calculation by making use of the mesh of rotor surface. Ahmed Kovacevic et al. [14] carried out a numerical simulation of a combined screw compressor-expander employed in high pressure refrigeration systems, and both the radial and axial forces were obtained. Based on multi-body dynamics, the cyclic variations of the gas-induced force and the torque on screw rotors were obtained with the rotating **angle** of the male rotor [15].

However, the previous studies of the axial force on the rotors in a screw compressor were all carried out by theoretical calculation. There is still lack of experimental research in this area. This paper presents an experimental investigation on the axial force acting on the rotors in a twin-screw refrigeration compressor. The study mainly focuses on the variations of the axial force under different operating and part-load conditions. The axial force on the rotor end face is also obtained through the experimental study. In addition, the experimental results of the axial force on the end face are compared with the simulation data calculated by using mean pressure model and sector pressure model, respectively.

2. Experimental set up

2.1. Compressor test rig

A standard refrigeration compressor performance measurement test rig was set up for the experimental investigation, as shown in Fig.1. It mainly consists of a twin-screw refrigeration compressor, an oil separator, an oil cooler, an internal cooler, a gas-liquid separator, condenser, evaporator and reservoirs. Data obtained from the measurement system mainly includes refrigeration capacity and power consumption of the compressor, temperatures and pressures at various positions.

Fig.1- A picture of twin-screw refrigeration compressor test rig

The compressor investigated in this paper is an oil-flooded open type twin-screw refrigeration compressor. A separate oil system is designed to supply the lubricating oil to the bearings and the balance piston. The detailed geometric parameters of the twin-screw refrigeration compressor are listed in Table 1.

Table 1-Geometric parameters of the twin-screw refrigeration compressor

parameter	value
Center distance	160mm
Outer diameter of the male rotor	215mm
Outer diameter of the female rotor	180mm
Length of the rotors	330mm
Built-in volume ratio of axial discharge port	5.0
Lobes of the male rotor	5
Lobes of the female rotor	6
Wrap angle of male rotor	300°
Theoretical displacement	1120 m ³ /h
Rotating speed	2960rpm

2.2. The measurement of p-V indicator diagram

The p-V indicator diagram plays an important role in the study of a twin-screw compressor. It presents the variations of refrigerant pressure with the rotating angle of the male rotor and can be used to estimate the indicated power consumption of the compressor. The average value of the axial force on the rotors helical surface can be calculated from this indicated power consumption.

Fig.2- The installation angles of the pressure transducers

The measurement of the p-V indicator diagram was conducted by using a few KULITE XTL-140M-17BARA pressure transducers. Their response frequency is as high as 500 kHz, while a temperature-compensating circuit is properly equipped to increase the accuracy of measurement. The maximum working pressure is 1.7 MPa and the range of working temperature is from -55 to +175 °C. The overall measurement error of the transducer is less than $\pm 0.5\%$ of full scale. Seven miniature pressure transducers are installed at the positions which are designed according to the built-in volume ratio of 5.0 for the twin-screw refrigeration compressor. The specific installation positions of the transducers are detailed in Fig. 2. Two of the transducers are used for measuring the refrigerant pressure at the suction chamber and discharge chamber, respectively, while the rest is used to measure the refrigerant pressure in the compression chamber. The angles among the transducers of the number one, number two, number three, number four and number five are 70.4°, 68°, 68° and 65° respectively. In addition, the transducer number one can measure the pressure of both the suction process and part of the compression process, and the transducer number five can measure the pressure of both the discharge process and part of the compression process. The other three transducers are merely for the measurement of part of the compression process.

Finally, the p - ϕ diagrams are obtained from the pressure data measured by the seven pressure transducers. Then, the p - ϕ diagram is converted into the p - V indicator diagram by the transformation between the rotating angle of male rotor and the volume of working chamber, as explained in literature [13].

