Research on Suction Performance of Two-Cylinder Rolling Piston Type Rotary Compressors Based on CFD Simulation

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ABSTRACT

In this paper, the transient refrigerant gas flow in a two-cylinder rolling piston type rotary compressor is simulated using STAR-CD, a general computational fluid dynamics (CFD) software. Three models with different suction piping systems are calculated respectively, one is two suction pipes were connected from accumulator to each cylinder separately (separate suction piping system), the other is suction pipes were connected commonly from accumulator to the cylinders (common suction piping system). Based on the results, the factors influencing compressor performance are studied, such as suction mass flow, suction throttle, suction pressure pulsation in the cylinder suction chamber, etc. Also the simulation results are proved by the bench test.

1. INTRODUCTION

The crankshaft of two-cylinder rotary compressors has two eccentric part with phase difference of π rad in each cylinder, so the compressors have the advantage of better dynamic balance, low vibration and relative even compression torque. Since the cooling capacity can be doubled only with a little increase of the outer size, the compressors meet the need of large capacity compressors and develop rapidly.

Between the upper and lower cylinders in two-cylinder rotary compressors for air conditioning, the suction, compression and discharge process has the phase difference of π rad. The separate suction piping system is two suction pipes connected from accumulator to each cylinder separately, while the common suction piping system is a suction pipe connected from accumulator to the pump, such as the main bearing, moderate plate or one cylinder, and then the cylinder suction hole connected to this suction pipe by a joint pipe. YANAGISAWA (YANAGISAWA, 1993) has studied the pressure pulsation in the suction pipes. The results show that the pressure pulsation of the separate piping system is larger than that of the common piping system. However, the pressure pulsation in the cylinder suction chamber and how it influence the actual gas enclosed into the cylinder haven't been investigated.

In this paper, the transient refrigerant gas flow in a two-cylinder rolling piston type rotary compressor is simulated using STAR-CD, a general computational fluid dynamics (CFD) software. Three models with different suction piping systems are calculated respectively, one is two suction pipes were connected from accumulator to each cylinder separately (separate suction piping system), the other is suction pipes were connected commonly from accumulator to the cylinders (common suction piping system). Based on the results, the factors influencing compressor performance are studied, such as suction mass flow, suction throttle, suction pressure pulsation in the cylinder suction chamber, etc. Also the simulation results are proved by the bench test.

2.MODEL AND BOUNDARY

The refrigerant in the rolling piston type compressor is three-dimensional compressible fluid and turbulent flow. In order to show the real fluid flow in the inner compressor and comprehensively analyze the compressor efficiency, this paper use the transient moving-mesh model to simulate the refrigerant working process (Geng, 2004 and Liu, 2003).

Fig 1 shows the model of compressors with separate piping system, the computational domain begins from the intake of cylinder suction pipe and ends of the discharge port in the upper muffler. Fig 2 is the draft of compressors with common piping system whose suction pipe is connected from accumulator to the middle of upper cylinder.

Model is assembled by diversified type mesh. The immovable mesh of muffler, cylinder suction hole and discharge line is accomplished using Proam package. The volume of suction chamber and compression chamber varies with the shaft rotates, which are moving mesh accomplished using Prostar package deforming mesh. The movement of rolling piston is controlled by .cgrd file. The motion of leaf valve is controlled by the pressure difference between compression chamber and muffler, which is also deforming mesh accomplished using Prostar package. This paper simplified the valve motion to pressure-activated motion. The valve opens when the pressure in the cylinder compression chamber is larger than that in the muffler. The higher the pressure difference, the larger the valve lifts. The valve begins to regress until the pressure in the cylinder compression chamber is less than that in the muffler. Since there are two leaf valves, the valve motion is controlled by multi condition event in STAR-CD.

The initial inlet and outlet boundaries of this model are all set to pressure boundary. The attachment between discharge port and compression chamber is arbitrary sliding mesh. The refrigerant's thermophysical property is defined based on simple PR equation, whose density is the function of both temperature and pressure as listed in equation (1). Turbulence in the flow is modeled using the standard k-ε high Reynolds model. Heat exchange is not considered in this analysis.

$$\rho = f(T, P) \tag{1}$$

3. SIMULATION RESULTS

The model shown in Fig 1, Fig2(a) and Fig2(b) has been analyzed respectively. The compressor's suction pressure is 0.625MPa. The suction temperature is 35°C. The discharge pressure is 2.146MPa. The rotating speed of compressor is 2840rpm. The refrigerant gas is R22. Since there is phase difference of π rad between upper and lower cylinder, the upper cylinder begins suction at 0 degree and the lower cylinder begins suction at 180 degree per shaft rotating cycle.

