Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

2008

Experimental Investigation of the Dynamics of Self-Acting Valve in a Reciprocating Compressor for Transcritical CO2 Refrigeration

Yuan Ma
Xi'an Jiaotong University

Yanan Gan
Xi'an Jiaotong University

Xueyuan Peng
Xi'an Jiaotong University

Ziwen Xing
Xi'an Jiaotong University

Experimental Investigation of the Dynamics of Self-acting Valve in a Reciprocating Compressor for Transcritical CO_2 Refrigeration

Yuan Ma*, Yanan Gan, Xueyuan Peng, Ziwen Xing
Institute of Compressors, School of Energy and Power Engineering,
Xi'an Jiaotong University, Xi'an, 710049, P.R. China
E-mail:mayuan@mail.xjtu.edu.cn

ABSTRACT

The self-acting valve has a significant influence on the efficiency and reliability of the reciprocating compressor. The large density and high pressure difference through the valve cause serious bending and impact stresses in the valve and herby great challenges for the valve design. Experimental investigation of the valve dynamics are required to design a self-acting valve with high efficiency and long life span for the transcritical CO_2 compressor. A semi-hermetic reciprocating compressor was developed for CO_2 refrigeration application, and a test system was incorporated into the compressor performance test rig, focusing on investigating the dynamics of the discharge valves. With the experimental results, the movement of the valve is described in detail for transcritical CO_2 compressor, based on which the thermodynamic performance of compressor is studied.

Keywords: Reciprocating compressor, Transcritical CO_2 refrigeration, Self-acting valve,

1 INTRODUCTION

As an environmentally friendly refrigerant, CO_2 has gained more and more attention. Different institutions and companies have developed CO_2 compressors for different applications. Some small CO_2 compressors have been developed in Japan for the use in domestic heat pump water heater and automobile air-conditioner. These compressors are mostly scroll and rolling piston types, with power around 1-2kW and COP around 4. Large and medium CO_2 compressors have been produced by Dorin and Bock for the use in commercial applications.

Compared with traditional fluorocarbon-based refrigerants, CO_2 has lower critical temperature (31.1℃) but higher critical pressure (7.37MPa). Due to the transcritical operation of the cycle, CO_2 compressor is working under much higher pressure, 5-10 times higher than the standard compressor. This high pressure causes large forces, which greatly challenges the design of key components such as crank, connecting rod, bearings and suction/discharge valves.CO_2 has a high volumetric capacity (22.6MJ/m^3 at 0℃), which is 1.58, 5.12 and 8.25 times as much as NH3, R22 and R12 respectively. Thus, the swept volume of CO_2 compressor is smaller than the standard compressor. So, it could be possible to design the compressor compact and cost effective compressor. However, it becomes more difficult to arrange the valves with sufficient flow area in a relatively small space.

The high pressure difference, combined with the large density of CO_2, bring great bending and impact stresses to the valves. Also, the high speed (2900rpm) of CO_2 compressor causes high impact velocities. Researchers have reported that the discharge valve and the spring have relatively short life and are easy to break due to improper material, design and manufacture. Robust design of the valves is crucial to improve the reliability of the CO_2 compressor. Junghyoun Kim (2006) analyzed the valve dynamics of a hermetic reciprocating compressor using R134a as refrigerant. The dynamic behavior of the valves and the pressure-volume diagram were obtained. However, the discharge pressure is much lower than the critical pressure and the property of the two refrigerants was quite different. Jeffrey J.NIETER (2006) analyzed the discharge port and valve and the results revealed excessive over-pressure loss. Detailed experimental research about the valve dynamics for transcritical CO_2 cycles is unavailable now.

To investigate the valve dynamics in the CO_2 reciprocating compressor, a simulation program is established. Also the pressure variation inside the cylinder and the movement of the discharge valve are measured to validate the program. In order to identify the key factors that greatly influence the valve dynamics, tests are conducted under different conditions by changing the pressure ratio, valve lift, and compressor speed.

2 DEVELOPMENT OF CO_2 COMPRESSOR

2.1 Prototype of reciprocating CO_2 compressor

A single stage semi-hermetic reciprocating CO_2 compressor with two cylinders is developed for the use in commercial refrigeration. The cooling capacity is 29kW under the evaporation temperature of -10℃.

