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NUMERICAL SIMULATION AND PERFORMANCE ANALYSIS OF ROTARY VANE COMPRESSORS FOR AUTOMOTIVE AIR CONDITIONER

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ABSTRACT

In order to improve the efficiency and reliability of the rotary vane compressors used in the automotive air conditioning systems, a comprehensive simulation model is developed. The simulated results are compared with measured data. The variation of pressure in compression chamber and clearance chamber at various rotational speeds and their effect are explained. The cooling capacity, power, COP, various efficiencies, loss and the reed vane behavior are analyzed. The effect of the change of pressure in papilionaceous oil chamber on compressor performance is discussed. This paper also presents some features of a rotary vane compressor for a CO_2 trans-critical cycle.

INTRODUCTION

The rotary vane compressor used for automotive air conditioner has many advantages, such as simple structure, smaller size, lighter weight, easier to start-up, rotating smoothly etc. But unfortunately there are some disadvantages in this kind of compressor. For example, at lower rotational speed, it has lower cool capacity; at higher rotational speed, it has higher frictional power losses. In order to improve the performance of the rotary vane compressor, a mathematical model of this kind of compressor is developed.

In this paper, the control volume, energy equation, valve model, leakage model and dynamic model of vane and rotor are described. The pressure in compression chamber and clearance chamber at various speeds are explained, and the effect of refrigerant expanding in clearance chamber on compressor performance is described. The behavior of reed valve is discussed. The variations of the valve closing angle and velocity of valve impacting on valve seat are shown. The leakage, frictional loss, volumetric efficiency and mechanical efficiency at various speeds are discussed. The effect of the change of pressure in papilionaceous oil chamber on compressor performance is shown. The comparison of simulated results with measured data is given.

Carbon dioxide (CO_2) is a natural refrigerant and is one candidate of alternatives to HFC refrigerants, so we also discuss the characteristics of a rotary vane compressor for a CO_2 trans-critical cycle using this mathematical model.

MATHEMATICAL MODEL

Control volume

The first step to conduct a compressor simulation is to analyze the working process inside the compressor, to define control volume and to determine the mass exchange between the control volume and its surroundings. Figure 1 shows a schematic view of the rotary vane compressor. As the rotor rotates, the sliding vanes which locate in the slots of the rotor separate the space between rotor and cylinder into several compression chambers. The discharge port and slot transfer constitute the clearance chamber of a rotary vane compressor. When the leading vane of the compression chamber passes the discharge port, the pressure in the clearance chamber is significantly different from that in the compression chamber since the internal compression ratio of a rotary vane compressor is lower. So the refrigerant in the leading chamber and clearance chamber quickly expand to the compression chamber. When the pressure in compression chamber is balanced with that in leading chamber and clearance chamber, the refrigerant in these chambers is compressed together. Considering the feature of rotary vane compressor above mentioned, in this time, the compression chamber, clearance chamber, leading chamber are respectively selected as a control volume.

After the discharge process of the compression chamber begins, the pressure in clearance chamber is below that in the compression chamber due to the throttle at the entrance of discharge hole. The compression chamber is separated from the clearance chamber and is taken as a control volume alone. The clearance chamber is treated as a throttling element instead of a control volume, because the flow velocity through the discharge port is so large that the refrigerant kinetic energy is not negligible. And the mass flow rate discharged from the compressor chamber and the gas force acting on the valve is calculated directly by the pressure difference between the compression chamber and discharge plenum.

For the above control volume, when the energy balance is studied, the internal energy of oil within the control volume is considered and the mass equation of oil is formulated correspondingly in order to accurately evaluate the heating effect of refrigerant and oil leaked on the refrigerant. However, the volume of oil is negligible. The mass equations and the energy equation are described in the reference [3].

Valve

When the discharge valve of a vane rotary compressor opens, the reed valve wraps about its curved backer plate. The free length of reed valve varies with the valve displacement and thus the valve stiffness and effective mass always varies too. In this paper the valve system is treated as a one-freedom degree vibration system with variable stiffness and mass and its differential equation of motion is shown in reference [3].

