doi:10.3772/j.issn.1006-6748.2018.01.001

Study on effect analysis and parameter optimizing of stepless capacity control system on reciprocating compressors①

Wang Yao (王 瑶)*, Zhang Jinjie②**, Zhou Chao*, Liu Wenhua*
(* Diagnosis and Self-recovering Engineering Research Center, Beijing University of Chemical Technology, Beijing 100029, P. R. China)
(** Compressor Health and Intelligent Monitoring Center of National Key Laboratory of Compressor Technology, Beijing University of Chemical Technology, Beijing 100029, P. R. China)

Abstract

An improved model of reciprocating compressor operation cycle with a stepless capacity control system is presented and influence of the key parameters of the system is evaluated. In the stepless capacity control system of a reciprocating compressor, mechanical unloaders are used to partially hold suction valves open for a certain time during the compression stroke. The typical working process of the reciprocating compressor is changed by capacity regulation apparatus. However, some critical parameters like the hydraulic force acting at the unloader have not been rigorously studied in previous researches. Here an improved numerical model of a double acting reciprocating compressor under the control stepless capacity is proposed and verified by experimental trials. Numerical simulations are carried out to select and evaluate the acting force which definitely has an influence on indicator diagrams of compressors. It is observed that the optimized range of 350N to 380N is preferable for the unloader force such that the intensity of opening and closing impacts are minimized.

Key words: reciprocating compressor, stepless capacity control, numerical model, parameters optimization

0 Introduction

Reciprocating compressor is a kind of large-scale and key equipment widely used in oil extraction, gas production, oil refining, chemical industry[1] and other fields. Nowadays, a current market demand exists for a variable exhaust volume of reciprocating compressor, which requires that the exhaust volume of compressors should be variable to adjust to the production needs. However, most compressors use a backflow valve refluxing method to change the exhaust volume which leads to high power consumption and waste. The stepless capacity control method is being studied and developed for the application constantly. The suction valves will be held open forcibly during a variable portion of the compression stroke. It uses unloaders controlled by computer numerical control hydraulic valves on suction valves to adjust the time of suction valves opening and closing[2]. The unloader is driven by a single-rod cylinder, which possesses a multitude of advantages, such as small size and simple structure, compared with a double-rod symmetric cylinder[3].

Counter-current flow will be produced during the initial portion of compression process and compressed gas volume will be reduced. As a result, power consumption will be decreased. The action of stepless capacity control system will change the movement of valve plate and working condition of suction valve significantly[4]. The working cycle of reciprocating compressor and capacity control effect will be influenced by the key parameters of the stepless capacity control system. Furthermore, improper parameters may lead to faults of suction valve as leakage, spring fracture and valve plate fracture.

Mathematical models of the working cycle under different operating conditions of a reciprocating compressor are proposed[5-7]. The working loads and time of the suction valve can be calculated by these models.

Valve dynamics and time dependent flow field through the valve are coupled[8]. Equations of gas flow and valve plate under the stepless capacity control system are established[9]. Equations of valve plate under normal operating conditions of a reciprocating compres-

① Supported by the National Key Research and Development Plan (No. 2016YFF0203305), the National High Technology Research and Development Programme of China (No. 2014AA041806) and the Fundamental Research Funds for the Central Universities (No. ZY1617).
② To whom correspondence should be addressed. E-mail: zjj87427@163.com
Received on May 25, 2017

sor are established[10-12]. Based on these equations, transient motion and stress of the plate are analyzed by numerical simulations.

Above all, most research reports focus on dynamic characteristics of reed valves or ring valves of the compressor with the stepless capacity control system. However, there are few studies on the behavior of mesh valves, which is also widely used in large-scale compressors, when a compressor is under capacity regulation condition. In addition, high-pressure hydraulic oil needs to be provided by the hydraulic system to provide enough force to hold the valve open in practical application. However, too large force may cause excessive impact stress on both valve plate and unloader which adversely diminishes the working life of the valve and the unloader. Therefore, an attempt is made in the paper to obtain the lowest hydraulic pressure requirement for the same performance by thoroughly studying the influence of different unloader force. An improved numerical model of a reciprocating compressor installed with the stepless capacity control system is reported. To verify the accuracies of the models, theoretical and experimental trials have been conducted. 'Object and models' section describes the research objectives and mathematical models. 'Simulation and analysis' section analyzes the motion law of the plate valve and the cylinder pressure. The effects of the acting force of the unloader are also be evaluated. A method of optimization design on the acting force through data verification is presented. The proposed models and method provide the basic technology to realize precise control of reciprocating compressor in different working conditions.

