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Dynamic characteristics of suction valves for reciprocating compressor with stepless capacity control system

Bin Tang[1], Yuanyang Zhao[1], Liansheng Li[1], Le Wang[1], Guangbin Liu[1], Qichao Yang[1], Haiping Xu[2], Feng Zhu[2] and Wenhui Meng[3]

Abstract

The dynamic characteristics of suction valves for reciprocating compressor with a stepless capacity control system are studied in this article. The self-acting valve model that is applicable for stepless capacity control condition is derived. For the suction valve movement controlled by actuator, a simulation of the hydraulic and mechanical system is conducted. An experimental platform is setup, which is used to test the valve dynamic when it is controlled by actuator. The results from the mathematical model show that the valve impact speeds are influenced by the valve lift, Mach number of the flow in the valve clearance and the initial crank rotation angle of the valve closing process. The simulation results agree well with experimental results. The simulated maximum speed is about 0.53 m/s, and the maximum speed tested by experiment is about 0.58 m/s, both of which are much lower than the speed of the automatic valve (3 m/s). In addition, simulation results show that the maximum speed is at constant when the hydraulic piston stroke increases, which is much different from the automatic valve. It becomes apparent from the aforementioned results that with stepless capacity control system the dynamic characteristics of the suction valve are changed. The valve can be limited at a low speed, and the valve impact speeds to seat and stopper decrease significantly, which would be helpful for extending the life of the valve plate.

Keywords

Reciprocating compressor, stepless capacity control, suction valve, dynamic characteristics

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Introduction

Large reciprocating compressors have been mainly used in the industries of chemical processing, gas transport and storage, refineries, etc. One of the new needs in these industries is better and more efficient load and capacity control methods, which need compressors to meet varied operating demands, flexible requirements and high efficiency goals.[1] The stepless capacity control method was firstly reported in White,[2] and its basic principle is that the suction valve is still kept to open after suction stroke end by an actuator, so that the redundant gas in cylinder flows back to the inlet chamber before the real compression stroke begins. Therefore, the amount of gas compressed can be adjusted by controlling the closing time of the suction valve, and the discharge capacity and power reduction are virtually proportional.[2,3] In recent years, the stepless capacity control system has become a hot topic of research, and key problems are controlling the suction valve motion precisely and making it operate reliably.[4–9] Steinruck et al.[7] developed a stepless capacity control system named HydroCOM and applied it successfully to the industrial processes. A fast-acting solenoid valve, one of key parts in HydroCOM, is applied to control motion of the suction valve. Hong et al.[4] studied the control strategy of stepless capacity regulation, developed a prototype, and tested it in a reciprocating compressor. In the literatures, the research of dynamic characteristics of suction valves with stepless capacity control has been rarely found.

[1]State Key Laboratory of Compressor Technology, Hefei General Machinery Institute, Hefei 230031, China
[2]Shanghai HuanTian Industrial Co., Ltd, Shanghai 200065, China
[3]Shenyang Blower Works Group Co., Ltd, Shenyang 110869, China

Corresponding author:
Bin Tang, State Key Laboratory of Compressor Technology, Hefei General Machinery Institute, Hefei 230031, China.
Email: Tangbinsoddy@126.com

The stepless capacity control method is implemented by actively controlling the suction valve motion as described in various references.[2–9] If the motion of suction valve is abnormal, the efficiency of compressor may be lower. And especially, the unscheduled shutdowns of compressor may increase.[10,11] Furthermore, the stepless capacity control system could change the valve impact speed to the stopper and seat, and influence the fatigue life of the valve plate.[11] Therefore, studies on the dynamic characteristics of suction valves in the stepless capacity control system are essential.

The actuator used to control the motion of suction valve is driven by a hydraulic system. Because the valve moves together with the actuator, the dynamic characteristics of actuator can be taken as that of the valve. This means that we only need to simulate the hydraulic and mechanical system so as to obtain the characteristics of actuator and valve. The simulation can be conducted conveniently by the AMESim platform.[12,13]

In this article, the dynamic characteristics of suction valves were investigated by setting the mathematic model and doing simulation based on the AMESim platform. An experiment was conducted and the experimental results were compared with the simulated ones.

Modeling of suction valves with stepless capacity control system

Working process of suction valves

Figure 1 shows the schematic layout of the stepless capacity control system and the state of closed and open valve. The flow paths are labeled with arrow lines. A 2/3 way solenoid valve controlled by a Programmable Logic Controller (PLC) is used to control the pressure of the hydrocylinder. The actuator moves back and forth under the action of hydraulic pressure. As solenoid valve powers up, the hydraulic cylinder connects with a high pressure oil circuit, the hydraulic pressure drives the actuator to move forth, and the valve in contact with the actuator is forced to open. As the solenoid valve powers down, the hydraulic cylinder connects with a low-pressure oil circuit, the actuator moves back under the force of reset spring, and the valve is closed by valve springs and gas force in the cylinder. With the action of actuator, the closing time of suction valve will be delayed, redundant gas in the cylinder flows out of cylinder through the suction valve, and this is called the back flow process. The *p–V* diagram of compressor operating with stepless capacity control is shown in Figure 2. The indicated power decreases with the decrease of capacity. Therefore, using this method could not only regulate the capacity, but also reduce the power consumption.

When the stepless capacity control system in reciprocating compressors is in use, there are two ways to open the suction valves. One is to open the valve by gas force at first, then launch the actuator which gives the action on the suction valve. In this condition, the valve open process is the same as that of the automatic valve. The other way is that the valve is forced open by actuator before it opens automatically, and in this condition, the valve moves together with the actuator. In the closing process of suction valve, if the speed of valve is lower than that of the actuator, the valve will move alone, otherwise the speed of the valve will be limited by the actuator and they will move together. From the above description, two kinds of movement patterns of the valve can be summarized:

1. *The suction valve moves alone*. In this pattern, the valve still works as a self-acting valve, and its movement is uncontrolled by the actuator. The dynamic characteristics of the valve are studied with a self-acting valve model that is applicable for the stepless capacity control condition.
2. *The suction valve moves together with the actuator*. The valve works like a forced valve and its movement is completely controlled by the actuator, i.e. the valve has the same dynamic characteristics as the actuator, which can be obtained by the simulation of hydraulic and mechanical system based on AMESim.

