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Study of Valve Motion in Reciprocating Refrigerator Compressors Based on the 3-D Fluid-Structure Interaction Model

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ABSTRACT

In this paper, valve motion in the refrigerator reciprocating compressor is studied based on a 3-D fluid-structure interaction model. Experiment of the P-V graph was carried out to validate the numerical model. Simulations showed that increased thickness of the valve would finally prolong delayed closure. It seems a little different from the traditional prediction that increased thickness would result in increased strength and advanced closure.

1. INTRODUCTION

The compressor is core of a refrigerating facility. Its reliability is very important to the whole system. It is well known that valves are main wearing parts in a reciprocating compressor (Danqing W., Jingtong C., 1979). In published references, many different methods have been adopted to simulate the working process and the valve motion of the reciprocating refrigerating compressors. Most of the estimation of inner state parameters of the compressor is based on energy balance and spring model of the valve motion (Junlong Z. *et al.*, 2012; Satyam B., 2008). One-dimensional method has been applied to simulate and design the compressor including the valve (Mithraratne P. *et al.*, 2000; Lee L. and Randall J., 2003). Experience parameters are necessary for the simulation program. And the one dimensional model fails to predict the detail of the motion of the valve. Multi-dimensional method is also used to simulate the compression and valve motion (Derek S. and Sorin M., 2002; Kim J. *et al.*, 2006). It is complex and costs big computer resources.

In this paper, a refrigerator compressor with refrigerant R600a is adopted to carry out the simulation of the valve motion based on a 3-D fluid-structure interaction model.

2. THE SIMULATION MODEL

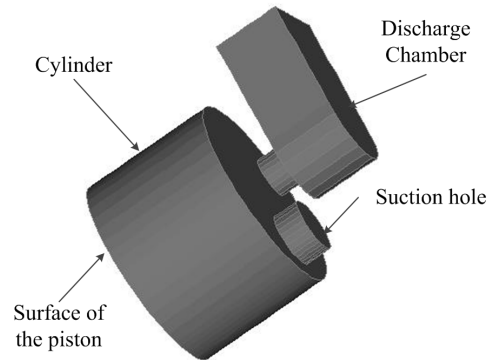
The geometric model is established using Unigraphics NX8 software, and the simulation is carried out based on the ADINA. Thermal physical parameter of the Refrigerant comes from NIST database.

2.1 The geometric Model

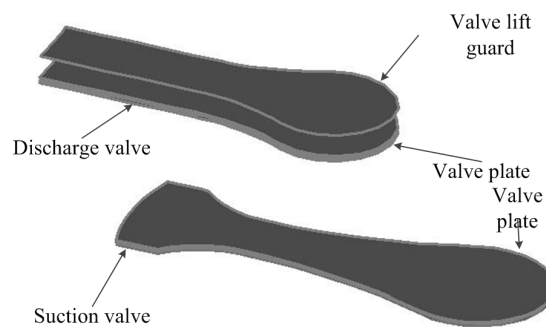
Geometric model of a refrigerator compressor is established. Displacement of the compressor is 15.3 cm³. Parameters of the compressor are shown in Table 1. Geometric model of the compressor is shown in Figure 1.

Table 1: Parameters of the compressor

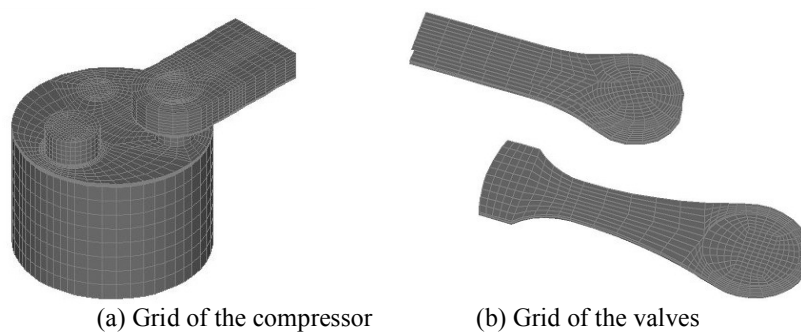
Parameters	Value
Diameter of the cylinder/mm	31
Stroke/mm	24
Diameter of the suction valve hole/mm	8.2
Diameter of the discharge valve hole/mm	5.6

**Figure 1:** Geometric model of the simulation model

The solid models of the suction valve and the discharge valve (Mei L. and Yongzang Y., 2000) are shown in Figure 2.

**Figure 2:** Solid model of the suction valve and the discharge valve

Grid of the model is shown in Figure 3.



(a) Grid of the compressor

(b) Grid of the valves

Figure 3: Grid of the compressor and the valves

2.2 Boundary Conditions

Temperature and pressure conditions of the suction gas are set as 305K and 0.068MPa. For the discharge port, pressure pulsation recorded by experiments is input into the simulation model as the boundary condition, to ensure the accuracy of simulation results of the valve motion.

