IOP Conf. Series: Materials Science and Engineering **232** (2017) 012005 doi:10.1088/1757-899X/232/1/012005

Investigation on the variation of pressure in the cylinder of the refrigerator compressor based on FSI model

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Abstract. The performance of refrigerator compressor has a significant influence on the coefficient of performance of the refrigerator. In order to find out the main cause for the variation of the performance, this paper presents a three-dimensional fluid structure interaction (FSI) model of the refrigerator compressor and an experimental investigation to verify the model. To record the p-V indicator diagram, a refrigerator compressor was modified and the pressure sensor was installed. Based on the FSI model, the variation of pressure in the cylinder that influenced the performance of compressor was identified by varying working condition, rotating speed and refrigerant. The FSI model and result could provide useful information for both performance testing and optimization of compressor performance.

1 Introduction

In recent years, domestic refrigerators with low power consumption and eco-friendly are in high demand because the amount of electricity consumed by refrigerators is among the highest for all consumer household products. The reduction of the power consumption of refrigerators is greatly dependent on a highly efficient compressor.

To develop a highly efficient compressor, it is important to have an in-depth research on the variation of pressure in the cylinder. A number of factors affect the variation of pressure in the cylinder of a reciprocating compressor. To fully understand the inner workings of a reciprocating compressor and to improve the compressor efficiency, the numerical simulation and experimental investigation are the main research methods.

The numerical tools give much help to understand the dynamic characteristics of compressor. Winandy et al. used a numerical model for the estimation of compressor performance [1]. Elhaj et al. applied the same compressor modeling to a two-stage reciprocating compressor for condition monitoring [2]. Yang et al. presented a comprehensive simulation model for a semi-hermetic CO_2 reciprocating compressor in which the leakages and the frictions are considered [3]. Farzaneh-Gord et al. developed a mathematical model based on real gas model to simulate natural gas reciprocating compressors [4].

With the development of Computational fluid dynamics (CFD) technology, CFD has been widely used as a design tool in many engineering applications. Yu wang et al presented a numerical simulation of the thermodynamic process in the cylinder and dynamics of the self-acting valves for an air reciprocating compressor [5]. Akira and Kenji showed the CFD applications in suction muffler and compressor valve design [6].

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The latest research hotspots are the interaction of different physical fields, especially fluid structure interaction, as that is a typical fluid structure interaction problem. Hyeong-Sik et al showed the FSI in impact analysis of compressor discharge valves [7]. Aditya and Josep simulate numerically the FSI for flow through valves of a hermetic compressor using immersed boundary method [8]. Qin Tan et al proposed a three-dimensional fluid–structure interaction model for the discharge valve movement in a rotary compressor [9].

All the above mentioned numerical researches need to be verified with experimental data. Yuan Ma et al conducted an experimental investigation of the discharge valve dynamics in a reciprocating compressor for trans-critical CO2 refrigeration cycle [10]. José Luiz GASCHE et al measured the instantaneous position of the reed by using a very accurate optical sensor [11]. Dr. Kemal Sarıoğlu et al performed detailed temperature and pressure measurements as the boundary conditions for numerical study [12].

In this work, a hermetic reciprocating compressor was modified to record the p-V diagram in the cylinder. A three-dimensional fluid structure interaction (FSI) model was built based on the hermetic reciprocating compressor. Comparison of simulation results with experimental data implies that present model can well describe the working process of compressor. This FSI model could be used for a new compressor development.

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2 The FSI model of refrigerator compressor

It is well known that the flow of refrigerant gas and the valve movement are coupled with each other in the working process of the valve in refrigerator compressor. Valve will produce bending deformation under refrigerant fluid load, and the deformation of valve in turn affects the flow of refrigerant. This kind of problems is much more complicated, because not only the equations of structure and CFD must be solved but also the interaction between them should be considered. The fluid and solid equations are solved individually using the latest data provided from each other. In this analysis, compressor model is divided into fluid domain and corresponding structural domain.

2.1 The fluid domain

Fluid domain of compressor model is divided into three primary parts, the cylinder part, the discharge volume and the suction pipe. Between those parts, gap boundary condition is imposed to segregate discharge volume and cylinder, also to segregate cylinder and suction pipe. FSI boundary is imposed on corresponding area where discharge valve and suction valve are placed in the coordinate system.

