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International Compressor Engineering Conference

School of Mechanical Engineering

2008

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Modeling and Experimental Investigation on the Internal Leakage in a CO_2 Rotary Vane Expander

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ABSTRACT

A rotary vane expander has been developed to replace the throttling valve for improving the COP of transcritical CO2 refrigeration system. Due to the serious internal leakage, the expander efficiency is not satisfactory. This paper presents the theoretical and experimental investigation on the internal leakage in a rotary vane expander prototype. Firstly, the leakage flow rate through the internal leakage paths with varied clearances were measured under various pressure differences as the expander was static. Then, the nozzle and friction pipe model was used to simulate the flow in the leakage gaps for calculating the leakage flow rate, and the commercial CFD software FLUENT was used to solve the equations in the model. Finally, the calculation results were compared with the experimental data. It was shown that the leakage through the seal arc between the cylinder and rotor is dominant and the percentage is 41%. The leakage through the end surface gaps accounts for 32% and the other leakage was mainly associated with the vane. The rotation speed has little influence on the leakage flow rate, though the volumetric efficiency increases with the speed directly. The seal arc length and the gap clearance has a remarkable effect on the leakage flow rate. Springs put in the vane slot were proven to be able to help the vane contact with the cylinder tightly. The increasing rotational speed can improve the volumetric efficiency of the expander to the value of 60% at the rotational speed of 3000 rpm.

1. INTRODUCTION

Recently CO2 becomes more and more popular in the refrigeration system. The system COP is lower than the traditional refrigeration system due to the high throttling loss associate the throttling valve. Many researchers have made much efforts to develop various expanders to replace the throttling valve for the COP improvement[1-7]. Rotary vane type expander has also been focused because of its good characteristics such as simple structure, small size, light weight and no need of inlet and outlet valves. Compared with the traditional refrigeration system, the pressure difference acting on the suction and discharge ports of the expander is much higher, and the flow rate is much lower. The leakage becomes one of the largest challenge that good expander performance is attempted to achieve, especially for the rotary vane expander. In this paper the leakage flow through the leakage paths on the rotary vane expander have been measured and mathematical mode was established to find appropriate way to solve the leakage problem.

2. THE EXPANDER PROTOTYPE

2.1 Basic structure

As shown in Fig. 1, the double acting rotary vane expander mainly consists of cylinder, rotor, a bearing house, a shaft, two ball bearings and seven vanes. There are two suction and discharge ports. As the vane number is 7, 14 working processes occur in one cycle. The suction process ends as the working chamber reaches the design maxim suction volume and the discharge process begins when the volume of the working chamber reaches the maxim value. Thus the suction and discharge valves are both omitted. Owing to the symmetrical nature of the structure, the radial forces can be balanced, and two ball bearings just need to endure the axial force acting on the shaft. A shaft seal is used to prevent the high pressure gas from leak into the ambient along the shaft.

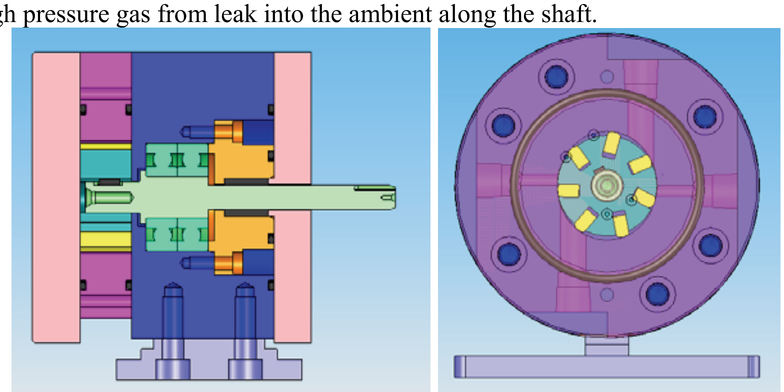


Fig. 1 The expander prototype.

2.2 Leakage paths

There are 5 main leakage paths in the rotary vane expander, as shown in Fig. 2. Leakage path 1 is the leakage directly from the suction port to the discharge port through the small gap between the cylinder and the rotor, i.e. the seal arc. Leakage path 2 is the clearance between the rotor and the end cover. If the vane can not contact the cylinder wall tightly, leakage path 3 exits between the top of vanes and the cylinder wall. Similar as leakage path 2, high pressure gas can pass the clearance between the end surface of the vane and the end cover to the low pressure chamber, called leakage path 4. Leakage path 5 indicates the gaps between the vanes and vane slots in the rotor.