2.1.1 Geometrical parameters

Stroke-to-bore ratio has important influences on the performance of the compressor. Large stroke-to-bore ratio could reduce the sealing length and thus reduce the leakage. Additionally, a small bore helps to limit the piston force, which would reduce the loads on the moving parts. However, there will be smaller space left to arrange the valves with sufficient flow area. From the leakage and space considerations, a stroke-to-bore ratio of 1.03 is chosen.

Table 1 Compressor specifications

Model	Reciprocating	Motor power	18.5kW
Type	Semi-hermetic	Swept volume	11.8m^3/h
Stage	single	Suction pressure	25.00~40.00bar
Number of cylinder	2	Discharge pressure	75~120bar
Rotational speed	2900r/min		

2.1.2 Design of the structure

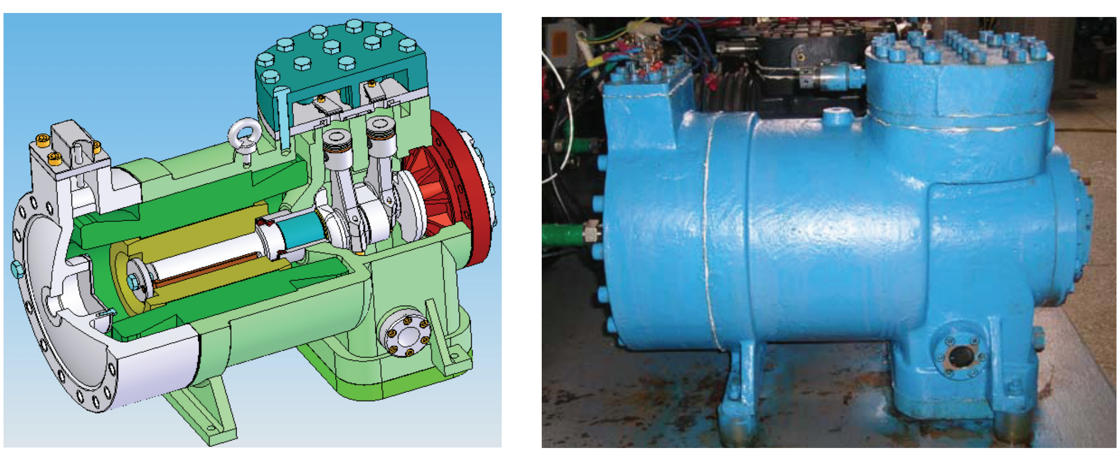


Figure 1: CO_2 compressor

Suction cooling is adopted to keep the motor running smoothly and efficiently. A combination valve is designed to acquire sufficient flow area in the limited space. Sandvik 7C27Mo2 is used as the valve material, which has good corrosion resistance, high toughness, and excellent fatigue strength properties. Hydrodynamic journal bearings are selected due to the large piston force.

2.1.3 Leakage and friction

The compressor is semi-hermetic and thus the shaft seal could be avoided. Two piston rings are arranged to limit the leakage between the cylinder and the piston. An oil pump driven by the crankshaft is designed to supply the lubricant to the bearings, interface between crankshaft and connecting rod, and interface between piston and connecting rod.

2.1.4 Lubricant

Polyolester type lubricant (POE) is completely miscible with CO_2, which helps the oil return to the compressor. Even through hydrolysis of POE remains a cause for concern in CO_2 systems, a desiccator in the system can effectively solve the problem.

2.2 Reliability issues for the discharge valve

The reliability of the compressor is checked at a standard test condition. During the test, the discharge valve and spring have ever broke several times. The structure of the discharge valve is shown in Figure2. Distance ring is used to control the valve lift.

