Numerical modeling of dynamic characteristics for combined valves in multiphase pump

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ABSTRACT

This research presents theoretical analysis and numerical simulation of combined valves’ dynamic characteristics in reciprocating oil–gas multiphase pump. Based on the mathematical model describing valves’ motion, a geometric model of pump cavity and combined valves is put forward for numerical simulation in multiphase pump. The simulation is conducted by computational fluid dynamics (CFD) method with its dynamic grid technique. The motion process of suction valve and discharge valve are obtained on the basis of boundary conditions and optimized numerical approaches. And the effects of gas volume fraction (0.1 \textasciitilde 0.9), suction pressure (0.20 \textasciitilde 0.40 MPa) and discharge pressure (1.0 \textasciitilde 3.0 MPa) on valves’ motion are reported. The results of pressure distribution, and valves’ lift and velocity show that valve plates may cling or rebound and then vibrate after reaching the lift limiter, and the lag angle of one cycle remains 3° under different working conditions. The study could lay theoretical foundation for the design of new type of multiphase pump.

ARTICLE HISTORY

Received 16 March 2016
Accepted 4 February 2017

KEYWORDS

Computational fluid dynamics (CFD); multiphase pump; oil–gas flow; dynamic grid technique; \(k\)-\(\varepsilon\) model

1. Introduction

The output of an oil well is often accompanied by a multiphase medium, such as natural gas, water, or solid particles in oil drilling (Räbiger, Maksoud, Ward, & Hausmann, 2008). At present, more than 70% of oilfield associated gas is vented or burned at home and aboard, which results in a massive waste. With the exploration and development of offshore oil–gas fields, multiphase transportation technology has gained more and more applications (Wang, Zha, Mcdonough, & Zhang, 2015). It provides oil–gas close-line transportation simultaneously. Therefore, it could not only make full use of existing facilities and reduce infrastructure investment, but also increase oil and gas production (Yang, Hu, Hu, & Qu, 2017).

Oil–gas multiphase pumps are the key facilities of transportation technology. They could be divided into rotodynamic and positive displacement pumps (Lieu, Chan, & Ooi, 2012; Pirouzpanah, Gudigopuram, & Morrison, 2017). The internal-compressed reciprocating multiphase pump belongs to the latter. It has good anti-gas-lock and compressive properties.

One of its innovative designs is the vertical combined valves at the pump outlet. The combined valves have both suction and discharge functions which intensify gas–liquid two-phase collection and pressurization. As the pivotal hydraulic component, the combined valves are directly related to instantaneous two-phase migration flow and discharge stability in internal-compressed multiphase pump (Wu, Wu, Li, & Wang, 2010). Hence, it is essential to study the working mechanism of combined valves to achieve the reliable operation of the multiphase pump and efficient recovery of oilfield associated gas.

Currently, many literatures on pump and compressor valves have been published. Yu, Dianbo, Jianmei, and Xueyuan (2010) and Ma, He, Peng, and Xing (2012) both presented the experimental results on the valve’s motion of compressor. After being opened, the discharge valve gains increasing velocity sharply and strikes the valve stop in a short period. In the closing process, the valve appears to have short-time bumpy vibrations near half maximum lift and closed position which prolongs the total time of one cycle. Tanaka and Tsukamoto (1999) analyzed the effect of a valve’s motion on the transient phenomena of centrifugal pump by experimental study. It was found that a rapid opening/closure of the valve results in the instantaneous fluctuations of pressure and flow rate in the pump.

What’s more, some researchers have focused on the study of valve’s internal flow. The advanced methods
of flow field measurement and numerical simulation are widely adopted (Cao, Gao, Li, Xing, & Shu, 2011; Shah, Chughtai, & Inayat, 2013), such as particle image velocimetry (PIV), computational fluid dynamics (CFD), etc. (King, Ölçmen, Sharif, & Presdorff, 2013). Bassi, Crivellini, Dossena, Franchina, and Savini (2014), and Li, Gao, and Yang (2013) both presented turbulent flow characteristics of valve clearance. It is found that velocity gradient gradually spreads from the center to sides of valve clearance, and the strong shear force and erosion action of fluid have effects on the clearance wall, especially on the corners of valve plate and seat. Masjedian Jazi and Rahimzadeh (2009) and Jiang, Wu, Wang, and Xu (2011) studied the distribution and migration of low-pressure area in valve clearance. It has been proved that the vortex and energy dissipation are much more obvious when valve’s lift is smaller. All of the above literatures could provide good theoretical and analytical basis for the study on combined valves of internal-compressed reciprocating multiphase pump.