When the force of actuator is relieved, if the valve speed is higher than the actuator reset speed, the valve and actuator will move together. Otherwise, the valve will move alone as a self-acting valve. So the speed of self-acting valve (valve moving alone) and the reset speed of actuator have to be calculated separately. By comparing these two speeds, the situation of valve movement could be confirmed.

Mathematic model for suction valves moving alone

As mentioned earlier, if the suction valve moves alone, its dynamic characteristics are investigated by using a mathematic model based on the theory of self-acting valve. Following assumptions are used in the derivation of the equations.[14]

1. The suction valve moves alone.
2. The equation of the valve movement is a one-degree of freedom system.
3. Ideal gas equation and one-dimensional gas flow are used without heat transfer.

The suction valve is closed until the end of the clearance volume expansion process of compressor. When the pressure differential of gases between cylinder and valve cavity is high enough to balance out the force of

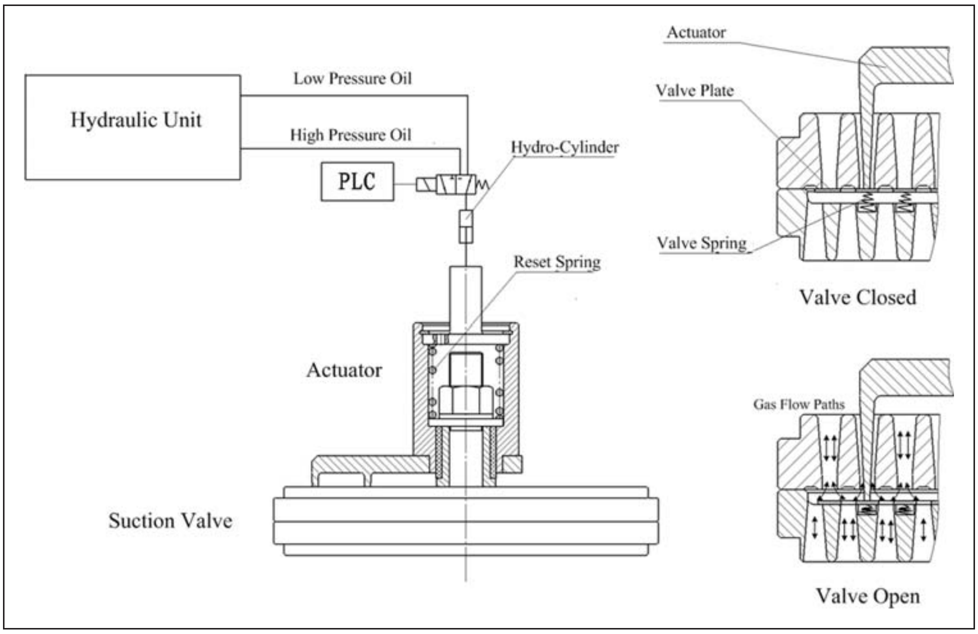

Figure 1. Structure of stepless capacity control system.

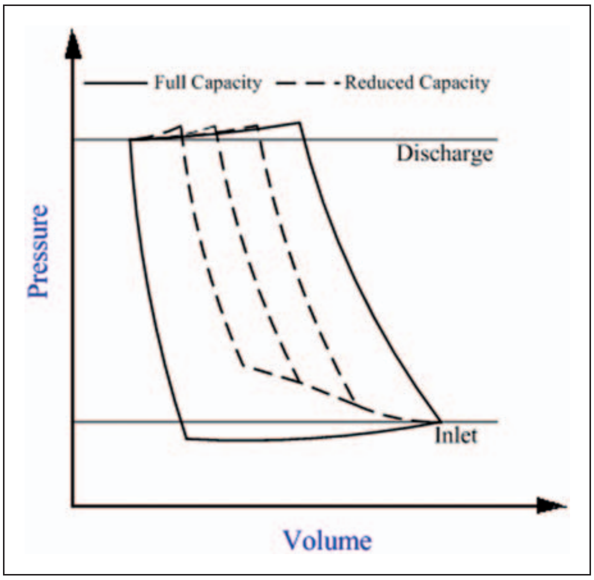

Figure 2. *p–V* diagram of compressor with stepless capacity control system.

valve springs, the valve will be opened. During the opening process, the equations of valve motion and gas flow are as follows[11,14]

$$\frac{\mathrm{d}^2h}{\mathrm{d}\theta^2}=\frac{1}{m\omega^2}\left[\beta p_s(1-\varphi)a_p-ZK(H_0+h)\right] \tag{1}$$

$$\frac{\mathrm{d}\varphi}{\mathrm{d}\theta}=-\frac{1}{\pi}\left(\alpha+\frac{1-\cos\theta}{2}+\frac{\lambda}{4}\sin^2\theta\right)^{-1}\times\left[\begin{array}{l}\pi k\varphi\left(\frac{\sin\theta}{2}+\frac{\lambda}{4}\sin 2\theta\right)\\ -\sqrt{\frac{2k^2}{k-1}\frac{1}{Mv}\frac{h}{H}}\varphi^{\frac{1}{k}}\sqrt{1-\varphi^{\frac{k-1}{k}}}\end{array}\right] \tag{2}$$

The initial and boundary conditions are as follows[14]

$$\left[\left.\begin{array}{r}(h)_{\theta_0}=0\\ (\mathrm{d}h/\mathrm{d}\theta)_{\theta_0}=0\\ (\mathrm{d}^2h/\mathrm{d}\theta^2)_{\theta_0}=0\end{array}\right\},\quad (\varphi)_{\theta_0}=1-ZKH_0/\beta p_s a_p,\right.$$

$$\left.\begin{array}{l}(\mathrm{d}h/\mathrm{d}\theta)_i=-C_R(\mathrm{d}h/\mathrm{d}\theta)_i\\ (h)_i=H\end{array}\right\}.$$

As suction process ends, the suction valve still keeps open due to the action of the actuator. The gas in the cylinder backflows to the suction pipe although compression process has begun. The backflow process is similar to the discharged process. But in backflow, the suction valve works as a discharge valve and opens completely under the action of the actuator. This process is different from the ordinary self-acting valve. The equation of gas flow can be derived from the theory of self-acting valve.[14]

$$\frac{\mathrm{d}\varphi}{\mathrm{d}\theta}=-\frac{1}{\pi}\left(\alpha+\frac{1-\cos\theta}{2}+\frac{\lambda}{4}\sin^2\theta\right)^{-1}\times\left[\pi k\varphi\left(\frac{\sin\theta}{2}+\frac{\lambda}{4}\sin 2\theta\right)+\sqrt{\frac{2k^2}{k-1}\frac{1}{Mv}}\varphi^{\frac{k-1}{k}}\sqrt{\varphi^{\frac{k-1}{k}}-1}\right] \tag{3}$$