3.1 Pressure Pulsation In Cylinder Suction Chamber

The average pressure in cylinder suction chamber with separate piping system varies with the shaft rotation is shown in Fig 3. Because of the independent suction of upper and lower cylinder, the pressure pulsation curve is similar to the one cylinder rotary compressors. The pressure curve of each cylinder shows the same tendency and equal pulsation amplitude. At the beginning of suction period, the pressure pulsation in suction chamber is larger because of gas back flow to the suction pipe and throttling. With the shaft rotation, the fluctuation is attenuated. While to the end of suction period, the gas pressure rises again since the supercharge effect caused by the gas inertial motion.

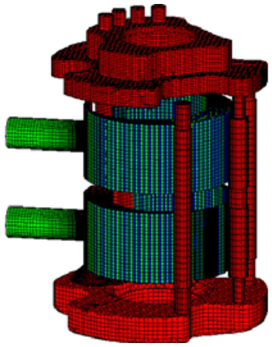
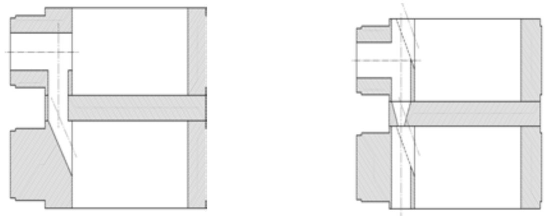


Figure 1 model of separate piping system



(a) (b)

Figure 2 draft and CAD model of common piping system

Fig 4 shows the average pressure in cylinder suction chamber with separate piping system shown in Fig 2(a) varies with the shaft rotation. The suction process is interactional between upper and lower cylinder with a suction joint pipe connected to each other, so the pressure curve is different. The pressure in upper cylinder shows the same tendency with separate piping compressors, such as fluctuating more at the beginning and end of the suction period. While the pressure fluctuates less at the beginning and the fluctuation is not attenuated during the suction middle period as that of the separate piping system. Since the common suction pipe of Fig 2(a) model is connected from accumulator directly to the upper cylinder, and a suction joint pipe is connected from the upper cylinder to the lower cylinder, the suction process of upper cylinder is influenced by the lower cylinder little. At the beginning and end of the suction period, the pressure pulsation is inspirited by the gas back flow of lower cylinder whose volume variational rate is maximal then. While during the middle suction period, the lower cylinder just begins suction at 180 degree and the pressure fluctuates much, which weaken the pressure pulsation attenuation in the upper cylinder.

Though the pressure in lower cylinder fluctuates large at the beginning and end of the suction period, the suction pressure shows a sunken tendency during the whole suction period. The suction line of lower cylinder is longer and pipe section area is the same as separating piping system. When the volume of suction chamber changes great, more suction throttling occurs and pressure descends much.

Fig 5 shows the average pressure in cylinder suction chamber with separate piping system shown in Fig 2(b) varies with the shaft rotation. This model uses a gradually reduced joint pipe connected from the upper cylinder to the lower cylinder. The pressure curve of the upper cylinder shows the same tendency as that of the separate piping system. The pressure pulsation decreases great and little fluctuation exits expect the beginning and end of suction period. The frequency of pressure pulsation in the lower cylinder is reduced much and the pressure curve is changed to a more smooth curve with sunken tendency. This model keeps the advantage of model Fig 2 to effectively attenuate the pressure pulsation of upper cylinder at the beginning of suction period. Simultaneously the gradually reduced joint pipe effectively restrains the interaction between two cylinders, which have the same advantage as that of the separate piping system. However, since the minimum diameter of the reduced joint pipe is designed too small, severe throttling occurs and much suction pressure loss produces, especially the bottom suction pressure of lower cylinder reduces about 3000Pa.

3.2 Results Of The Beginning And End Per Suction Cycle

The gas enclosed into the cylinder at the end of suction cycle means the maximum suction gas per cycle, while the gas enclosed into the cylinder at the beginning of compression cycle means the actual suction gas per cycle. Table 1 lists the maximum suction gas and actual suction gas respectively. As to the upper and lower cylinder, the maximum suction gas has little difference, which is influenced little by the suction piping system. Compared to the maximum suction gas, the actual suction gas varies. The back flow ratio of each cylinder is different, which means the suction piping system will influence the actual suction gas. Table 2 lists the pressure in each cylinder at the beginning of suction cycle and the end of compression cycle respectively, which shows the same results as the suction gas mass.

