ICR07-B2-431

# TESTING AND ANALYZING ON P-V DIAGRAM OF $CO_2$ ROLLING PISTON EXPANDER

**Xianyang ZENG , Yitai MA , Shengchun LIU , Hongli WANG**
Thermal Energy Research Institute, Tianjin University,
Tianjin, 300072, China
+86-22-27404741, ytma@tju.edu.cn

## ABSTRACT

Developing high efficient expander replacing throttle valve is one of the effective methods of improving performance of $CO_2$ trans-critical refrigeration cycle. Testing on P-T and P-V diagram of $CO_2$ rolling piston expander is discussed. The results show that expansion work in P-V diagram agree with practical recovery power. It is also shown that three different expansion processes can be occurred in expander. Owing to the theoretical expansion $V_r$ in expander is constant after manufacturing, the expander will normal expand if the expansion ratio in practical working conditions equals to $V_r$; while the expansion ratio is larger than $V_{r,}$ over-expansion will be produced; under-expansion may be generated as the expansion ratio is lower than $V_{r.}$ Therefore, some measures that overcoming over-expansion and under-expansion to match the practical work condition of expander are discussed.

## INTRODUCTION

For the deterioration of ozone depletion and greenhouse effect, research on the application of natural working fluid $CO_2$ has been a considerable increase in the interest and development activity internationally. Thermodynamic analysis shows that throttling loss of the expansion valve is the largest percentage of total irreversibility of the $CO_2$ transcritical cycle system. Several cycle modifications, such as staging of compressor and expansion, internal heat exchanger, ejector, vortex pipe and expander, are taken to replace the expansion valve to decrease the system total irreversibility loss and improve the cycle performance. Among the state of the arts, replacing the throttling valve with an expander is considered to be one of the most effective ways.

According to the thermodynamic and transport properties of $CO_2$, Prof. Lorenzen (1993) suggested that replacing the throttling valve with an expander to extract and make use of the expansion work in $CO_2$ transcritical cycle. Maurer et al (1999) investigated the axial piston and gear pump expanders, and shown its volumetric efficiency was about 30% and 68%, and the isentropic efficiency was about 41% and 55%, respectively. Beak et al (2005) carried out experiments with a piston-cylinder expansion device, the isentropic efficiency reported approximately 11% and increased the COP of the system up to 10.5%. Huff et al (2003) presented their experiment on scroll expander, its maximum volumetric efficiency was 68% at 2200 RPM and the maximum isentropic efficiency was 42% at 1800 RPM. Nickl et al (2002, 2003, 2005) devoted themselves in the development of free piston expander and first publish their work in 1994. In their third generation design of $CO_2$ expander, three-stage expansion process was realized and the expected efficiency is 68%. Ma et al (2006) have been researched on $CO_2$ expander in 2000 and the third generation

$CO_2$ rolling piston expander has been manufactured, the adiabatic efficiency is more than 30% and the maximal efficiency reached to 46%.

In a word, the efficiency of the expander is low in the open literature, only about 40% presently. Therefore, it is necessary that further improving the efficiency of expander and promote the $CO_2$ transcritical cycle into practice. In this paper, based on the $CO_2$ transcritical cycle water-water heat pump system with expander, test rig of P-V diagram of the rolling piston expander is designed and measured, some useful results and operational regularity of the expander are attained.

## 1. EXPERIMENTAL SET-UP

### 1.1 Water-water heat pump system with expander

Fig.1 shows the $CO_2$ transcritical cycle water-water heat pump system with expander, it is composed of $CO_2$ compressor, gas cooler, throttle valve, evaporator, gas-liquid separator, rolling piston expander, cooling water system, chiller water system and data acquisition system.

