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A Numerical Simulation of Fluid-Structure Interaction for Refrigerator Compressors Suction and Exhaust System Performance Analysis

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ABSTRACT

To analyze the performance of refrigerator compressor suction and exhaust system , fluid-structure interaction-based compressor suction and exhaust system analysis method was researched, and fluid structure interaction simulation model was built. The valve motion, mass flow rate, pressure-volume diagram, indicated power, cooling capacity, etc were obtained. The simulation results were validated by experiments. The fluid-structure interaction simulation method has been used in new compressor development.

1.INTRODUCTION

Developing a small hermetic reciprocating compressor is a complex process. Traditionally, the process was driven by experience and experiments, it takes lots of time and costs. Nowadays, customers pay more attention to details of products, they are not only sensitive to efficiency and price, but also sensitive to acoustics, reliability, etc. The new product development difficulty increases significantly. Experiment-driven method can not afford the hermetic reciprocating compressor development properly.

In recent years, numerical tools give much help to product development. Compared to experiments, doing numerical analysis costs less and is time-saving. For example, it can picture inside the compressor, but doing experiments inside the compressor is very difficulty and costly. More and more numerical tools are used in compressor engineering.

To understand the dynamic characteristics of compressor, Yong-Yeoun *et al.*(2004) showed the dynamic analysis with DADS. Rinaldo *et al.*(2006) showed the use of numerical structure analysis tools in compressor development. Akira and Kenji(2008) showed the CFD applications in suction muffler and compressor valve design. The latest research hotspots are the interaction of different physical fields, especially the interaction between fluid and solid domains (fluid structure interaction, FSI), as the valve dynamics influences the compressor performance greatly. Hyeong-Sik *et al.*(2010) showed the FSI in impact analysis of compressor discharge valves. Aditya and Josep(2012) simulate numerically the FSI for flow through valves of a hermetic compressor using immersed boundary method.

In this work, hermetic reciprocating compressor suction and exhaust system fluid structure interaction simulation method was researched. The Pressure-Volume diagram of cylinder, valve leaf motion, discharge pressure pulsation, mass flow rate etc were acquired. Simulation results were compared with experimental data. This method has been used in new compressor development.

2. THEORETICAL BASIS OF THE METHOD

To simulate the suction and exhaust process of compressor, three aspects should be considered. The first is how to solve the structure domain and get valve motion, the second is how to solve the fluid domain, the third is how to transfer data between structure solver and fluid solver.

Suction and discharge valve leaves are the mainly motion parts, the basis equation for kinetic analysis of valve leaves is:

$$[M]\{\ddot{X}\}+[C]\{\dot{X}\}+[K]\{X\}=\{F(t)\} \tag{1}$$

where $[M]$ is mass matrix of valve leaf, $\{\ddot{X}\}$ is acceleration, $[C]$ is damping matrix, $\{\dot{X}\}$ is velocity, $[K]$ is stiffness matrix, $\{X\}$ is displacement, $\{F(t)\}$ is the resultant force on the valve. To get the motion of valve leaf, Finite Element Method(FEM) was used, the structure domain was meshed into many sub-domains. In every sub-domain , algebraic equations approximate above kinetic differential equations.

Finite volume method (FVM) was used to solve fluid flows problems in the suction and exhaust system, the volume occupied by the refrigerant gas is divided into discrete cells, the governing partial differential equations (the Navier-Stokes equations, the mass and energy conservation equations, and the turbulence equations) are recast in a conservative form, and then solved over discrete control volumes.

During the fluid structure interaction, fluid dynamic solver transfer pressure to structure solver, between couple surfaces, the structure solver solve the structure equations to get the new position of the structure parts (valve leaves), when structure solving is finished , updated displacement data is transferred to fluid solver.

When the valve leaf is opening in the process, the physical bounds around valve leaf are changing with time, the mesh quality deteriorated rapidly. To guarantee the mesh quality, moving mesh is applied, and a mesh control strategy is built .