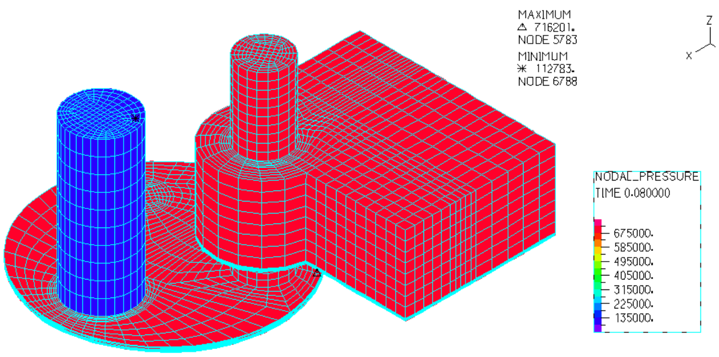

Fig.1. model of compressor fluid domain

The working fluid is R600A, and material properties of fluid are shown in Table 1. The fluid is assumed as low speed compressible flow model. In the fluid domain, the initial pressure and temperature in the cylinder and discharge chamber are given as 0.68MPa and 308.15 K, respectively. The pressure at the suction pipe (suction pressure) is given as 0.0672MPa.

The RNG model is employed to simulate the turbulent flow in the valve and the cylinder. The fluid domain and the structural domain are all discredited using 8 node 3D element. Solve time step is 1E-5 second.

2.2 The structural domain

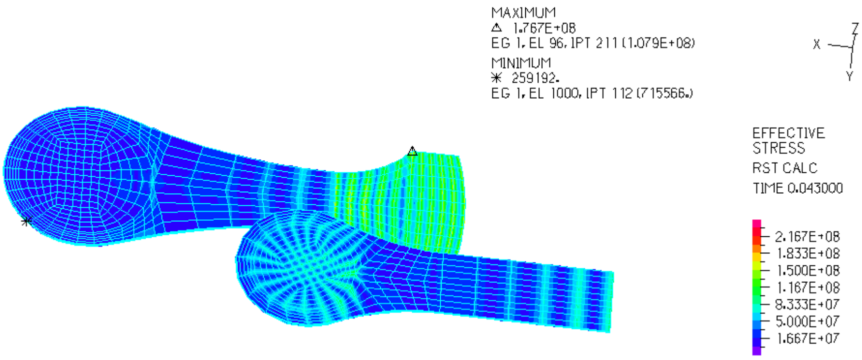

Fig.2. Finite element model of compressor structural domain

The structural domain of compressor model includes the suction valve, the discharge valve, the valve seat and the retainer with the discharge valve. The contact condition is introduced between the reed valve and the valve seat. FSI boundary is imposed on the surface of the discharge valve and suction valve. Material properties used in this model is constant properties with E = 210GPa, v = 0.25, and p = 7800 kg/m^3.

2.3 The moving wall

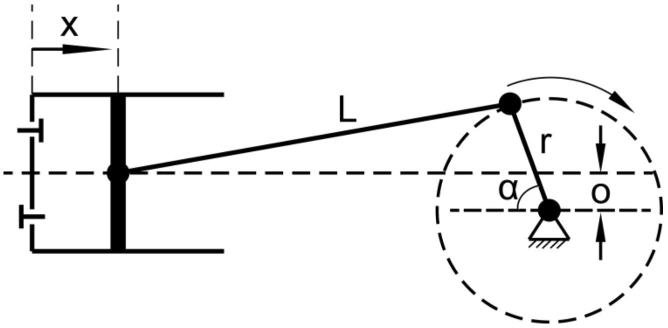

Fig.3. Schematic diagram of the refrigerator compressor

The piston movement is determined by the compressor crank as Figure 3. The piston position in dependence of crank drive parameters as equation (1-1) is considered as moving wall in the FSI model. In this paper, the offset of the piston which makes the lateral force applied on the piston well-distributed has been toke into account, and the piston move from the TDC (top dead center) to the BDC (bottom dead center) at start time.

$$x = r\left[\left(1-\cos\alpha\right)+\frac{\lambda}{4}\left(1-\cos 2\alpha\right)-k\lambda\sin\alpha\right] \qquad (1\text{-}1)$$

Where x is the piston displacement from the TDC, α is the crank angle of crank, r is crank radius, λ is ratio of crank radius to length of connecting rod, k is eccentric distance.