Nomenclature			
h	Valve displacement (m)	ω	Angular velocity
H	Maximum displacement (m)	λ	Ratio between the crankshaft radius and connecting rod length
L	Compression length of the springs (m)	φ	Pressure ratio
v	Velocity ($m \cdot s^{-1}$)	δ	Coefficient of effective force action
V	Volume (m^3)	η	Relative capacity
m	Mass (kg)	Δ	Change quantity
P	Pressure (Pa)	ξ	Loading coefficient
A	Force area on the valve (m^2)	s	Suction process
Z	Number of valve springs	sv	Suction valve
K	Equivalent stiffness ($N \cdot m^{-1}$)	sp	Suction valve plate
k	Ratio of specific heat of gas	sbp	Suction buffer plate
αA	Flow cross section area of valve (m^2)	ss	Suction valve spring
R	Gas constant	svo	Suction valve opening
T	Temperature (℃)	bp	Buffer plate
t	Time (s)	r	Removal of actuator force
C	Rebound coefficient	svc	Suction valve closing
F	Force (N)	m	Maximum
q	Gas load	cy	Cylinder
α	The valve flow coefficient	str	Stroke volume
β	Coefficient of applied force of gas	cle	Clearance
θ	Angular	exp	Expansion
		ac	Actuator

1 Object and models

The objective of this work is a two-dimensional (2D) type model of a double-acting reciprocating compressor with two cylinders, as shown in Fig. 1. The operating parameters of the compressor and symbols of the numerical models are shown in Table 1.

The typical stepless capacity control system of a reciprocating compressor is mainly composed of a hydraulic unit, a computer-controlled unit, mechanical actuators and the monitoring system, as shown in Fig. 2. Mechanical unloader is the key part of the mechanical actuator driven by the hydraulic system. The suction valves of a reciprocating compressor are opened and closed by the mechanical unloader during part of the compression to control the compressor capacity.

The process is controlled by the computer-controlled system. At present, the stepless capacity control systems is widely used because of high energy-saving effectiveness. Moreover, the suction valves working under the stepless capacity regulation condition have complicated stress states and are easily damaged by shocks.

The motion of an actuator decided by the unloader force has a major influence on the performance of a stepless capacity regulation system.

In the capacity regulation condition, the unloader force needs to hold open of the suction valve for a certain time during the compressor stroke. If not, the required appropriate capacity load could not be achieved.

Fig. 1 The double-acting reciprocating compressor

Table 1 Operation parameters of the compressor

Unit number	Structural form	The number of cylinders	Driven machine	Number of suction valves
K101	Horizontal type	2	Motor	8
Suction pressure(MPa)	Exhaust pressure(MPa)	exhaust volume(m^3/min)	Cooling form	Number of exhaust valves
0.1	0.3	12	Water cooling	8

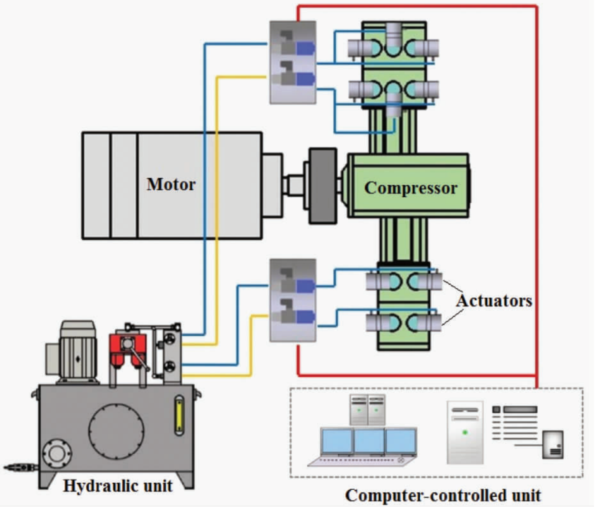

Fig. 2 Schematic diagram of a stepless capacity control system

The movement of valve plate is mainly decided by the unloader force when the capacity control system is operated. If the force is not large enough, the valve plate will be partially opened instead of fully opened in the reverse stroke resulting in the increment of pressure loss.

When the unloader is pushed down to force the valve plate into an open position, it creates an undesirable loading impact on the valve plate. Too large force may cause excessive impact stress on both valve plate and the unloader which will adversely diminish the working life of the valve and the unloader.

The unloader force provided by the hydraulic system is also one of the major determinants of the type selection of a pump in the process of system design.

It needs to establish a working cycle model to analyze the adjusting effect of the capacity control system and motion law of the valves. The differences of the working dynamic characteristics between a plate valve and a ring valve under full load condition are listed as follows:

(1) The plate needs to be taken as a whole to calculate the force and flow area of the plate valve.

(2) After overcoming part of stroke, the valve plate impacts on a damping plate and is thereby braked due to the enlarged moving mass. Then the valve plate overcomes the remaining stroke together with the damping plate.

(3) In addition to the gas force and spring force, the deformation force resulting from the plate deformation is also an important factor to valve plate movement.

2 Theoretical analysis

A mesh valve usually includes a valve seat, a valve plate, a valve lift guard, a buffer plate and some springs. Different suction and discharge pressures have different requirements on the structural design of the valve, particularly on the material of the valve plate and the spring stiffness.