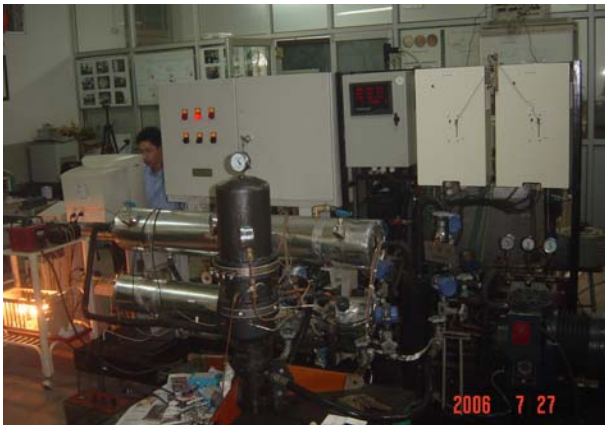



Fig.1 Water-water heat pump system with expander

### 1.2 Test rig of P-V diagram system

The test rig of P-V diagram system mainly includes pressure sensor, charge amplifier and data collector. The pressure sensor is 601A piezoelectric sensors made in Kistler and 5039A212 charge amplifier is selected to match for the pressure sensor. The A/D data collector's maximal sampling frequency is 100KHz, it can pick 10,000 data in one second. The connection schematic map of test rig system can be shown in Fig.2, and the P-V diagram of the rolling piston expander test rig system, schematically shown in Fig.3.


Fig.2 Connection schematic map of P-V diagram test rig system

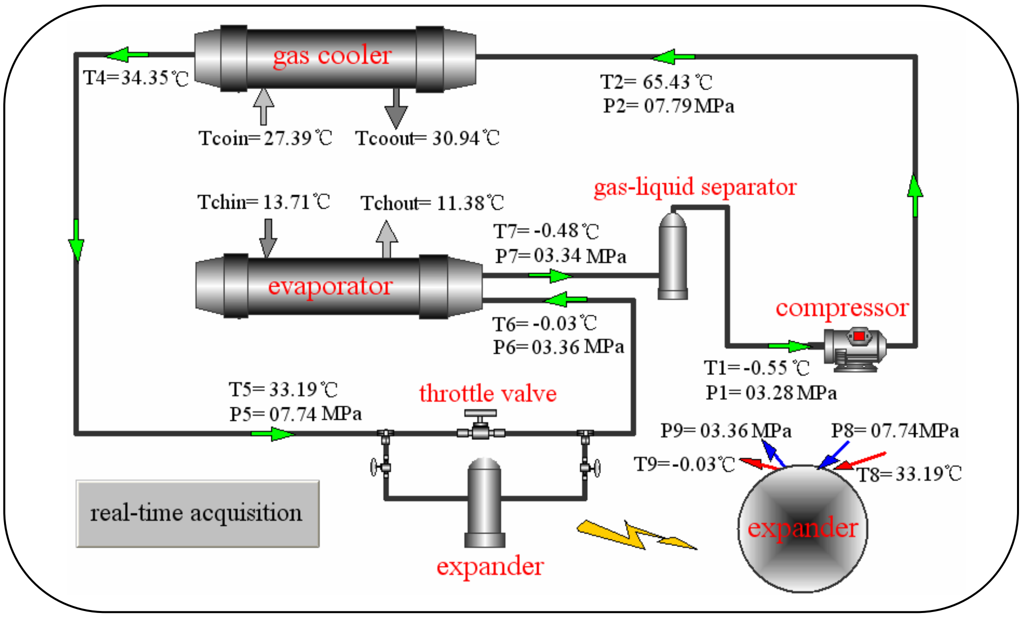



Fig.3 Schematic view of P-V diagram of rolling piston expander test rig system

## EXPRIMENT RESULTS ANALYSIS

### 2.1 Comparisons of the P-T diagram

In order to analysis P-T diagram in different expansion process, three typical working conditions of the rolling piston expander are given in Table 1.

As shown in Fig.4, there are 29.5 periodic variation curves in the P-T diagram in working condition of Ⅰ, agreeing with the rotational speed in one second in expander (1770/60=29.5). Therefore, the pressure sensor and data acquisition system are responded and recorded the pressure change of expander entirely. In Fig.4, pressure increasing slowly before refrigeration suctioned into expander, when suction valve is opened, inlet pressure reaching to maximum quickly, and pressure pulses are also come into being. While suction valve is closed and expansion process is beginning at the same time, the inlet pressure is decreased rapidly. Because of the rotational speed of expander is higher than designed speed (1500rpm), over-expansion occurred at the anaphase of expansion process. When expansion chamber is connected with exhaust vent, pressure of expansion chamber (outlet pressure) raised to exhaust pressure as the refrigerant in evaporator partly backflow.

