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# Simulation analysis of a two rolling piston expander replacing a throttling valve in a conventional refrigerant heat pump system

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## ABSTRACT

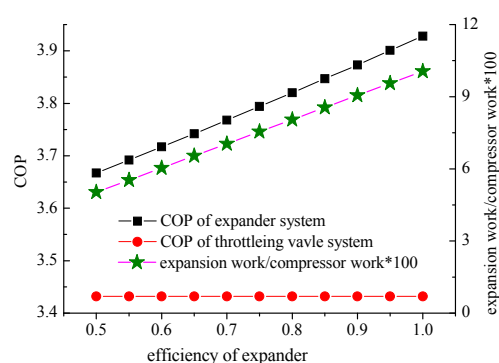
An expander is employed to replace a throttling valve in a heat pump system of the conventional refrigerant in order to further improve the efficiency of the system by recovering partial expansion work to decrease energy consume of the system. Due to large expansion ratios of conventional refrigerants, a two-rolling piston one-stage expander was proposed and designed. Simulation analysis was developed by analyzing irreversible losses in the expansion work process. The simulation results show that, friction losses of a conventional refrigerant expander are 80.2% of total losses, which are the main factors to influence the efficiency of expander. Then the measures of reducing friction losses are presented, which would be expected to improve the expander efficiency efficiently.

## 1. INTRODUCTION

In recent years, refrigeration and heat pump units have been widely applied in China with the development of national economy and improvement of people's living standards, which also led to huge energy consumption simultaneously. Under the background of global energy shortage, it is of great importance to heighten system efficiency to save energy and meet the increasingly stringent national energy efficiency standards.

Many efficient efforts have been investigated to improve system efficiency, including the design and analysis of cycle principle, high efficient compressors (screw, centrifugal etc.), throttle mechanism, high efficient heat exchanger etc.[N. Stosic, 2003]. For example, isentropic efficiency of a medium-sized compressor by optimizing study can be higher than 80%[DORIN, 2011]. The heat transfer coefficient of a plate heat exchanger can reach up to 2000W/(m<sup>2</sup>·K)[Anjun Li, 2008]. And the control of system has been described to be close to perfect[Craig Cordell, 2003]. While, in comparison with those designations, expansion part is to be considered less[Yitai Ma, 2008].

Based on the thermodynamic calculation of R22 cycle COPs (coefficient of performance) of a system with various efficiency of expander are shown in Figure 1. The calculation condition is that the evaporation temperature and condensation temperature are 7.2°C/54.4°C, with a superheat and supercooling of 11.1°C/8.3°C. When the expander efficiency reaches 50%, the COP of a system with an expander can get an amplification of 6.84% compared with the system with a throttling valve, and the recovering work of expander is equal to 5.03% of input work of a compressor. When the expander efficiency reaches 70%, the system COP increases by 9.79% and the expander recovering work is equivalent to 7.05%, of input work of a compressor. Therefore, using an expander to replace a throttling valve can improve COP of a refrigeration system. The key technology is to develop a high-efficient expander.



**Figure 1** variations of system COP and expansion work ratio to expander efficiency

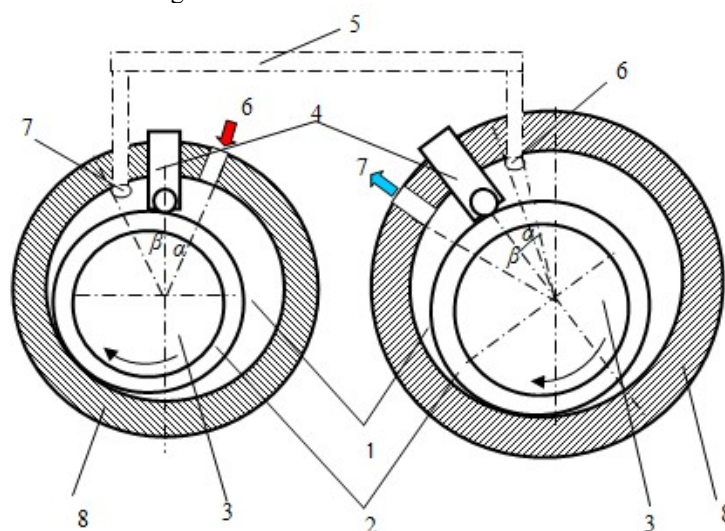
The expansion ratios of traditional refrigerants are 20-30 in common and are much larger than compression ratio. The structure of an expander with such large expansion ratio is difficult to design. Considering the large expansion ratio of conventional refrigerants, authors considered that a two-rolling piston expander is proper type

expander to develop, which is proposed by some researchers (Katsumi et al. 2006) for a CO<sub>2</sub> expander. In this paper, theoretical analysis and simulation calculation was carried out to investigate internal losses of conventional refrigerant expander in order to find the main factors which affect the efficiency of expander.