The gas backflow process lasts until the action of actuator is removed, which is driven by hydraulic pressure, then the suction valve is closed by gas pressure and valve springs. In the process of valve closing, the equations of valve motion and gas flow can be expressed as follows[11,14]

$$\frac{d^2h}{d\theta^2} = \frac{1}{m\omega^2}\left[\beta p_s(\varphi - 1)\alpha_p + ZK(H_0 + h)\right] \tag{4}$$

$$\begin{aligned}\frac{d\varphi}{d\theta} = &-\frac{1}{\pi}\left(\alpha + \frac{1-\cos\theta}{2} + \frac{\lambda}{4}\sin^2\theta\right)^{-1} \\ &\times\left[\pi k\varphi\left(\frac{\sin\theta}{2} + \frac{\lambda}{4}\sin 2\theta\right)\right. \\ &\left.+\sqrt{\frac{2k^2}{k-1}}\frac{1}{Mv}\frac{h}{H}\varphi^{\frac{k-1}{k}}\sqrt{\varphi^{\frac{k-1}{k}} - 1}\right]\end{aligned} \tag{5}$$

The initial and boundary conditions are as follows[14]

$$\left.\begin{array}{l}(h)_{\theta_0} = H \\ (dh/d\theta)_{\theta_0} = 0 \\ \left(d^2h/d\theta^2\right)_{\theta_0} = \frac{1}{m\omega^2}\left[\beta p_s(\varphi_{\theta_0} - 1)a_p + ZK(H_0 + H)\right]\end{array}\right\},$$

$$\left.\begin{array}{l}(dh/d\theta)_i = -C_R(dh/d\theta)_i \\ (h)_i = 0\end{array}\right\}.$$

When the backflow process stops, the valve begins to close. Thus, the initial condition values θ_0 and φ_0 are the final values of the backflow process which can be calculated by equation (3). This process is also much different from the ordinary self-acting valve.

Equations (1) to (5) are solved by the Runge–Kutta method, and the solution logic is shown in flow charts presented in Figures 3 and 4. The results vary with the change of the Mach number, valve lift and the initial rotation angle of the valve closing process. The Mach number of valve clearance, defined as $Mv = u_a/C_s$, is a characterization of valve clearance flow loss and influenced by piston area, valve opening area, compressor speed, gas properties, etc.[14] The larger the valve opening area is, the smaller the Mach number.

Simulation modeling for suction valves moving together with actuator

It has been pointed out earlier that when the suction valve moves together with the actuator, the dynamic characteristics can be investigated by the simulation of hydraulic and mechanical system based on the AMESim platform.

In the modeling of the hydraulic and mechanical system, the following assumptions are considered[12, 13]

1. No thermal analysis is incorporated in the hydraulic system.
2. The flow resistance of oil in pipes and the friction between movement parts are ignored.
3. Control signals and differential phases between these signals are simulated by function generators that are modeled by using available components in the control libraries of AMESim. The frequency of signals is equal to the operating frequency of compressor.
4. The gas force and the force of valve springs are equivalent to one spring force.
5. Because all the movement parts have same speed, the mass of movement parts can be replaced by one object M.
6. The relief valve is used to control the maximum pressure of the oil circuit. The oil filter, oil cooler and check valves are modeled by using available components in the hydraulic libraries of AMESim.

A two-stage L-type air compressor has been researched in this paper, and its four suction valves are controlled by the stepless capacity control system. The hydraulic and mechanical systems are shown in Figure 1. The schematic layout of the model based on AMESim is shown in Figure 5. For each controlled valves, the mass M is the total mass of movement parts that contains the hydraulic piston, actuator and valve plate. The spring in the hydraulic cylinder acts as a reset spring. The other spring is used to replace the valve springs. The gas force during the suction and backflow process is varying with the crank rotation angle, and it can be calculated from equation (6). By referring to Tang et al.,[5] the changing regularity of the gas pressure can be obtained. During the suction process, the largest gas force is about 25 N, and the hydraulic force is about 2411 N (hydraulic pressure 12 MPa). Hence, the influence of gas force can be ignored when the valve opens. During the valve closing process, the largest gas force is about 98 N. The reset spring and valve springs force can reach to 1100 N. According to equation (6), the gas force at different valve closing time is approximately interpreted by a linear function, which can be defined by a force source in AMEsim force libraries.

$$F_g = \beta(p - p_s)a_p = \beta p_s(\varphi - 1)a_p \tag{6}$$

Experimental research

Figure 6 shows the experimental setup designed to test the performance of the stepless capacity control system and the dynamic characteristics of suction valve, and Figure 7 shows the picture of the actuator. The experimental system includes a two-stage L-type air compressor whose four suction valves are controlled by actuators, hydraulic unit, PLC control systems and data acquisition systems. Each side of cylinders has one suction valve that is controlled by the actuator. The suction valve dynamic of the cover side of the first stage is measured by an eddy current displacement sensor (±0.1% uncertainty), and the pressure of this side is also measured by a pressure sensor (±0.2% uncertainty). The speed of the suction