3.SIMULATION AND INENTIFICATION

3.1 Suction and discharge system model

The model include Finite Element Model of structural domain and Finite volume model of fluid domain. Fluid domain is divided into three regions, the cylinder, the suction line, the discharge line. The baffle boundary conditions are used to segregate the three regions. Moving mesh is used to simulate the variable volume in the cylinder.

Two wall boundaries are built around the area where suction valve and discharge valve are placed, they are used to exchange data with structural model. The fluid domain is meshed using 4-node tetrahedral and 8-node hexahedral 3-D fluid element, it is shown in Figure 1.

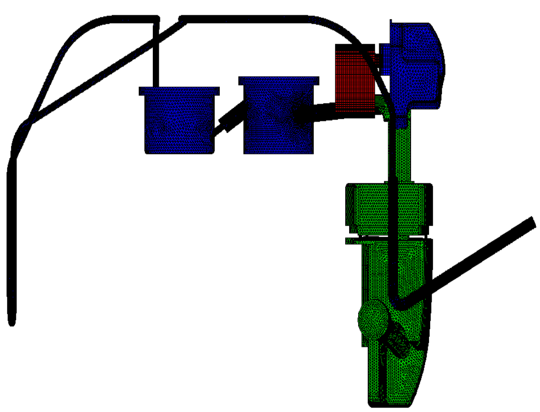

Figure 1: Finite volume model of fluid domain

Suction and discharge valve leaves are the main parts of structure, there is a limiter with the discharge valve. Structural domain is meshed using 8-node hexahedral and 6-node 3-D solid elements, it is shown in Figure 2. The leaf surfaces adhering to fluid domain are meshed with shell elements, they are used to exchange data with fluid model. Fixity is imposed at the root of suction and discharge valve.

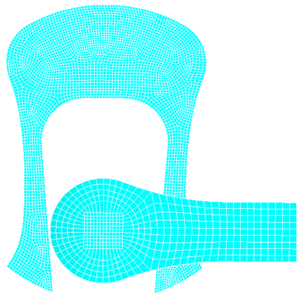

Figure 2: Finite element model of structural domain

The working fluid is refrigerant gas R600a. The ideal gas law is used to solve gas flow equation ,incompressible and viscous flow are used. The compressor rotation speed is 3000 RPM, solve time step is 5.0E-6 second. Initial pressure in discharge line is 0.77MPa , pressure in suction line and cylinder is 0.0632MPa. The suction muffler inlet temperature is 55 degree, gas superheat is considered.

3.2 Motion of suction and discharge valve

During cylinder movement, suction and discharge valves open and close periodically. Figure 3 plots the suction valve leaf motion. The horizontal axis represents crank angle. When piston is at lower dead center, the crank angle is 0 degree. The vertical axis represents the motion at valve head center node. Suction valve closed after 360 degree, maybe backflow occurs.

