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The Modification of the Valves of a Refrigeration Piston Compressor due to the Refrigerant Change

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Abstract

Compressor and refrigerant play key roles in a refrigeration system. Owing to the demands of environment protection, the CFCs and HCFCs refrigerants will be phased out in a short time, and new refrigerants such as R600, R134a and R407C will be used. As the new refrigerant is selected, the motion regularity of the valves will be changed. The motion regularity of the ring valves of a 2-stage refrigerating piston compressor is simulated, and the calculations are done respectively with refrigerants R717 (ammonia), R22 and R134a. The results show that refrigerants and working conditions of a compressor have a direct effect on the motion of the ring valves. Comparing the original design parameters of the ring valves with the modified ones, the former are effective for the HCFC refrigerants, the latter have perfect motion of the valves for the new refrigerants, and the refrigeration system performance parameters, such as input power, the refrigerating quantity and the COP value, are also better. Service life test of the compressor system using the ring valves has been done for R717. The compressor using the modified ring valves has a life exceeding 8000 hours. Hence, the valve design should consider the influences of the refrigerant.

Key words: ring valve, compressor, refrigerant, design, simulation

1. Introduction

As people found CFCs (chlorofluorocarbons) and HCFCs (hydro chlorofluorocarbons) refrigerants have a destructive effect on the ozone layer (Molina et al., 1974) and a greenhouse effect, scientists began to seek new refrigerants. At present, two main groups of refrigerants are potential replacement candidates for the CFCs and the HCFCs, namely, the HFCs (hydro fluorocarbons) and the natural refrigerants. In response to the Montreal Protocol and consequent regulations (1987 and 1995), the air-conditioning and refrigerating industries are now in the process of evaluating, introducing and adapting the new refrigerants.

In the process of replacing refrigerants, the performances of the old refrigeration system will be changed. A key reason is that change of refrigerant in a compressor causes change of the motion regularity of refrigerating compressor valves. The old valves cannot meet the requirements of the new refrigerant.

The valve is a key component of a refrigerating piston compressor. Its performance has a great effect on the behavior of the refrigeration system. Hence, the motion regularity of the ring valves of a 2-stage refrigerating piston compressor is simulated in this paper, and the calculations are done respectively with refrigerants R717, R22 and R134a.

2. System Simulation

2.1 Model assumptions

The main purpose of the simulation is to find the relations between ring valves of the refrigerating piston compressor and the refrigerants; the main points of the simulation model are to simulate the working process of compressor and the motion regularity of the ring

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valves. So, we can use the following assumptions:

1. All cycles were operated in steady-state condition. The steady state is correct for the performance of the compressor in the refrigeration system.
2. There was no heat loss from the compressor surfaces to its surroundings.
3. The refrigerants are in the same condition on any two points of the control volume, which means the conditions of all refrigerants in the compressor cylinder are uniform.
4. The mechanical efficiency of a compressor was fixed at 0.85.
5. The condensation and evaporation temperatures were fixed at 35°C and -40°C ,respectively.
6. The suction temperature of the first stage compressor was fixed at -20°C.

2.2 Development of the model

To find the relations between the ring valves of the refrigerating piston compressor and the refrigerants, some basic equations are needed in the simulation. These equations are: the energy conservation equation, the mass conservation equation, the kinematic equation of the valve plate, and the volume change equation. All of them are discussed in the following.

2.2.1 Energy conservation equation

The compressor is working in unsteady conditions. Selecting the cylinder of the piston compressor as the control volume, using the First Law of Thermodynamics, we can obtain the equation for the inner energy of gas in the control volume in a minute time interval:

$$d(m\,u) = dQ + dH_i + dH_o + dW \qquad (1)$$

where $d(mu)$ is the change of the inner energy of the gas in the control volume, dQ is the heat absorbed by the gas in the control volume, dH_i is the introduced enthalpy of the gas entering the control volume, dH_o is the carry-over enthalpy of the discharge gas, and dW is the power working on the gas.

Equation (1) can also be written as follows:

$$\begin{aligned} d(m\,u) = dQ + \sum dm_i\ h_i\ + \\ + \sum dm_o\ h_o + dW \end{aligned} \qquad (2)$$

where dm_i is the mass of gas entering the control volume, h_i is the enthalpy of gas entering the control volume, dm_o is the mass of gas discharged from the control volume, and h_o is the enthalpy of thegas discharged from the control volume.