2.3. The measurement of the axial force

The twin-screw compressor studied in this paper has four journal bearings to support the radial force on the rotors, and one pair of rolling bearings installed at the discharge end of the male and female rotors to support the axial force on the rotors. **The axial force on the male rotor is much larger than that on the female rotor, so the current experimental research was only conducted on the male rotor. The axial force acting on the female rotor can also be investigated with the same method as the male rotor.**

The axial force was measured by several Tecsis F1818 micro piezoresistive force sensors as shown in Fig. 3. It can not only measure the fluctuation amplitude, but also the value of the axial force. The measuring range is 20 kN and the overall error is less than 1% of full scale.

Fig.3- A picture of the force sensor

Fig.4- A picture of the component of force sensor

Due to the compact structure of the bearing bore, there is not enough room for the installation of the force sensors. In order to install the force sensors into the compressor, a specially manufactured component with the force sensors were fabricated as shown in Fig. 4. There are two components of this kind named positive component and negative component, respectively. Each component includes three force sensors. The specific installation positions of the two components are shown in Fig. 5. The positive component was supposed to measure

the positive composite axial force, and the negative component was to measure the residual pre-tightening force of the bolts.

Fig.5- The installation positions of the two components with the force sensors

2.4. The data acquisition system and the experimental operating conditions

The signals of the pressure and force sensors are recorded by a DEWE-1201 data acquisitions system as shown in Fig. 6. It is capable of voltage acquisition, current acquisition and electrical bridge acquisition according to measurement requirement. Electric signal can then be converted into pressure or force signals for analysis.

Fig.6- A picture of the data acquisition system

To measure the value and the variations of axial force under different operating and part-load conditions, the compressor is operated under three different operating conditions with five different slide valve loadings as shown in Table 2.

Table 2-The experimental operating conditions

Condensing/ Evaporating temperature (°C)	Discharge/ suction pressure (bar)	Superheat (°C)	Built-in volume ratio of the slide valve	Slide valve loading (%)				
+30/-15	11.9/2.96	10	3.2	100	95	90	85	75
+35/0	13.548/4.976	10	2.6	100	95	90	85	81
+40/-40	15.33/1.05	10	5.0	100	95	90	85	75

3. Measurement and calculation of the axial force

3.1. Measurement of the composite axial force of gas and end face

The axial forces on the male rotor bearings are composed by axial force on the helical surface of the rotor expressed as F_g , axial force on the end face F_e , balance piston axial force F_b and shaft shoulder axial force F_{ss} . The axial force measured by the positive component is F_p , and the negative component is F_n . The schematic diagram of the axial forces on the male rotor bearings is shown in Fig. 7. These axial forces can be expressed as:

$$F_p = F_g + F_e + F_s - F_b + \quad (1)$$

therefore, the composite axial force of gas and end face is:

$$F_g + F_e = F_p + F_b - F_s - \quad (2)$$

F_b and F_{ss} can be calculated using the pressure in the relevant compression chamber and the diameters of balance piston and the shaft shoulders.

According to the above equations, the composite force of gas and end face can be obtained from the experimental data F_p and F_n . The detailed description of calculation process of the F_g and F_e is shown in the following sections.

Fig.7-The schematic diagram of the axial forces on the male rotor bearings

3.2. Calculation of the average gas axial force from p-V indicator diagram

According to the signal of each pressure sensor, an entire p-V indicator diagram can be obtained. Fig. 8 shows the p-V indicator diagram under the operating condition of +30/-15 °C with full-load of slide valve. The V0 shown in the Fig. 8 represents the maximum volume of the working chamber of the compressor.

Fig.8-The p-V indicator diagram of the +30/-15 °C operating condition

The indicated power N_i can be calculated from the p-V indicator diagram. Then the average

value of gas axial force can be expressed as [13]:

$$F_{\text{gav}} = 9545.2 \cdot \frac{2\pi}{T} \cdot \frac{N_i}{n} \quad (3)$$

where N_i represents the indicated power of compressor, T represents screw lead of the male rotor and n represents rotating speed of the male rotor.