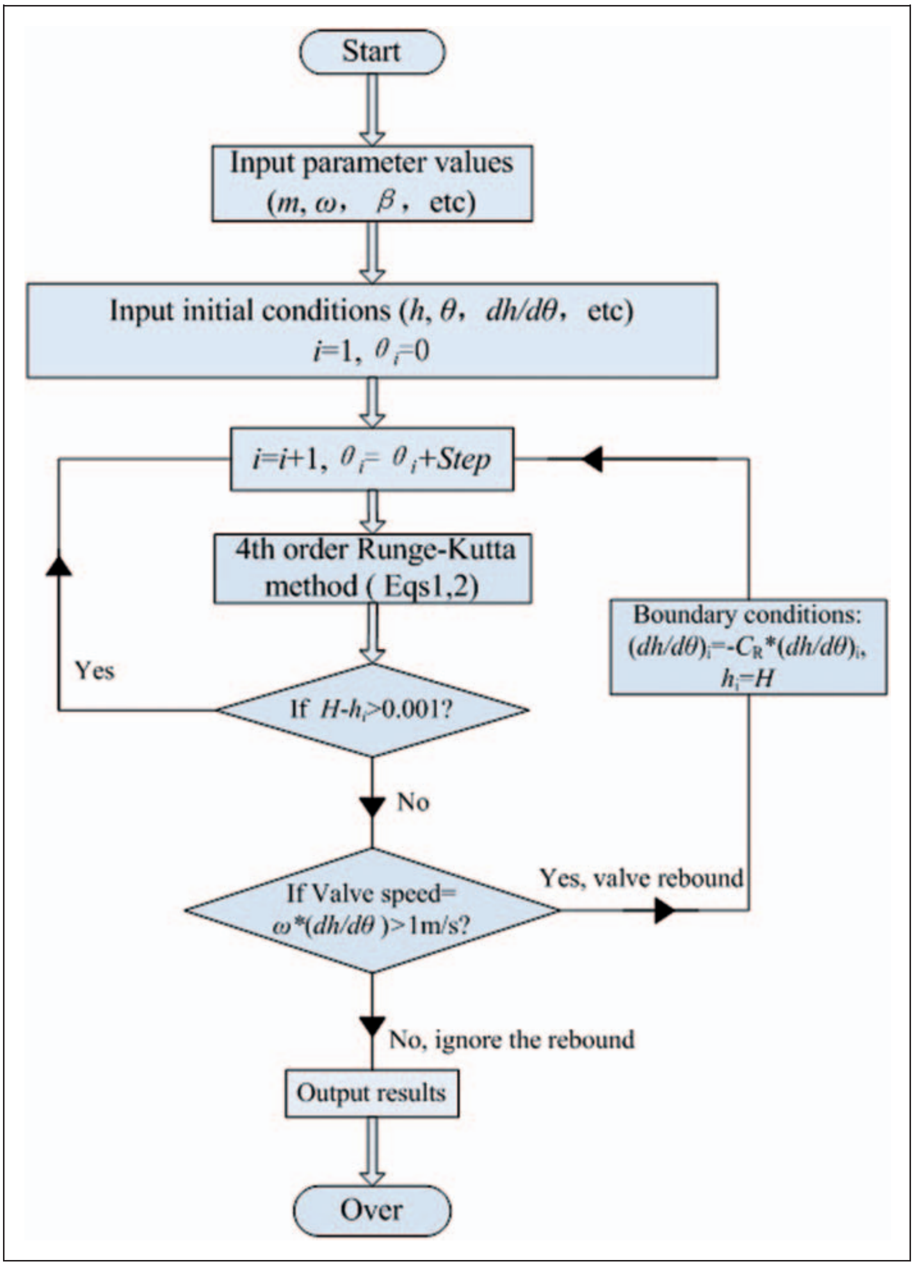

Figure 3. Flow diagrams of solution logic for equations (1) and (2).

valve was calculated by taking a derivative of valve displacement with respect to time, and the absolute accuracy of this process is 0.01 m/s. The results show that the capacity of the compressor can be regulated steplessly, and the energy consumption is reduced proportionally with respect to the reduction of the capacity.

Main specifications of stepless capacity control system are shown in Table 1.

Results and discussion

Results from mathematical model

When $Mv = 0.2$, $H = 1.5$ mm and the initial rotation angle of valve closing process $\theta_0 = 4.5$, the variations of valve speed and displacement with the crank rotation angle during a compression cycle are shown in Figures 8 and 9. From the figures it can be seen that, in the valve opening process, the initial acceleration of the valve is zero and then increases rapidly. But in the closing process, the valve moves with an almost constant acceleration from the beginning of the valve closing. The valve impact speed against the valve seat is even higher than the impact speed to the stopper. It is because that when the suction valve closure is delayed, the valve is closed by gas pressure and valve springs.

The impact speed is calculated with the change of the valve lift, Mach number and initial angle of valve closing process. The results are shown in Figures 10 and 11 and Table 2. Both the impact speeds to the