According to the researches, pump valves are mostly liquid valves or uniform gas-liquid valves, and compressor valves are gas valves at present. However, under multi-factor influences, the motions of combined valves are not exactly the same as that of common pump valve or compressor valve.

On one hand, the vibration forms of combined valves are affected greatly by the characteristic waveform of time-frequency domain in reciprocating multiphase pump (Pei et al., 2016). That is, research on the opening-closing motion of combined valves can simulate suction and discharge process more realistically under actual rotation of crank shaft in reciprocating multiphase pump. On the other hand, in view of the complexity of operation conditions, especially gas volume fraction, the motion of combined valves might face great uncertainty and instability. It is valuable to study the effects of operating parameters on dynamic characteristics of combined valves.

In this paper, the mathematical model describing the motion of combined valves is established to analyze its working mechanism in reciprocating multiphase pump. The modeling of pump cavity and combined valves are done in reciprocating multiphase pump during numerical simulation. The CFD software package Fluent is used to simulate the whole stroke of reciprocating multiphase pump, supported by dynamic grid technique and user defined function (UDF). The research could provide foundations for design of high-efficiency multiphase pumping technology.

2. Working principle of combined valves

The combined valves are arranged at the top of cavity vertically in three-cylinder double-acting reciprocating multiphase pump. The structure of combined valves is shown in Figure 1. It mainly consists of suction and discharge valve plates, valve body, and accessories, such as springs, bolts, sealing rings, etc. The valve plates play the crucial roles in connecting multiphase pump with external pipelines. In valve body, the equispaced outer and inner circular runners provide suction and discharge passages, respectively.

The working process of combined valves is as follows: with the piston’s working, the pump cavity expands and the cavity pressure declines. Due to the downward total force of differential pressure, spool gravity, and spring force on suction plate, gas–liquid flow goes through the outer runners of valve body, and the suction valve begins to open. The suction plate is pushed down quickly to the maximum lift, and clings to the lift limiter below over time. As the rotational angle of crank shaft $\theta > 180^\circ$, the

![Figure 1. Structure of combined valves: (a) 2-D diagram; (b) 3-D diagram.](image1.png)
fluid is compressed in the pump cavity, and the suction valve begins to close and rises again under the action of opposite total force. After the suction valve closes completely, the suction process finishes and the mixture continues to be compressed at the bottom of combined valves. In the roles of outlet pressure, spring force, and spool gravity, discharge plate will remain closed until the cavity pressure increase to a certain value. Once the discharge process begins, gas and liquid flow through the inner runners and the discharge plate is pushed up to the above limiter in a short time. As \( \theta \) gradually increases to 360\(^\circ\), the discharge plate tends to be fully opened until the mixture almost flows out. When the cavity pressure is low enough, the discharge plate falls back, stays closing again, and the discharge process finishes. Then a suction-discharge working cycle is completed.

3. Mathematical model of gas–liquid valves’ motion

To describe the dynamic characteristics of pump valves, continuity equations and mechanics equilibrium equations need be established. Because the multi-hole gas–liquid mixer is installed in the front of multiphase pump, assumptions are as follows:

(a) initial gas volume fraction at the inlet is constant, and gas phase is uniformly distributed in the form of bubble flow;
(b) suction and discharge pressure both are constant;
(c) hydraulic loss and friction loss can be ignored in valve clearance due to its small proportion of pump cavity.

3.1. Continuity equations

In suction and discharge process, the cavity pressure changes with the rotational angle of crank shaft, and the volume of compressible gas–liquid mixture changes correspondingly. However, the fluid masses are identical whether the mixture is compressible or not (Zhang, Zhang, & Wang, 2009). So the continuity equations of reciprocating multiphase pump could be established in terms of the conservation of mass instead of that of volume flow rate.