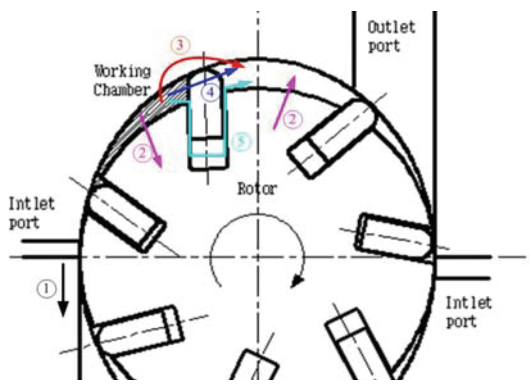


Fig.2 Leakage paths

3. EXPERIMENTAL INVESTIGATION

3.1 Experimental setup

As shown in Fig. 3, the performance test system consists of one high pressure vessel and one low pressure vessel from which the suction and discharge pressure have been attained. The valves before the suction port and after the discharge port can control the working pressure. A nozzle set is used to get the flow rate of the air and a generator is set for the output power of the expander. And the rotation speed of the expander can be adjusted by the load on the generator.

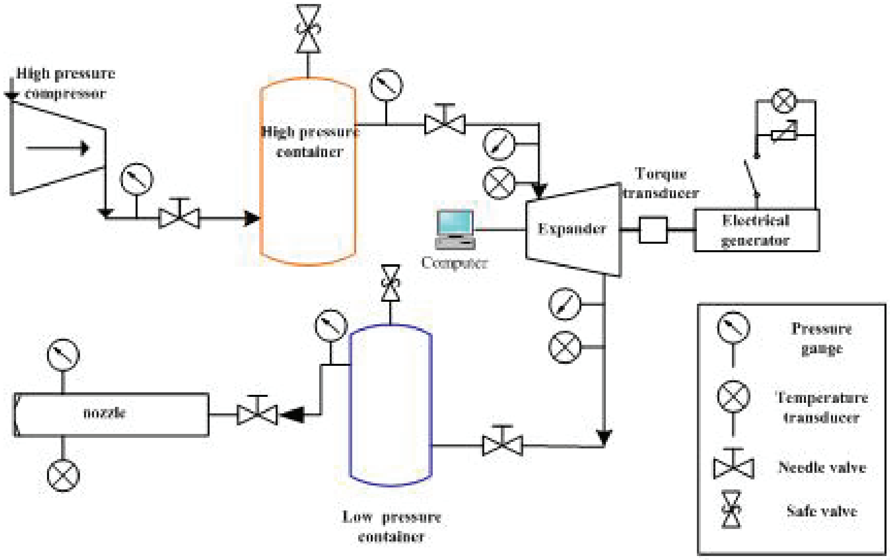


Fig.3 Schematic diagram of the experimental system

In order to get the amount of the leakages, the shaft of the expander is fixed when the expander is filled with air in different pressure and springs are added between the vanes to ensure the contact between the top of the vane and the cylinder wall. The flow rate got by the nozzle set is the amount of the total leakage. As shown in fig.4, seal vane is arranged between the suction and discharge port, which prevent the direct leakage from the suction port and the discharge port. The difference of the measured flow is the leakage from the seal arc. And an aluminum foil is used to forbid the leakage form the gap between the rotor and the end cover, also the difference value is the leakage from the end gap.

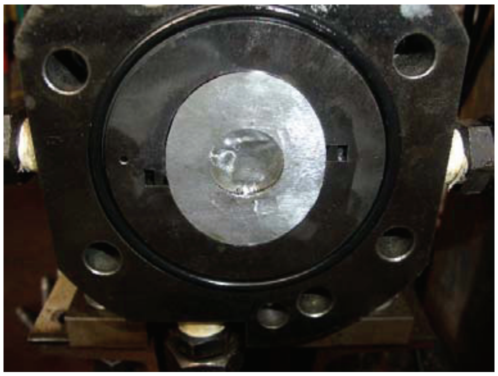


Fig.4 sealing measures

3.2 High Speed Imaging System

A high speed imaging system is used to find whether the top of the vane is contacting with the cylinder all the time. The type of high speed video recorder is Phantom V 7.2 which can get 6888 pictures per second and picture size is 800×600. A transparent end cover is used to replace the steel one in order to find the inner working process.