Table 1 Gas enclosed into cylinder at the end of suction cycle and beginning of the compression cycle（%）

model	Upper cylinder			Lower cylinder		
	End of suction A1	Beginning of compression A2	A2/A1	End of suction A1	Beginning of compression A2	A2/A1
Fig 1	100%	98%	98%	100.4%	98.6%	98.2%
Fig 2(a)	99.6%	97.6%	98%	99.5%	97.1%	97.6%
Fig 2(b)	99.7%	97.3%	97.6%	100.3%	98.1%	97.8%

Table 2 Average pressure in suction chamber at suction end and compression beginning (MPa)

model	Upper cylinder		Lower cylinder	
	End of suction A1	Beginning of compression A2	End of suction A1	Beginning of compression A2
Fig 1	0.6466	0.6381	0.6496	0.6423
Fig 2(a)	0.6440	0.6356	0.6429	0.6264
Fig 2(b)	0.6443	0.6329	0.6486	0.6386

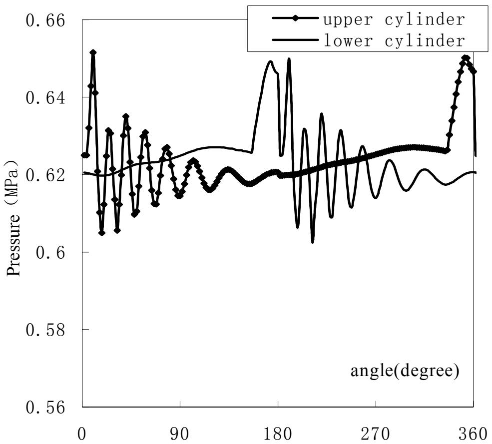


Figure 3 Pressure pulsation of separate pipe

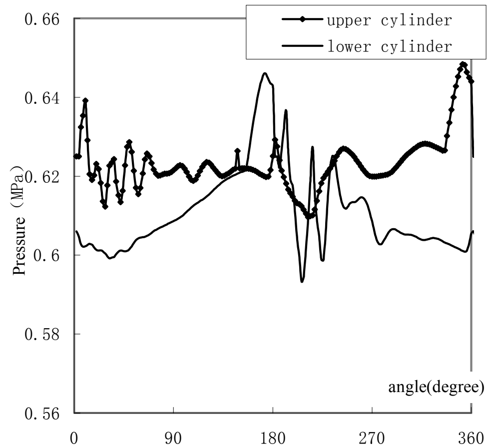


Figure 4 Pressure pulsation of common pipe (Fig 2a)

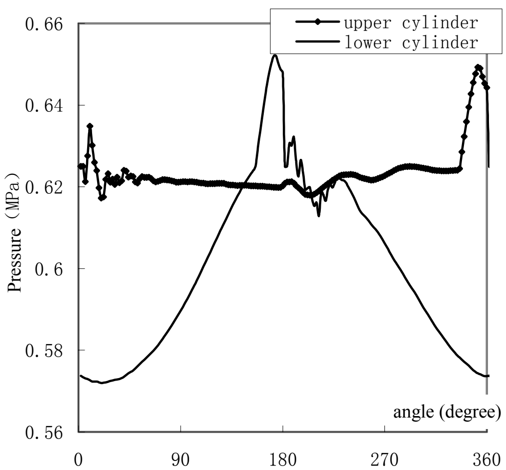


Figure 5 Pressure pulsation of common pipe (Fig 2b)

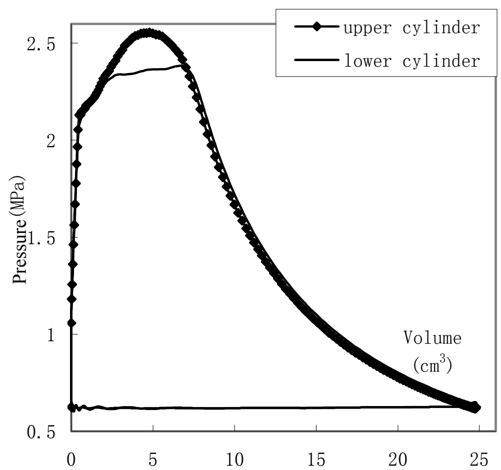


Figure 6(a) P-V diagram of separate pipe

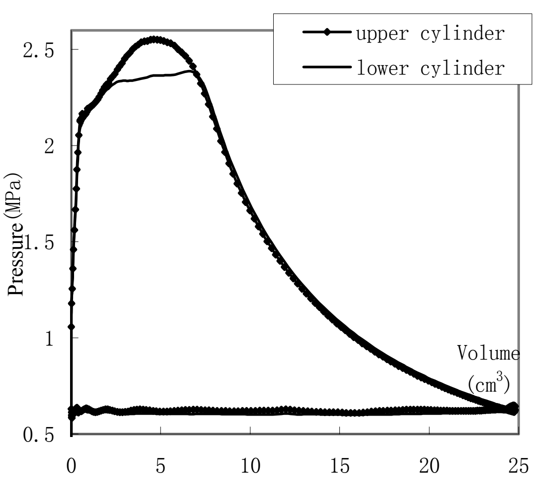


Fig6(b) P-V diagram of common pipe (Fig 2a)