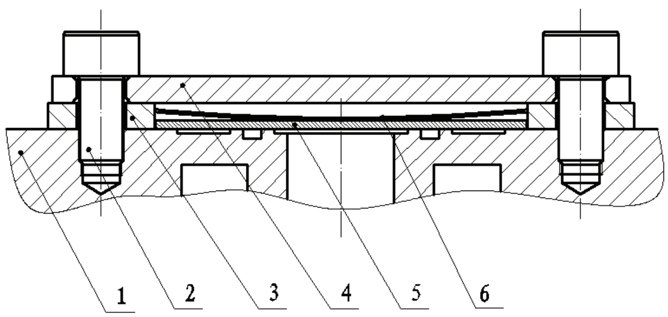


Figure2: The discharge valve

（1-Valve Plate，2-Bolt，3-Distance ring，4-Vlave stop，5-Discharge valve，6-Spring）

During the reliability test, the discharge valves broke firstly after about 100 hours, as shown in Figure3. After replacing the initial valve material with Sandvik 7C27Mo2, the life of discharge valves was prolonged to 500 hours. The broken valves are shown in Figure4.

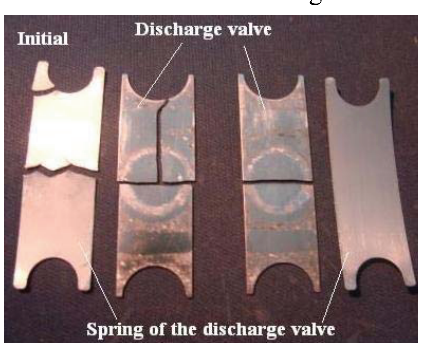


Figure3: The initial broken discharge valve

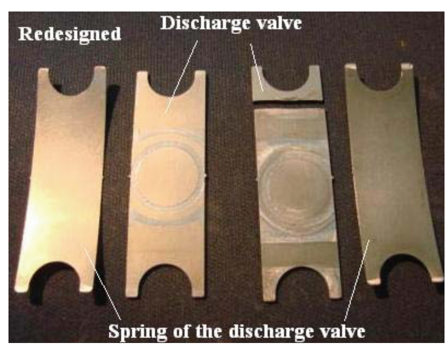


Figure4: The redesigned broken discharge valve

Table 2 Experimental Configuration for Discharge Valve

Model	Reciprocating CO_2 Compressor
Discharge Valve	Reed Valve
Thickness of Valve	1 mm
Thickness of Spring	0.203, 0.305, 0.457 mm
Material	Sintered Steel, 7C27Mo2, Sandvik
Operating Fluid	R774
Operating Discharge Pressure	7.5~12.0MPa

Although the discharge valve has broken several times, the suction does not have such problem because of the relatively low suction pressure. Thus, the movement of the discharge valve is measured in order to investigate the valve dynamics [3, 4].

3 EXPERIMENTAL METHOD AND FACILITIES

3.1 Rig for performance of CO_2 compressor

A transcritical CO_2 refrigeration test rig was built to do the performance experiments of CO_2 compressor and other special tests. Three major circuits compose such parts as the circuit of transcritical CO_2 refrigeration cycle, the water cycle in the side of gas cooler and evaporator. The cooling capacity in evaporator can be balanced by a heater in cold water tank and its maximum power is 50kW. Also, the heating capacity in gas cooler is balanced using a water chilling unit affording cooling capacity 63kW. A manual needle valve is installed in experimental system to regulate the flow of CO_2 fluid. Material of the pipeline in system is stainless steel, and it can bear up to 14MPa pressure (a pressure-head switch is used). A blast plate is designed in the discharge chamber of the compressor to ensure the safe operation of system.