Fig.5 High speed video recorder

4. THEORETICAL MODEL FOR SEAL ARC LEAKAGE

The leakage through the seal arc is a directly leakage between suction and discharge pressure, and it is depended on two arcs as shown in fig. 6. In this paper, nozzle and friction pipe model is used to analysis the leakage mass flow rate[8].

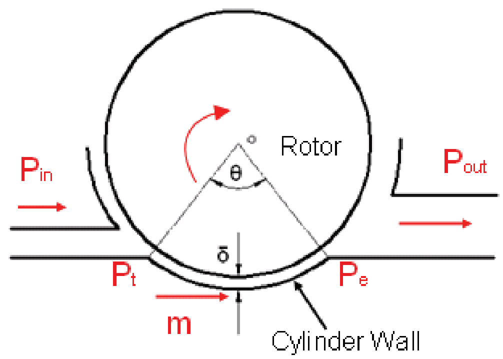


Fig.6 High speed video

According to the gas dynamic theory, Mach number, M and the length of the friction pipe l_f , satisfies Equation (1).

$$\lambda \frac{L_f}{2\delta} = \frac{1}{kM_t^2}\left(\frac{M_e^2 - M_t^2}{M_e^2}\right) + \frac{k+1}{2k}\ln\left[\frac{M_t^2\left(1+\frac{k-1}{2}M_e^2\right)}{M_e^2\left(1+\frac{k-1}{2}M_t^2\right)}\right] \tag{1}$$

Where, λ is the resistance coefficient and $l_f = R \times \theta$.

If the fluid velocity at the exit, e, is equal to the velocity of sound. It can be simple as Equation (2) as follow.

$$\lambda \frac{L_f}{2\delta} = \frac{1-M_t^2}{kM_t^2} + \frac{k+1}{2k}\ln\frac{(k+1)M_t^2}{2+(k-1)M_t^2} \tag{2}$$

Then the pressure rations of $\frac{P_t}{P_e}$ and $\frac{P_c}{P_t}$ can be given as follows:

$$\frac{P_t}{P_e}=\frac{1}{M_t}\left(\frac{k+1}{2+(k-1)M_t^{\ 2}}\right)^{0.5} \tag{3}$$

$$\frac{P_c}{P_t}=\left(1+\frac{k-1}{2}M_t^{\ 2}\right)^{\frac{k}{k-1}} \tag{4}$$

If the total pressure $\frac{P_c}{P_e}$ is less then the given pressure ratio of the expander $\frac{P_{in}}{P_{out}}$, the flow chokes. The Mach number M_e is equal to 1, and the mass flow rate can be gotten by Equation (5).

$$m=\delta L_c P_e v_e \Big/ \left(R_g T_e\right) \tag{5}$$

Where, $Te=Tc/[1+(k-1)M_e^2/2]$, $v_e=M_e\sqrt{kR_gT_e}$.

If there is no choke, the inlet Mach number $M_t^{'}$ should be assumed, and the outlet Mach number can be given by Equation (6).

$$\lambda\frac{L_f^{'}-L_f}{2\delta}=\frac{1-M_e^{\ 2}}{kM_e^{\ 2}}+\frac{k+1}{2k}\ln\frac{(k+1)M_e^{\ 2}}{2+(k-1)M_e^{\ 2}} \tag{6}$$

Where, $l_f^{'}$ can be gotten by Equation（2）using assume value of $M_t^{'}$ .If the new pressure ration $\frac{P_c}{P_e}$ is equal to the actual given pressure ratio $\frac{P_{in}}{P_{out}}$, the right Mach numbers of M_t and M_e can be calculated. Then the leakage mass flow rate is on the way. If the pressure ratios are not equal, the inlet Mach number $M_t^{'}$ will be changed until the right result is gotten.

5. RESULTS AND DISCUSSION

5.1 Leakage distribution in different leakage paths

Fig. 7 shows the flow rates in various leakage paths under different pressure differences. It can be found that leakages increases with the pressure differences, but the increase becomes smaller and smaller. The leakage through the seal arc is the largest one. Under the lower pressure difference, the leakage from the end cover is smaller, but the increasing speed is faster than in the other leakage paths. When the pressure difference is 0.3MPa, the leakage percentages through the end cover, seal arc and the rest leakage paths are 21.89%, 41.76% and 36.35%, respectively. And when the pressure difference reaches 1.7MPa, the percentages change to 32.06%, 41.16% and 26.77%, correspondingly.