The rotational speed of the compressor is constant. Hence, the angle of rotation can be written $\theta = \omega t$ .The relationship of $d\theta$ and dt is linear. Hence, equation (2) can be written as follows:

$$\begin{aligned} \frac{d(mu)}{d\theta} = \frac{dQ}{d\theta} + \sum\left(\frac{dm_i}{d\theta}\cdot h_i\right) + \\ + \sum\left(\frac{dm_o}{d\theta}\cdot h_o\right) + \frac{dW}{d\theta} \end{aligned} \qquad (3)$$

Because h can be considered as the function of temperature T and specific volume v, we can write the total derivative of $h = h(T, v)$ as follows:

$$\frac{dh}{d\theta} = \left(\frac{\partial h}{\partial v}\right)_T \frac{dv}{d\theta} + \left(\frac{\partial h}{\partial T}\right)_v \frac{dT}{d\theta} \qquad (4)$$

Considering the thermodynamic equations $u = h - pV$ and $v = \frac{V_c}{m}$, Eq. (5) can be obtained:

$$\frac{d(mu)}{d\theta} = \frac{d[m(h - pV_c/m)]}{d\theta} = \frac{d(mh) - d(pV_c)}{d\theta} \qquad (5)$$

where V_c is the instantaneous volume of the control volume, p is the gas pressure, and m is the gas mass in the control volume.

The flow of gas into the compressor can be considered instantly as steady-flow, so the equation $h = h_o$ can be obtained.

Considering equations (4) and (5) and the thermodynamics' equation $dW = -pdV_c$, equation (3) can be written as follows:

$$\begin{aligned} \frac{dp}{d\theta} = \frac{1}{v}\left[\left(\frac{\partial h}{\partial v}\right)_T \cdot \frac{dv}{d\theta} + \left(\frac{\partial h}{\partial T}\right)_v \cdot \frac{dT}{d\theta}\right] - \\ - \frac{1}{V_c}\left\{\sum\left[\frac{dm_i}{d\theta}(h_i - h)\right] + \frac{dQ}{d\theta}\right\} \end{aligned} \qquad (6)$$

Using equations

$$\frac{dp}{d\theta} = \left(\frac{\partial p}{\partial v}\right)_T \frac{dv}{d\theta} + \left(\frac{\partial p}{\partial T}\right)_v \frac{dT}{d\theta}$$

and

$$\frac{dv}{d\theta} = \frac{d(V_c/m)}{d\theta} = \frac{1}{m}\cdot\frac{dV_c}{d\theta} - \frac{V_c}{m^2}\cdot\frac{dm}{d\theta}$$

,

equation (6) can be written as following:

$$\frac{dp}{d\theta}=\frac{\frac{1}{v}\left[\left(\frac{\partial h}{\partial v}\right)_T-\frac{\left(\partial h/\partial T\right)_v\cdot\left(\partial p/\partial v\right)_T}{\left(\partial p/\partial T\right)_v}\right]\frac{dv}{d\theta}}{1-\frac{1}{v}\cdot\frac{\left(\partial h/\partial T\right)_v}{\left(\partial p/\partial T\right)_v}}-\frac{\frac{1}{V_c}\left\{\sum\left[\frac{dm_i}{d\theta}\left(h_i-h\right)\right]+\frac{dQ}{d\theta}\right\}}{1-\frac{1}{v}\cdot\frac{\left(\partial h/\partial T\right)_v}{\left(\partial p/\partial T\right)_v}} \tag{7}$$

Equation (7) is the equation for the rate of pressure change in the control volume. It is an important equation used in the simulation.

To solve equation (7), we should know some other information as follows:

1. The real gas state equation $f(p,v,T)=0$. From this equation, we can get the partial derivative of pressure p, the enthalpy h of gas, and the partial derivative of enthalpy h, which are in equation (7).
2. The gas mass m and the change rate of mass $dm/d\theta$ in the control volume. These parameters can be obtained using equations (10) and (11), which are given below.
3. The working volume of compressor V_c and the rate of its change $dV_c/d\theta$. These parameters can be calculated on the base of the geometrical relationship of the piston compressor.