3.3. Calculation of the axial force on the end face

The calculation of the axial force on the suction end face is quite simple since the pressure on the suction end face can be considered as the suction pressure. However, the pressure on the discharge end face is quite different. Various pressures, including suction pressure, pressure in the compression process and discharge pressure, are acting on different positions of the discharge end face as shown in Fig. 9. So the calculation of the axial force on the discharge end face is much more complicated. On the other hand, accurate calculation of the axial force on the end face is very important to the calculation of the total composite axial force on the rotor. The mean pressure model and sector pressure model are proposed to calculate the axial force on the end face.

3.3.1. Mean pressure model

The mean pressure model is a conventional model widely used to calculate the axial force on the end face as given in the literature [13]. In this model, it is assumed that half of the discharge end face is under suction pressure and the other half under the average value of the suction pressure and the discharge pressure. Therefore, the axial force on the suction end face, the axial force on the discharge end face and the composite axial force on the end face can be calculated by equations (4) to (6), respectively.

$$F_{\text{es}} = p_s (A_r - A_j) \quad (4)$$

$$F_{\text{ed}} = \frac{1}{4} (p_s + p_d) (A_r - A_j) + \frac{1}{2} p_s (A_r - A_j) \quad (5)$$

$$F_{\text{eav}} = F_{\text{ed}} - F_{\text{es}} = \frac{1}{4} (p_d - p_s) (A_r - A_j) \quad (6)$$

where A_r represents the area of the end face of the male rotor, A_j represents the area of the journal of the male rotor, F_{es} represents the axial force on the suction end face, F_{ed} represents the axial force on the discharge end face and F_{eav} represents the average value of the composite axial force on the end face.

3.3.2. Sector pressure model

In the mean pressure model, the influences of the built-in volume ratio and the area of discharge port on the axial force are not taken into account, which may lead to significant error in the calculation of the axial force on the discharge end face. Therefore, a sector pressure model was introduced in this paper. In the sector pressure model, the discharge end face is divided into several sectors according to the number of lobes of the male rotor. The pressure on each sector equals to the pressure in the cavity between two lobes of the rotor as shown in Fig. 9. Then the axial force on the discharge end face can be calculated based on the areas of sector and the pressure on the corresponding sectors which can be obtained from the p-V indicator diagram recorded. The axial force on the discharge end face can be expressed as:

$$F_{ed} = \sum_{k=1}^5 p_k A \quad (7)$$

where A represents the area of each sector as shown in Fig.9 and can be expressed as:

$$A = \frac{A_r - A_j}{Z_1} \quad (8)$$

The composite axial force on the end face calculated by this model can be expressed as:

$$F_{eav} = F_{ed} - F_{es} = \sum_{k=1}^5 p_k A - p_s (A_r - A_j) \quad (9)$$

Fig.9- The divisions of the discharge end face

4. Results and discussion

4.1. Variations of the composite axial force of gas and end face

The composite axial force of gas and end face is calculated using equation (2) from the experimental pressure values of the positive component, the negative component, the axial force of the shaft shoulder and the axial force of the balance piston. The results are shown in Fig. 10 for data under three operating conditions with full loading of the slide valve.

Fig.10- Variations of the composite axial force of gas and end face

As shown in Fig. 10, the variation frequency of the composite axial force of gas and end face under the three operating conditions was all 250 Hz. This was because the rotating speed of the male rotor was 2960rpm and the number of lobe of the male rotor was 5. The average values of the composite axial force were 20187.95 N, 22022.81 N and 20248.42 N under operating conditions of +30/-15 °C, +35/0 °C and +40/-40 °C, respectively. And the average axial force of the +35/0 °C operating condition was larger than those of the other two operating conditions. This is due to the higher suction pressure and the small built-in volume ratio. Moreover, the fluctuation amplitudes of the axial forces of the three operating conditions were about 800 N, 1500 N and 3000 N, respectively. Due to the significant under-compression and large pressure difference between suction and discharge, the fluctuation amplitude of the +40/-40 °C operating condition was larger than those of the other two operating conditions. Therefore, the balance piston design and bearing selection should take the fluctuation of the axial force into consideration according to practical operating conditions.