Leakages

In a rotary vane compressor, the leakage flow paths include the radial clearance between rotor and cylinder, rotor end face clearance, vane end face and side face clearance. The vane tip comes into full contact with the cylinder wall so there is no leakage across the vane tip. Figure 2 illustrates the leakage flow paths in this kind of compressor.

For the radial clearance, leakage is assumed to be homogeneous oil-refrigerant mixture. The mass ratio of oil to refrigerant is proportional to that in the high pressure chamber. The velocity and mass rate of leakage flows is obtained by the Fanno flow equation. Meantime the indirect refrigerant leakage caused by solubility difference between high pressure chamber and low pressure chamber also is considered.

Since the ratio of vane height to clearance height is greatly larger than one, the leakage flow through the vane end face clearance also is assumed as a Fanno flow of refrigerant containing a little oil.

The leakage through vane side face clearance is refrigerant-dissolved oil, so it can be treated as a laminar flow of an incompressible Newtonian fluid, of course the indirect refrigerant leakage also is considered.

The clearance between rotor end face and side plate also has a great effect on rotary vane compressor performance. In this case a gas leakage flow occurs from a high pressure gas chamber to a low pressure gas chamber which are divided by vanes, while an oil leakage flow occurs from a papilionaceous oil chamber which is located in the center of the side plate to a low pressure gas chambers. Free boundary lines are formed between the two leakages flow. In order to obtain the mass rate of leakage flow, a two-phase flow model is used to analyze this leakage flow.

Dynamics

The vane tip gas force has a great effect on vanes behavior especially during the discharge process. In order to accurately evaluate this gas force and the inertia force of vane, the closest point between vane tip and cylinder wall is obtained. Before the point, the pressure at vane tip is the leading chamber pressure and after the point, which is the trailing chamber pressure.

The papilionaceous oil chamber not only supply oil for the motion parts, but also can produce throttling effect during discharge process which increases the vane back pressure avoiding the vane tip detach from the cylinder wall. The continuity equation and Bernoul equation are used to obtain the vane back pressure.

A constant frictional coefficient is used to estimate the vane side frictional forces and a frictional coefficient relationship[6] and a hydrodynamic lubrication model are used to estimate the vane tip frictional force. The pressure distribution on the vane side in the vane slot of the rotor is assumed to be linear from vane back pressure to chamber pressure. The forces acting on the vane, rotating torque and frictional losses at vane and bearing are calculated.
In the dynamic analysis, the chamber pressure is calculated by solving the thermodynamic model equations.

RESULTS AND DISCUSSIONS

Table 1 shows the testing and calculation conditions for the rotary vane compressor. Table 2 shows the comparison of the cooling capacity values, shaft power and COP calculated by the above model with the measured data of a rotary vane compressor with a displacement of 96cc per rotation. The simulated results are consistent with the measured data quite well. Most deviations are less than 3%.

Figure 3 shows the variation of pressure in compression chamber and clearance chamber with the rotational angle of rotor when the rotational speed is 2000rpm. When the leading vane of compression chamber pass the discharge port of rotational angel about 168 degree, the pressure in clearance is about discharge pressure, while the pressure in compression chamber is about 0.56Mpa since this time the internal compression ratio only reach 1.67, so the refrigerant in clearance chamber and leading chamber quickly expand to compression chamber. Correspondingly the pressure in clearance decreases quickly and the pressure in compression chamber increases quickly. From the P-V diagram, shown in Figure 4, it can be seen that this process of a rotary vane compressor has great effect on indicated power but has a little effect on cooling capacity because at this time the compression chamber has passed the suction port. This feature of the rotary vane compressor is different from the reciprocating piston compressor and other type compressors.