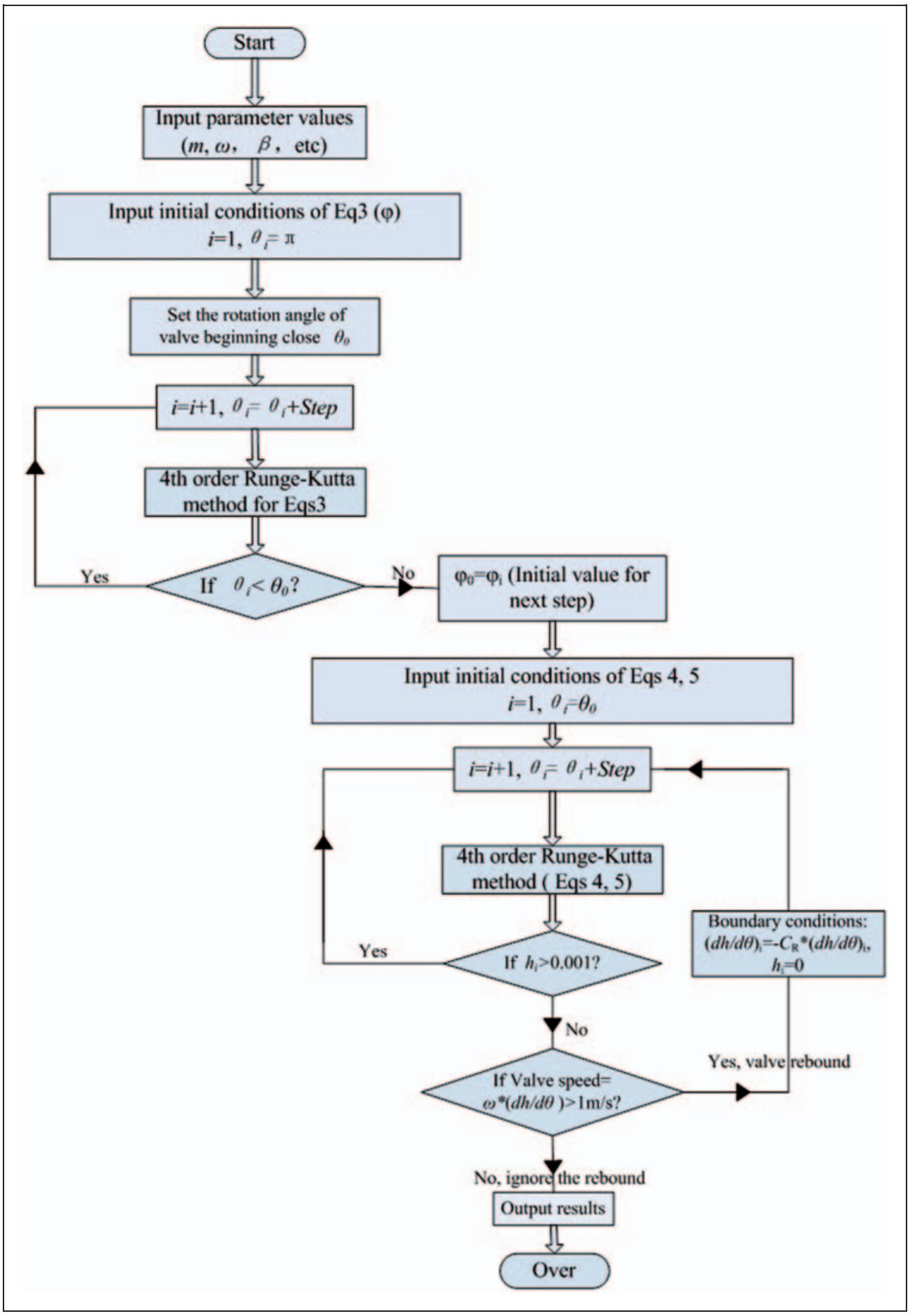

Figure 4. Flow diagrams of solution logic for equations (3) to (5).

stopper and to the seat increase with the valve lifts. From Table 2, it can be seen that the Mach number and the initial rotation angle of the valve closing process have a large effect on the valve impact speed to stopper. Furthermore, it can be concluded that the essential factor affecting the valve impact speed is the initial pressure of the valve closing process. The results in Table 2 are plotted in Figure 11. For a constant valve lift, the valve impact speed to seat increases with the increase of initial pressure φ_0 of the valve closing process, which can be attributed to that the acceleration of valve closing process increases with the initial pressure.

Results from AMESim

The valve speed is studied by changing the length of the hydraulic piston stroke, which also means the valve lift. Ignoring the compressibility of oil and the oil leakage in the hydraulic cylinder, the valve speed V can be described in equation (7), where dq/dt is the oil

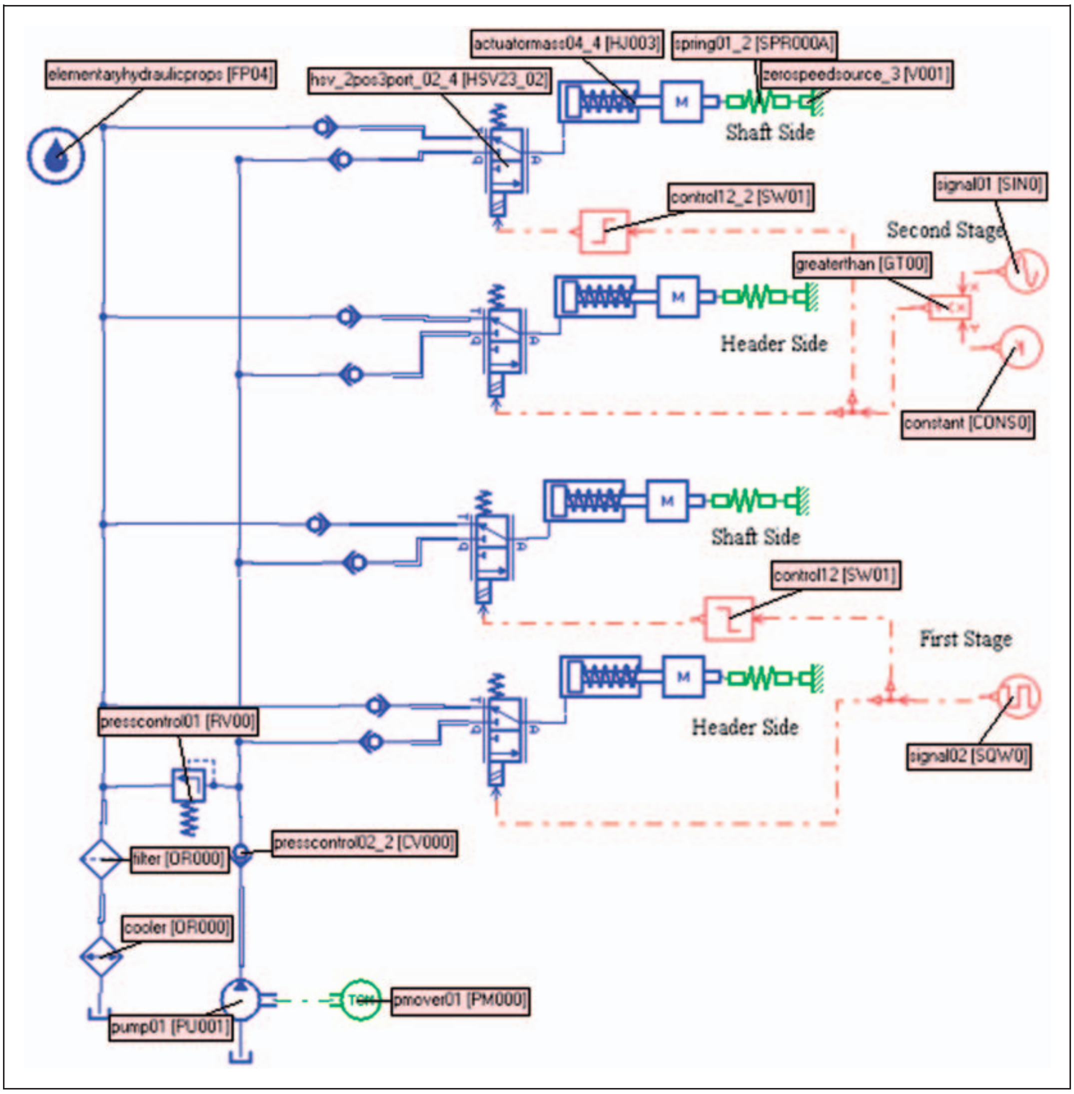

Figure 5. Simulation model of stepless capacity control system based on AMESim.

volume flow rate in the hydraulic cylinder, and A is the piston area

$$V = \frac{\mathrm{d}q}{A\mathrm{d}t} \tag{7}$$

The hydraulic piston stroke is the distance piston could move. The simulation results for different hydraulic piston stroke length are shown in Figures 12 and 13. It is obvious that the peak speed during the valve opening is lower than 0.58 m/s no matter how long the stroke is. It is likely that, when the diameter of the hydraulic piston is constant, the piston speed is determined by the oil volume flow rate of the cylinder that is approximately constant as the hydraulic system reached a steady state. But the reset and valve springs force changes as the actuator displacement changed. Thus, the hydraulic system cannot reach a complete steady state, which brings on a speed decrease after the peak speed when the stroke is longer than 3 mm. During the valve closing process, the initial acceleration increases with the piston stroke length increase. It is because that the reset spring forces are different at the beginning of this process.