Table1 Operational parameters of the rolling piston expander

| Working condition | Rotational speed (rpm) | Inlet pressure (MPa) | Inlet temperature (℃) | Outlet pressure (MPa) | Expansion Pressure (MPa) | Expansion ratio | Expansion type |
|---|---|---|---|---|---|---|---|
| I | 1770 | 7.50 | 32.70 | 3.75 | 3.25 | 2.72 | over-expansion |
| II | 1540 | 7.66 | 33.20 | 3.76 | 3.65 | 2.53 | normal-expansion |
| III | 650 | 7.54 | 33.30 | 3.06 | 3.85 | 2.23 | under-expansion |

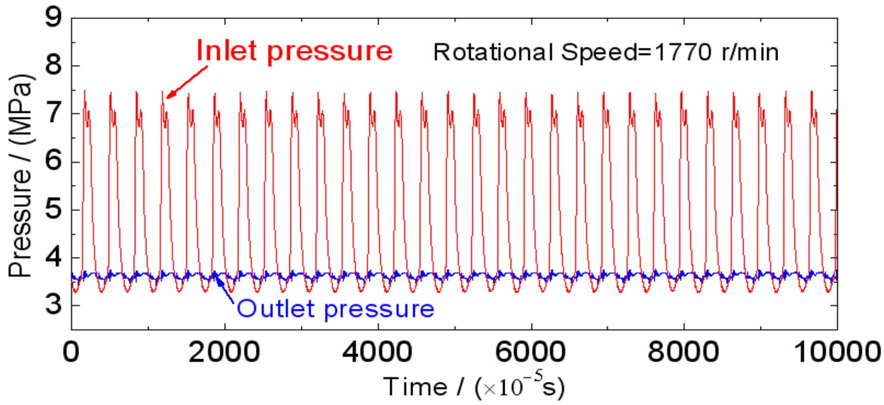



Fig.4 P-T diagram in working condition of Ⅰ

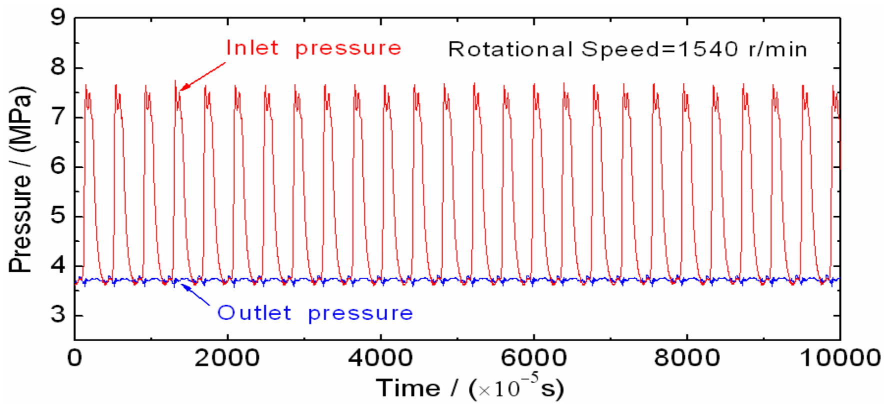



Fig.5 P-T diagram in working condition of Ⅱ

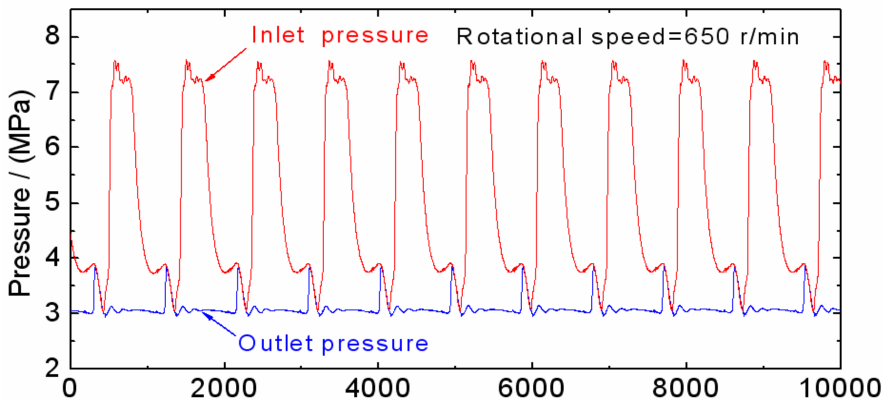



Fig.6 P-T diagram in working condition of Ⅲ

The P-T in diagram working condition of Ⅱ is shown in Fig.5, the pressure change of suction and exhaust vent just like Fig.4, but over-expansion is not occurred as the rotational speed of the expander primary reached to designed speed. It is shown that in this working condition the expansion process primarily attained to the designed value, normal-expansion occurred in expander.