Figure 5 shows the variation of volumetric efficiency with rotational speed. Figure 6 shows the variation of leakage loss, suction resistance, and suction heating with the rotational speed. With the rotational speed increases, the volumetric efficiency also increases, but the increasing becomes slowly at high rotational speed due to the suction resistance enhances. And the leakage loss is the main factor affecting the volumetric efficiency. The suction heating has a small effect on the volumetric efficiency and it will decrease slightly at high rotational speed. Figure 7 shows the variation of the cooling capacity loss caused by leakages with rotational speed. It can be seen that the leakage though the rotor radial clearance produces a significant impact on the cooling capacity, while the effect of the leakage through the rotor end face clearance is small as the clearance of tested compressor is small. Since the velocity of vane relative to slot becomes large at high rotational speed, the leakage through the clearance of vane side increases with rotational speed.

As shown in Figure 8, the closing angle of reed valve becomes bigger at the high rotational speed and the velocity of the valve impacting on valve seat increases quickly with the rotational speed.

Figure 9 shows the main frictional loss in the rotary vane compressor. The frictional loss at vane bottom and vane slot edge is directly proportional to the rotational speed basically. The loss at vane tip where mainly is in boundary frictional condition increases quickly with rotational speed especially at high rotational speed. The frictional loss of oil film resistance and bearing where mainly is in hydrodynamic lubrication condition is small, and they have the same amplitude. Figure 10 shows the variation of total frictional loss, suction and discharge loss. Figure 11 shows the variation of indicated efficiency and mechanical efficiency with rotational speed. The mechanical efficiency decreases slightly with rotational speed since the frictional loss at vane tip shown in Figure 9. The indicated efficiency falls at the high rotational speed due to the discharge loss shown in Figure 10.

Figure 12 shows the variation of the cooling capacity, shaft power and COP with the rotational speed. The cooling capacity is proportional to the rotational speed basically. The shaft power at the high speed is relatively large due to the frictional loss and discharge loss. The COP falls at the high rotational speed due to the increase of shaft power.

From the above analysis, it can be seen that the mechanical efficiency of the rotary vane compressor is low. The main reason is that the contact force between vane tip and cylinder wall is large, correspondingly the frictional loss becomes large. Reducing the pressure in papilionaceous oil chamber is an effective method to improve the mechanical efficiency. Figure 13, and 14 show the variation of contact force between vane tip and cylinder wall when the pressure in papilionaceous oil chamber is 1.0, 0.6 times of the discharge pressure respectively. With the pressure decreases, the contact force reduces quickly. When the papilionaceous oil chamber pressure is 0.6 times of the discharge pressure, the contact force becomes zero during discharge process. If the pressure in papilionaceous oil chamber continues to reduce, the leakage will appear between the vane tip and cylinder wall. Figure 15 shows the variation of mechanical efficiency and volumetric efficiency when the papilionaceous oil chamber pressure changes. It can be seen that when the papilionaceous oil chamber pressure is 0.6 times of the discharge pressure, the mechanical efficiency is improved more than 5% and the volumetric efficiency also is improved slightly since the leakage through rotate end face clearance and vane side face clearance decreases.

Based on the mathematical model, we estimate the performance of a rotary vane compressor for a CO_2 trans-critical cycle. The displacement of the CO_2 compressor is 13.5cc, and its suction and discharge port area are assumed to be 1/4 of the R134a compressor, correspondingly the clearances also are reduced. The calculating conditions are shown in Table 1.

Figure 16 shows the volumetric efficiency versus rotational speed. The volumetric efficiency increases with rotational speed, since the leakage loss becomes small at high rotational speed while the suction resistance loss is small even at high rotational speed. This feature can make CO_2 compressor run at high rotational speed with high volumetric efficiency. When the number of vanes increases, the volumetric efficiency can be improved obviously, since the pressure difference of two adjacent compression chambers falls, shown in Figure 17.

For a CO_2 compressor, the moment per unit length acting on vane side out of the rotor slot caused by gas force of two adjacent compression chambers is a main factor affecting the compressor reliability. By increasing the number of vanes, this moment can be decreased obviously, shown in Figure 18. The maximum value of this moment at a 7 vanes CO_2 compressor is about 2 times of the R134a compressor, if the material of vanes is replaced by alloyed steel at a CO_2 compressor, the strength of vanes should be sufficient.