## 2. BRIEF INTRODUCTION OF TWO-ROLLING PISTON EXPANDER

### 2.1 Structure of Expander

A two-rolling piston expander of conventional refrigerant is equivalent to two sets of single rolling piston expander system, each of which includes a slide, a rolling piston, a cylinder, a eccentric wheel etc[Ma Guoyuan, 2003]. Its basic structure is shown in Figure 2. The left side is the first stage cylinder system, and the right is the second stage. There is an angle between two slides. A connecting pipe connects the outlet of first stage cylinder and the inlet of second stage cylinder. The first stage cylinder acts as an inlet control device. As the first stage cylinder is always connected with inlet pipe, the inside pressure fluctuation can be reduced, and the pressure during inlet process is stable. When the first stage piston rotates one circle, a suction process ends. Meanwhile, the fluid begins to expand. Expansion process completes when the second stage piston rotates to the outlet. The second rolling piston continues to run one circle to accomplish the exhaust process. Therefore, in order to complete one work cycle, the first rolling piston whirls one circle and the second rolling piston whirls two circles. The two-rolling piston one-stage expansion structure is one of effective methods to solve large expansion ratio of conventional refrigerant.



**Figure 2** Conventional refrigerant two-rolling piston expander  
 1-cylinder 2-rolling piston 3-eccentric wheel 4-slide  
 5-connecting pipe 6-inlet port 7-outlet port 8-cylinder wall  
 $\alpha$ -end of inlet angle  $\beta$ -start of outlet angle

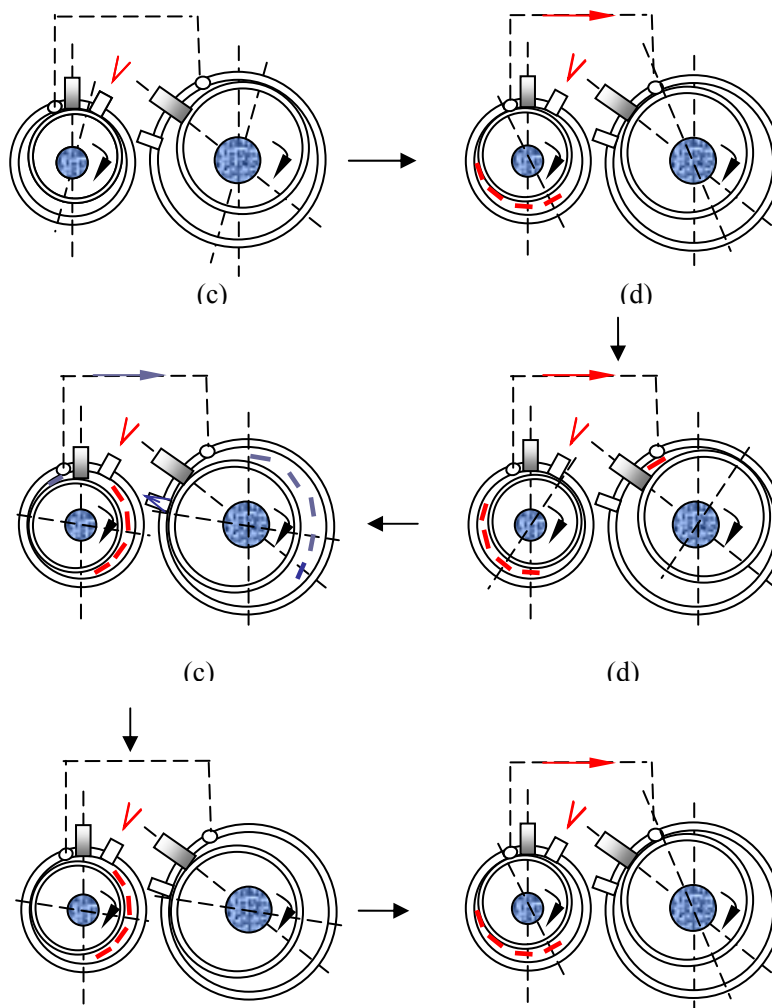
### 2.2 Movement Characteristics of Expander

There are some characteristic angles of expander, including: 1) $\alpha_1$ : the inlet rear edge angle of the first cylinder, 2) $\beta_1$ : the outlet front edge angle of the first cylinder, 3) $\alpha_2$ : the inlet rear edge angle of the second cylinder, 4) $\beta_2$ : the outlet front edge angle of the second cylinder, 5) $\alpha$ : the angle between two slides, 6) $\delta$ : the sharing angle of inlet channel. The inlet and outlet angles are shown in Figure 2.

This expander is equivalent to two sets of expander system, and the movements are more complicated. A complete working cycle includes suction process, expansion process and exhaust process. Base on that, the movements of expander have been analyzed and diagramed as shown in Figure 3.

In Figure 3(a), the rotation angle is  $(\alpha_1 - \delta)$ , which represents the begin of suction process of the 1<sup>st</sup> cylinder of this cycle.