The P-T diagram in working condition of Ⅲ is shown in Fig.6, the rotational speed is lower than designed speed obviously, and under-expansion occurred at the anaphase of expansion process. When expansion chamber is connected with exhaust vent, pressure of expansion chamber (inlet pressure) dropping abruptly, and pressure of exhaust vent (outlet pressure) of expander increased clearly, then the two pressure equivalent until next expansion process is begin.

## 2.2 Comparisons of the P-V diagram

### 2.2.1 Analysis of three kinds of expansion process

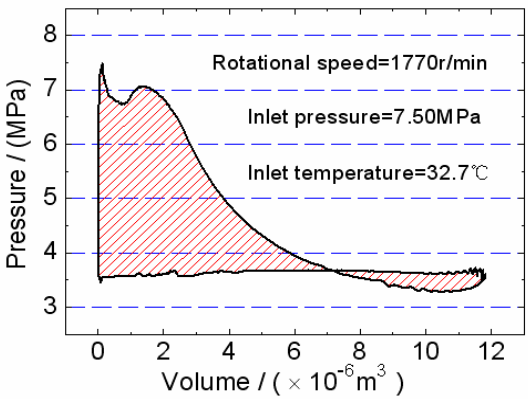



Fig.7 P-V diagram in working condition of Ⅰ

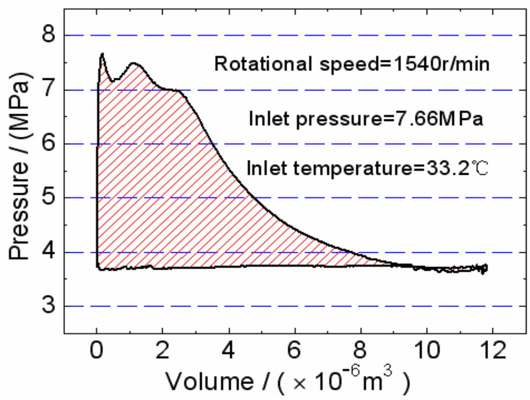



Fig.8 P-V diagram in working condition of Ⅱ

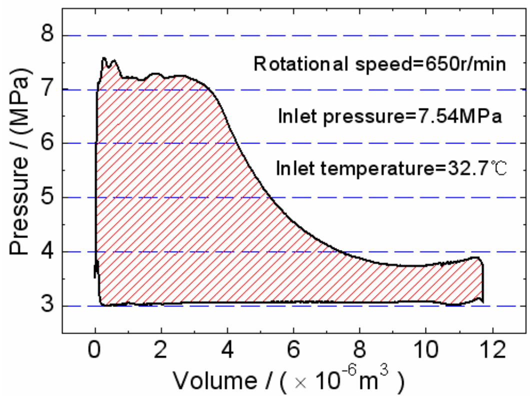



Fig.9 P-V diagram in working condition of Ⅲ

Fig.7~Fig.9 are shown the P-V diagram in three typical working conditions of the expander, and their operational parameters are given in Table 1. It can be seen from Fig.7 to Fig.9 that three kinds of probable expansion processes inside the expander are divided, namely over-expansion, normal-expansion and under-expansion. This can be determined by the structural parameters of the expander. The designed working conditions of rolling piston expander are as following:

- evaporation temperature: 5℃
- inlet pressure : 9MPa
- inlet temperature: 35℃.