2.2.2 Mass conservation equation

From the Law of Conservation of Mass, we get the following equation:

$$dm = dm_i - dm_o \tag{8}$$

where dm is the change of mass in the control volume.

If the leakage of the compressor is neglected, the mass change just occurs in the suction and discharge processes. In the four processes of the refrigeration piston compressor, i.e. expansion, suction, compression and discharge, the mass change in the control volume can be written as:
expansion and compression: $dm = 0$,
suction: $dm = dm_i$,
discharge: $dm = -dm_o$.

The operating principles of the suction valve and the discharge valve are identical. Their structures usedin this paper are similar. Hence, the discussions following do not differentiate between the suction valve and the discharge valve in suitable position. *Figure 1* shows the diagrammatic sketch of the valves.

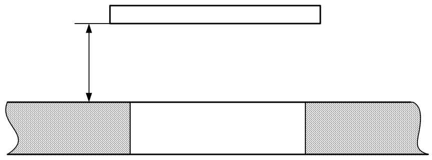

Figure 1. The diagrammatic sketch of the valves

For the suction valve, zone 1 in Fig. 1 connects with the suction pipe and zone 2 connects to the cylinder of the compressor, and the gas from the suction pipe flow to the cylinder through the suction valve. For the discharge valve, zone 1 connects with the cylinder of the compressor and zone 2 connects to the discharge pipe, and the gas flow to the discharge pipe through the discharge valve.

We can consider the flow through the valves (including the suction and discharge valves) is as one dimensional adiabatic steady-flow. The gas flow rate through the gap of the valve is calculated as follows:

$$\begin{aligned}&\text{Suction valve: } w=\sqrt{2(h_2-h_1)}\\&\text{Discharge valve: } w=\sqrt{2(h_1-h_2)}\end{aligned} \tag{9}$$

From equation (9) we can obtain the mass flow rate through the gap of the discharge valve:

$$dm = C_f\cdot A_y\cdot w\cdot\frac{1}{v_2}\cdot dt$$

This equation can also be written:

$$\frac{dm}{d\theta}=\frac{C_f\cdot A_y}{\omega\cdot v_2}w \tag{10}$$

where C_f is the flow coefficient, $C_f = 0.7$ from experiments, ω is the angular velocity of the compressor, v_2 is the specific volume of gas in zone 2 (shown in *Figure 1*), h_1 and h_2 are the enthalpy of the gas in zones 1 and 2 respectively, A_y is the effective through-flow area, which has the linear relationship with the displacement of the valve plate y.

From equations (8) and (10), we can obtain the mass of the gas in the control volume and the mass flow rate through the suction valves and discharge valves.

2.2.3 Kinematic equation of the valve plate

The forces acting on the valve plate are the gas pushing force, the valve spring force, the gravitational force of valve plate and springs, the gas damping force, and so on. But when the valve plate moves, the two main forces acting on it are the gas pushing force F_g and the valve spring force F_s. We will onlyconsider these two main forces in the system simulation.

Figure 2 shows the force diagram of the valves.

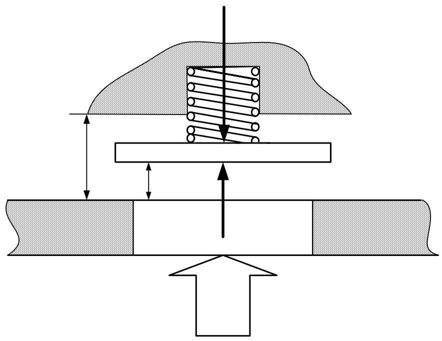

Figure 2. The force diagram of the valves

The motion of the valve plate can be simplified to the single-degree-of-freedom translation. From Newton'sSecond Law, we can obtain the equation of the valve's kinematics:

$$M_v \cdot \frac{\partial^2 y}{\partial t^2} = F_g - F_s \tag{11}$$

where M_v is the equivalent mass of the ring valves plate and y is the displacement of the valve plate.

$$F_g = C_l \cdot \Delta p \cdot A_v \tag{12}$$

where C_l is the lift coefficient of the valve plate, its value was obtained from experiments, in this paper, C_l=0.7; Δp is the pressure difference between two sides of the valve; A_v is the area of the valve plate.

$$F_s = \lambda_{st}(y + y_0) \tag{13}$$

where λ_{st} is the summation of all springs' rigidity action on the valve plate, y_0 is the length decrement of spring when the valves shut down and y is the displacement of the valve plate.