In Figure 3(b), the rotation angle is  $(360 - \beta_1)$ . At this moment, the connecting pipe begins to connect the working chambers of two cylinders and the suction process of 2<sup>nd</sup> cylinder of this cycle begins.



**Figure 3** movements of the conventional refrigerant expander

In Figure 3(c), the rotation angle is  $(360+\alpha_1)$ . Right now, the whole suction process finishes, and the expansion process begins.

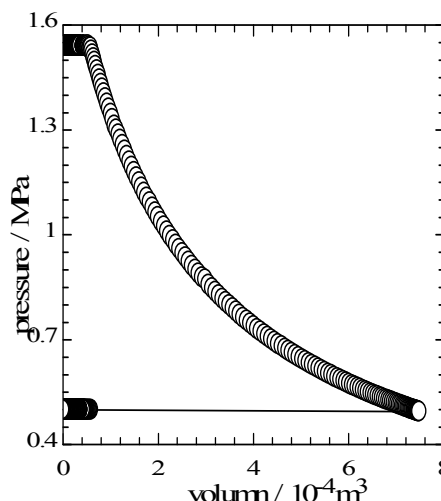
In Figure 3(d), the rotation angle is  $(720-\beta_2-\alpha)$ , which represents the finish of expansion process and the start of exhaust process of this cycle.

In Figure 3(e), the rotation angle is  $(1080-\beta_2-\alpha)$ , which represents the finish of whole exhaust process of this cycle. At this moment, a whole working process has been completed.

In Figure 3(f), the rotation angle is  $(1080-\beta_1)$ , which represents the start of suction process of the 2<sup>nd</sup> cylinder of the next cycle. The next working process is going on.

**2.3 P-V Diagram of the Ideal Expansion Process**

Considering the ideal expansion process, the variation of pressure and volume in working chamber is shown in Figure 4. Figure 4 shows the whole working process of expander. During the suction process, pressure keeps constant, when the piston run around one circle, the expansion process starts, the pressure decreases quickly with the increase of volume. The area enclosed by the curve is the ideal expansion work of the expander.



**Figure 4** the variation of pressure and volume in working chamber of ideal expansion process

### 3. FORCE ANALYSIS OF EXPANDER MAIN COMPONENTS

The main components of expander include a slide (sliding valve), a rolling piston, and an eccentric wheel. In the calculation of the friction losses of expander, frictions between main components are considered through the force analysis.

Basing on the reference of Rotary compressor[Ma Guoyuan, 2003], forces and pressure distributions on a slide is shown in Figure 5. Forces on the slide include: 1)the elastic force  $F_k$  of spring at the end of slide, which's direction is along the slide centerline to cylinder center; 2)the inertial force  $F_{iv}$ , representing the resultant force of slide;3)positive pressure  $F_{r1}$ ,  $F_{r2}$  on both sides of slide and the friction  $F_{rt1}$ ,  $F_{rt2}$  between the slide and slide grooves, in which friction is opposite to the moving direction of slide, and change to the reverse every 180 degrees of rolling piston; 4)positive pressure and friction force between slide and rolling piston, in which positive pressure  $F_s$  is along the connect line of connect point(slide end and rolling piston) and rolling piston center pointing to slide, while friction force  $F_t$  is opposite to the speed direction of connect point; 5)differential pressure force  $F_c$  at both ends of slide and 6)force  $F_h$  on the part of slide which inserted into cylinder caused by pressure difference between the suction chamber and expansion chamber. The direction of this force points to low pressure side from high pressure side.