As shown in Fig. 3, there are multiple flow channels machined on the valve seat and valve lift guard. The direction of the gas flow in the suction valve is indicated by arrows. In order to calculate the equivalent flow area of the plate valve, each resistance area is considered as an orifice and the equivalent flow area is calculated by the equations of series-parallel effective flow area.

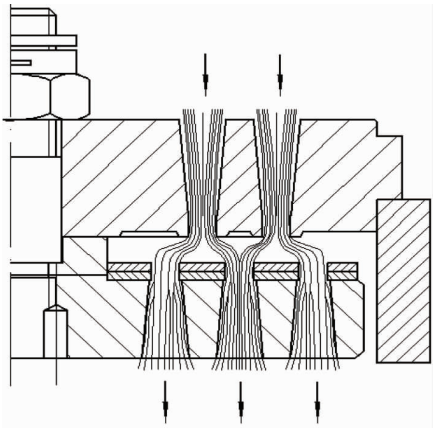

Fig. 3 The structure section view of a plate valve

In full load condition, mesh valves of a reciprocating compressor are automatic. When the pressure is larger than the spring force, the plate valve will open and the cylinder is connected with the inlet chamber or the discharge chamber. When the pressure is less than the spring force, the valve will be forced into a close position. The alternating impacting of an actual valve plate on a valve seat and/or valve guard causes the impact loading.

Moreover, the model can be also able to describe the valve behavior under two typical fault conditions. The valve spring's fatigue and valve leakage by reducing the spring stiffness and adding a variable cross-sectional leakage area respectively.

In this paper, the model is used to show the valve behavior and gas flow under normal condition. The mathematical model of valve motion under full load condition is shown in Eq. (1), where h_s, v_s, m_s are the displacement, velocity and mass of the suction valve respectively, P_s is the suction pressure and T_s is the suction temperature.

$$\begin{cases} \dfrac{\mathrm{d}h_s}{\mathrm{d}\theta} = \dfrac{v_s}{\omega} \\ \dfrac{\mathrm{d}v_s}{\mathrm{d}\theta} = \dfrac{1}{m_s\omega^2}\left[P_s(1-\varphi_s \quad A_s\beta_s - Z_sK_{ss}(L_0+h_s) - K_{sp}h_s - k_{sbp}K_{sbp}(h_s - \dfrac{H_{sp}}{2}) \right] \\ \dfrac{\mathrm{d}\varphi_s}{\mathrm{d}\theta} = -\dfrac{1}{\omega}\dfrac{k\varphi_s}{V_{cy}}\omega\dfrac{V_h}{2}\left(\sin\theta + \dfrac{\lambda}{2}\sin 2\theta\right) + \dfrac{k}{\omega}\dfrac{\alpha_sA_s}{V_0+\dfrac{V_h}{2}\left(1-\cos\theta+\dfrac{\lambda}{2}\sin^2\theta\right)}\varphi_s^{\frac{1}{k}}\sqrt{\dfrac{2k}{k-1}RT_s[1-(\varphi_s)^{\frac{k-1}{k}}]} \end{cases} \tag{1}$$

The parameters of the initial condition include opening angle of the valve, initial displacement of the valve plate, initial velocity of the valve plate, initial acceleration of the valve plate and initial pressure in the cylinder. The initial displacement, velocity and acceleration of the valve plate are all zero, as shown in

$$\begin{cases} h_s \mid_{\theta=\theta_{svo}} = 0 \\ \left.\dfrac{\mathrm{d}h_s}{\mathrm{d}\theta}\right|_{\theta=\theta_{svo}} = 0 \\ \left.\dfrac{\mathrm{d}^2h_s}{\mathrm{d}\theta^2}\right|_{\theta=\theta_{svo}} = 0 \end{cases} \tag{2}$$

The initial pressure in cylinder P_0 can be calculated by force equilibrium formulas in Eq. (3) and Eq. (4).

$$(P_s - P'_0A_p - Z_sK_{ss}L_0 = 0 \tag{3}$$

$$P_0 = \frac{1}{A_p}(P_sA_p - Z_sK_{ss}L_0) \tag{4}$$

where A_p is the area of port, Z_s is the number of springs, K_{ss} is equivalent stiffness of the spring and L_0 is the initial length of the spring.

When the valve plate moves to $h_s = \dfrac{H_{sp}}{2}$, the valve plate will impact the buffer plate and the moving mass will be enlarged, as shown in Eq. (5), causing a further decrease in the acceleration of the valve plate.

$$m_s\left(\left.\frac{\mathrm{d}h_s}{\mathrm{d}\theta}\right|_{\theta=\theta\frac{H_{sp}}{2}}\right)_{\text{before}} = (m_s + m_{bp})\left(\left.\frac{\mathrm{d}h_s}{\mathrm{d}\theta}\right|_{\theta=\theta\frac{H_{sp}}{2}}\right)_{\text{after}} \tag{5}$$

where m_{bp} is the mass of buffer plate.