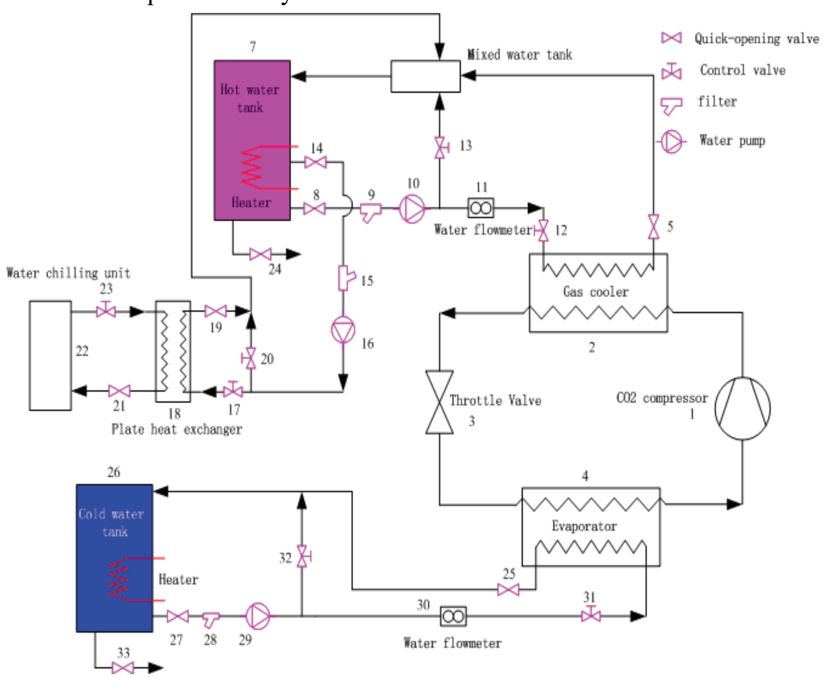


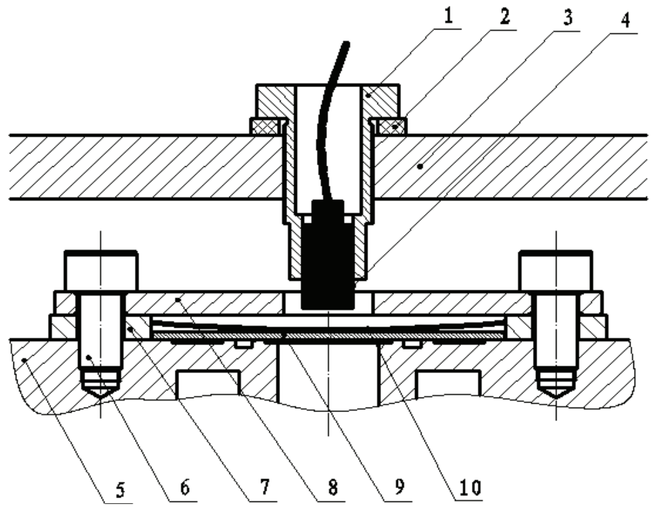
Figure5: Transcritical CO_2 refrigeration system

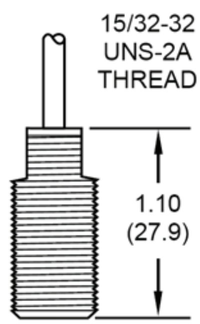
3.2 Measuring system to record valve displacement

Due to the operating characteristic of transcritical reciprocating CO_2 compressor, some key factors need to be considered for selecting the suitable postion sensor, such as high pressure, seal in high pressure and limited setting space. An eddy current sensor was selected and it is manufactured by an international aerospace corporation. The selected sensor can satisfy the requirement of high pressure (8.0MPa~12.0MPa), temperature (90℃~130℃) and frequency response (50 kHz). Moreover, the size of sensor is small enough to be installed in limited space. The method of installing the sensor is illustrated in figure 5.

Major challenge for installing the sensor is the sealing problem caused by the high pressure. Through the installing structure, shown in figure6, this issue is divided into two leakage paths which are easier to seal: one is between the sensor and the tube, and the other is between the tube and cylinder lid. For the former one, some epoxy sealant are filled into the gap of screw thread between sensor and tube, and two locknuts are fitted in both sides of sensor. A red copper gasket is used to solve the leakage for the latter leakage path, and the height of the gasket also adjust the distance between the top of sensor and measured valve.

In order not to damage the real stiffness of the spring, we did not drill the hole in the spring as in the valve stop. The sensor measures the displacement of spring actually. Because of the enough pretightening force for spring, we suppose that the spring can move at the same pace with the valve. So, the measured results can be regarded as the right motion rule of discharge valve. Moreover, the bending radius of spring is very big, about 230mm so the testing surface is considered as a flat face which is the basic test requirement for sensor.





Eddy current sensor

Figure6: Position sensor and installed structure
(1-Tube, 2-Sealing plate, 3-Cylinder lid, 4-Position sensor,
5-Valve plate, 6-Bolt, 7-Distance ring, 8-Vlave stop, 9-Discharge valve, 10-Spring)

4 RESULTS AND DISCUSSION

Through varying design parameters of the discharge valve, movement of the valve is acquired to explore the factors that greatly influence the valve dynamics. Tests are conducted under different pressure ratio, valve lift, spring thickness, and compressor speed. Additionally, p-t diagram inside the cylinder is analyzed with the valve movement to get more information about the valve dynamics.