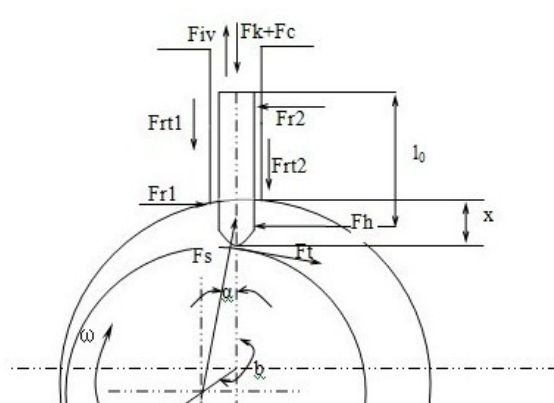


Figure 5 force analysis diagram of slide

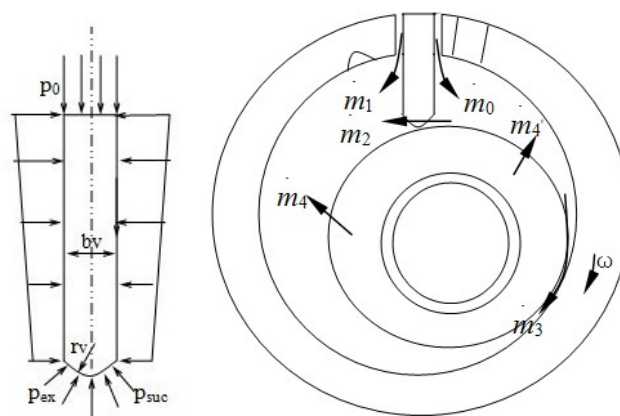


Figure 6 leakage analysis within expander

### 4. LOSSES ANALYSIS IN EXPANSION PROCESS

#### 4.1 Friction Losses of Expander

On base of the relative moving components of expander, the friction losses of conventional refrigerant expander include: 1) $L_1$ , the loss between main shaft and bearing, 2) $L_2$ , the loss between eccentric wheel and cylinder end covers, 3) $L_3$ , the loss between eccentric wheel and rolling piston, 4)  $L_4$ , the loss between rolling piston and cylinder wall, 5)  $L_5$ ,  $L_6$ , the loss between rolling piston and the upper and lower end caps of cylinder, 6)  $L_7$ , the loss between rolling piston and slide and 7)  $L_8$ , the loss between slide and slide grooves.

So, all friction losses of cylinder are:

$$W_{fric} = L_1 + L_2 + L_3 + L_4 + L_5 + L_6 + L_7 + L_8 \quad (1)$$

#### 4.2 Leakage Losses of Expander

According to the relative motion between components during expansion process, leakage positions can be obtained, which mainly include 1)gap between slide and slide groove  $\delta_1$ , 2)gap between the two sides of slide and cylinder end covers (referred as slide side gap)  $\delta_2$ , 3)gap between rolling piston outer surface and cylinder inner wall of the cut point (referred as radial gap)  $\delta_3$ , 4)gap between rolling piston and cylinder end covers (referred as end face gap)  $\delta_4$ , and 5)gap between slide end and rolling piston  $\delta_5$ , as shown in Figure 6.

In general, as the position of slide and slide grooves are relatively fixed, the gaps  $\delta_1$  between them do not vary with external environment, and can be treated as fixed clearances. Slide side gap,  $\delta_2$ , results in leakage between two working chambers, and gas refrigerant leaks from high pressure chamber to low pressure chamber through this gap. In addition, radial gap,  $\delta_3$ , changes slightly with the rotation of rolling piston. The existence of this gap causes the gas refrigerant leaking from high pressure working chamber to low pressure chamber. Rolling piston

end gap,  $\delta_4$ , may cause lubricant leakage within the bearing leaking to the working chamber. This gap changes little and is considered as a fixed value in the calculation. At the contact of slide and rolling piston,  $\delta_5$ , existing lubricant film can decrease the leakage greatly, which means the leakage there is very small and can be ignored. The expander leakage characteristics are shown in Table 1.

**Table 1** leakage characteristics of different gaps

Leakage rate	Leakage gap	Gap characteristics	Leakage characteristics	leakage model
$m_1$	Gap between slide and slide grooves $\delta_1$	Fixed gap	Mainly lubrication oil	Liquid leakage model
$m_2$	Slide side gap $\delta_2$	Fixed gap	Mainly gas refrigerant	Gas leakage model
$m_3$	Racial gap $\delta_3$	Change with the moving of rolling piston	Mainly gas refrigerant	Gas leakage model
$m_4$	End face gap $\delta_4$	Fixed gap	Mainly lubrication oil	Gas leakage model
$m_5$	Gap between slide and rolling piston $\delta_5$	Small, can be ignored	Mainly gas refrigerant	Liquid leakage model

The leakage rate of liquid leakage model is:

$$\dot{m} = \left( \frac{\Delta p \delta^3}{12 \mu l} + \frac{1}{2} v_x \delta \right) H \rho_0 \quad (2)$$

The leakage rate of gas leakage model is:

$$\dot{m} = \varphi_0 A \dot{g} \quad (3)$$

$$\dot{g}(p_{in}, \rho_{in}, \rho_{out}) = \begin{cases} \sqrt{p_{in} \rho_{in} \left( \frac{2}{k+1} \right)^{\frac{k+1}{k-1}}} & \left( \frac{p_{out}}{p_{in}} \right) < \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} \\ \sqrt{\frac{2k}{k-1} p_{in} \rho_{in} \left( \frac{p_{out}}{p_{in}} \right)^{\frac{2}{k}} \left[ 1 - \left( \frac{p_{out}}{p_{in}} \right)^{\frac{k-1}{k}} \right]} & \left( \frac{p_{out}}{p_{in}} \right) \geq \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} \end{cases} \quad (4)$$

The leakage mass flow rates can be calculated through corresponding model. The corresponding leakage losses are generally caused by enthalpy changes. So, the leakage loss is the product of leakage mass flow rate and enthalpy difference. And the total leakage loss is:

$$W_{leak} = LL_1 + LL_2 + LL_3 + LL_4 + LL_5 \quad (5)$$

### 4.3 Other Losses of Expander

For a rolling piston expander, when a exhaust process is completed and rolling piston turns over the outlet port, a small amount of refrigerant between rolling piston and slide cannot be discharged completely, which would do negative work to expander. In the design of an expander, it is necessary to adjust the angle between discharge point and the slide to make it as small as possible to get smaller clearance volume loss.