2.2.4 Volume change equation

The state parameters of the gas in the cylinder also have relation to the working volume and its change rate. Hence, we must use the equation about the working volume.

According to the geometrical relationship of the piston compressor, the working volume of the piston compressor cylinder can be shown as:

$$V_c = \frac{\pi D^2 S}{8}\left[1 - \cos\theta + \frac{1}{\lambda}(1 - \sqrt{1 - \lambda^2 \sin^2\theta})\right] + V_0 \tag{14}$$

where D is the diameter of the cylinder of compressor, λ is the ratio of the radius of the crank and the length of the connecting rod, S is the range of the compressor piston movement, and V_0 is the clearance volume of the compressor.

From equation (14), we can obtain the rate of change of the compressor working volume:

$$\frac{dV_c}{d\theta} = \frac{\pi D^2}{8} S\left(\sin\theta + \frac{\lambda \sin\theta}{2\sqrt{1 - \lambda^2 \sin^2\theta}}\right) \tag{15}$$

2.2.5 Gas equation of state and state parameter

In the thermodynamic process of the compressor, the changes of mass and energy are realized by the change of the gas state parameters. Hence, the study of the compressor thermodynamic process is related to the gas equation of state.

Refrigerants are real gas. The equation of state of real gas should be used. In this paper, the state equations for R717, R22 and R134a are shown as follows (Wu et al. 1997):

R717:

$$\frac{pv}{RT} = 1 + \sum_{i=1}\sum_{j=0}\left(b_{ij}\frac{1}{v^i T_r^j}\right)$$
$$C_v^0 = \sum_{i=0}^{4} C_i T^i \tag{16}$$

R22:

$$\frac{p}{J} = \frac{RT}{v - b_0} + \sum_{i=2}^{5}\left[\frac{A_i + B_i T + C_i e^{-KT/T_c}}{(v - b_0)^i}\right] + \frac{A_6 + B_6 T + C_6 e^{-KT/T_c}}{e^{\alpha v}(1 + c' e^{\alpha v})}$$
$$C_v^0 = a + bT + cT^2 + dT^3 + f/T^2 \tag{17}$$

R134a:

$$p = \frac{RT}{(v - b)} + \frac{A_2 + B_2 T + C_2 e^{-KT_r}}{(v - b)^2} + \frac{A_3 + B_3 T + C_3 e^{-KT_r}}{(v - b)^3} + \frac{A_4}{(v - b)^4} + \frac{A_5 + B_5 T + C_5 e^{-KT_r}}{(v - b)^5}$$
$$C_p^0 = c_1 + c_2 T + c_3 T^2 + c_4 T^3 + c_5 T^{-1} \tag{18}$$

In equations (16)-(18), all parameters' values can be obtained from Wu et al. (1997).

Using thermodynamics differential equation, we can obtain the enthalpy h of gas from equations (16)-(18). The local derivatives of pressure p and enthalpy h can also be obtained.

2.3. System Simulation

2.3.1 Basic parameters

In the system simulation, there are some basic parameters needed, which include compressor parameters and valve parameters. The piston compressor basic parameters used in the simulation are as follows:

The diametric of the cylinder of compressor D = 100 mm;

The range of the compressor piston movement S = 100 mm;

The ratio of the radius of the crank and the length of the connecting rod λ = 0.2273;

The clearance volume of compressor V_0 = 251.3 mm^3;

The rotational speed of compressor n = 1440 rpm.

The structure of the suction valve and the discharge valve are similar. Their original design parameters are shown in TABLE I:

TABLE I. THE ORIGINAL PARAMETERS OF THE VALVES

Parameters	Suction valve	Discharge valve
Lift range (mm)	2.2	1.7
Length decrement of spring when the valves shut down y_0 (mm)	4.3	6.8
Equivalent mass of the ring valves plate M_v (kg)	0.034	0.025
Area of the valve plate A_V(mm^2)	55.13	40.06
Number of springs action on valve plate	6	8
Single spring rigidity (N/m)	1410	1410

From TABLE I we can calculate the summation of all springs rigidity action on the suction valve plate λ_{st} as 8460 N/m, and for the discharge valve plate λ_{st} as 11280 N/m.