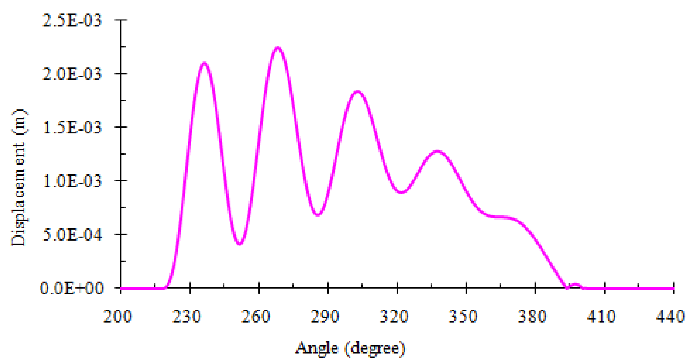

Figure 3: Motion of suction valve

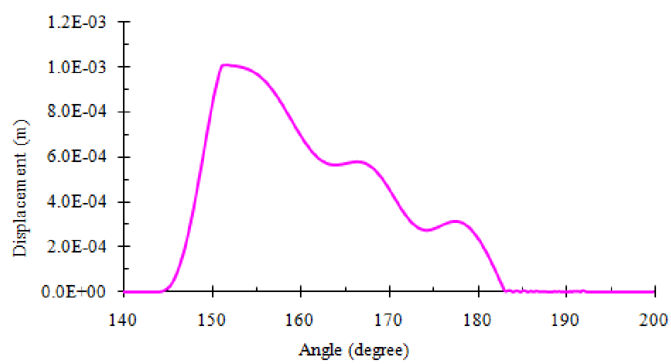

Figure 4: Motion of discharge valve

Figure 4 shows the motion of discharge valve. it is different from the suction valve motion. There is a limiting stopper in the discharge valve, and discharge period is shorter than suction period. When valve closed ,the critical angle is more than 180 degree, maybe backflow occurs. Backflow in discharge valve is more critical, because gas density increased a lot after being compressed, a little backflow can influence cooling capacity evidently.

3.3 Pressure inside cylinder and indicated power

Figure 5 shows the variable pressure inside cylinder. The horizontal axis represents the volume inside the cylinder. The vertical axis represents the pressure inside the cylinder. As piston moves to the top dead center, the pressure inside cylinder increases. When the pressure inside cylinder is larger than pressure in discharge line, the discharge valve is opening. When piston goes back, pressure inside cylinder decreases, discharge valve will close, then suction valve opens. The pressure-volume diagram can be used to calculate indicated power and COP. Integrating the product of pressure and cylinder cross section area, we can get the indicated power is 56.51W.

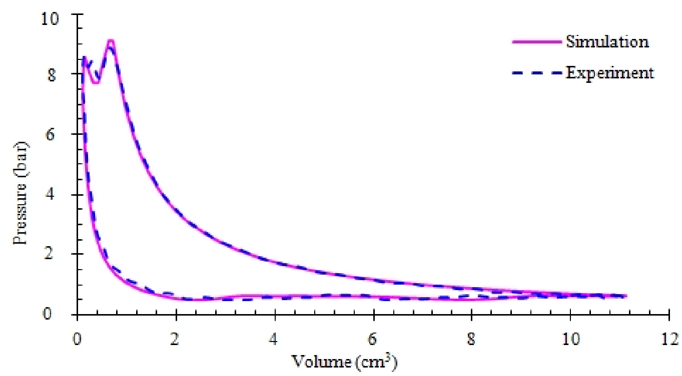

Figure 5: Pressure inside cylinder

3.4 Pressure pulsation in discharge line

Figure 6 shows the pressure pulsation in the discharge tube outside the compressor. The pressure pulsation is usually much smaller than that in the cylinder head. When the high pressure gas goes through the long discharge tube inside the compressor, the pressure pulsation decreased obviously. The gas pressure pulsation is important to refrigerator, if it is too large, it may cause the condenser pipe vibration and noise.

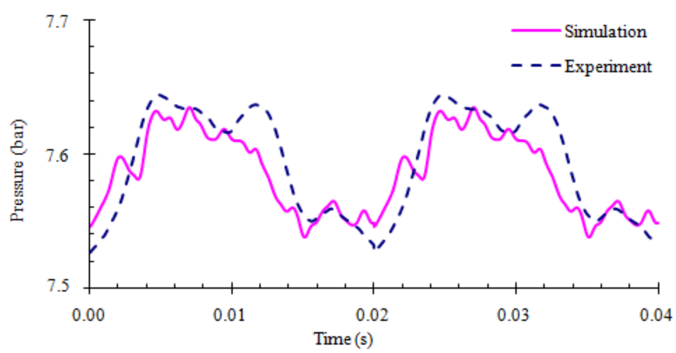

Figure 6: Pressure pulsation in discharge line

3.5 Mass flow rate and cooling capacity

Figure 7 shows the mass flow rate through the suction port. It shows that the flow is not stable. Figure 8 shows the mass flow rate through the exhaust port. By calculation, we can get the average mass flow rate, the average mass flow rate of suction port and that of the exhaust port are the same, they are 0.448g/s. The enthalpy difference is 336.984kJ/kg for R600a refrigerant in ASHRAE condition (evaporating temperature -23.3℃, condensing temperature 54.4℃, suction temperature 32.2℃, supercooling temperature 32.2℃, ambient temperature 32.2℃), so the cooling capacity is 151.12W. Cooling capacity is the measure of a cooling system's ability to remove heat, here it means the heat that the cooling system can remove at the given mass flow rate in the ASHRAE condition, it equals the product of mass flow rate and enthalpy difference. We have calculated the indicated power before, it is 56.51W, then the indicated efficiency(the ratio of the indicated power of adiabatic compression cycle to the indicated power of actual compression cycle) is 92.2%.