3 Experimental method

To verify the FSI model of the refrigerator compressor, experimental verification was carried out. The p-V indicator diagram, the pressure pulsation in suction pipe and discharge pipe and the suction valve displacement have been obtained by the experimental facilities. The valve dynamics is a strong fluid-structure coupling because the valve motion is a response to the pressure force it experiences, and the pressure evolves based on the mass influx/efflux through the valve at a particular lift.

3.1 Refrigeration system

The refrigeration system has been built up to adjust the operation condition of the compressor, as Fig.4 showed. The main refrigeration loop consisted of a compressor, condenser, manual throttling valve, evaporator and other additional components. There was a heating belt to heating the evaporator. The balance valve was used to connect the evaporator and the condenser when charging refrigerant. The operation condition can be coordinative adjusted by the heating belt, the throttle valve and the fan.

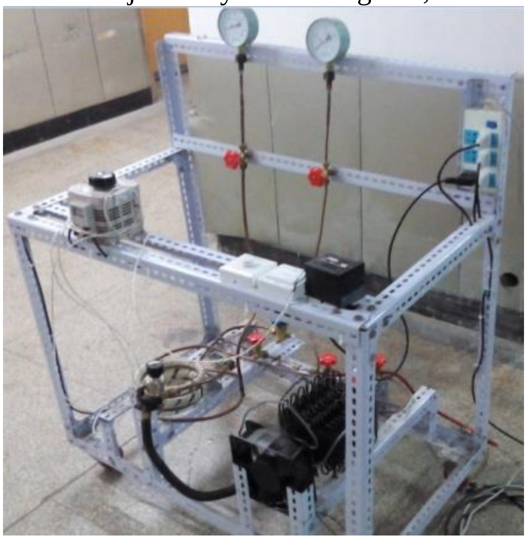

Fig.4. Test facility for refrigeration system

3.2 Modification of compressor to record the p-V indicator diagram

Conventionally, a hermetically sealed compressor shell comprises of two semi-shells welded together. Since a frequent modification of the system takes place, a modified flange shell is recommended for such experimentations. Due to the small size of the refrigerator compressor, the experimental study on recording the p-V diagram was complex. In order to measure the transient pressure together with the rotation angle, a high precision pressure sensor was adopted. Considering the practical compressor structure and sensor dimensions, the sensor was installed as shown in Fig. 5(a). The pressure sensor was embedded in the valve seat Fig. 5(b). And the signal wire was led out.