In suction process, the masses of mixture in pump cavity at the moments of \( t \) and \( t + dt \) can be expressed respectively as

\[
M_t = (Ax_p + V_0 - V_1)\rho \\
M_t + dM_t = [A(x_p + dx_p) + V_0 - (V_1 + dV_1)](\rho + d\rho)
\]

where \( A \) is the cross-sectional area of piston; \( x_p \) is the piston displacement at the moment of \( t \); \( V_0 \) is the clearance volume; \( \rho \) is the average density of mixture in pump at the moment of \( t \), expressed as \( \rho = f(P) \); \( P \) is the average cavity pressure at the moment of \( t \); \( V_1 \) is the changing volume of mixture in pump caused by suction plate’s motion, expressed as

\[
V_1 = A_f h_1
\]

where \( A_f \) is the area of suction valve; \( h_1 \) is the lift of suction plate.

By means of subtraction of Equation (1) and Equation (2) and ignoring the higher-order infinitesimal items, the mass increment \( dM_t \) of mixture at \( dt \) time can be expressed as

\[
dM_t = (Ax_p + V_0 - V_1) d\rho - \rho dV_1 + A\rho dx_p
\]

The mass \( dM_{1t} \) of mixture flowing through the clearance of suction valve can be expressed as

\[
dM_{1t} = \varepsilon_1 C_s A_1 \sqrt{\frac{2|P_s - P|}{\rho_1}} \rho_1 dt
\]

where \( P_s \) is the suction pressure; \( C_s \) is the flow coefficient of suction valve; \( \rho_1 \) is the mixture density in suction valve; \( A_1 \) is the flow area of clearance in suction valve, expressed as

\[
A_1 = \pi d_f h_1 \sin \alpha
\]

where \( d_f \) is the equivalent diameter of suction plate; \( \alpha \) is half of cone angle, \( \alpha = 90^\circ \) for flat valve.

\[
\varepsilon_1 = \begin{cases} 
-1 & P > P_s \\
1 & P \leq P_s 
\end{cases}
\]

Based on mass conservation, the continuity equation describing mixture’s flow in pump cavity and suction valve can be expressed as

\[
(Ax_p + V_0 - V_1) d\rho - \rho dV_1 + A\rho dx_p
\]

Equation (8) can be simplified as

\[
\frac{dP}{dt} = \frac{\varepsilon_2 C_s \pi d_f h_1 \sqrt{2|P_s - P|\rho_1} + \rho A_{1f} (dh_1/dt)}{(Ax_p + V_0 - A_{1f}h_1)(d\rho/dP)}
\]

Similarly, the continuity equation describing discharge process can be expressed as

\[
\frac{dP}{dt} = \frac{\varepsilon_2 C_d (\sqrt{2}/2) \pi d_f h_2 \sqrt{2|P - P_d|\rho_2} - \rho A_{12} (dh_2/dt) - \rho A (dx_p/dt)}{(Ax_p + V_0 + A_{12}h_2)(d\rho/dP)}
\]
where \( C_d \) is the flow coefficient of discharge valve; \( d_{i2} \) is the equivalent diameter of discharge plate; \( h_2 \) is the lift of discharge plate; \( P_d \) is the discharge pressure; \( \rho_2 \) is the mixture density in discharge valve; \( A_{i2} \) is the area of discharge plate; \( \varepsilon_2 \) is the coefficient of discharge valve, shown as

\[
\varepsilon_2 = \begin{cases} 
1 & P \geq P_d \\
-1 & P < P_d 
\end{cases}
\] (11)

### 3.2. Mechanics equilibrium equations of valve plates

During the opening–closing process of valve plates, the forces on valve plates, such as inertial force, spring force, and differential pressure are considered to establish the mechanics equilibrium equations.

For suction valve,

\[
m_1 \ddot{h}_1 = (P_s - P)A_{i1} + m_1g - c_1h_1 - F_1
\] (12)

For discharge valve,

\[
m_2 \ddot{h}_2 = (P - P_d)A_{i2} - m_2g - c_2h_2 - F_2
\] (13)

where \( m_1, m_2 \) are the masses of suction and discharge plates, respectively; \( c_1, c_2 \) are the stiffness of suction and discharge springs; \( F_1, F_2 \) are pre-tightening forces of suction and discharge springs; \( g \) is gravitational acceleration. In this paper, the maximum lifts of suction and discharge valves are limited in reciprocating multiphase pump.

Based on the theoretical analysis, 3-D geometric model of pump cavity and combined valves is established to do numerical study in the paper. User defined function is used to define the valves’ motion.

### 4. Dynamic simulation of combined valve

#### 4.1. Geometric model

For double-acting reciprocating pump, the working of each pump cavity is independent which is identical to that of single-acting reciprocating pump. So only a set of pump cavity and combined valves are studied within a stroke of piston.