CONCLUSIONS

A comprehensive simulation model of a rotary vane compressor is developed. The simulated results agree well with the measured data for the rotary vane compressor. The variation of the pressure in compression chamber and clearance chamber with the rotational speed are discussed. The valve behavior with the rotational speed is also discussed. The velocity of valve impacting on valve seat rises quickly with the rotational speed. The leakage loss is the dominant factor influence the cooling capacity of the rotary vane compressor. The frictional loss between vane tip and cylinder wall is the main factor affecting the mechanical efficiency. Reducing the pressure in papilionaceous oil chamber can obviously improve the mechanical efficiency, further more the volumetric efficiency also will increase slightly. For a rotary vane compressor for a CO_2 trans-critical cycle, increasing the number of vanes can obviously improve the volumetric efficiency and decrease the moment per unit length acting on vane side out of rotor slot.

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Table 1 the testing or calculating conditions

Refrigerant	Suction pressure	Superheat	Discharge pressure	Subcooling
R134a	0.3Mpa	10℃	1.57Mpa	5℃
CO_2	3.48Mpa	10℃	10.0Mpa	

Table 2 Comparison between the calculated results and the measured data

Speed (Rpm)	Capacity (w)		Power (w)		COP	
	Measured	Calculated	Measured	Calculated	Measured	Calculated
1000	2307.6	2301.9	1122.7	1112.0	2.06	2.07
2000	4816.9	4849.7	2221.5	2213.4	2.17	2.19
3000	7250.5	7346.8	3434.1	3423.0	2.11	2.15