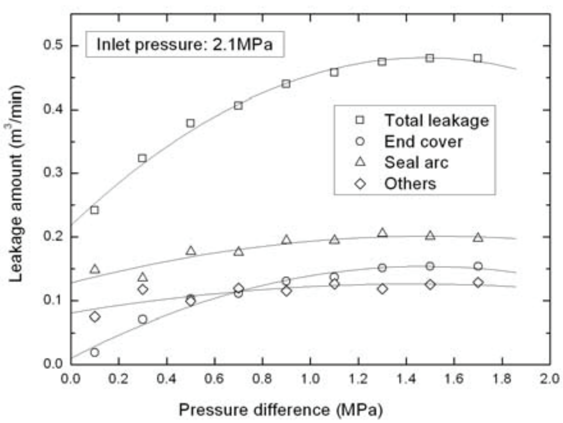


Fig. 7 leakage amounts for different pressure difference

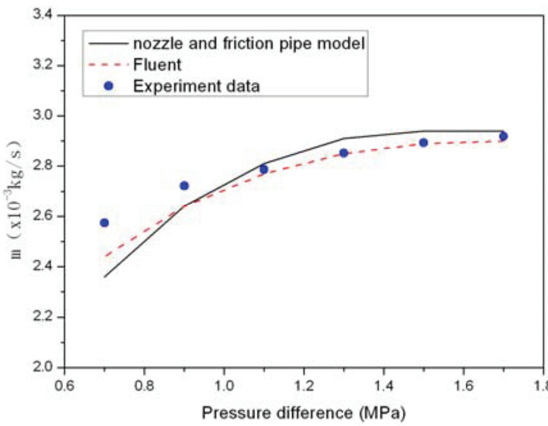


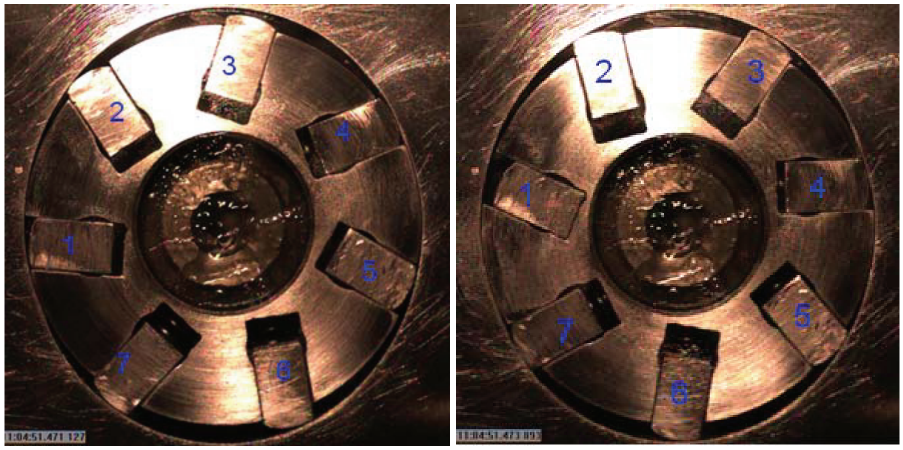
Fig. 8 leakage through the seal arc

5.2 Leakage through the seal arc

The calculated and the experimental results of the leakage flow rate through the seal arc are shown in Fig.8. The solid line is the calculated result from the nozzle and friction pipe model and the dash line is attained by the Spalart-Allmaras model. The experiment results are represented by the dot circle. From this fig, it is shown that the leakage flow rate grows with the pressure differences, but when the pressure difference is larger than 1.3MPa, there is little change in the mass flow rate. Further compared with the results from the three different ways, good agreement has been gotten in the high pressure difference. When the pressure difference is lower than 1.0Mpa, the experimental data is a little larger than the calculated results. But fortunately, the expander prototype is usually operating at high pressure difference conditions. Further simulation is performed by the two different models, focusing the influence of the gap clearance and the length of the seal arc. It is found that the increased angle of the seal arc from 5 degree to 30 degree, the mass flow rate will be decreased about 40%, and the reducing gap clearance from 30 μm to 10 μm the flow rate will be lessen to about 30%. The influence on the rotation speed of the rotor is analyzed, and the decrease in the mass flow of about 3% is found when the rotational speed is 5000rpm, so the concern on the varied rotating speed can be neglect.