The values of the phase angles of the male rotor were not measured in this investigation, but it could be calculated according to the experimental results, as it is shown below. It has

been theoretically illustrated that the axial force is the biggest at the angle where discharge port opens. On the other hand, according to the screw compressor profile data and designation parameters, the phase angle of the peak point could be obtained, which equals the phase angle when the discharge port opens.

4.2. The axial force on the end face

Fig. 11 shows the variations of the axial force acting on the end face versus the rotating angle of the male rotor calculated using the sector pressure model. The variation frequency of the axial force on the end face was also 250Hz because of the same rotating speed and number of lobe of the male rotor. The average values of the axial force on the end face were 9636.92 N, 11136.6 N and 9925.55 N under operating conditions of +30/-15 °C, +35/0 °C and +40/-40 °C, respectively as also shown in Table 3. The average value of the axial force on the end face can also be calculated by the mean pressure model as shown in Table 3. In addition, N_i can be obtained from the experimental p-V indicator diagram. The average value of the axial force of gas, F_{gav} , can then be calculated by the above equation (3). With the experimental results of the composite axial force of gas and end face shown in Fig.10, the average value of the composite axial force of gas and end face was computed as shown in Table 3. At last, the experimental results of the average value of the axial force on the end face were obtained.

Fig.11-The axial force of the end face variations versus the revolving angle of the male rotor

It can be found from the experimental results in Table 3 that the axial force F_{eav} on the end face under three operating conditions are higher than the axial force F_{gav} on the helical surface due to gas pressure. This is quite different from previous understanding, and has significant influence on the design of balance piston and the selection of bearings. Therefore, enough attention should be paid to the calculation of the axial force on the end face during the compressor design period.

Table 3-The axial forces acting on the end face obtained through experiment and different models

Condensing/ Evaporating temperature (°C)	Built-in volume ratio	$F_{\text{gav}}+F_{\text{eav}}$ (N)	F_{gav} (N)	F_{eav} (N)				
				Experimental results	Mean pressure model		Sector pressure model	
					Calculated with model	Deviation from the experimental results	Calculated with model	Deviation from the experimental results
+35/0	2.6	22022.8 1	8978.5	13044.22	4003.12	69.31%	11136.6	14.62%
+30/-15	3.2	20187.9 5	7462.7	12725.25	4174.98	67.19%	9636.92	24.26%
+40/-40	5.0	20248.4 2	6571.2	13677.22	6668.76	51.24%	9925.55	27.43%

The comparison results shown in table 3 indicated that the average value of the axial force on the end face calculated using the sector pressure model had a better agreement with the experimental results than that of the mean pressure model. This meant that the sector pressure model was more accurate in predicting the axial force and hence could be used during the compressor design period. Furthermore, it was found that the deviation of the force calculated by the mean pressure model from the experimental results was getting smaller and smaller as the built-in volume ratio increased from 2.6 to 5.0. This was attributed to the fact that the high pressure area of the discharge end face was much smaller at a higher built-in volume ratio. Therefore, it is concluded that the mean pressure model would be more suitable for the operating condition with high built-in volume ratio than that with low built-in volume ratio.

It was also seen from Table 3 that the results calculated by the sector pressure model was still smaller than the experimental values. This could be partly explained by the fact that the

oil on the end face might have influence on the pressure distribution on the end face.