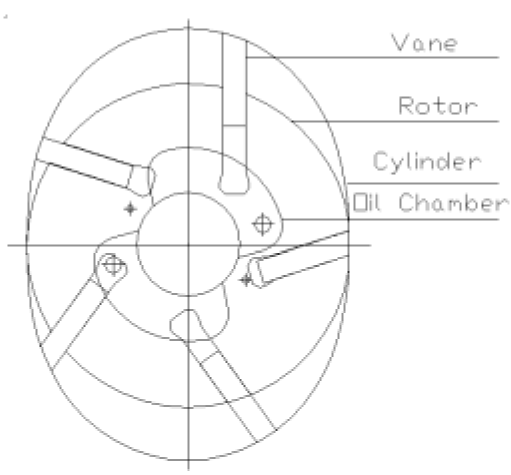

Fig.1 Sectional view of compressor

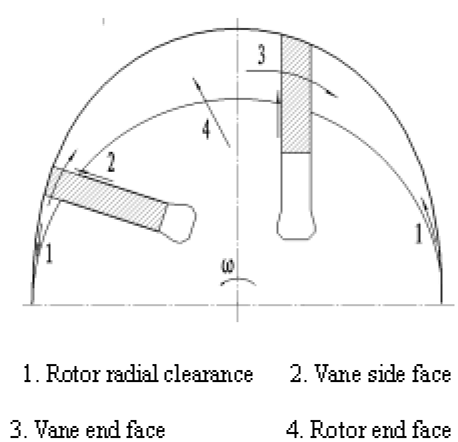

Fig.2 leakage flow paths

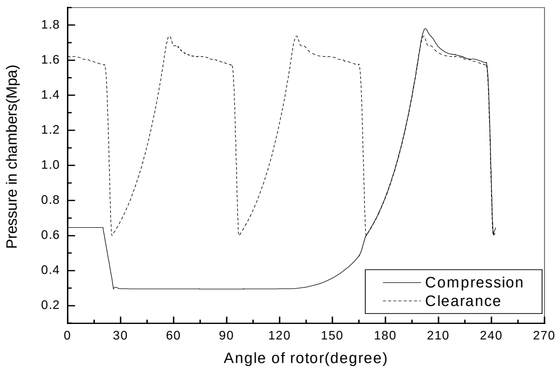

Fig.3 Pressure in working chambers at 2000rpm

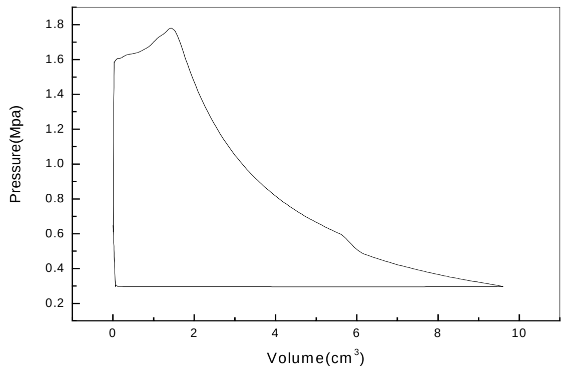

Fig.4 P-V diagram at 2000rpm

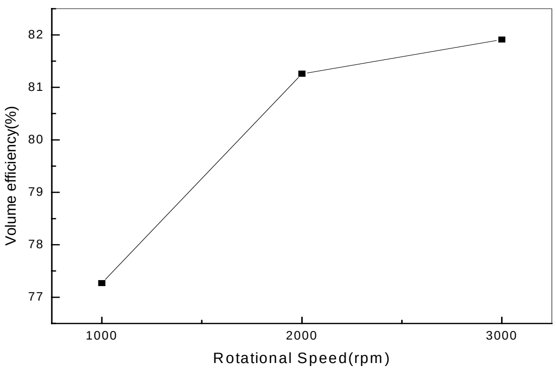

Fig.5 Volumetric efficiency of compressor

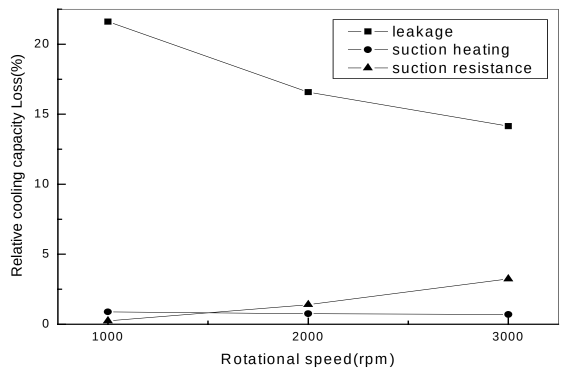

Fig.6 Cooling capacity loss

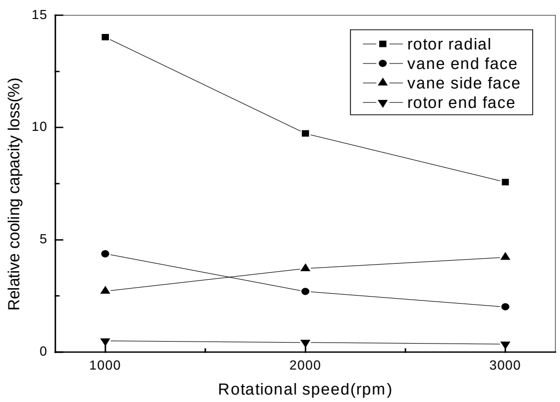