2.3.2. Solving process of the model

Considering the compression of the gas as adiabatic process (Shu and Tramschek, 1984), the quantity of heat transfer through the wall of cylinder is zero and equation $dQ/d\theta = 0$ can be deduced.

Substituting equations (14)–(18) for equations (7) and (10), and considering kinematic equation of the valve plates equation (11), a system of three differential equations is obtained on the base of equations (7), (10) and (11). Solving this system of three differential equations , we can obtain the state parameters of gas and the state of valves at any instance. From these parameters we can calculate the macroeconomic parameters of a system such as volumetric efficiency, indicated work, refrigerating COP, and so on.

When the refrigerant of the refrigeration system is changed, the performances of the system and the refrigeration piston compressor decreases. The main problem is that the motion regularity of the ring valves worsens, and the service life of the valves reduces.

If we do not want to change the main structure parameters of the ring valves and the piston compressor, there are two effective methods to modify the ring valves. One is to change the spring rigidity of the ring valves. The other is to change the lift range of the valve plate parallel with the first method.

The motion regularity of the ring valves is mainly determined by the kinematic equation of the valve plates, equation (11). The gas pushing force F_g and the valve spring force F_s are the main influencing factors of the motion regularity. Without changing the main structure of the ring valves and compressor, changing the valve spring rigidity and the lift range can improve the stress state, and the motion regularity of the valve plates.

To simplify the manufacturing technique of the compressor and the valves, all the valves (including the suction and discharge valves) have the same valve spring. The numbers of the springs are not the same but the single spring rigidity is the same for the suction and discharge valves. Hence, we change the single spring rigidity at the same time for all valves but do not change the number of springs for the suction and discharge valves.

According to the results of the simulation, the ring valves have very bad motion regularity when the refrigerant is R717 and relative good for the other two refrigerants, i.e. R22 and R134a. Hence, the modification of the valves is implemented just for R717.

4. Results and Discussion

The motion regularity of the ring valves of a 2-stage refrigerating piston compressor is simulated, and the calculations are done respectively with the refrigerants R717, R22 and R134a. The motion regularities of the ring valves are different for different refrigerants.

4.1. Original design of the ring valves

Figures 3 and *4* show the motion regularity of the ring valves with the original design parameters. In the figures, the first stage bulges of the curves (about a rotation angle 0-180°) are the motion regularities of the suction valves and the second bulges (about a rotation angle 180-360°) are for the discharge valves.

From *Figures 3* and *4*, we can see that the ring valves have excellent motion regularity when the refrigerant is R22 or R134a. But when R717 is the refrigerant, the ring valves have very bad motion regularity. The valve plates frequently impact the lift limiters and the valve seating. The impact frequency is very high for suction valves of the first stage compressor. These deeply affect the reliability of the ring valves and the refrigeration system.

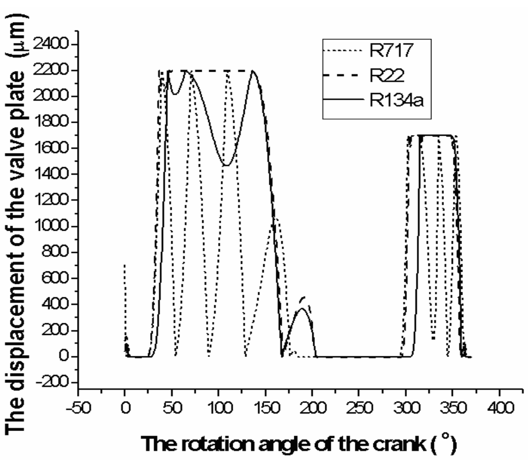

Figure 3. Valve motion regularityof the first stage compressor for three refrigerants

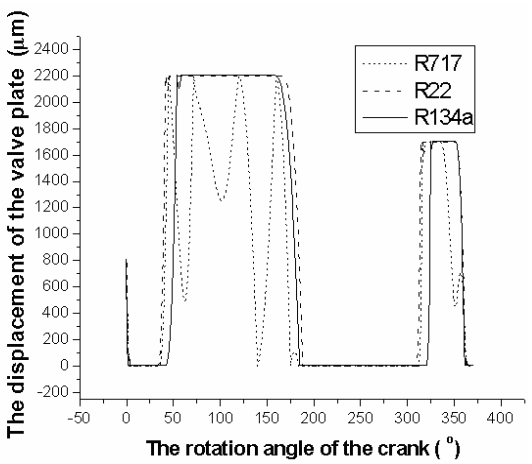

Figure 4. Valve motion regularity of the second stage compressor for three refrigerants

TABLE II shows the performance parameters of the compressor and the refrigeration system. The volumetric efficiency in TABLE II is defined as:

$$\eta_V = \frac{V}{V_t} \tag{19}$$

where V is the real delivery of compressor, and V_t is the theoretical delivery of compressor.