The geometric model of pump cavity and combined valve is established according to basic structural parameters (shown in Table 1). Due to the symmetrical cavity and combined valves, half of their geometric model is adopted in simulation. On the premise of ensuring little effect on simulated values, grids’ remeshing, and calculation speed could be accelerated.

For combined valves, suction valve is flat valve, and discharge valve is cone valve. When discharge stroke finishes, the flow zone of reciprocating multiphase pump is shown in Figure 2.

| Parameters                          | values           |
|-------------------------------------|------------------|
| Inner diameter of cylinder block \( D \) (mm) | 115              |
| Length of stroke \( S \) (mm)         | 90               |
| Crank radius \( r \) (mm)            | 45               |
| Length ratio of connecting rod \( \lambda \) | 1/8              |
| Angular velocity of crank \( \omega \) (rad / s) | 8 \pi            |
| Clearance volume \( V_0 \) (mm\(^3\)) | \( 1.9 \times 10^5 \) |

#### 4.2. Grid generation

The grids of geometric model are conducted by the CFD pre-processing software, Gambit. Due to the complex structure, multiple grids are adopted in the whole model. Specifically, the flow regions near suction and discharge valve plates are discretized by tetrahedral grids, while the hexahedral structured grids are used in the other regions. The detailed grids around suction valve and discharge valve are shown in Figure 3.

Along with the piston and valves’ motion, the grids of changing fluid domain are rezoned in each time step by using dynamic grid technique. The motion of piston can be obtained by importing the geometric and motion parameters through ‘In-cylinder’ option. The motions of surfaces intersecting with valve plates and the end and side faces of piston are defined as those of corresponding deformation. The grid interface and remeshing technologies are used in simulation.

#### 4.3. Numerical method

The standard commercial CFD code, Fluent, is employed. The coupled, implicit calculation option, standard \( k-\varepsilon \) turbulence model and PISO algorithm are chosen as
numerical approaches. The method of standard wall function is used to deal with the boundary conditions near solid. From the results of CFD simulation, the value of $y^+$ is about 53 which is in the reasonable range of 30 to 300. So that it is proper to adopt $k$-$\varepsilon$ turbulence model and wall function in the study.

For multiphase flow, the numerical simulation method can be divided into three categories: (1) continuous medium mechanics method, which studies the law of multiphase flow migration at the macroscopic level; (2) molecular dynamics simulation method, which is based on the statistical molecular dynamics; (3) the lattice–Boltzmann method at the mesoscopic level. The most-used first method in engineering application includes Euler–Lagrangian and Euler–Euler methods (Baltussen, Kuipers, & Deen, 2017). And the Euler–Euler method is usually adopted to simulate the gas–liquid flow with wide range of gas content. In reciprocating multiphase pump, the gas volume fraction changes constantly with cavity pressure, and the two-phase coupling and pulsation characteristics need to be considered. Compared with the VOF model and Eulerian model, the Mixture model is more suitable for studying the complex two-phase flow in reciprocating multiphase pump.

In addition, User defined function (UDF) is adopted to define valves’ motion before the iteration of each time step. The resultant forces on valve plates are calculated first, and valves’ velocity and lift at current time step are obtained on the basis of those at previous time step, with the flow chart shown in Figure 4. The rigid motions of valve boundaries are defined further, which is written to a specified file for next time step.

According to the operational environment of Changqing oilfield in China (Zhao, 2012), the liquid and gas phase are set as crude oil and methane respectively, with the parameters shown in Table 2.

The numerical simulation is conducted under the initial conditions of suction pressure $P_s = 0.40$ MPa, discharge pressure $P_d = 3.0$ MPa, and gas volume fraction $GVF = 0.5$. Based on the hole size of the inlet mixer in front of reciprocating multiphase pump, the bubble diameter of gas is set as 2 mm in the simulation.
And the step size of crank angle is 0.5° in calculation. According to the piston stroke, connecting rod length, and crank shaft speed, the time step size could be obtained as 0.000347 s. The convergence criterion of energy residual is set as $10^{-6}$, while those of other variables’ residuals are set as $10^{-3}$.

4.4. Validation of the results

4.4.1. Verification of grid independence

It is important to investigate the effect of grid size on the results of simulated model (Khandelwal, Dhiman, & Baranyi, 2015). The test of grid independence is carried out by studying the characteristics of flow rates in multiphase pump.