Gap condition is set for the suction valve and the discharge valve. The piston surface is set as the moving wall. Eccentric installation of the cylinder is considered in the simulation model. The motion equation of the piston is

$$x = R[(1 - \cos \alpha) + \frac{\lambda}{4}(1 - \cos 2\alpha) - \lambda\eta \sin \alpha + \frac{\lambda^2\eta^2}{2(1 + \lambda)}]. \quad (1)$$

Here, λ is the ratio of rotating radius, R , of the crankshaft to the length, L , of the rod. η is the eccentric ratio of the eccentric distance to the rotating radius, R .

To simulate the deformation and motion of the valve, the leader-follower condition is set between the valve and its contact surface as shown in Figure 4.

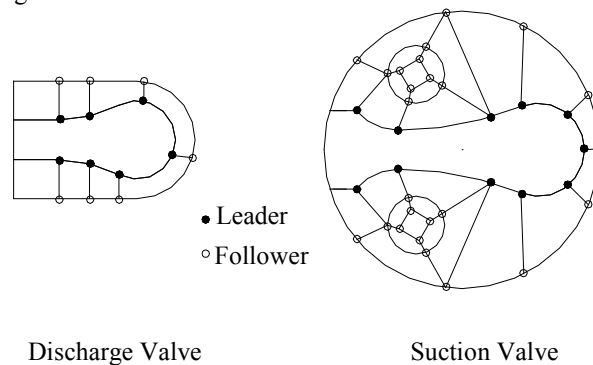


Figure 4: Leader-follower condition of the valve

3. VALIDATION OF THE 3-D FLUID-STRUCTURE INTERACTION MODEL WITH EXPERIMENTS

To validate the 3-D fluid-structure interaction model, experiments of the inner working process was carried out.

3.1 The experiment system

An experiments system of the inner working process of the compressor was established as shown in Figure 5.

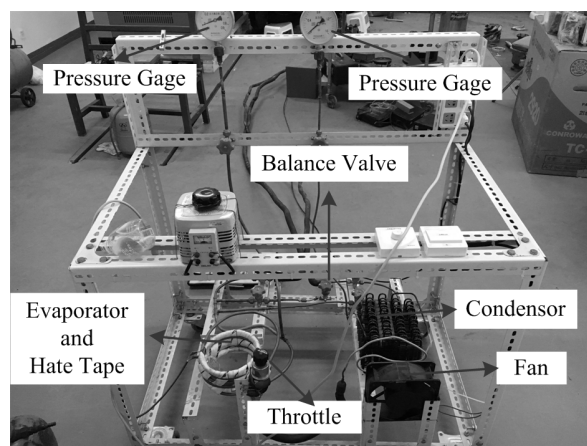


Figure 5: Experiment system of the refrigerator compressor

The inner pressure is tested using a kulite pressure sensor with the type of XTL-AC-190M. Rotating angle of the crankshaft is tested using a Proximity Switch produced by Omron. All the data is recorded by a NI system.

3.2 Comparisons of the simulation results to the experiment results

A comparison of the working process obtained by simulation and experiment is shown in Figure 6. The rotation speed of the compressor is 2970 rpm, the suction pressure is 0.068MPa, and the discharge pressure is 0.78MPa. Well consistency between the simulation results and the experiment results could be found.

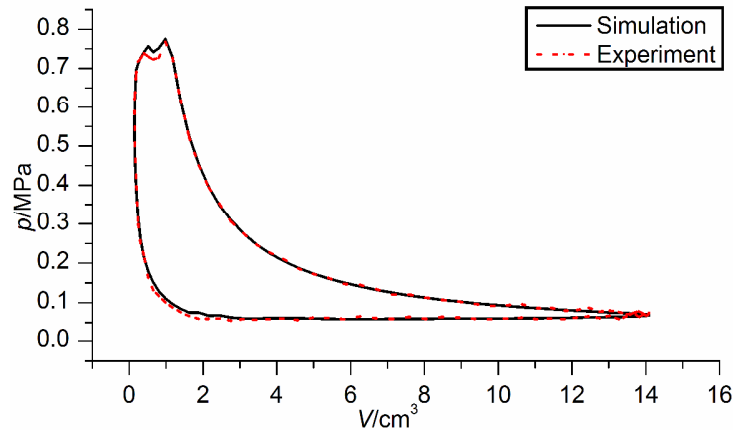


Figure 6: P-V diagram obtained by the experiment and simulation

The difference of the power between the simulation and the experiment is 1.9%. It means 3-D fluid-structure interaction model proposed in this paper is valid.