It can be seen that the COPs of the refrigeration system are different using refrigerants R717, R22 and R134a, respectively. The volumetric efficiency is low when the refrigerant is R717, because the motion of the suction valve is bad. Also we can see the refrigerating outputs are different. The main influencing factor of the refrigerating output is the refrigerating output per cubic meter of the refrigerant. Because the refrigerating output per cubic meter of R134a is much lower than that of the others, the refrigerating output of the refrigeration system using R134a is less than the others.

TABLE II. PERFORMANCE PARAMETERS OF THE COMPRESSOR AND THE REFRIGERATION SYSTEM FOR THREE REFRIGERANTS

Refrigerant	R717	R22	R134a
Loss of suction and discharge processes (kW)	4.37	3.57	3.35
Volumetric efficiency (%)	88.04	93.47	91.26
Indicated power of compressors (kW)	28.93	31.49	16.06
Refrigerating output (kW)	63.36	72.64	36.89
Input power (kW)	34.06	37.05	18.89
COP of the refrigeration system	1.86	1.96	1.95

According to the results of the simulation using the original design parameters, the ring valves have very bad motion regularity when the refrigerant is R717 and relatively good for the other two refrigerants, i.e. R22 and R134a. Hence, the modification of the valves is implemented just for R717 and the main method is reducing the valves spring forces.

4.2 Change the spring rigidity

The ring valves have very bad motion regularity when the refrigerant is R717. This is because the gas pushing forces and the valve spring forces are not matched. The valve spring forces are too large for the gas pushing force when the refrigerant is R717. Hence, the most effective method is to reduce the value of the valve spring force.

Figures 5 and *6* show the motion regularities of the ring valves when the single spring rigidity is reduced from 1410 N/m (original design parameter) to 400 N/m. From *Figures 5* and *6*, it can be seen that the motion regularity of the ring valves is improved when reducing the spring rigidity. The impact times of the valve plate decreases, which is very important to increase the reliability of the compressor and the refrigeration system.

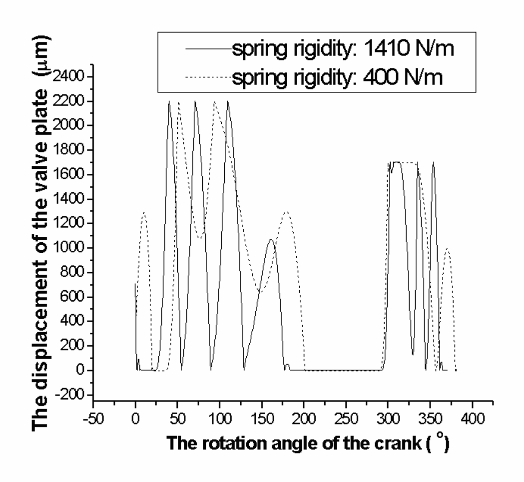

Figure 5. Valve motion regularity of the first stage compressor for R717

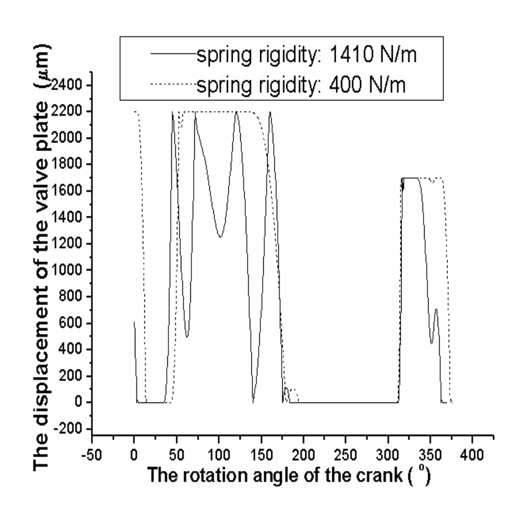

Figure 6. Valve motion regularity of the second stage compresso for R717

Figure 7 shows the parameters of the refrigerant R717 in the control volume. We can realize that in the suction and discharge stages, i.e. the two-terminal of the curves, the change of the frequency and amplitude is big. When the spring rigidity is decreased to 400 N/m, this situation is improved. The fluctuation of the parameters, such as pressure, temperature and mass, is caused by the bad motion regularities of the valve plates. The suitable spring force can improve the motion regularities and avoid the fluctuation of the gas parameters.