In addition, the connection pipe would be filled with working fluid all the time, while a part of refrigerant in this pipe cannot be discharged. The size and length of connection pipe is designed based on the location of two cylinders. And it cannot be too short or small for the probably throttling loss.

### 4.4 The Total Losses and Efficiency Calculations of Expander

Above all, the expander losses  $W_t$  is:

$$W_t = W_{fric} + W_{leak} + W_{else} \quad (6)$$

Expander adiabatic efficiency  $\eta$  can be calculated as follows:

$$\eta = \frac{W_e - W_t}{W_e} \quad (7)$$

Where,  $W_e$  represents the ideal recovering power (W),  $W_{else}$  is other losses (W).

### 5. SIMULATION RESULTS AND ANALYSIS

The simulation involves the suction and expansion process of a R22 expander. The inlet and outlet condition of expander is 40°C/1.53MPa and 0°C/0.498MPa. The radius and height of two cylinders are 55.5/28mm and 82.5/107mm. And the eccentricity of two cylinders is 4.4/11mm.

#### 5.1 Friction Losses of Expander

Figure 7 shows the variations of friction forces against rotation angle of two slides. The first slide suffers less friction force, while the second slide suffers more. To the first slide, friction forces between slide and slide grooves  $F_{rt1}$ ,  $F_{rt2}$  change in the range of 2N, and that between slide and rolling piston  $F_t$  changes within 5N. To the second slide, friction force between slide and rolling piston  $F_t$  has a maximum of 800N. Friction force between slide and the right slide groove  $F_{rt2}$  is within the range of 200~400N, and differs large to the force between slide and the left groove  $F_{rt1}$ , which mainly results from the rotation direction of rolling piston(clockwise). In addition, the breakpoints at 180° in Figure 6 are caused by the moving direction change of slide.

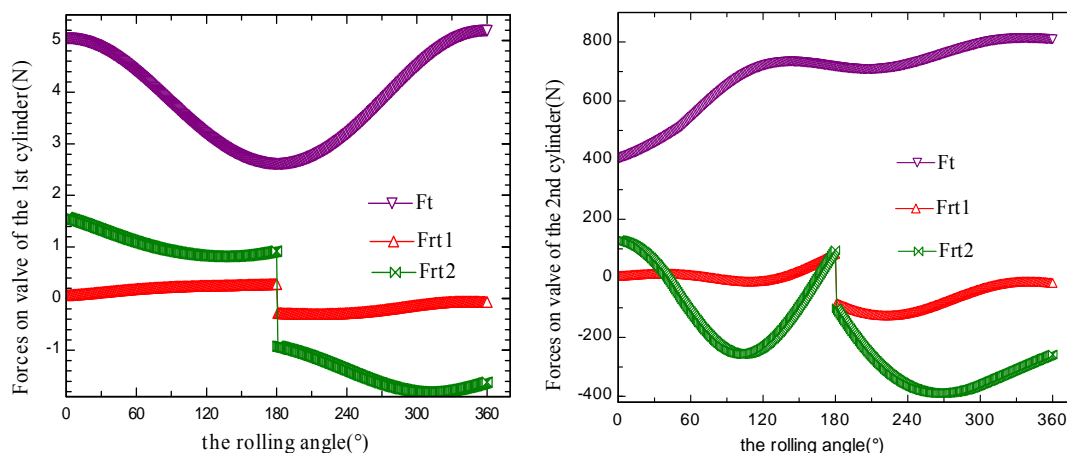


Figure 7 variations of friction forces with rotation angle of two slides

Figure 8 and Figure 9 are the changes of friction losses with rotation angle of two cylinders respectively. Besides the fixed losses of bearings, the larger share of friction losses distributes between slide and rolling piston  $L_7$  and between slide and slide grooves  $L_8$ . As the moving direction change of slide at 180° and the beginning of expansion process at 52°, relative turning points appeared in the figures.