4. SIMULATION RESULTS OF THE VALVE MOTION

The 3-D fluid-structure interaction model could predict the three dimensional motion of the valve. For the reed valve in the refrigerator compressor, a decline of the valve would result in difference lift at different place of the valve. Thus motion of the center point, which is located at the center of the area covering the suction hole or discharge hole on the valve, is chosen as motion of the valve. Motion of the valves is then shown by the lift-angle line in the Figure 7.

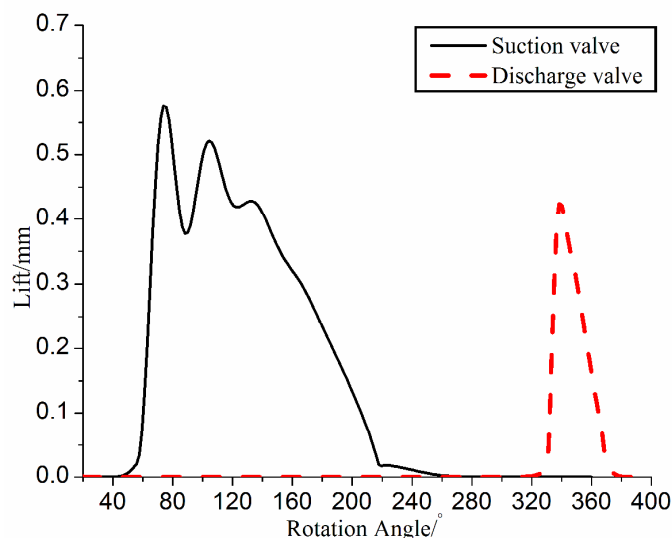


Figure 7: The lift-angle line of the valve motion

It could be found that, there are time delayed closure, both of the suction valve and the discharge valve. It is normal for a small reciprocating compressor. To obtain the highest efficiency, the suction should be as full as possible, and

the discharge should be as empty as possible. A little delayed closure could enlarge the lift-angle section, which means low suction resistance. And such a small delay results in almost no leakage. It is similar for the discharge valve.

Simulation of the valve with different thickness shows that an increasing thickness of the valve would result in prolong the delayed time of the closure.

4.1 Influence of the thickness on the closure of the suction valve

Motions of the suction valve with three different thicknesses are shown in Figure 8. It could be found that the lift-angle section of the valve motion is decreased with the increasing of the valve thickness. That means increased resistance of the suction. However, the delayed time of the closure point is prolonged with the increasing of the valve thickness. For the thickness of 0.152mm, 0.203mm and 0.254mm, the closure point of the valve located at the angle of 203.4°, 214.2° and 219.6° respectively. This conclusion seems conflict to traditional predictions that the increased strength of the valve would result in advanced closure.

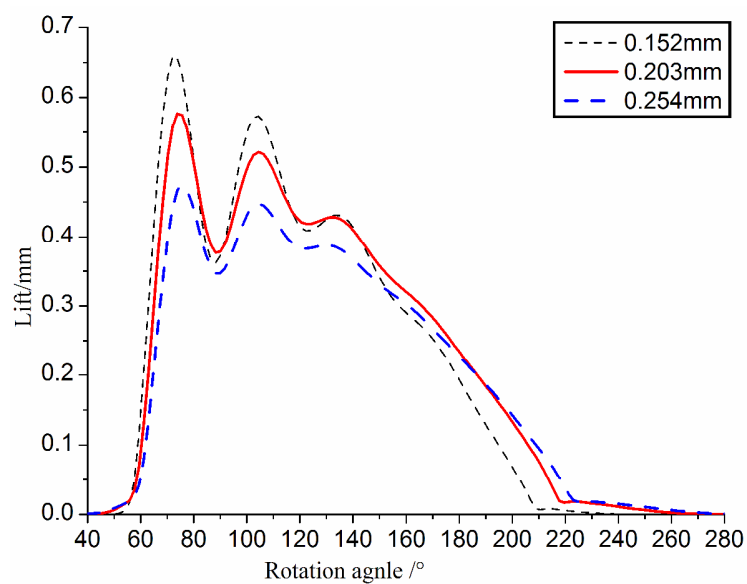


Figure 8: Motion of the suction valve with different thickness

This motion could be explained by investigating the forces acting on the valve. The simulation results show that higher gas load acting on the thicker valve. For this suction valve, the closure point is after the downside dead point, which means the closure is after the 180°, where the piston is moving toward the suction valve. This could result in quick close. However, the most important reason is the lift of the valve. For a thinner valve, the valve lift is bigger. That means a bigger deformation. During the close process, this bigger deformation means quicker recovery. Furthermore, the lift of the valve is longer, which could result in a higher final velocity of the valve. Because of these reasons, the thinner valve results in an advanced closure than that of the thicker valve. Using traditional one dimensional method fails to get this conclusion. This is the advantage of the 3D method because of its details in calculating the gas force on the valve plate, the bending of the valve plate and its related rebound force which drives the valve plate fall back on the valve hole.