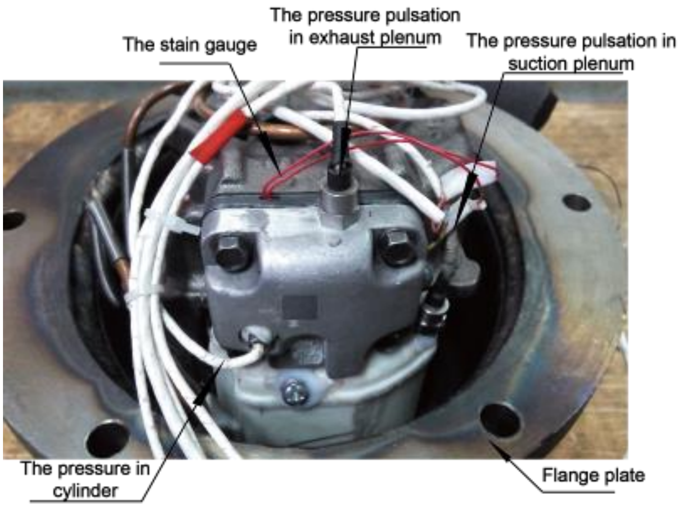

Fig.5. The modification of Compressor

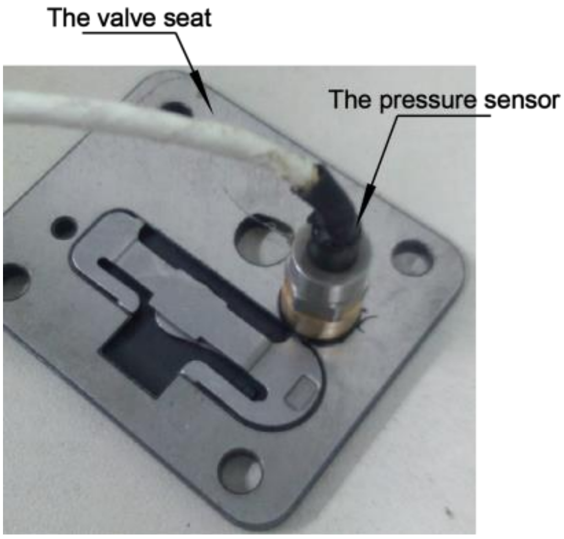

Fig.6. Schematic of sensor installation

It was difficult to obtain the positions of the piston directly, but the position of the piston is determined by the compressor crank as the equation (1-1). Thus, an approach switch was mounted on the frame directly facing to the big end of connecting rod, as shown in Fig.5 and Fig.6. When the piston was at the BDC (bottom dead center), the big end met the approach switch during the compressor was running, and a pulse signal was generated.
The dynamic pressure sensor uncertainty was less than ±0.1% and its response frequency was as high as 1400 kHz. The maximum working pressure was 1.9MPa, and the range of working temperatures was between -55℃ and 175℃

4 Results and discussion

4.1 Comparison of experiment and simulation

Firstly, for validation the FSI model in this study, the results of numerical method have been compared with available experimental results; generally, there are a good agreement between the measured and numerical values. One way to verify the valve efficiency is through the p-V diagram.

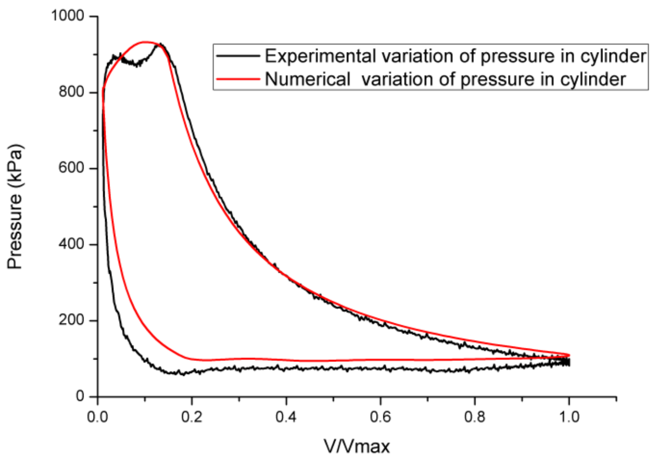

Fig.7. p-V indicator diagram

The understanding the power consumption on the refrigerating cycle and its losses can contribute to obtain better compressors. By the p-V indicator diagram it is possible to obtain the work exerted on the gas during the compression cycle and also the losses on the suction and discharge valves.

Comparison of the simulation and experimental p-V, the result of the FSI model was essentially in agreement with that obtained by experiment, in particular, the compression process completely overlap. According to the calculation mothed of power compression, numerical integral experiment and simulation of the compression work is 209.67W and 184.36W, respectively. It is proved that the calculation result of the FSI model is reliable.

4.2 Effect of variable working condition on compressor performance

The output of the simulation model includes the characteristic parameters' trend during the compression cycle. Fig.8 shows the compression cycle on the p-V diagram under the different operative conditions studied: it is possible to observe the compression phase, the discharge phase with the characteristic pressure fluctuation, the re-expansion of the refrigerant in the clearance volume and, finally, the suction phase.