According to the experimental results of the discharge valve movement, the discharging process lasts about 0.2 T. After being opened, the valve is blown up quickly, while the pressure in the cylinder decreases immediately. Then the valve is rebounded back to the valve plate after hitting the valve stop. During the rebound process, there is a fluctuation when the valve is rebounded to about the half height. Finally, the valve clings to the valve after a little rebound. Under different tests, the valve hits the valve stop only once. Although the trends of the valve movement are the same under different tests, detail differences can be observed.

4.1 Valve movement under different discharge pressures

Suction pressure is kept constantly at 3.0MPa, while discharge pressure is adjusted to 8.0MPa, 10MPa and 12.0MPa. Figure7 indicates the valve displacement curves under different discharge pressure. Common to all the curves is listed as follows: Firstly, the valve is opened when the pressure in the cylinder reaches the maximum. Then the valve hits the valve stop and is rebounded to the valve plate. And we can see the fluctuation during the rebound process.

Careful examination of closing process (as show in A) indicates that the discharge valve is closed more rapidly when the discharge pressure becomes larger. And the rebound process (as show in B) tells us that the valve is blown higher by the discharge gas with higher pressure.

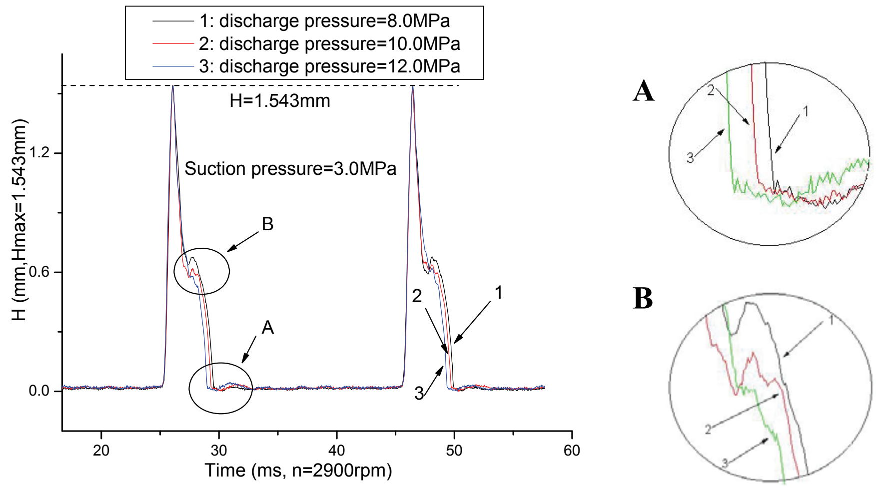


Figure7: Displacement curves of discharge valve under different discharge pressures

4.2 Valve movement under different lift

Through changing the distance rings, valve lift is adjusted to 2.5mm, 3.0mm, and 3.5mm. The displacement curves are shown in Figure8. When the valve lift becomes larger, the rebound curve becomes smoother, as curve 3 shows. Otherwise, the rebound curve would fluctuate intensely, as curve 1 shows.