However, the COP of the compressor is decreased with the increasing of the valve thickness.

4.2 Influence of the thickness on the closure of the discharge valve

For the discharge valve, the situation is similar. The valve motions with three valve thickness are shown in Figure 9.

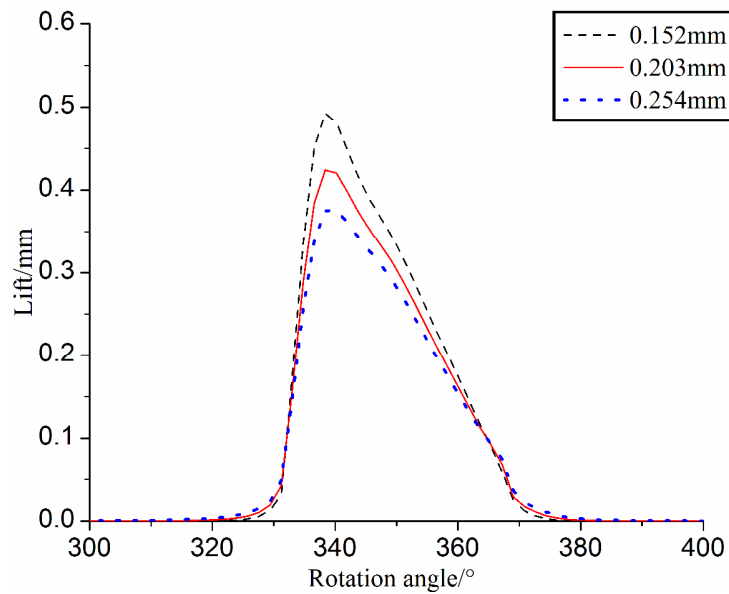


Figure 9: Discharge valve motion with three different thickness

The enlarged figure of the lift-angle line at the closure point is shown in Figure 10. A clearly delayed closure is shown. For different thickness of 0.154mm, 0.203mm and 0.254mm, the crankshaft angle of the closure point are 379.8° , 387° and 408.6° respectively.

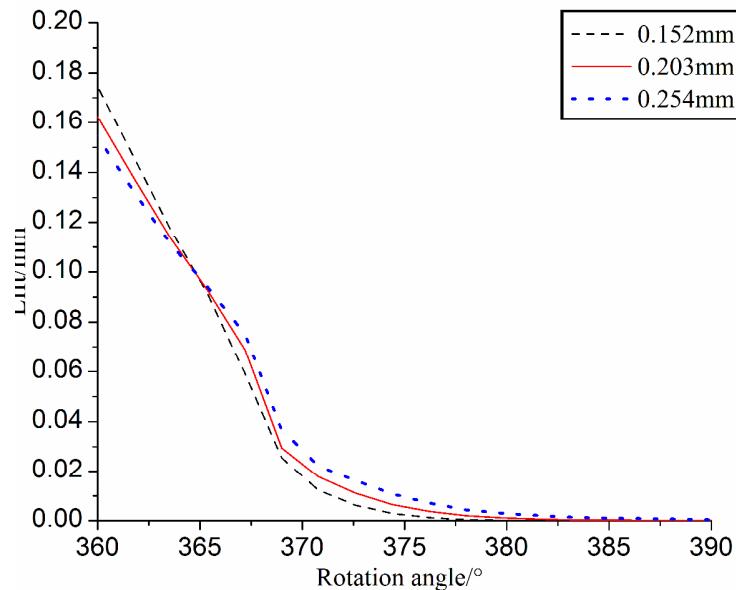


Figure 10: Enlarged figure of the lift curve at the closure point

It is same with the suction valve that the large lift of the valve with the thinner thickness finally results in its quick closure.

5. CONCLUSIONS

A three dimensional fluid-structure interaction model of small refrigerator reciprocating compressor was established in this paper. The simulation model was validated by experiments of investigating its inner working process. Then simulation of the motion of both the suction valve and the discharge valve was carried out.

Obtained results show that increased thickness of the valve could result in delay of its closure. The situation is same for the suction valve and the discharge valve. Reason for this situation is a little complicated. It is a comprehensive result of the motion of the valve plate, due to the changing rebound force, the gas force on the valve plate and the lift. Overall, the higher lift of the thinner valve is the main reason for the quicker recover of the valve plate. This is advantage of the 3-D Fluid-Structure Interaction simulation method.

NOMENCLATURE

P	Pressure	(MPa)
V	Stroke Displacement Volume	(cm^3)
R	Crankshaft Rotating radius	(m)
α	Rotation angle	($^\circ$)
L	length of the rod	(m)
λ	R/L	
η	ratio of the eccentric distance to R	
x	Displacement of the piston	(m)

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