When the valve moves to $h_s = H_{sp}$, the valve plate and the buffer plate will impact the lift limiter. A rebound model is applied to analyze the collisions, as shown in Eq. (6).

$$\left(\left.\frac{\mathrm{d}h_s}{\mathrm{d}\theta}\right|_{\theta=\theta_{H_{sp}}}\right)_{\text{before}} = -C_R\left(\left.\frac{\mathrm{d}h_s}{\mathrm{d}\theta}\right|_{\theta=\theta_{H_{sp}}}\right)_{\text{after}} \tag{6}$$

where C_R is the rebound coefficient.

Suction valves of a reciprocating compressor are kept open for a certain time by unloaders to delay the closing process of valve plates and the following compression stroke until the gas remaining in the cylinder meets the production demands. After the hydraulic force is released, valve plates begin to move toward the valve seats and away from the valve guards. The movement of the valve plates of suction valves is controlled by unloaders acting on the valve plates in capacity regulation conditions. Therefore, the action of unloaders should be taken into account for mathematical models of variable loads. Coefficient ξ_{ac} is defined to control the certain action time of unloaders on the valve plates.

$$\xi_{ac} = \begin{cases} 1, & \theta_{svo} \leqslant \theta \leqslant \theta_r \\ 0, & \text{otherwise} \end{cases} \tag{7}$$

$$\eta V_s + V_{\exp} + V_{cle} = V_{cle} + \frac{V_{str}}{2}\left(1 - \cos\theta_r + \frac{\lambda}{2}\sin^2\theta_r\right) \tag{8}$$

$$\begin{cases} \dfrac{dh_s}{d\theta} = \dfrac{v_s}{\omega} \\ \dfrac{dv_s}{d\theta} = \dfrac{1}{M_s\omega^2}\left[P_s(1 - \varphi_s \quad A_s\beta_s + \xi_{ac}F_{ac} - Z_sK_{ss}(L_0 + h_s) - K_{sp}h_s - k_{sbp}K_{sbp}(h_s - \dfrac{H_{sp}}{2})\right] \\ \dfrac{d\varphi_s}{d\theta} = -\dfrac{1}{\omega}\dfrac{k\varphi_s}{V_{cy}}\omega\dfrac{V_{str}}{2}\left(\sin\theta + \dfrac{\lambda}{2}\sin 2\theta\right) - \dfrac{k}{\omega}\dfrac{\alpha_s A_s}{V_{cle} + \dfrac{V_{str}}{2}\left(1 - \cos\theta + \dfrac{\lambda}{2}\sin^2\theta\right)}\varphi_s^{\frac{k-1}{k}}\sqrt{\dfrac{2k}{k-1}RT_s[(\varphi_s)^{\frac{k-1}{k}} - 1]} \end{cases} \tag{9}$$

When ξ_{ac} equals to 1, the unloaders open the suction valves forcibly and the gas in the cylinder flows back to the suction chamber. When ξ_{ac} equals to 0, the unloaders withdraw and suction valves will close automatically. ξ_{ac} is defined as Eq. (7). The volume of gas that needs to be compressed is calculated as Eq. (8), where V_s is the effective volume of suction process, $V_{\exp}$ is the expansion volume of gas in the clearance space, V_{cle} is the volume of clearance space and V_{str} is the stroke volume of the compressor.

The general motion characteristic for a valve plate and dynamic change of cylinder pressure is then described by Eq. (9), where F_{ac} is the hydraulic force on the unloader. The flow chart of the part-load model is plotted in Fig. 4.