The valve speeds from the self-acting valve model and the actuator controlled valve model are compared in Figure 14. For actuator controlled valve, the valve is opened by the actuator before it opens automatically. It is obviously that the self-acting valve speed is much higher than that of the actuator controlled valve.

Results from experiment and comparison

Figure 15 shows the displacement of the valve and the pressure in the compressor cylinder which are measured by sensors. It can be seen that the gas in the cylinder begins to be compressed and the pressure increases rapidly after the suction valve is closed. Figure 16 shows the experimental and simulation results of the actuator controlled valve speed (valve

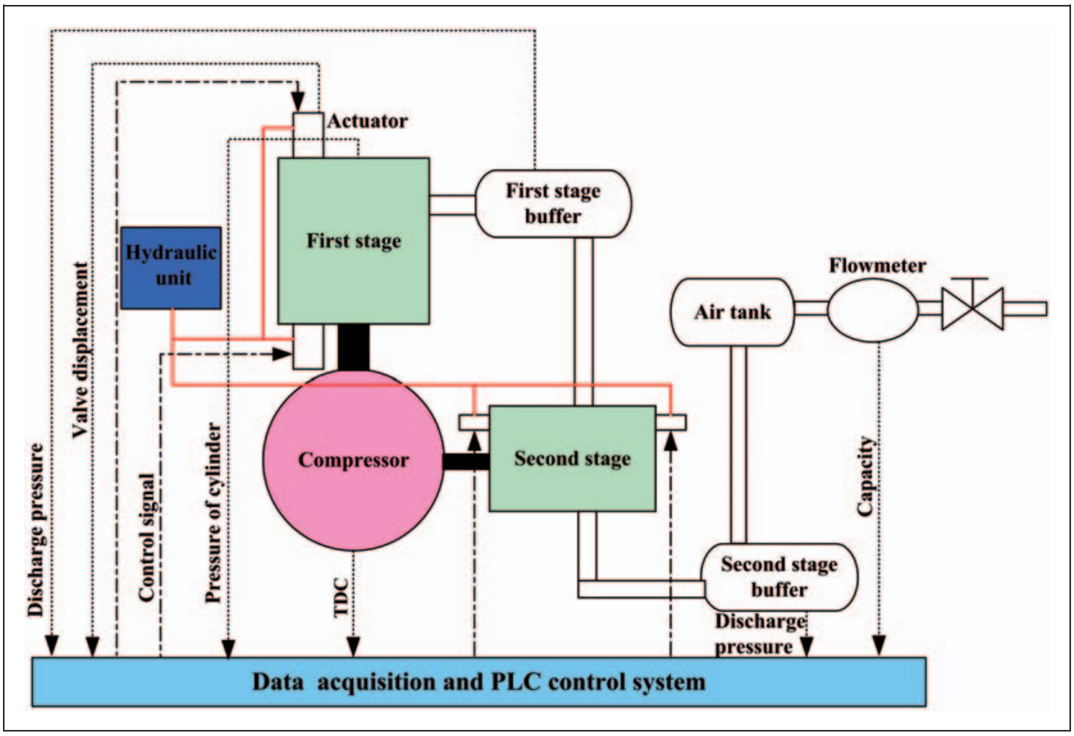

Figure 6. Schematics of experiment setup.

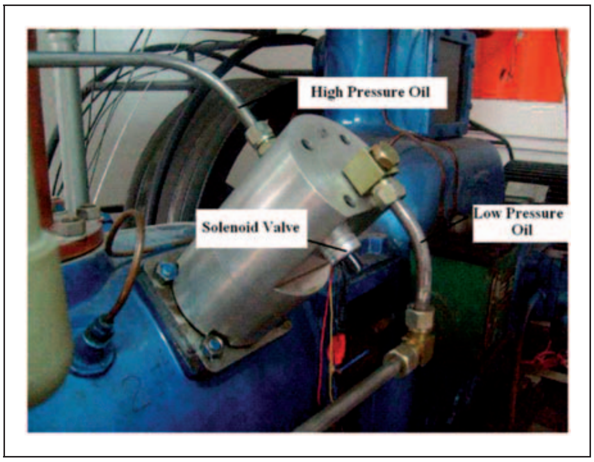

Figure 7. Picture of actuator.

lift $H = 1.5$ mm). The experimental peak speed during the suction valve opening process is almost 0.58 m/s, but during the closing process the peak speed is only 0.39 m/s. It is partially attributed to that the return spring force is a little too low in the experimental setup. Comparing the simulation results with the experimental results, the difference between the simulation and experiment is small (peek speeds differ by 0.06 m/s), which can be attributed to the assumptions made in the simulation model.

Comparing the results shown in Figures 14 and 16, it can be seen that the self-acting valve speed (higher than 2.5 m/s) is much higher than that with the actuator controlled (about 0.58 m/s). It could be concluded that during the suction valve closing process, due to the lower speed of the actuator, the valve speed will be limited by the actuator and they will move together. During the suction valve opening process, if it is opened by actuator, its speed will be much lower than the speed when the valve opens automatically. As shown in Figures 10 and 11, valve impact speeds increase rapidly with the increase of the valve lift. But Figure 13 shows that the peak speed is almost constant with the increase of the piston stroke. If the valve movement is controlled by the actuator, even increasing the valve lift, the valve would still be limited at a low speed. Therefore, with the controlling of actuator, not only the flow loss of valve clearance could be reduced by increasing the valve lift, but also the valve life could be extended by limiting the valve impact speed.