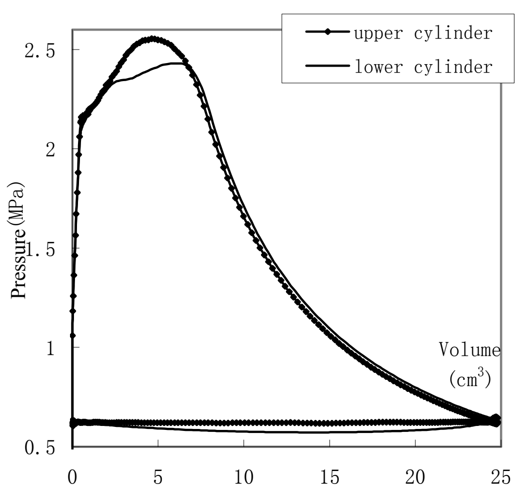


Fig6(c) P-V diagram of common pipe (Fig 2b)

3.3 Compressor Performance

Based on the results, the suction and compression P-V diagram of above models is shown Fig 6(a), (b) and (c) respectively. The thin line represents the P-V diagram of lower cylinder and the line with point represents the P-V diagram of upper cylinder. As to the compressors with separate piping system, the P-V diagram of upper and lower cylinder is same. While to the compressors with common piping system, the P-V diagram of lower cylinder shows the sunken tendency, especially to the Fig 2(b) model with larger suction throttling.

Since the upper muffler has smaller discharge line, higher flow resistance occurs. The over-compression in upper cylinder is severer than that in lower cylinder. In addition, the clearance volume of lower cylinder is a little larger than that of upper cylinder in the simulation model, which means the maximum suction volume of lower cylinder is a little larger. So from the P-V diagram, the compression curve of lower cylinder is higher than that of upper cylinder.

On the P-V indicator diagram, the indicator work of each compression cycle, ω, is identified as the enclosed area by the suction and compression chamber pressure curves. The actual mass flow rate of delivered refrigerant per cycle, m, is obtained from the simulation results. Then the cooling capacity of per cycle, q, can be calculated from equation (2). J represents the cooling capacity per weighing. We use the COP defined in equation (3) as the parameter to evaluate compressor performance.

$$q = m \times J \tag{2}$$

$$COP = q / \omega \tag{3}$$

Based on the compressor performance lists in table 3, the performance of upper cylinder with common piping system can be improved by changing the figure of suction joint pipe. While since the longer suction pipe and larger suction throttling of lower cylinder, the compression power of cylinder increases and cooling capacity decreases, especially the Fig 2(b) model. As to the calculated model, the overall efficiency with separate suction piping system has higher performance than that of the common suction piping system. If we want to improve the overall efficiency of common suction piping system equal to that of the separate suction piping system, the pipe connected to the accumulator should be optimized, such as enlarge the diameter.

Table 3 Compressor efficiency

model	Upper cylinder			Lower cylinder			Total efficiency
	q	ω	COP	q	ω	COP	
Fig 1	100%	100%	100%	100%	100%	100%	100%
Fig 2(a)	99.6%	99.7%	99.9%	98.3%	100.3%	98.0%	99.0%
Fig 2(b)	99.6%	99.6%	100%	99.5%	104.2%	95.6%	97.8%

Note: Assume the q, ω and COP of separate piping system is 100%.

4. TEST

The compressor performance test and noise test of two-cylinder compressors as Fig 1 and Fig 2(a) shown have been done respectively. The performance test is done according to the ASHARE test specification same as the calculating condition. Results show the efficiency with separate suction piping system is a little higher than that of the common suction piping system, same as the simulation result. The noise test results show the compressors with common suction piping system has better noise level, the total noise is reduced 2-3 dB(A). Mainly because the pressure pulsation attenuates and suction noise in accumulator is reduced.

5. CONCLUSIONS

In this paper, three models with different suction piping systems are calculated respectively using STAR-CD package. The calculated results and bench test results both show that the noise level with common suction piping system is improved by reducing 2-3dB(A) because of the pressure pulsation in suction chamber attenuating. Since the compressor with common suction piping system has not been optimized, the efficiency is a little lower than that with separate suction piping system.

Table 4 Cooling capacity test

Model	capacity	power	COP
Fig 1	100%	100%	100%
Fig 2(a)	99.86%	101.0%	99.4%

Note: Assume the capacity, power and COP of separate piping system is 100%.

Table 5 Noise test

Model	X direction（dB）	Y direction（dB）
Fig 1	69.0	68
Fig 2(a)	65	66

NOMENCLATURE

ρ	density	kg/mm^3
T	temperature	^oC
P	pressure	MPa
ω	indicator work of each compression cycle	J/rev
m	mass flow rate of delivered refrigerant per cycle	kg/rev
J	cooling capacity per weighing	J/kg

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