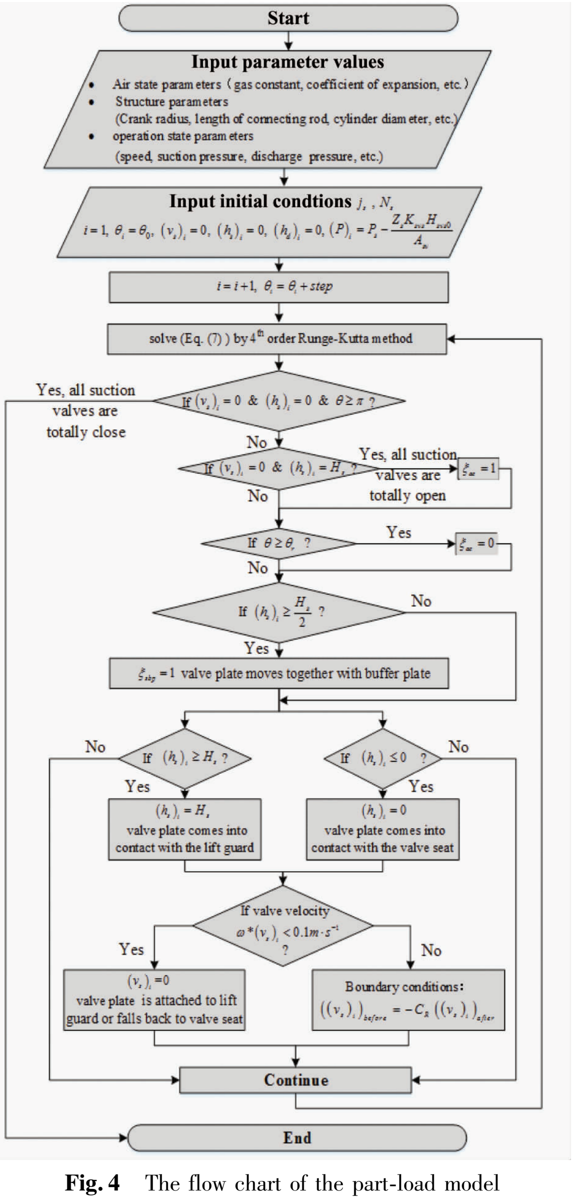

Fig. 4 The flow chart of the part-load model

3 Simulation and analysis

3.1 The pressure simulations of variable loads

Considering that making mechanism reconstruction of the valve structure on different compressors is difficult, it is not possible to measure valve plate displacement with a sensor directly for every reciprocating compressor. An accurate model is definitely needed for optimization design of the stepless capacity control system.

Numerical solution of stepless capacity regulation models is conducted and the results are compared with the experimental results. As shown in Fig. 5, the simulated dynamic pressures under different loads show good consistency with the measured results in the whole working process of the compressor.

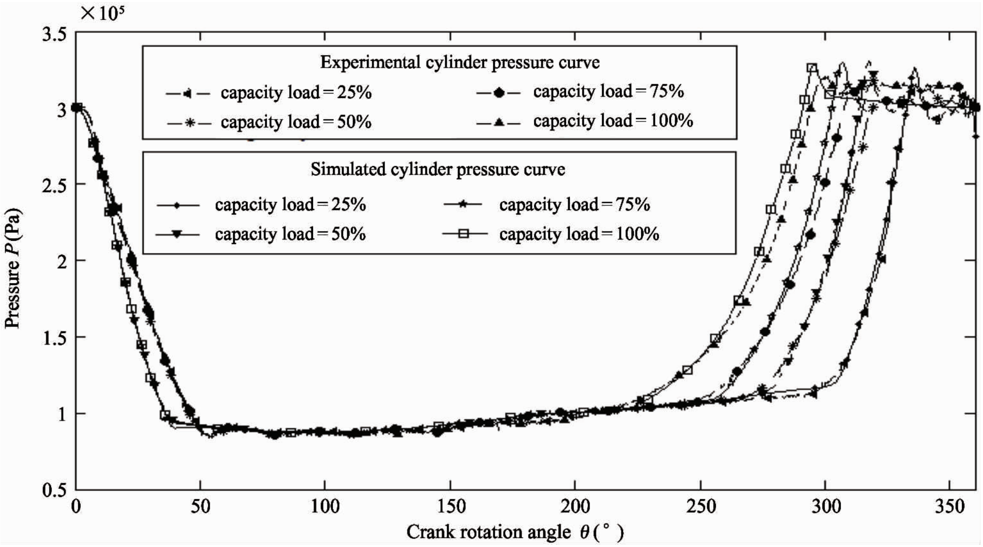

Fig. 5 Dynamic pressure in the cylinder under different load

3.2 The plate motion of the mesh valve under variable loads

Fig. 6 shows the measured and predicted valve displacement for suction stage. The cylinder pressure drops as the piston moves in the angular range from 0° to 50°. When the cylinder pressure drops sufficiently below that of the inlet manifold (0.1MPa), the suction valves open and suction process begins. The closing of the valve plate takes place in a crank angle ranging from 168° to 180°.

The simulated result is found to be good agreement with measured displacement in Fig. 6. Based on the comparison of measured and predicted results of cylinder pressure and valve displacement. It is confirmed the mathematic models of the compressor under variable loads conditions are acceptable. Therefore, further analysis about the influence of critical parameters on performance of the stepless capacity control system is possible.