Conclusions

In this article, the dynamic characteristics of the suction valve are clarified by a changed self-acting valve model which is applicable for the stepless capacity control condition and by a simulation model based on AMESim. The accuracy of the simulation results is verified by the experimental measurement. Without the control of the actuator, the impact speed of the valve is almost 3 m/s, and it increases rapidly with the increase of the valve lift. But with the control of the actuator, the predicted valve impact speeds reduce to about 0.53 m/s, and the experiment value is about 0.58 m/s which verified the accuracy of the prediction. During the suction valve closing process, the valve

Table 1. Main specifications of stepless capacity control system.

Compressor		Thrust coefficient, β	0.901
Ratio of crank radius to length of connecting rod, λ	0.2	Mach number in valve clearance, Mv	0.2
Adiabatic compression coefficient, k	1.4	ZK	10 kN/m
Speed	300 r/min		
Gas	Air	Hydraulic unit	
Relative clearance volume, α	0.16	Maximum pressure, P	220 bar
Inlet pressure of first stage, p_s	0.1 MPa	Capacity of pump, Q_v	14.4 L/min
Suction valve		Diameter of cylinder, D	16 mm
Mass of movement parts, m	2.72 kg	Stiffness of reset spring, K_R	50 kN/m
Area of holes in valve seat, a_p	0.0056 m^2	Preload of reset spring, F_{pre}	1 kN
Valve lift, H	1.5 mm		

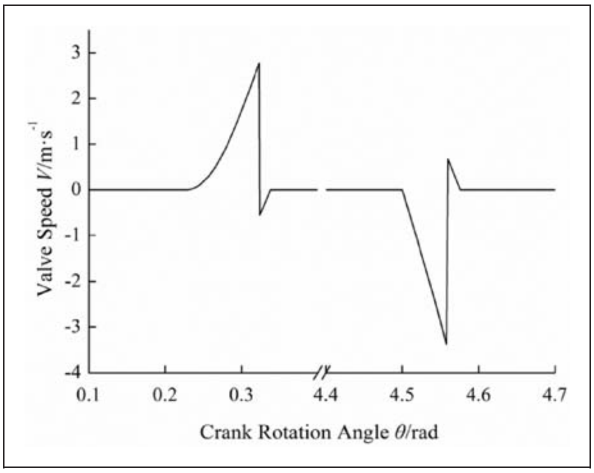

Figure 8. Valve speed as a function of crank rotation angle.

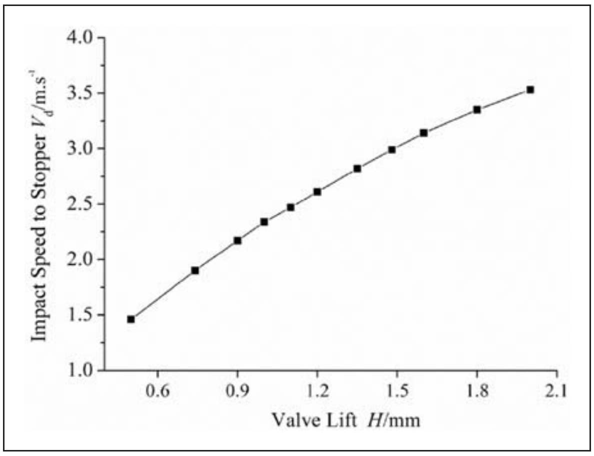

Figure 10. Valve impact speed to stopper at different valve lift.

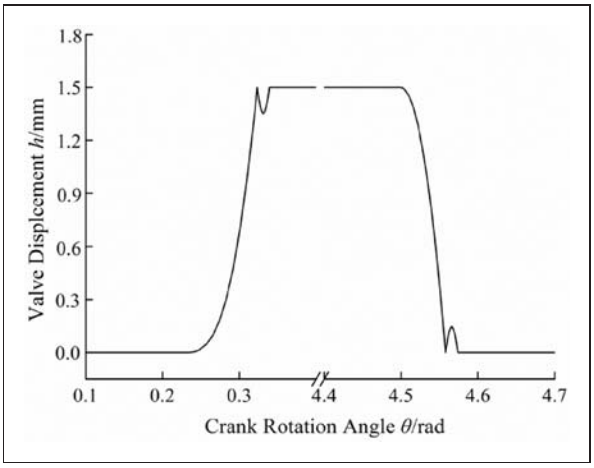

Figure 9. Valve displacement as a function of crank rotation angle.

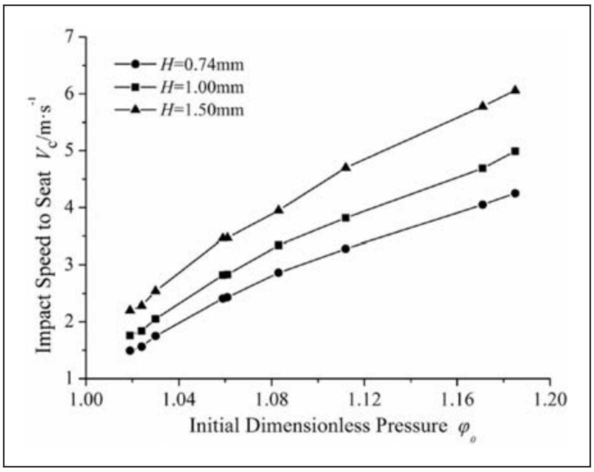

Figure 11. Valve impact speed to seat at different initial pressure and valve lifts.

speed is limited by the actuator and it moves together with the actuator. To limit the suction valve at a low speed, one practical way is to make the valve depressed by the actuator before it opens automatically. When suction valves are controlled by the actuator, the peak speed of the suction valve will still keep at a low value with the increase of .the valve lift, which is only 0.58 m/s. Therefore,

Table 2. Impact speed to seat at different Mv and θ_0.