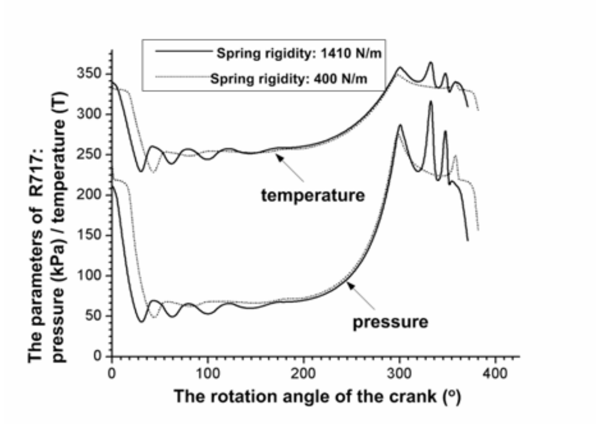

Figure 7. Change of the gas parameters in the first stage compressor

TABLE III. PERFORMANCE PARAMETERS OF THE COMPRESSORS AND THE REFRIGERATION SYSTEM FOR DIFFERENT LIFT RANGE WHEN THE SPRING RIGIDITY IS 400 N/M AND THE REFRIGERANT IS R717

Lift range of valve (mm)	2.2, 1.7*	2.0, 1.5	1.8, 1.3
Loss of suction and discharge processes (kW)	2.71	2.73	2.97
Volumetric efficiency (%)	86.38	86.65	88.70
Indicated power of compressors (kW)	25.02	25.34	26.22
Refrigerating output (kW)	62.17	62.38	63.85
Input power (kW)	29.44	29.81	30.84
COP of the refrigeration system	2.11	2.09	2.07

*The first number is the lift range of the suction valves; the second number is the lift range of the discharge valves

When the spring rigidity is changed, the performance parameters of the compressors and the refrigeration system are shown on the second column of TABLE III. The refrigeration system maintains high performance when the spring rigidity decreases.

4.3 Changing the lift range

Decreasing the lift range a little parallel to changing the spring rigidity, the motion and reliability of the ring valves can be improved and the economical performances of the compressors and the system fall a little.

As decreasing the lift range, the acceleration of the valve plate does not change under the condition that forces F_g and F_s are not changed. But the time trequired by the valve plate moving from the lift limiter to the valve seating or reversed is reduced. Hence, the speed of the valve plate and the force acting on it are reduced as it impacts the lift limiter or the valve seating. So, the reliability of the compressor and the refrigeration system is improved. *Figures 8* and *9* show the motion regularity of the ring valves with the different lift ranges.

The performance parameters of the compressor and the refrigeration system are changed when changing the lift range. Table 3 shows these performance parameters with different lift ranges. The system performance parameters have not changed largely when decreasing the lift range minutely, but the reliability of the system will be improved.

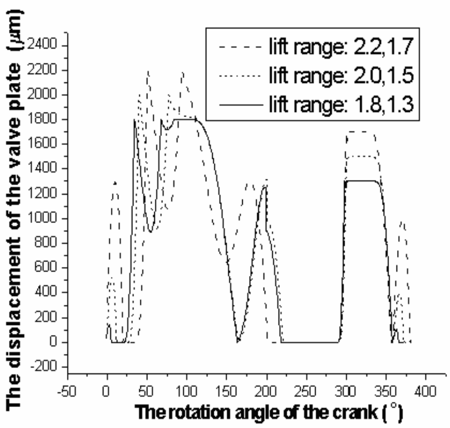

Figure 8. Valve motion regularity of the first stage compressor for different lift range