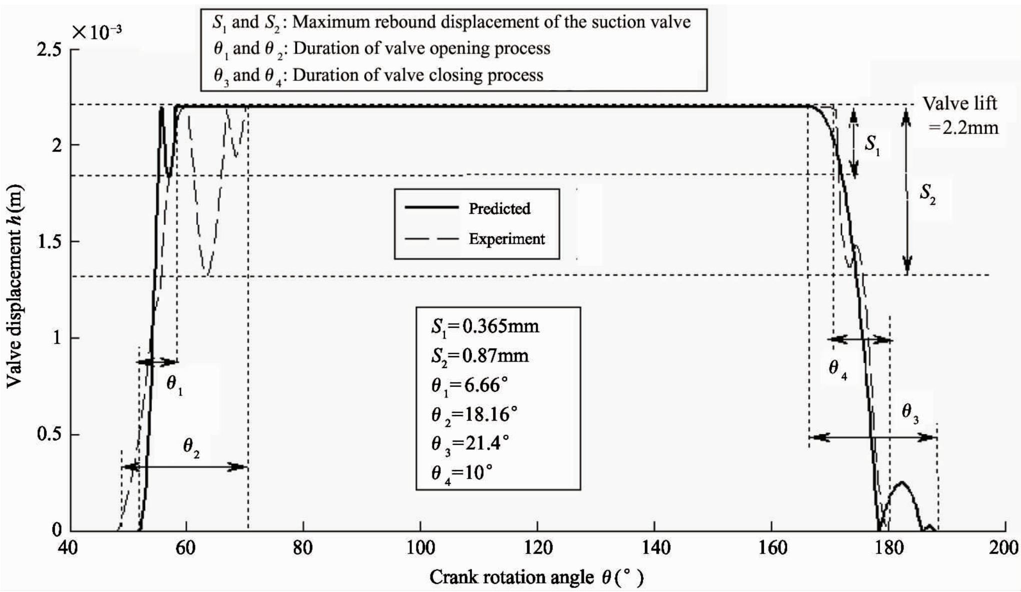

Fig. 6 The motion of suction valve plate under full load

The predicted speeds of the valve plate at different loads are shown in Fig. 7. Both the crank angle and the maximum speed of the valve plate in its closing process increase as the load decreases. When the load is less than 90%, the valve plate and buffer plate will impact the valve seat in the closing process. If the load continues to fall, the closing speed of the plate will be larger than the opening. This result is caused by the increase of cylinder pressure in the reverse stroke.

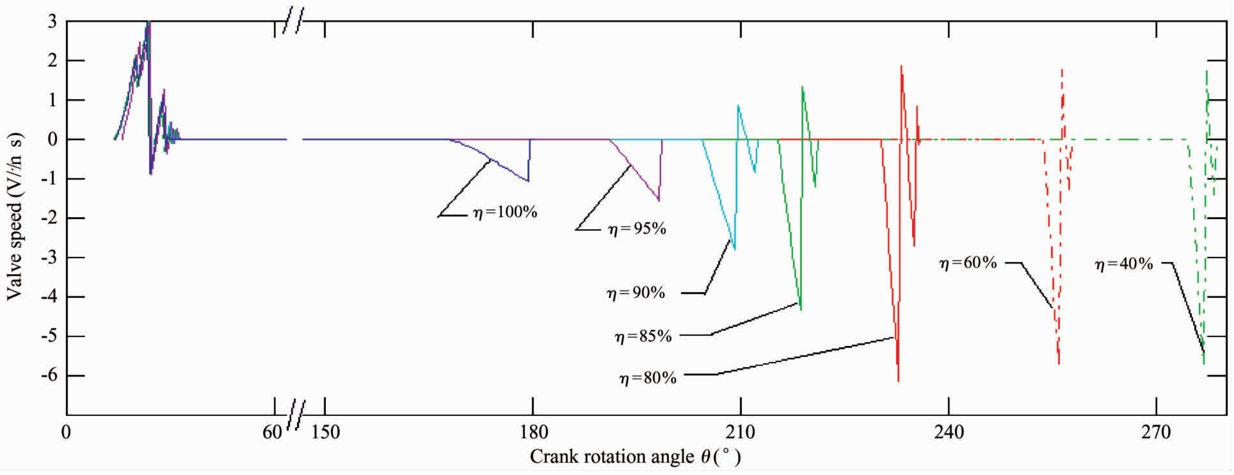

Fig. 7 The motion of suction valve plate under different loads

As shown in Fig. 8, the velocity of the valve plate motion after withdraw of the unloader initially increases as the load decreases from 100% to 80% and then decreases slightly when the load is less than 80%, representing the largest velocity of 6.2m/s at 80% load. The larger velocity means the greater impact stress of valve plate when it impacts the valve seat in the closing process, which in turn causes premature sealing element fracture. As a result, it is necessary to control the action of the unloader to decrease the impact velocity of the valve plate on reaching the valve seat in the system design.

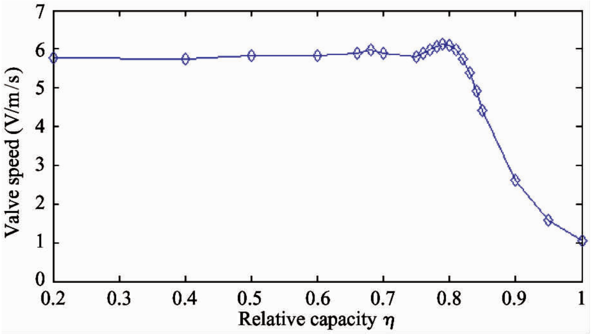

Fig. 8 The largest velocity of the valve plate motion after withdraw of the unloader

3.3 Analysis on the effect of unloader force

The unloader force that acts on the valve plate has major influence on the performance of the stepless capacity regulation system. For the same capacity load of 50%, the difference of the indicator diagram under different unloader forces is analyzed in this section, as the *P-V* diagram in Fig. 9 shows.