			V_c (m/s)		
Mv	θ_0	φ_0	$H = 0.74$ mm	$H = 1.00$ mm	$H = 1.50$ mm
0.12	4.5	1.024	1.56	1.84	2.28
	5.0	1.030	1.75	2.05	2.54
	5.5	1.019	1.49	1.76	2.20
0.20	4.5	1.061	2.43	2.83	3.47
	5.0	1.083	2.86	3.34	3.95
	5.5	1.059	2.41	2.82	3.46
0.30	4.5	1.112	3.28	3.82	4.70
	5.0	1.185	4.25	4.99	6.06
	5.5	1.171	4.05	4.69	5.78

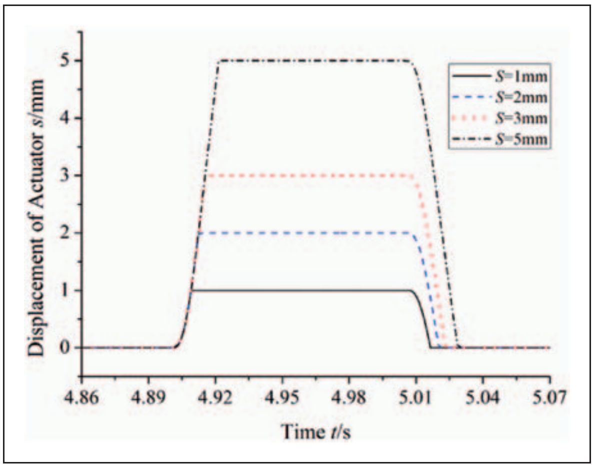

Figure 12. The variation of displacement with different piston stroke in a compression period.

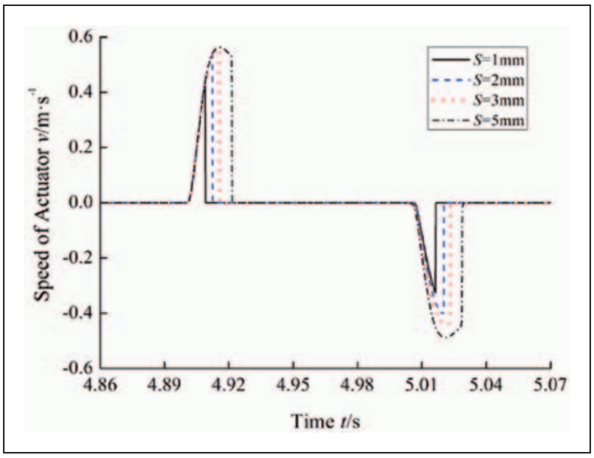

Figure 13. The variation of speed with different piston stroke length in a compression period.

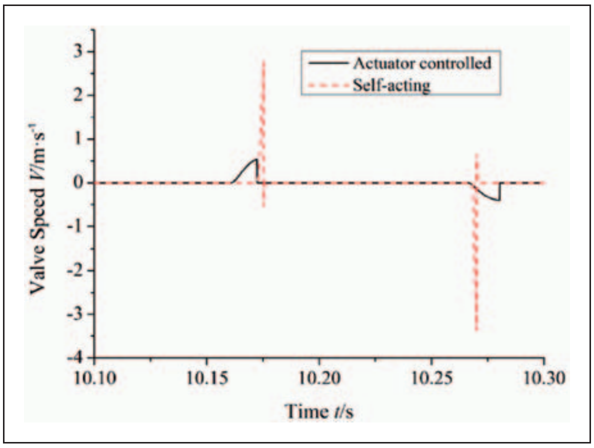

Figure 14. Comparison of speeds between self-acting valve and actuator controlled valve.

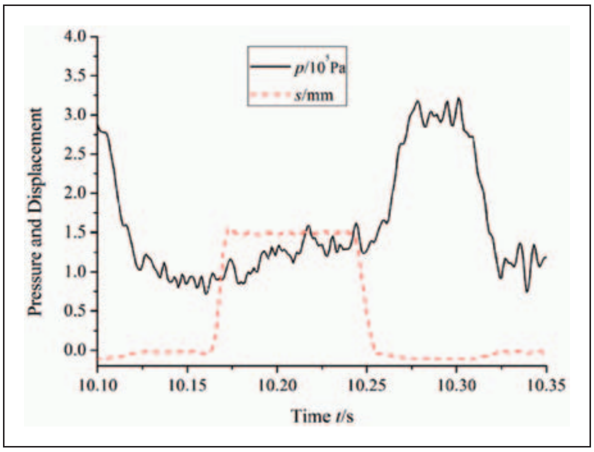

Figure 15. Displacement and pressure tested by experiment.

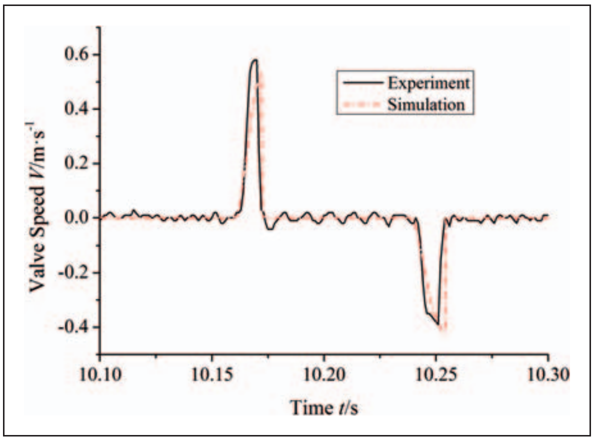

Figure 16. Comparison of valve speeds between simulation and experiment.

with the stepless capacity control system, the dynamic characteristics of suction valve are changed, the valve impact speed to seat and stopper decrease significantly, which would be helpful for extending the life of valve plates.

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Appendix

Notation

a_p	area of holes in valve seat
C_R	valve rebound coefficient
C_s	local speed of sound
D	diameter of hydraulic cylinder
Fg	gas force
F_{pre}	preload of reset spring
h	valve displacement
H	valve lift
H_0	preload distance of valve spring
K	adiabatic compression coefficient
K	valve spring stiffness
K_R	reset spring stiffness
m	mass of valve movement part
Mv	Mach number in valve clearance
p	pressure in cylinder
p_s	inlet pressure
P	hydraulic pressure
Q_v	capacity of hydraulic pump
s	displacement of actuator
S	hydraulic piston stroke
u_a	gas equivalent velocity in valve opening area
v	speed of actuator
V	speed of valve
V_c	impact speed of valve to seat
V_d	impact speed of valve to stopper
Z	valve springs amount
α	relative clearance volume
β	thrust coefficient of valve
λ	ratio of crank radius to length of connecting rod
ω	rotation speed of compressor
φ	dimensionless pressure p/p_s
φ_0	initial dimensionless pressure
θ	crank rotation angle
θ_0	initial rotation angle of valve closing