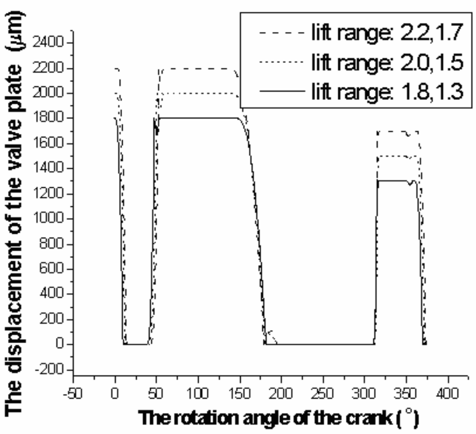

Figure 9. Valve motion regularity of the second stage compressor for different lift range

4.4 Experimental result

We experimented with the compressor systems using the original and modified valves respectively. The main purpose of the experiment was to test the service life because the performance parameters of the compressor and the refrigeration system changed very little for different designs of the ring valves. Hence, the most important performance of the modified valves is service life.

From the service life test of the compressor system using the ring valves and R717, as we have done, it was found that the compressors using the modified valves have longer life than those using original valves. When using the original valves, the compressor can run normally for less than 500 hours because the plates of the valves have disintegrated by frequent impact. But using the modified valves, the can run n can run normally exceeding 8000 hours.

5. Conclusions

Compressors and refrigerants play key roles in refrigeration systems. Owing to the demands of environment protection, the CFCs and HCFCs refrigerants will be phased out in a short time. As a new refrigerant is selected, the motion regularity of the ring valves changes. The motion regularity of the ring valves of a 2-stage refrigerating piston compressor is simulated, and the calculations are done respectively with refrigerants R717 (ammonia), R22 and R134a. Also the performance parameters of the compressor and the refrigeration system for three refrigerants were calculated.

Using the ring valves with original design parameters, the motion of the valves and the performance parameters of the compressor and the refrigeration system are effective when the refrigerant is R22 or R134a, but ineffective with R717. As the motion of the valve plate was bad, the parameters of the compressors and the system descended, i.e. refrigeration COP and the volumetric efficiency.

The gas pushing force was changed and it was not matched with the valve spring force. The valve spring forces are too large for the gas pushing force when the refrigerant is R717. Hence, the impact frequency of the valves plant is high for R717. The most effective method is to reduce the value of the valve spring force.

When using the valves of the modified parameters by changing the spring rigidity or/and the lift range, the valves have perfect motion for the refrigerant R717 and the refrigeration system performance parameters, such as the input power, the refrigerating output and the COP value, are also better. The impact times of the valve plate decreases, which is very important to increase the reliability of the compressor and the refrigeration system, because when changing the spring rigidity, the valve spring force changed and it was matched with the gas pushing force for R717.

When the motion regulation of the valve plant is bad, the parameters of gas R717 in the cylinder fluctuated frequently during suction and discharge. But when the spring rigidity is decreased to 400 N/m, the motion is improved and the fluctuation of the parameters, such as pressure, temperature and mass, is reduced too. The suitable spring force can improve the motion regularities and avoid the fluctuation of the gas parameters, which were important to the refrigeration systems.

The service life test of the compressor system using the ring valves is done for R717. The compressor using the modified ring valves has service life exceeding 8000 hours.

The results show that refrigerants and working conditions of compressors have a direct effect on the motion of the ring valves. Hence, the valve design should consider the influences of the refrigerant.

References

Decisions by the seventh meeting of the parties to the Montreal protocol on substances that deplete the ozone layer, Vienna, 1995.

Molina, M. J., Rowland, F. S., 1974, " Stratospheric sink for chlorofluoromethane: chlorine atom catalyzed destruction of ozone." *Nature*, 249: 810 - 812.

Montreal protocol on substances that deplete the ozone layer, Final Act, United Nations Environment Programme, 1987.

Shu, P. C., Tramschek, A. B.,1984, "Optimization of Compressor Valve Design Using the "Nelder-mead" Simplex Method", *Proceedings of 7th International Compressor Engineering Conference*, USA.

Wu, Yezheng, Han, Baoqi. , 1997, *The principle of refrigeration and its equipment*, Xi'an Jiaotong University Press, Xi'an

Yasuda, H., Touber, S., Machielsen, C. H. M., 1983, "Simulation model of a vapor compression refrigeration system", *ASHRAE Trans*; 98 (2a): 408-425.