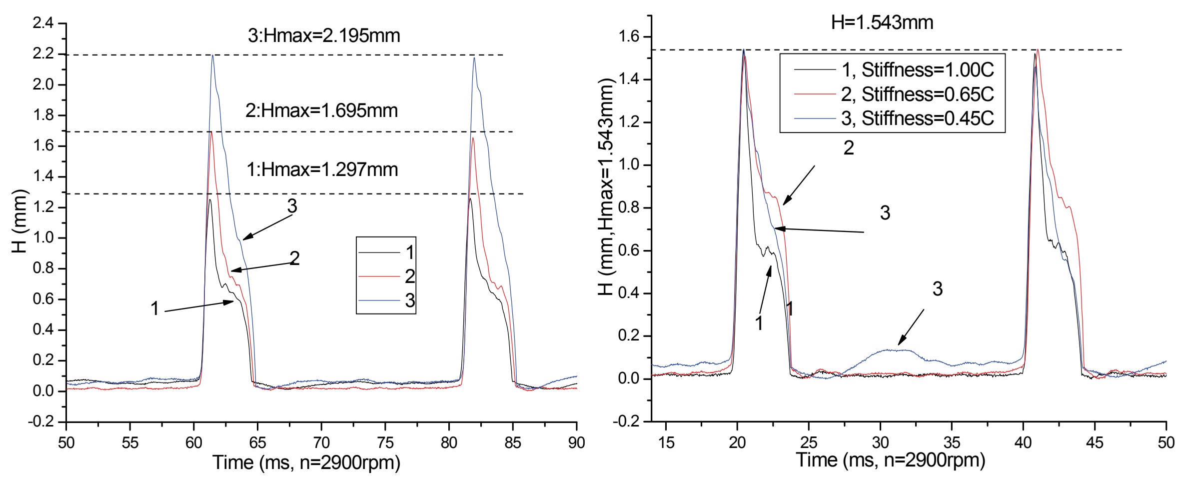


Figure8: Displacement curves of discharge valve under different lift (left)

Figure9: Displacement curves of discharge valve under different spring stiffness (right)

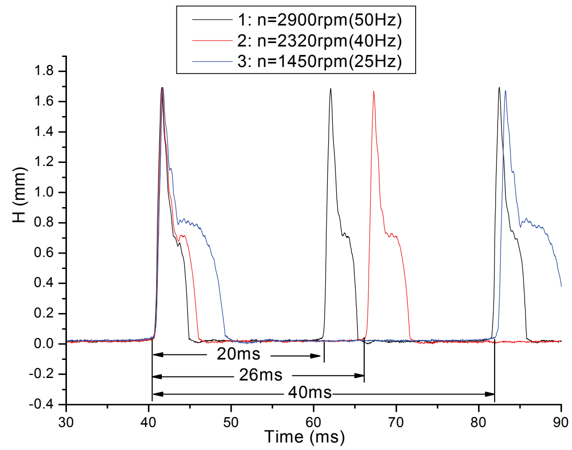
4.3 Valve movement under different spring stiffness

For the constant valve lift (H=1.543mm), different springs are installed in the discharge valve to investigate the influence of the spring stiffness. The displacement curves of discharge valve under different spring stiffness are presented in Figure9. From curve 1, we can see the valve is always kept on the valve plate, which means the sealing effect is quite satisfying. However, the valve fluctuates intensely when it comes to curve 3, which will increase the leakage from the discharge valve.

The area under the displacement curve reflects the pressure loss at the discharge valve. The larger the area is, the less pressure loss there will be. The relatively small area under curve 1 demonstrates that the pressure loss would be more when the spring is thicker. It is necessary to select a spring with proper thickness in order to strike a balance between pressure loss and leakage.

4.4 Valve movement under different compressor speed

Through adjusting the frequency of the current, the compressor speed is changed from 2900rpm-1450rpm. As shown in Figure 10, the discharging period increases after decreasing the compressor speed. Also, the area under the displacement curve becomes larger, which means the pressure loss would decrease. However the isentropic efficiency is reduced when the compressor speed is lowed because of the increasing leakage through the piston rings, as shown in Figure 11.



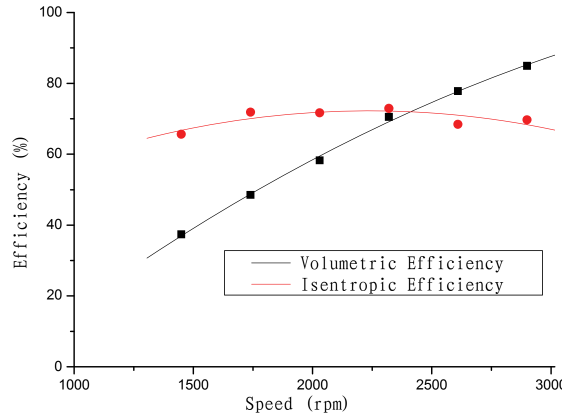


Figure10: Comparison of different speed for displacement of discharge valve (left)

Figure11: Volumetric efficiency and isentropic efficiency for different speed (right)

4.5 Valve movement with p-t diagram inside the cylinder

Figure12 presents the p-t diagram inside the cylinder, together with the displacement curve of the discharge valve. The valve opens when the pressure inside the cylinder reaches the maximum. However, the valve is still opening when the pressure inside the cylinder becomes lower than the discharge pressure, as shown in A. This indicates that the valve later closing than expected, which will reduce the volumetric efficiency. It can be optimized by adjusting the stiffness of spring or the height of the valve lift to resolve.