4.3. Variations of the axial force under part-load conditions

The axial force variations under part-load conditions were also investigated. Fig. 12 shows the variations of the composite force of gas and end face under +30/-15 °C under different part-load conditions. It can be seen that, with the loading decreasing from 95% to 75%, the composite axial force of gas and end face was also decreasing. The average values of the composite axial forces under different part-load conditions were 16227 N, 14930.1 N, 14544.09 N, and 13835.85 N, respectively as the slide valve loading decreased from 95% to 75%. This could be explained by the fact that the rotor length for compression is getting shorter as the loading of slide valve decreased. Moreover, it can be also observed that the fluctuation amplitudes of the composite axial force were gradually increasing with decrease in the loading of slide valve. The fluctuation amplitude was less than 1000 N under 95% loading condition. However, as the loading of the slide valve decreased to 75%, the fluctuation amplitude of the composite axial force rose even up to 2000 N. The main reason was that the built-in volume ratio became smaller as the loading of the slide valve decreased.

Fig.12- Variations of the composite axial force under different part-load conditions

In addition, the average values of the composite axial force of gas and end face and the average values of the axial force on the end face under three operating conditions could also

be obtained according to equation (9) and the experimental results as shown in Fig.13 and 14, respectively.

Fig.13 shows the average values of the composite axial force of gas and end face under three operating conditions with different slide valve loadings. It is found that the average values of the composite axial force decreases as the loading of the slide valve decreases. For example, under +30/0 °C operating condition, with the loading decreasing from 95% to 81%, the average values of the composite axial force decreased from 20922 N to about 17500 N. This can be explained by the fact that the rotor length for compression is getting shorter with the decrease of the slide valve loading. It can be concluded that, with the slide valve loading becoming lower and lower, the axial force F_b of the balance piston becomes larger than the composite positive axial force on the male rotor. Then the direction of the composite axial force on the male rotor could be reversed. Therefore, the variations of the composite axial force of gas and end face versus the slide valve loading are very important to the balance piston design and should be fully considered according to the practical operating conditions of a twin-screw refrigeration compressor to avoid the opposite axial force.

Fig.13- Variations of the average values of the composite axial force of gas and end face under part-load conditions

Fig. 14 shows the average values of the axial force on the end face under three operating conditions with different slide valve loadings. It can be seen that the average values of the axial force on the end face were also decreasing with the decrease of loading of the slide valve.

For example, under +30/0 °C operating condition, the average values of the axial force on the end face decreased from 12600N to about 10600N with the loading decreasing from 95% to 81%. But unlike that of the average values of the composite axial force of gas and end face, the average values of the axial force on the end face under +40/-40 °C operating condition was the highest among the three operating conditions. This could be attributed to the big pressure difference between the suction and the discharge pressure and severe under-compression process of +40/-40 °C operating condition with the built-in volume ratio of 5.0.

Fig.14- Variations of the average values of the axial force on the end face under part-load conditions

5. Conclusions

This paper conducted an experimental study on the rotor axial force under different operating conditions. The mean pressure model and the sector pressure model were used to calculate the axial force on the end face of the male rotor. Through the comparison between the calculated and the experimental results, the following conclusions were obtained:

(1) The variation frequency of the composite axial force of gas and end face depended on the number of lobes and the rotating speed of the male rotor. The suction pressure and the compression ratio had significant influence on the composite axial force. The fluctuation amplitudes of the composite axial forces were mainly affected by the compression ratio of the compressor.

(2) The axial forces F_{eav} on the end face are larger than the axial forces F_{gav} on the helical surface due to gas pressure. This is quite different from previous understanding, and would have significant influence on the design of balance piston and the selection of bearings.

(3) The calculation results of the sector pressure model had better agreement with the experimental results than that of the mean pressure model. The mean pressure model was more accurate for the operating conditions with large built-in volume ratio than that with small built-in volume ratio.

(4) The values of the composite axial force of gas and end face reduce with the decrease of the slide valve loading under part-load conditions. The force direction of the composite axial force on the male rotor may be reversed because axial force of the balance piston will remain the same under part-load conditions, which might be even larger than the composite axial force.

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