The following results are obtained:

When the unloader force is zero, the largest indicator diagram is displayed, indicating the largest energy consumption.

As the increment of the force, the indicator diagram decreases indicating that the compression stroke is delayed due to the gas backflow through the opened suction valves. The more the compression stroke is delayed, the larger the cylinder pressure will be in the backflow process.

If the unloader force is not large enough, suction valves are forced into a close position prematurely by the increased cylinder pressure and the actual load is larger than required.

When the force is more than 400N, suction valves close at the set target angle value and the indicator diagram remains unchanged.

It can be concluded that the unloader force should be large enough to meet the required capacity load. The hydraulic force of 400N is possibly not an optimal parameter value. Too large unloader force creates a forceful and undesired impact between the valve plate and the actuator. An optimization design of the hydraulic force should be carried out to avoid too large force.

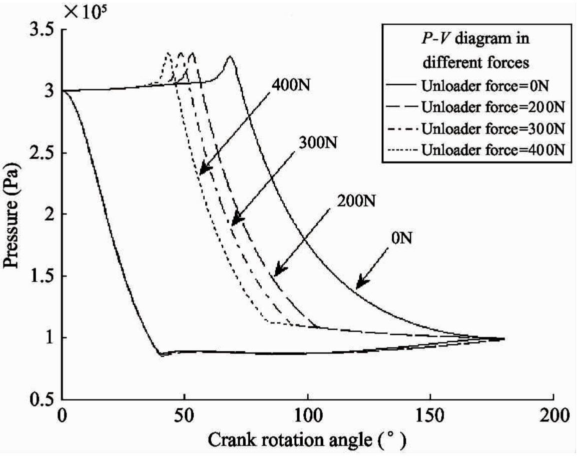

Fig. 9 The indicator diagrams with different forces of the unloader

3.4 Optimization design of the unloader force

As described in the previous sections, the force of the unloader has certain effects on the performance of the capacity regulation system. The results show that optimum value for the unloader force should be found to improve the performance and reliability of the system.

For the same capacity load of 50%, the force of the unloader is changed from 0 to 400N with a step size of 60N. The comparison of valve closing angle and relative capacity error under different hydraulic forces is shown in Table 2. As the force increases, the closing angle of the plate initially increases and then remains unchanged.

Table 2 Comparison of valve closing angle and relative capacity error under different unloader forces

Parameters \ Numbers	1	2	3	4	5	6	7	8	9
$F_{ac}(N)$	500	440	360	300	240	180	120	60	0
$\theta_r(°)$	268	268	268	261	252	244	232	221	180
q/q_0	1.0	1.0	1.0	1.027	1.071	1.108	1.155	1.223	1.458

In Fig. 10, it can be seen that the closing angle of the plate reaches the setpoint of 268° when the force is larger than 350N. Therefore, 350N is the ideal force for the actuator to hold the suction valve open. However, an acceptable force range is more applicable to engineering than a certain value because the oil pressure range is easy to be controlled for the former. Finally, a force range of 350N to 380N is chosen for practical use and the corresponding oil pressure range is from 5.5MPa to 6MPa. However, there are few studies on the optimization of the unloader acting force of the stepless capacity control system. The oil pressure of a stepless capacity control system can reach up to more than 14MPa and the unloader acting force could break the valve plate.

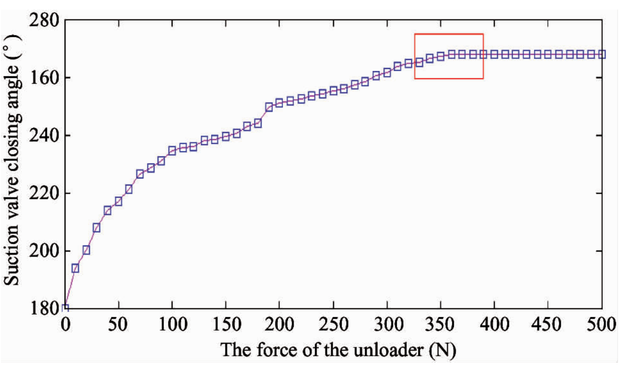

Fig. 10 Valve closing angles under different unloader forces

The vibration of valve's opening and closing events under optimized oil pressure is compared with that without optimization in Fig. 11 and Fig. 12. The intensity of opening and closing impacts are minimized, indicating that the optimization can effectively decrease the impact stress on both valve plate and the unloader. The hydraulic pressure requirement for the existing system has been reduced by 25%.