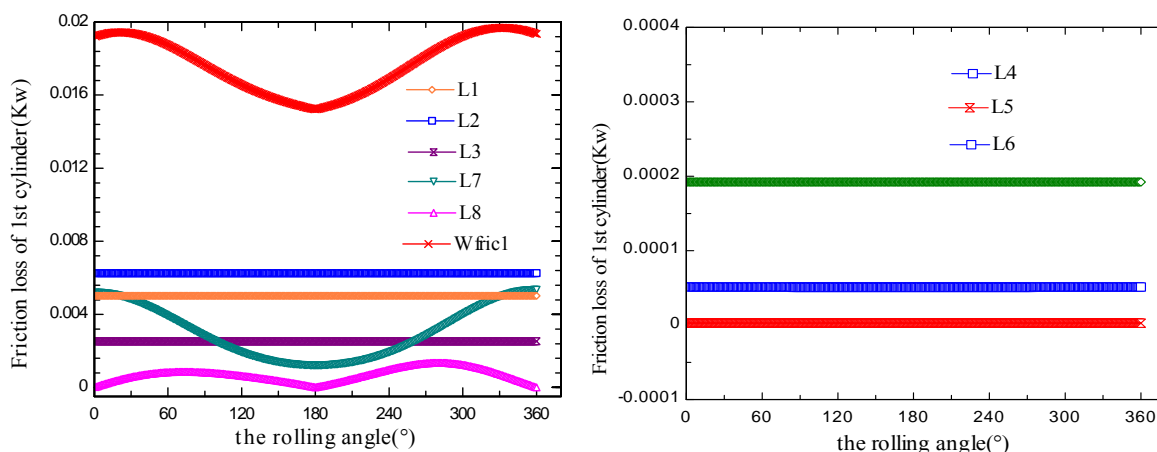


Figure 8 changes of friction losses with rotation angle of the first cylinder

In the first cylinder, maximum friction loss between a slide and slide grooves reaches 1.5W, and that between the slide and the rolling piston gets 5.2W. Besides, friction loss caused by the movement between main shaft and bearings, that between eccentric wheel and cylinder end covers, and that between eccentric wheel and rolling

piston are fixed values of 5.0W, 6.4W, 2.4W respectively.. The total friction loss of first cylinder is less than 20W.

In the second cylinder, maximum friction loss between slide and slide grooves is of 1.01KW, and that between slide and rolling piston gets 1.8KW, which are caused by the large friction forces on them. In addition, friction losses between main shaft and bearings, that between eccentric wheel and cylinder end covers, and that between eccentric wheel and rolling piston are 0.1KW, 54W, 11W respectively. While losses in other parts are close to zero.

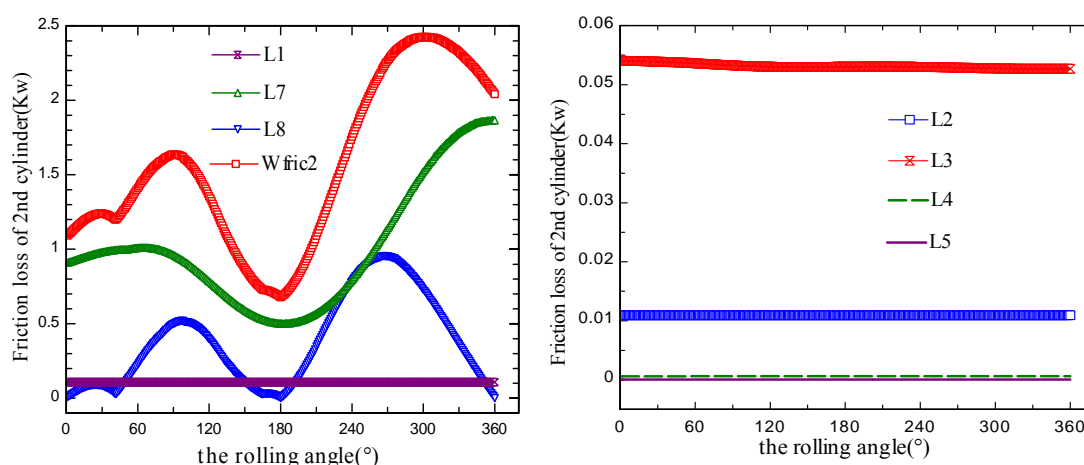


Figure 9 changes of friction losses with rotation angle of the second cylinder

Based on the above, proportions of different friction losses to the total friction losses can be obtained, as shown in Table 2. It is obvious that the friction losses between slide and slide grooves and that between slide and rolling piston are large. Among them, the friction loss between slide and rolling piston of second cylinder is about 65.9% of the total friction losses, which means that reducing this part of loss would significantly improve expander efficiency.

Table 2 proportions of different friction losses to the total friction losses

Position of friction loss	First cylinder		Second cylinder	
	Loss (W)	Proportion (%)	Loss (W)	Proportion (%)
Between main shaft and bearings	5.02	19.3	107.4	6.02
Between eccentric shaft and cylinder end covers	6.27	24.1	29.59	1.66
Between eccentric shaft and rolling piston	2.53	9.74	53.28	2.99
Between rolling piston and inner cylinder	0.512	1.97	13.64	0.764
Between rolling piston and lower end cover	0.120	0.462	8.345	0.468
Between rolling piston and upper end cover	0.193	0.743	14.63	0.820
Between rolling piston and slide	7.62	29.3	1175.9	65.9
between slide and slide grooves	3.71	14.3	382.1	21.4
The total friction losses	25.98	100	1784.9	100

## 5.2 Leakage Losses of Expander

Figure 10 is the changes of leakage rates with rotation angle of first cylinder. The leakage rates of first cylinder are small. Among them, leakage rate between rolling piston and inner cylinder  $m_3$  is larger, which has a maximum of 0.0072kg/s and almost linearly decreases with the increase of rotation angle.