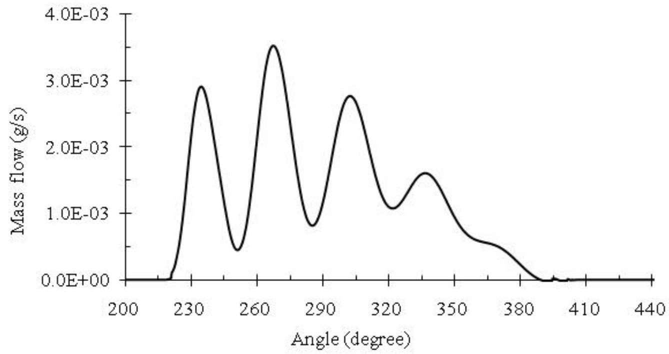

Figure 7: Mass flow rate through suction port

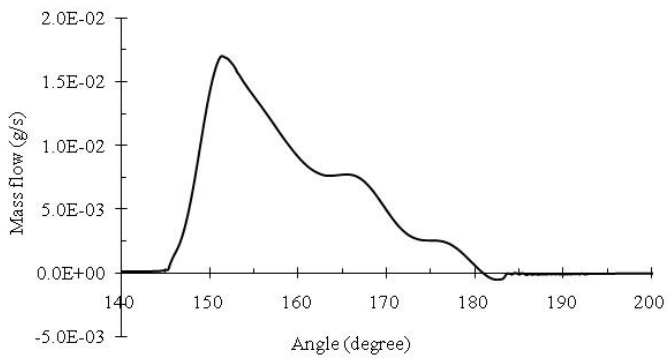

Figure 8: Mass flow rate through exhaust port

3.6 Gas velocity in suction and exhaust ports

Figure 9 shows the gas velocity in suction port, the maximum speed can reach 45m/s. Figure 10 shows the gas velocity in exhaust port, the maximum speed can reach 70m/s. For the density of exhaust system is more higher than suction system, and the maximum speed is higher, the pressure pulsation of exhaust is higher than suction side.