According to the structural parameters and thermodynamic calculation of the expander, its theoretical expansion ratio $V_r$ is 2.52, which equals to the volumetric ratio between the volume of expansion chamber beginning exhaust and the volume of suction chamber when suction valve is closed. So the rotational angle of the expander is reached to beginning expansion angle, the suction valve is closed and suction chamber turned into expansion chamber. The hour of closed or opened of the suction valve are decided by $V_r$, that is to say, by the lift range of disc cam of the suction valve. Once the lift range is fixed, the volumetric ratio keeps fixedness, even if the operational working condition deviated from designed working condition. When the expansion ratio of the expander equals to $V_r$ in practical working conditions, the expander is normal-expansion; if the expansion ratio large than $V_r$ in practical working conditions, over-expansion phenomenon can be occurred, that is to say, the discharge pressure of the expander is lower than evaporator pressure; while the expansion ratio lower than $V_r$ in practical condition, under-expansion phenomenon may be generated, that is to say, the outlet pressure of the expander is higher than evaporator pressure.

The expansion ratio of the expander in practical working conditions is shown in Table 1, assuming efficiency of the expander is 40%, and expansion pressure refers to the pressure value of zero rate of change in quick decompression of the expander. Due to the expander cannot operate strictly in designed working condition; the practical expansion ratio always deviated from theoretical expansion ratio $V_r$. When the rotational speed of the expander is 1770rpm (Fig.7), the expansion ratio is 2.72, larger than $V_r$, over-expansion of the expander can be occurred. When the rotational speed of the expander is 1540rpm (Fig.8), the expansion ratio is 2.53, reaching to $V_r$ primarily, normal-expansion come into being. While the rotational speed of the expander is 650rpm (Fig.9), the expansion ratio is 2.23, lower than $V_r$, under-expansion phenomenon is generated.

### 2.2.2 Comparison of expansion work

As shown in Fig.7~Fig.9, replacing the throttle valve in a transcritical $CO_2$ refrigeration cycle by the expander recovers expansion work, which can be calculated directly from the dashed area in P-V diagram ($\int VdP$). As shown in Table 2, the expansion work calculated in P-V diagram are smaller than the tested one, this may be the systematic error existed in the test rig.

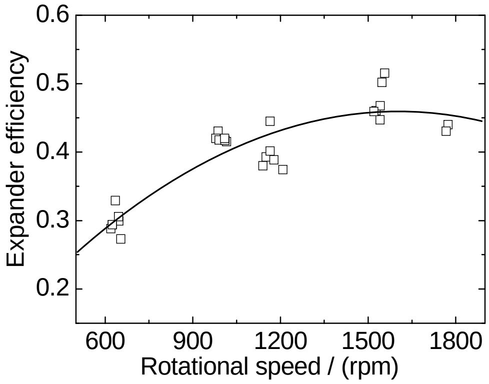
As seen from the single P-V diagram of the expander in Fig.7 to Fig.9, the expansion work of under-expansion is larger than over-expansion or normal-expansion one. But the under-expansion occurred in low speed and the times of work are less than over-expansion or normal-expansion in unit of time. Therefore, the expansion work of under-expansion is smaller than others in certain time, and the tested values of expansion work in Table 2 are also illustrated it.

Table 2 Expansion work between in P-V diagram and tested

| Working condition | Rotational speed (rpm) | Inlet pressure (MPa) | Inlet temperature (℃) | Outlet pressure (MPa) | Expansion Pressure (MPa) | $-\int VdP$ (W) | Tested (W) | Errors (%) |
|---|---|---|---|---|---|---|---|---|
| I | 1770 | 7.50 | 32.70 | 3.75 | 3.25 | 391.7 | 427.0 | -8.3 |
| II | 1540 | 7.66 | 33.20 | 3.76 | 3.65 | 403.2 | 450.5 | -10.5 |
| III | 650 | 7.54 | 33.30 | 3.06 | 3.85 | 250.2 | 268.0 | -6.6 |

### 2.3 Influences of rotational speed of expander on system performance

The target of development expander is makes the efficiency of expander highest, meanwhile, the COP of the $CO_2$ transcritical cycle system is optimal. The rotational speed of expander synthetically reflects all kinds of operational parameters, such as inlet pressure, inlet temperature, mass flow rate and load，so rotational speed is treated as variable in the following discussion on system performance.