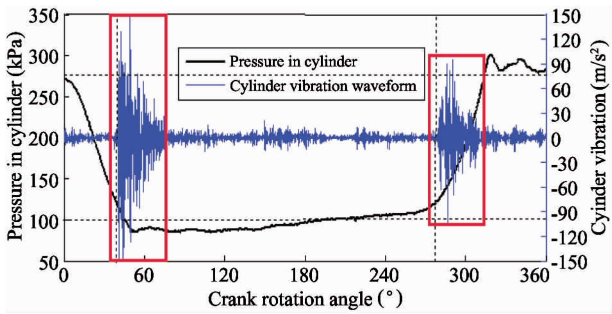

Fig. 11 The cylinder vibration waveform under about 50% load with oil pressure of 8 MPa

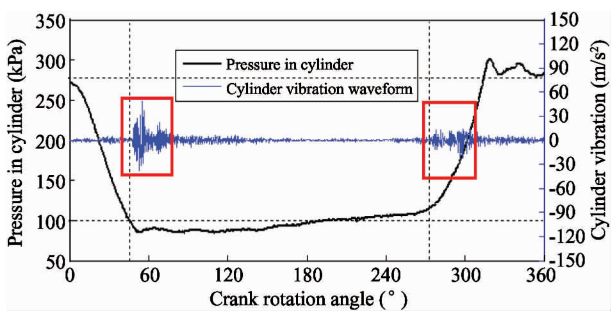

Fig. 12 The cylinder vibration waveform under about 50% load with optimized oil pressure of 6MPa

4 Conclusions

In this work, a new mathematical model is proposed to illustrate the working cycle of the reciprocating compressor under different capacity regulation conditions with the aid of a stepless capacity control system. The motion law of the valve plate is obtained and the influence of the parameters on the performance of capacity regulation is analyzed.

The following conclusions of the study are obtained:

Compared with the measured results, the predicted dynamic pressure and valve motion law show good consistency over different cooling load conditions. The new model has an acceptable accuracy.

The motion laws of the valve plate in different capacity conditions are obtained. Both the crank angle and the maximum speed of the valve plate in its closing process increase as the load decreases. The largest closing velocity of 6.2m/s occurs at 80% load.

The indicator diagrams under different unloader forces are obtained. As the increment of the force, the indicator diagram decreases, indicating that the compression stroke is delayed due to the gas backflow through the opened suction valves. The unloader force should be large enough to meet the required capacity load.

The optimized range of 350N to 380N is selected for the unloader force such that the intensity of opening and closing impacts are minimized.

References

[1] Xing C H, Xu F T, Yao Z Y, et al. A fault diagnosis method of reciprocating compressor based on sensitive feature evaluation and artificial neural network [J]. *High Technology Letters*, 2015, 21(4): 422-428

[2] Steinrueck P, Ottitsch F, et al. Better reciprocating compressor capacity control [J]. *Hydrocarbon Processing*, 1997, 76(2): 79-83

[3] Shen W, Huang H L, Li Y, et al. Work characteristic analysis of single-rod symmetric cylinders[J]. *High Technology Letters*, 2017, 23(3): 286-292

[4] Liu G, Zhao Y, Wang L, et al. Dynamic performance of valve in reciprocating compressor used stepless capacity regulation system [J]. *Journal of Oral Rehabilitation*, 2014, 28 (1): 26-32

[5] Sun S Y, Ren T R. New method of thermodynamic computation for a reciprocating compressor: computer simulation of working process [J]. *International Journal of Mechanical Sciences*, 1995, 37(4): 343-353

[6] Hong W, Jin J, Wu R, et al. Theoretical analysis and realization of stepless capacity regulation for reciprocating compressors[J]. *ARCHIVE Proceedings of the Institution of Mechanical Engineers Part E-Journal of Process Mechanical Engineering* 1989-1996, 2009, 223 (4): 205-213

[7] Zhang J J, Jiang Z N, et al. Numerical simulation and experimental study on plate valve transient motion and fatigue fracture principles [J]. *International Journal of COMADEM*, 2014, 17(2): 17-27

[8] Deng Z A, Xiao X. Experimental study on pressure fluctuation characteristics of slug flow in horizontal curved tubes[J]. *High Technology Letters*, 2017, 23(1): 23-29

[9] Jin J, Hong W. Valve dynamic and thermal cycle model in stepless capacity regulation for reciprocating compressor [J]. *Chinese Journal of Mechanical Engineering*, 2012, 25 (6): 1151-1160

[10] Tang B, Zhao Y Y, Li L S, et al. Dynamic characteristics of suction valves for reciprocating compressor with stepless capacity control system[J]. *Proceedings of the institution of mechanical engineers part E-Journal of process mechanical engineering*, 2014, 228 (2): 104-114

[11] Wang Y, Zhang J J, Liu J N, et al. Transient motion simulation and stress analysis of reciprocating compressor valve[J]. *Journal of Mechanical Strenth*, 2016, 38 (3): 543-548

[12] Zhang J J, Wang Y. A simulation study on the transient motion of a reciprocating compressor suction valve under complicated conditions[J], *Journal of Failure Analysis and Prevention*. 2016,16 (5): 790-802

Wang Yao, born in 1992. He received his Ph. D degree in Diagnosis and Self-recovery Engineering Research Center of Beijing University of Chemical Technology. His research interests include reciprocating compressor fault diagnosis and stepless capacity controlling.