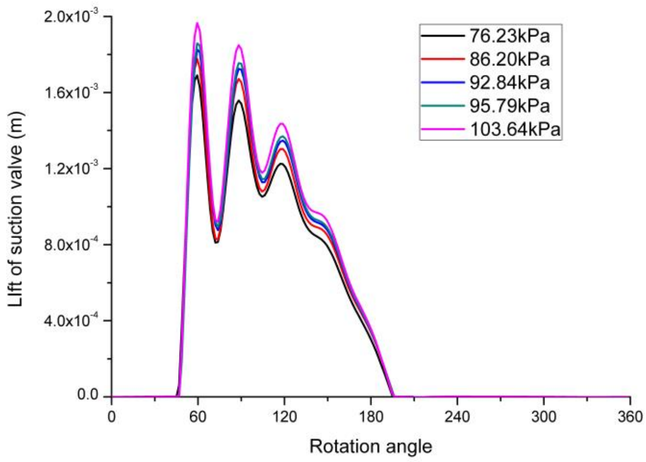

Fig.8. The lift of suction valve under different suction pressure

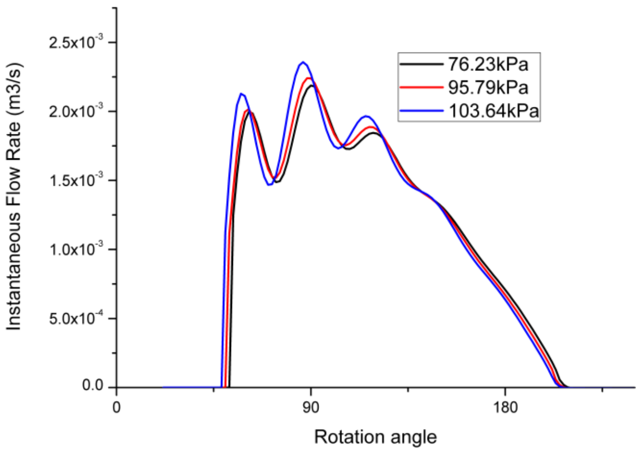

Fig.9. Instantaneous flow rate in suction phase under different suction pressure

The suction pressure significantly influences the motion of the suction reed valve. Under the same pressure ratio, the movement of the suction valve with different suction conditions were shown in Fig.8. As the thickness and stiffness of the valve plate remains the same, the time of valve opening and closing is consistent. With the increase of suction pressure, the maximum displacement of suction valve increased. And the refrigerant flow gets a larger flow area. When the suction pressure decreases, the gas force is smaller and the valve requires a relatively larger pressure difference to open, which leads to a larger flow loss and reduces the pressure of the gas refrigerant in the cylinder at the suction end, resulting in lower volumetric efficiency.

Variation of instantaneous flow rate with suction pressure was shown in the Fig.9, The instantaneous flow rate increases with the increase of suction pressure, since the suction valve get a larger flow area under the larger suction pressure.

4.3 Effect of rotating speed on compressor

Based on the FSI model, the effect of rotating speed on compressor has been studied at the same operating condition. As shown in the Fig.9, the speed has little effect on the suction phase. And the higher speed leads to higher pressure in compression phase, since the discharge valve flow area is relatively insufficient at the higher speed.

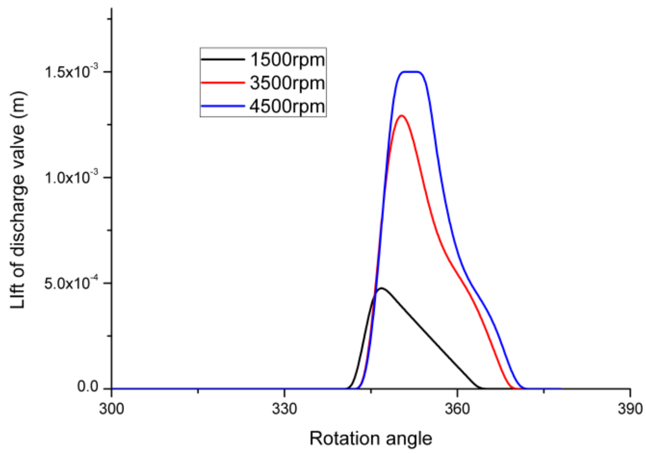

Fig.10. The lift of discharge valve under different rotational speed

The lifts of the discharge valve under different rotational speed were shown in Fig.10. Due to the higher rotating speed, the pressure in the cylinder increases rapidly while the flow capacity of the discharge valve is relative deficiency, which lead to higher pressure difference on the discharge valve in compression phase.