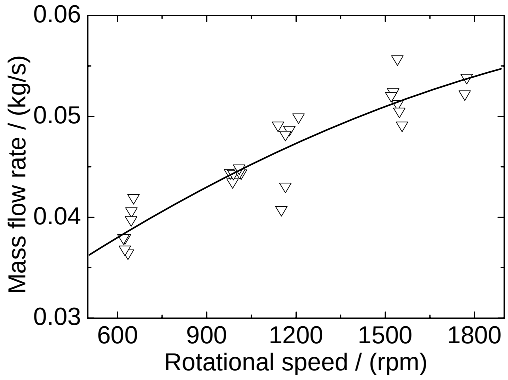
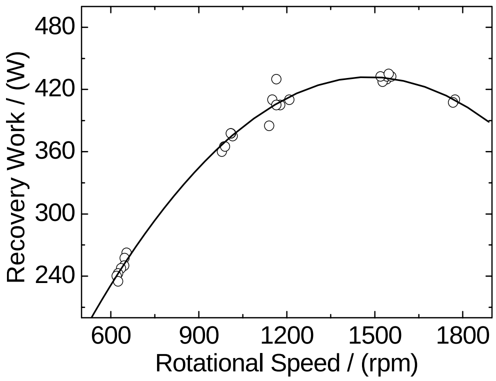




Fig.10 Influence of rotational speed on efficiency of expander

Fig.11 Influence of rotational speed on mass flow rate of expander



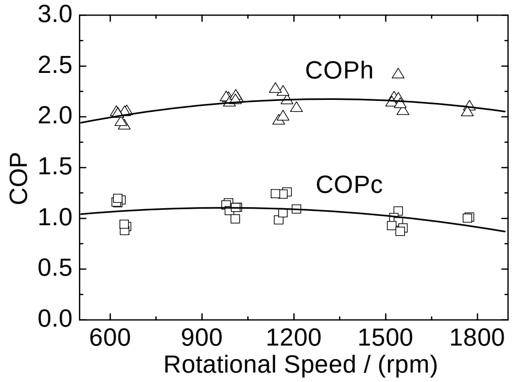




Fig.12 Influence of rotational speed on recovery work of expander in practical working conditions

Fig.13 Influence of rotational speed of expander on COP of system

Fig.10 shows the influence of rotational speed on efficiency of expander. With the increasing of rotational speed, efficiency of the expander also increases. However, when the rotational speed is exceeded by designed speed, efficiency of the expander decreases. The main reason is that under-expansion is occurred when the rotational speed of expander is low. The lower of the rotational speed, the more serious of under-expansion, and the more work-performing capacity lost of the refrigerant after expansion, and the lower of the efficiency of the expander. When the rotational speed is increasing, normal-expansion occurred in the expander, more expansion work produced and the efficiency of expander increases. Over-expansion created when rotational speed is higher than designed speed, parts of expansion work are wasted by offsetting over-expansion, and friction loss is also enlarged, thus leading to the efficiency of expander decreases. It is proved by the experimental results that the efficiency of the expander is optimal while it is operated near the designed rotational speed.

The influence of rotational speed on mass flow rate of expander is shown in Fig.11. With the increasing of rotational speed, the mass flow rate is also increasing forward, but when the rotational speed is exceeded by designed speed of expander, the range of increasing of rotational speed is decreasing. The main reason is that the designed mass flow rate of expander match to the displacement of compressor, when the rotational speed of expander is exceeded by designed speed, as the rotational speed of the compressor is fixed, that is to say, the compressor's displacement is constant, so the mass of refrigerant is limited, the refrigerant after expansion will be stored in the evaporator and can not wholly enter into refrigeration cycle, thus the range of increasing of mass flow rate inevitably decreased.

Fig.12 shows the influence of rotational speed on recovery work of expander. With the increasing of rotational speed, under-expansion is turned into normal-expansion in the expander, the total loss of friction and leakage is smaller, the mass flow rate of expander is also enlarged and recovery work of the expander enlarged clearly. But the rotational speed exceeded by designed ones, over-expansion is occurred, parts of expansion work are gone to waste by offsetting over-expansion and the total loss of friction and leakage also enlarged, the efficiency of the expander decreasing. It is proved by the experimental results that the recovery work of the expander is maximal as it is operated near the designed rotational speed.