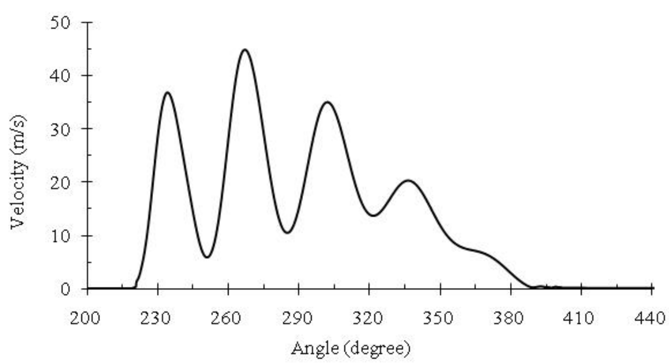

Figure 9: Gas velocity through suction port

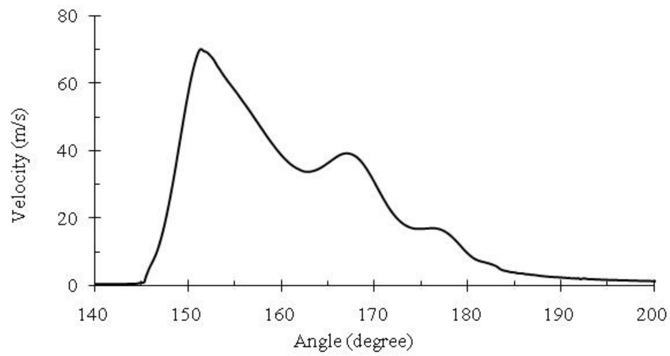

Figure 10: Gas velocity through exhaust port

3.7 Velocity of the valve leaves

Figure 11 shows the velocity of suction valve leaf, the maximum speed is 2.0m/s, and the speed that suction valve leaf impacts the valve plate is about 0.5m/s. Figure 12 shows the velocity of exhaust valve leaf, the maximum speed reaches 4m/s, the speed that exhaust valve leaf impacts valve plate is nearly 1.5m/s. Usually, the impacting speed is critical for valve leaf reliability, in valve design, usually it can't exceed 2~3m/s in limit working condition.

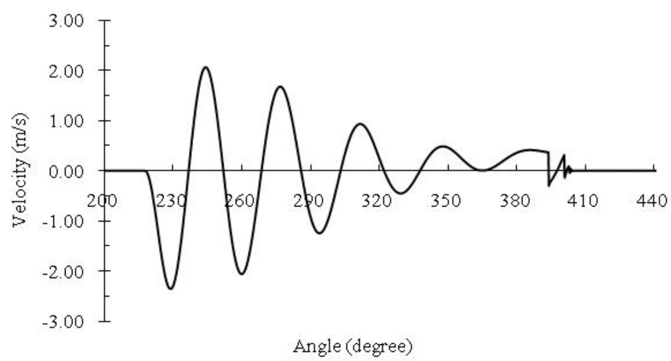

Figure 11: Velocity of suction valve leaf

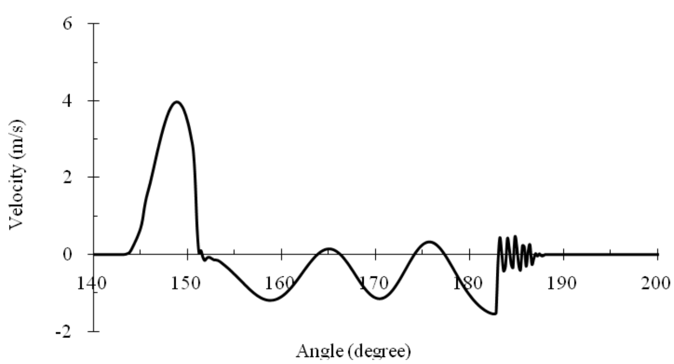

Figure 12: Velocity of exhaust valve leaf

4.EXPERIMENTAL VALIDATION

To validate the simulation, experiments are studied. Pressure inside cylinder, the discharge pressure pulsation, and cooling capacity are tested. Figure 13 shows the pressure inside cylinder test bench. A small hole is drilled at the top of cylinder, pressure sensor is installed in the hole. Rotary encoder is used to test the crank angle, which is installed on the top of shaft.

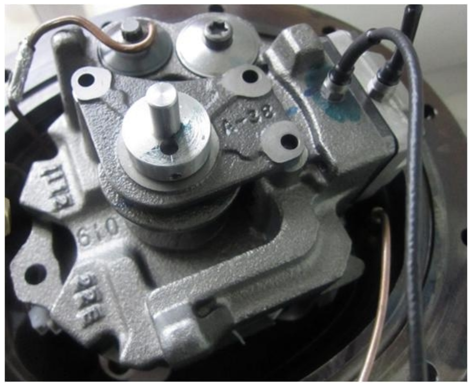

Figure 13: Pressure inside cylinder test system

Figure 5 and Figure 6 show the comparison between the simulation results and test data, the simulation data is close to test data. The simulation and test methods are used to optimize the refrigerator compressor suction and discharge systems.

5.CONCLUSIONS

Fluid structure interaction analysis gives a good way to picture the inside of refrigerator compressor. Pressure inside cylinder, valve motion, pressure pulsation, mass flow rate, indicated power, cooling capacity etc are obtained. The simulation results are compared with test data. The FSI method is very useful for compressor suction and discharge system optimization.

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