Fig.7 The leakage losses

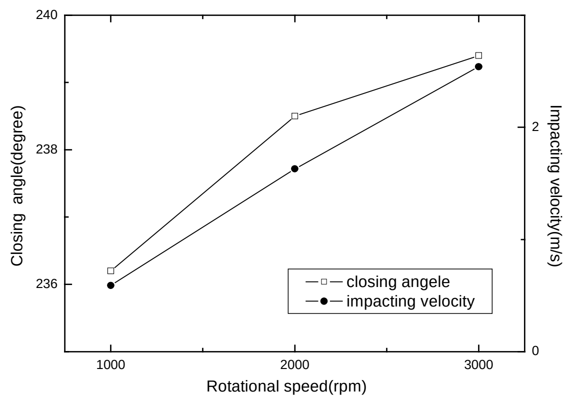

Fig.8 Closing angle and impacting velocity

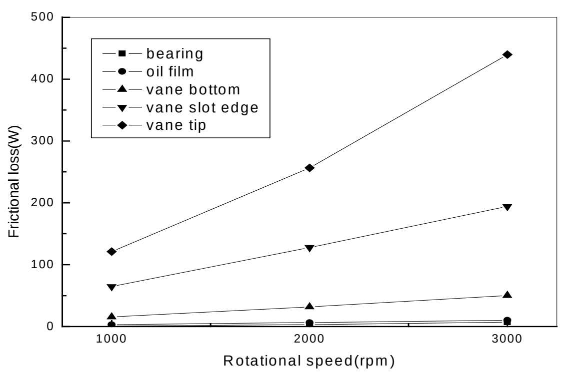

Fig.9 Main friction losses

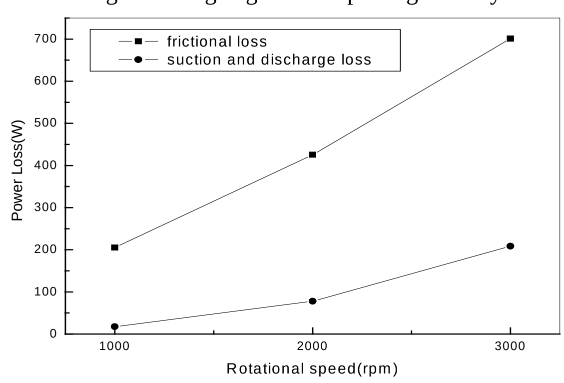

Fig.10 Frictional loss and discharge loss

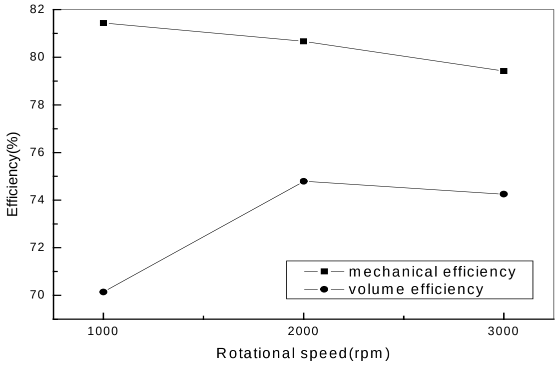

Fig.11 Volume and mechanical efficiency

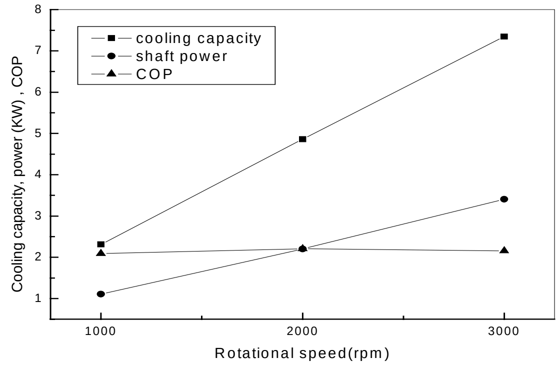

Fig.12 Cooling capacity power and COP

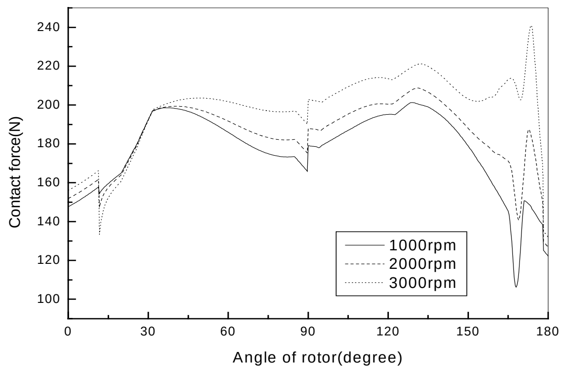