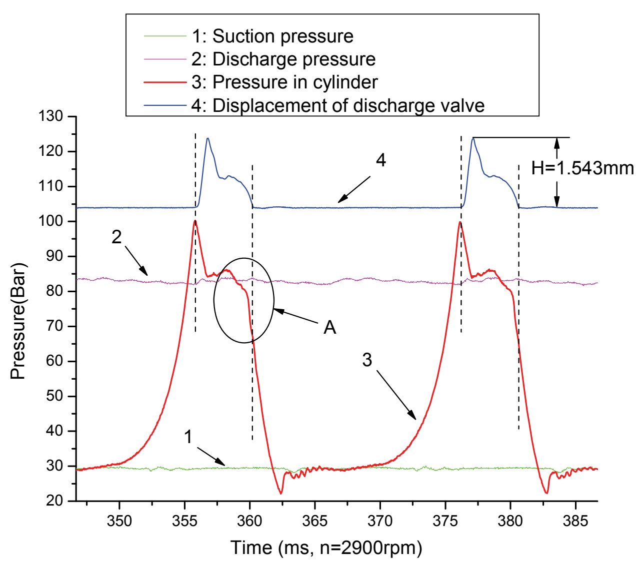


Figure12: Relation between displacement of discharge valve and p-t diagram

5 CONCLUSIONS

Through varying design parameters such as valve lift, spring thickness, pressure ratio and compressor speed, the movement of the valve is acquired to investigate the factors that greatly influence the valve dynamics. The results could be used for the optimization of the valve.

1. When the pressure ratio becomes larger, the discharge valve is closed more rapidly and the valve is blown higher by the discharge gas during the rebound process.
2. When the valve lift increases, the rebound curve becomes smoother but the valve would fluctuate on the valve plate, which would reduce the volumetric efficiency.
3. Proper spring with optimized thickness should be selected in order to strike a balance between pressure loss and leakage.
4. Low compressor speed could increase the discharging period and the area under the displacement curve, which there will be less pressure loss. However the isentropic efficiency would be reduced when the compressor speed is lowed.
5. The eddy current sensor is effective to measure the valve displacement of the valve for transcritical CO_2 compressor.

NOMENCLATURE

Symbols

P	Pressure	[MPa]
H	Height of the distance ring	[mm]
n	Speed of the CO_2 compressor	[rpm]
t	time	[ms]

REFERENCES

Junghyoun Kim, et al.Valve Dynamic Analysis of a Hermetic Reciprocating Compressor， International Compressor Engineering Conference. Purdue, 2006

Jeffrey J. NIETER. Experiences with Application of a CO_2 Reciprocating Piston Compressor for a Heat Pump Water Heater, International Compressor Engineering Conference. Purdue, 2006

Hsinheng. Li, T. E. Rajewski. “Experimental study of lubricant candidates for the CO_2 refrigeration system.” The proceedings of the 4th IIR - Gustav Lorenten Conference on Natural Working Fluids at Purdue, P409 – 416,

Li Hsinheng, Rajewski T E. Experimental study of lubricant candidates for the CO_2 refrigeration system//The proceedings of the 4th IIR-Gustav Lorentzen Conference on Natural Working Fluids. Purdue, 2000: 409-416.

Jurgen Sub, Horst Kruse. “Efficiency of the indicated process of CO_2 compressor.”Int J . Refrig. Vol. 21 , No. 3. pp194-201 , 1998.

Officinal Mario Dorin SPA. Semi-hermetic CO_2 Compressors for commercial applications. In: IEA Heat pump centre/ IIR Workshop on CO_2 technology in refrigeration, heat pump & air conditioning systems, Trondheim Norway. 1997. 5

Neksa.P, et al. Development of two stage semi hermetic CO_2 compressors. In: 4th IIR Gustav Lorentzen conference on natural working fluids, Purdue University, USA. 2000 :355～362