Figure 11 is the changes of leakage rates with rotation angle of second cylinder. The leakage rates between



rolling piston and inner cylinder is larger. That leakage rate mainly results from the changes of pressure in expansion chamber and gets a maximum at the beginning of expansion process(52°) as the maximum pressure difference of two working chambers. And rapidly decrease with the increase of rotation angle occurs till about 150°, and then its decline begins to be smooth, and gradually close to zero, which is mainly caused by the changes of pressure in expansion chamber. The leakage rate of end face gap,  $m_4$ , increases greatly from the beginning of expansion process to 120° and then change gently to 280° where get a slight increase. The maximum of that leakage rate is 0.018kg/s. The leakage through the gap between slide and slide grooves from back pressure chamber,  $m_1$ , also increases with the increase of rotation angle, and has a maximum leakage rate of 0.015kg/s. In addition, the leakage rate through slide side gap decreases with the pressure difference of two chambers, and gets a maximum rate of 0.0023kg/s at the beginning of expansion process.

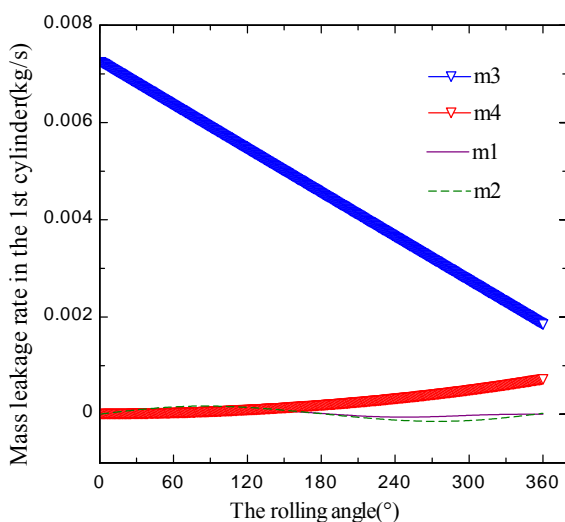


Figure 10 changes of leakage rates with rotation angle of first cylinder

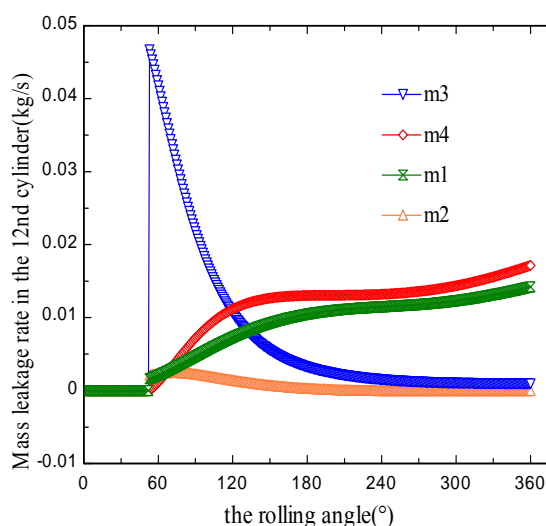


Figure 11 changes of leakage rates with rotation angle of second cylinder

Figure 12 is the changes of leakage losses with rotation angle of two cylinders. For the first cylinder, the mainly loss is caused by racial gap, and is within 1W. This is also the reason why the pressure changes little in the working chamber of first cylinder. For the second cylinder, the site of main loss is also through racial gap, and its biggest loss is close to 80W, which reduces rapidly with the increase of rotation angle. This part of leakage is caused by the pressure difference between expansion chamber and discharge chamber, which gradually reduces during the expansion process. The leakage loss through gap between slide and slide grooves and end face gap change similarly. And that of slide side gap keeps constant and its value is also small.