The influence of rotational speed of expander on COP of the system is shown in Fig.13. With the increasing of rotational speed of expander, the COP at the heating condition is increasing and reaching to maximal when the rotational speed around the designed one. But the COP at the cooling condition of the system is decreasing slightly with the increasing of rotational speed of expander. This may be done by the gas cooler of the system. As for the transcritical cycle, temperature and pressure are not coupled at supercritical state. The temperature and pressure of refrigeration located in the left or right of critical point are important for the COP of the system. Apparently, it is much higher when the temperature and pressure of refrigeration located in the left of the critical point. As heat transfer area of the gas cooler in test rig is small, the temperature of outlet refrigerant from gas cooler is high, thus leading to the COP at the cooling condition is low, this is a shortage and needs for improving at next step.

## 3. DISSCUSSION

In fact, over-expansion and under-expansion of expander are unsteady process in certain time. Under the condition that the structural parameters of expander are designed appropriately and long time operation of the expander, the operational parameters in refrigeration system such as temperature and pressure are varied correspondingly, the rotational speed and mass flow rate are also changed, the abnormal-expansion process such as over-expansion and under-expansion of expander can be tended to normal-expansion process gradually.

The displacement of the rolling piston expander is fixed on a certain rotational speed, if the suction valve's opening and closing time were designed variable and matched to the change of operational parameters in refrigeration system, normal-expansion can be occurred in the expander.

If compressor and expander integrated on a common shaft, and displacement volumes of compressor and expander are designed appropriately, Huff (2003) indicated that the expander cycle could be operated at or close to the optimum high-side pressure without variable expander displacement or additional control valves. Therefore, normal-expansion can be occurred in the expander.

## CONCLUSIONS

From the P-V diagram, it can be seen that three kinds of expansion process in expander inside the expander are divided, namely over-expansion, normal-expansion and under-expansion. When expander is operated around designed rotational speed, normal-expansion is occurred; and over-expansion is generated when the expander rotational speed is higher than designed speed; but under-expansion is produced when the expander rotational speed is lower than designed speed.

Three kinds of probable expansion processes can be determined by the structural parameters of the expander. When expander is operated around designed speed, expansion ratio of the expander equals to $V_r$ in practical working condition, the expander is normal-expansion; When expander is operated larger than designed speed, the expansion ratio large than $V_r$ in practical working condition in practical condition, over-expansion phenomenon can be occurred; while expander is operated lower than designed speed, the expansion ratio lower than $V_r$ in practical condition, under-expansion phenomenon may be generated.

The rotational speed of expander synthetically reflects of all kinds of operational parameters. It is proved by the experimental results that the efficiency and recovery work and COP of the expander is maximal as it is operated near the designed rotational speed.

## REFERENCES


Lorentzen G. Large heat pumps using $CO_2$ refrigerant. *IIR Commission B1/2*, Ghent, Belgium, 1993,191-197.

Maurer T., Zinn T. Experimentelle Untersuchung von Entspannungsmaschinen mit mechanischer Leistungsauskopploung für die transkritische $CO_2$-Käeltemaschine. *DKV-Tagungsbericht 26*, Berlin, 1999, 304-318.

H.J.Huff, R.Radermacher, M.Preissner. Experimental investigation of a scroll expander in carbon dioxide air-conditioning system. *International Congress of Refrigeration*, Washington, D.C. USA, 2003: ICR0485.

Nickl J, Will G, Kraus W E, Quack H. Design considerations for a second generation $CO_2$-expander. *5th IIR-Gustav Lorentzen conference on natural working fluids*, Guangzhou, China, 2002, 189-196.

Nickl J, Will G, Kraus W E, Quack H. Third Generation $CO_2$ Expander. Proceedings of the *21th International Congress of Refrigeration*, Washington, September, 2003, 17-22.

J. Nickl, G. Will, H. Quack, W.E. Kraus. Integration of a three-stage expander into a $CO_2$ refrigeration system. *International Journal of Refrigeration*, 2005, 28 (8):1219-1224.

Haiqing Guan,Yitai, Ma; Minxia Li. Some design features of $CO_2$ swing piston expander. *Applied Thermal Engineering*, 2006, 26(2-3): 237-243.

H.J.Huff, R.Radermacher. $CO_2$ Comperssor-expander analysis. 2003, *ARTI-21CR Final Report*, 611-10060-01.