4.4 Effect of refrigerant on compressor

The variation of pressure in the cylinder with different refrigerants is studied, and the p-V diagram is shown in Fig.11. Since the polytropic exponent of R600a is close to that of R134a which is 1.0944 as shown in Fig.12, the compression and expansion processes of R600a coincide with that of R134a. The polytropic exponent of CO_2, namely 1.2602, is bigger than that of R600a and R134a , therefore, the indicated work of CO_2 is more than that of R600a and R134a.

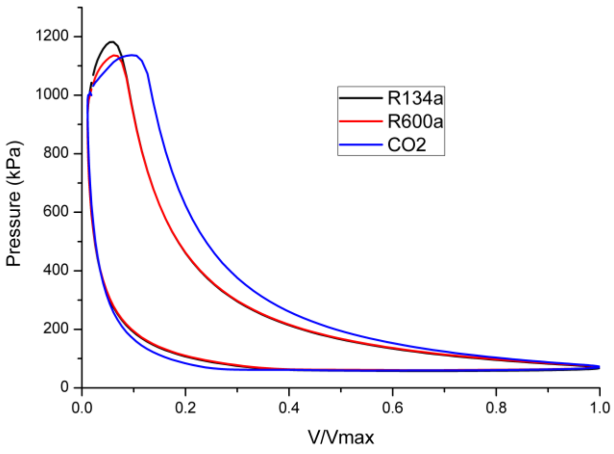

Fig.11. The p-V diagram with different refrigerants

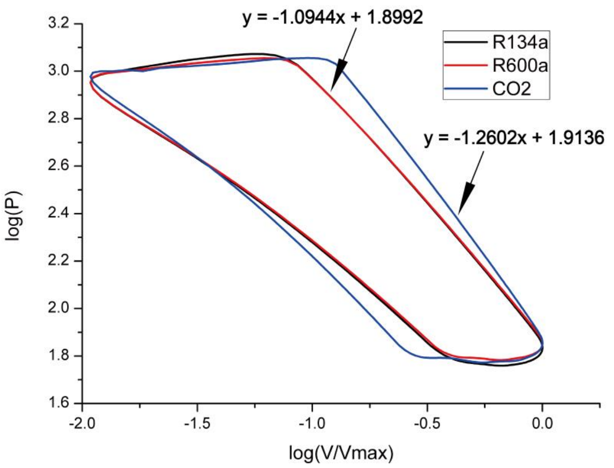

Fig.12. The log(p)-log(V) diagram with different refrigerants

Due to the density of R134a is larger than R600a, the pressure in the compression phase of R134a is higher than that of R600a. The property of refrigerant is shown in the Table 1.

Tab.1. The property of refrigerant (at P=101.23kPa, T=328.2K)

refrigerant	Density kg/m3	Viscosity (Pa·s)	Heat conductivity mW/(m·K)	Coefficient of volume expansion (1/K)	Cp kJ/(kg·K)	Cv kJ/(kg·K)
R600a	2.1985	8.2308E-6	20.152	0.0032426	1.8247	1.6689
R134a	3.8402	12.987E-6	15.793	0.0032102	0.89758	0.80972
CO_2	1.6386	16.374 E-6	19.103	0.0030846	0.87922	0.68701

5 Conclusions

This study provides an understanding of the Fluid Structure Interaction model of compressor. The simulation and experimental tools presented in this study offer promising opportunities for the development of high performance compressor.

Based on the FSI model, by varying the parameters such as the working condition, rotating speed and refrigerant, the main factors that influence the variation of pressure in the cylinder were identified and several conclusions have been made.

1. The three-dimensional fluid structure interaction (FSI) model which taking account of the offset crank mechanism has been established and experimental verified. And the FSI model is a useful tool for compressor performance analysis.
2. The instantaneous flow rate increases with the increase of suction pressure, since the suction valve get a larger flow area under the larger suction pressure.
3. The higher speed leads to higher pressure in compression phase, since the discharge valve flow area is relatively insufficient at the higher speed
4. The effect of polytropic exponent and density of refrigerant on the working process of compressor is studied.

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