5.3 Leakages through the top of the vane

For the rotary vane expander, the closure of the working chamber requires that the adjacent vanes which contact the cylinder wall tightly. Fig. 9 shows two pictures caught by the video which records two of the working process. In the test, springs were put in four slots. It can be seen that the vanes without springs do not contact with the cylinder wall tightly. When the vanes pass the inlet port, the high pressure gas coming through the port pushes the vanes back to the vane slot. If there are springs under the vane, it will come back quickly to contact the cylinder. But for the vane without spring, it will come back much slowly and impact the cylinder wall and rebound back, and the impacting and rebounding processes will last until the discharge port is open. By examining the three vanes without springs carefully, the movement characters are found to be slightly different. Vane1 and vane 3 almost has the same working condition as discussed above. But vane 2, similar to vanes with springs, can contact the cylinder wall tightly almost all the time, which agrees with the p-t diagrams analysis[7].



a) rotational angle1 b) rotational angle 2

Fig. 9 vane position in the working process

6. CONCLUSIONS

In this paper, leakages through the three main leakage paths of the rotary vane expander have been investigated by mathematical models and experiment. Some conclusions can be drawn as follows:

1) Among various leakages paths, the leakage from the seal arc is the most serious, and the percentage is about 41%, when the leakage from the end cover is about 32% and the rest is 27%.

2) The mass flow rate through the leakage paths increase with the pressure difference, but the increasing speed becomes smaller and smaller.

3) The nozzle and fiction pipe model and Fluent can satisfactorily predict the leakage flow rate through the seal arc especially at high pressure difference. By the two models, it can be found that enhancing the length and reducing the gap clearance of the seal arc can decrease the leakage mass obviously. And the rotational speed has little influence on the leakage.

4) The vanes without springs' help can not contact the cylinder wall tightly, but the pressure difference between adjacent chambers can make the middle vane between the three vanes work well.

5) When the rotational speed is 500 rpm, the volumetric efficiency of the expander is about 10%. But the increasing rotational speed can improve the volumetric efficiency of the expander to the value of 60% at the rotational speed of 3000 rpm.

REFERENCES

[1] Baek J S, Groll E A, and Lawless P B. Piston-cylinder work producing expansion device in a transcritical carbon dioxide cycle. Part I: experimental investigation [J]. International Journal of Refrigeration, 2005, 28 (2): 141-151.

[2] Huff H. and Radermacher R. Experimental Investigation of a Scroll Expander in a Carbon Dioxide Air-Conditioning System [C] // Proceedings of International Congress of Refrigeration. 2003. Washington, D.C: [s. n.], 2003

[3] Nickl J, Will G, Quack H, et al. Integration of a three-stage expander into a CO2 refrigeration system [J]. International Journal of Refrigeration, 2005, 28 (8): 1219-1224.

[4] Fukuta M. and Yanagisawa T. Performance prediction of vane type expander for CO2 cycle [C] // Proceedings of International Congress of Refrigeration. 2003. Washington, D.C: [s. n.], 2003

[5] Zhang Bo, Peng Xue-yuan, Zhang Fang-xi, et at. Development and Experimental Validation of Novel Free Piston Expander [J] Journal of Xi'an Jiaotong University. 2006, 40 (7): 776-780. (Chinese)

[6] Li Min-xia, MA Yi-tai, SU Wei-chen, et al. Research and Development of Rolling Piston Expander in CO2 Transcritical Cycle. [J] Journal of Tianjin University. 2004, (9). (Chinese)

[7] Yang Bingchun, Guo Bei, Peng Xueyuan, et al. Experiment Research on the Internal Working Process of a Rotary Vane Expander in the CO2 Refrigeration System [J]. Journal of Xi'an Jiaotong University. 2008, (3) (Chinese)

[8] T.yanagisawa and T.shimizu Leakage losses with a rolling piston type rotary compressor. I. Radial clearance on the rolling piston [J]. International Journal of Refrigeration, 1985, 8: 75-84.