Fig.13 The contact force at 1.0 times p_d

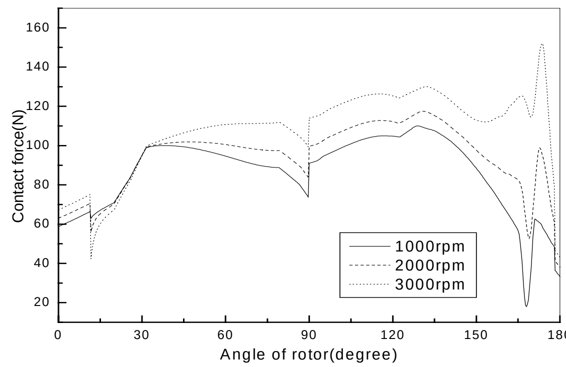

Fig.14 The contact force at 0.6 times p_d

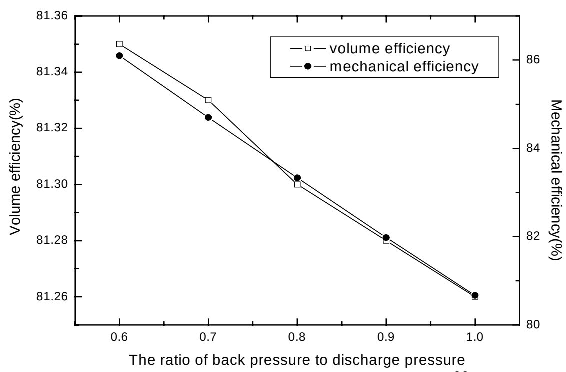

Fig.15 Volumetric and mechanical efficiency

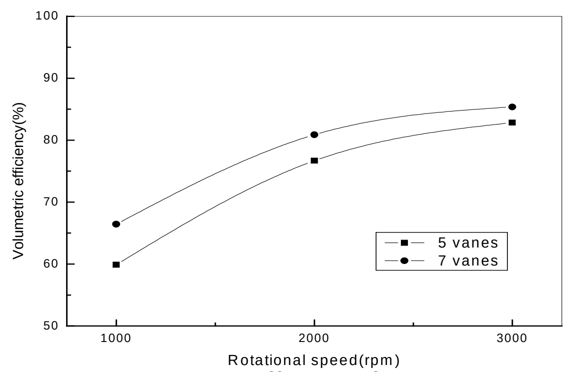

Fig.16Volumetric efficiency of CO_2 compressor

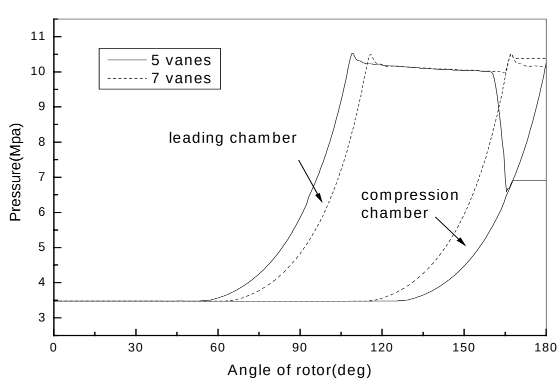

Fig.17 Chambers pressure of CO_2 compressor

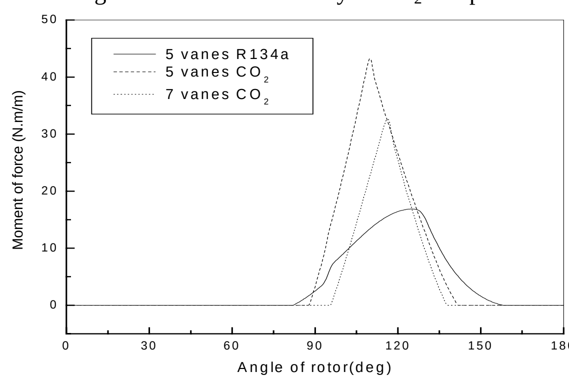

Fig.18 The vane side moment of CO_2 compressor