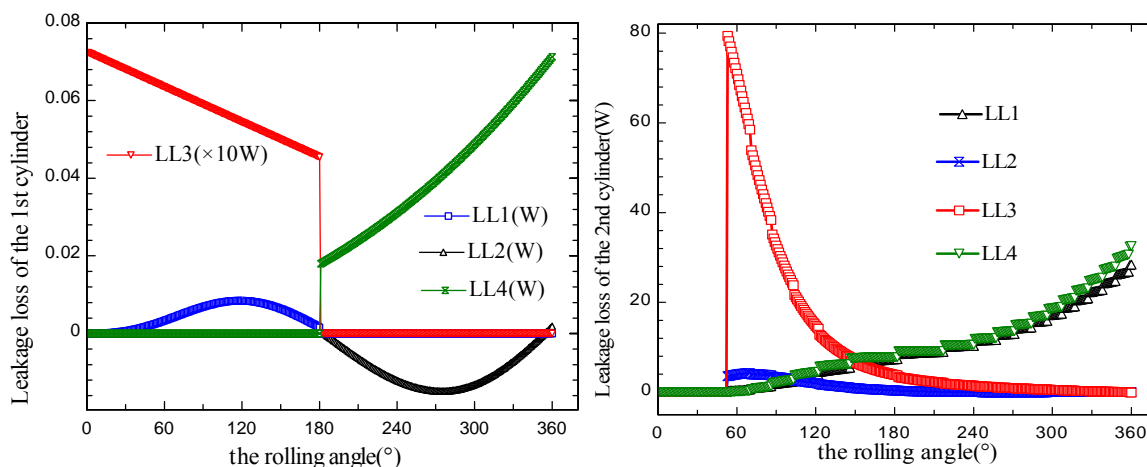


Figure 12 changes of leakage loss with rotation angle of two cylinders

As shown in Table 3, the proportion of leakage losses to the total leakage losses is cleared up. Leakage loss caused by racial gap is serious both in two cylinders.

**Table 3** proportion of leakage losses to the total leakage losses

Position of leakage loss	First cylinder		Second cylinder	
	Loss (W)	Proportion (%)	Loss (W)	Proportion (%)
Gap between slide and slide grooves	0.00746	1.12	72.05	21.0
Slide side gap	0.00562	0.845	28.85	8.39
Racial gap	0.481	72.3	142.5	41.4
End face gap	0.171	25.7	100.4	29.2
The total leakage losses	0.665	100	343.8	100

### 5.3 Discussions on Expander Efficiency

The total losses and expander efficiency are shown in Table 4. The simulation calculated expander efficiency is 48.5%. From Table 4 it can be seen obviously, that the friction loss is a very important factor affecting expander efficiency. The friction loss takes 41.3% of total ideal expansion work and about 80% of total losses.

**Table 4** expander losses and efficiency

item	value (W)	proportion (%)
Ideal expansion work	4188.0	100
Total loss	2156.0	51.5
Friction loss	1728.4	41.3
Leakage loss	344.47	8.23
Other irreversible losses	83.762	2.00
Real recover work	2032.0	48.5

Through the above analysis and simulation calculation, we find that the friction loss is about 41.3% of ideal expansion recovery work, while leakage loss and other irreversible loss are relatively small. To improve expander efficiency, reducing the friction losses between slide and piston and between slide and slide grooves are the key. The followings are some practical methods applied in the expander to reduce friction losses.

In order to decrease the friction loss caused by the movement between slide and slide grooves, rolling-needles are considered to add to join the slide and slide grooves which would change sliding friction to rolling friction and reduce their friction and friction loss. As rolling friction coefficient is far less than sliding friction coefficient and rolling-needles are lubricated well, the friction loss between slide and slide grooves could be decreased by 90.4%. For the decreasing of the friction loss between slide and rolling piston, the mainly measure is to improve the end part of slide, and make it to have well lubrication and sealing. Then, the friction coefficient between slide and piston could be declined to one third of the original value. The friction loss of this part is decreased by 67.8%. Lubrication between these components has a good work to reduce this loss, It is considered that a designed independent lubrication system of expander can improve the lubrication of main shaft, piston, eccentric wheel and other parts and the friction losses between these parts have a certain extent diminution separately. In final, the total friction losses now can be decreased by 63.7% compared with the previous value in Table 4.

In addition, in order to improve the sealing of expander, O-ring seal can be used to reduce expander external leakage and a higher machining accuracy can help to control the internal leakage. And these measures could decrease the leakage losses by 20% at least.

So, after these efficient measures, the expander efficiency could be increased greatly and has a theoretical value of 76.0%.

## 6. CONCLUSIONS

1. A two-rolling piston one-stage expander has been designed to fit for the large expansion ratio of conventional refrigerants. Besides, this structure can eliminate an additional suction control system and is able to ensure the inlet pressure stability.

2. Irreversible losses analysis and simulation calculates have been carried out by analyzing the moving parts of expander,. The results show that friction losses are accounting for about 80.2% of the total expander losses, and leakage losses are about 16.0%.

3. Improvement measures were discussed which would be efficient to improve expander efficiency. The friction losses decrease by 63.7%, and the simulation calculated expander efficiency increase greatly to 76.0%.

## NOMENCLATURE

The nomenclature referred in this paper is shown in the following:

$\alpha$	end of inlet angle	( $^{\circ}$ )	<b>Subscripts</b>	
$\beta$	start of outlet angle	( $^{\circ}$ )	1	the first cylinder
$\delta$	sharing angle of inlet channel/the leakage gap	( $^{\circ}/m$ )	2	the second cylinder
$\rho$	density	( $kg \cdot m^{-3}$ )	in	the inlet status
m	leakage rate	( $kg \cdot s^{-1}$ )	out	the outlet status
W	power	( $W/J \cdot s^{-1}$ )		
F	force	(N)		
m	leakage rate	( $kg \cdot s^{-1}$ )		
W	power	( $W/J \cdot s^{-1}$ )		
k	adiabatic coefficient	(-)		

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