Under $P_s = 0.40$ MPa, $P_d = 3.0$ MPa, and $\text{GVF} = 0.5$, the instantaneous flow curve of one stroke in a working cavity is simulated, which could be extended and superimposed to obtain that of three-cylinder double-acting multiphase pump, plotted in Figure 5.

So the average flow rates of simulated results could be got further by using the same types of grids with four different numbers. The details are shown in Table 3. It is found that the simulated results have little deviations for the last three grids. In view of simulation speed and precision, the model of $23.4 \times 10^4$ girds is used to generate the rest of the results presented in this paper.

![Figure 5. The instantaneous curve of flow rate in a three-cylinder double-acting reciprocating multiphase pump.](image_url)

![Table 3. Average flow rates of multiphase pump with different grids.](table_image)

| Number | grid number $N$ (10⁴) | average flow rate $q_s$ (m³·h⁻¹) |
|--------|-----------------------|----------------------------------|
| 1      | 10.9                  | 58.1                             |
| 2      | 23.4                  | 65.0                             |
| 3      | 41.1                  | 65.9                             |
| 4      | 46.2                  | 66.3                             |

4.4.2. Comparison with actual results

To verify the feasibility of numerical method, the average flow rates are compared with that of actual results from the factory tests of Pump 3DP90/3.0.

The test system is shown in Figure 6. The gas phase is supplied by the compressor 1, and the liquid phase is transported by the rotor pump 6 from line 18. The two phases flow through the mixer 10 before entering into the reciprocating multiphase pump 13, which is driven by three-phase electric machine with frequency converter 12 to adjust the test speed of pump. Finally, the pressurized mixture flow out from line 17. The gas, liquid, and mixture circuits are all equipped with the pressure gauges, flow meters, and regulating valves.

The operation of test is referred to the requirements of SY/T 6534-2002 'Twin-screw oil-gas multiphase pump.' And according to the suction pressure, the gas volume fraction at the inlet of pump is controlled by regulating the flow rates of gas and liquid circuits before the mixer. At the rated speed, the discharge pressure of reciprocating pump is gradually increased to the rated value in testing.

The comparisons of simulated and actual results under different discharge pressures are shown in Figure 7. With
the increase of discharge pressures from 1.0 MPa to 3.0 MPa, the simulated values are generally lower than actual values. The causes may be that the gas volume fraction under multiphase condition is difficult to maintain precisely in prototype testing, and the simulated values of three-cylinder reciprocating multiphase pump are obtained by dealing with that of a working cavity. However, the changing trends with discharge pressure match well for two kinds of results, and the maximum deviation is less than 5%. Thus, the results in the current simulation are in good agreement with actual results.

5. Results and discussion

5.1. Pressure distributions on pump cavity and valves

Pressure distributions on pump cavity and valves are shown in Figure 8 under the initial conditions.

In suction process (Figure 8(a) and 8(b)), the mixture flow into the pump, thus the valve plate is pushed to the maximum lift and falls back until the rotational angle of crank shaft $\theta$ is near 200°. It is founded that the differential pressure between valve plate and piston end decreases from $6.4 \times 10^4$ Pa to $1.1 \times 10^4$ Pa, so the pressure in pump cavity become more uniform with the increase of $\theta$.

In discharge process (Figure 8(c) and 8(d)), the piston has direct act on the mixture, and the pressure on piston end is much larger than that on the surface of valve plate, which reaches the minimum on the lower corner. The differential pressure on these two surfaces remains about $3.1 \times 10^5$ Pa, which has small change with $\theta$.

Furthermore, the pressure distributions on the driving surfaces of suction and discharge plates are shown in Figure 9 and Figure 10. On direct-contact surfaces, the high-pressure regions both expand from the 10 or 8 circular runners to the whole surfaces. The relative non-uniform degrees of maximum differential pressure are calculated at different eight times of these two figures, which are 4.7%, 7.0%, 20.6%, 2.4% for the upper surface of suction plate, and 1.3%, 5.8%, 23.8%, 4.7% for the lower surface of discharge plate, respectively. That is, the differential pressure inside and outside the runners reach the maximum when valve lift $h_1$ or $h_2$ reaches its maximum (4.5 mm), and then the radial pressure differences weaken with $\theta$’s increase.

5.2. Basic motion process of valve plate

Take the opening process of discharge valve, for example. The curves of valve plate’s lift, velocity, and average cavity pressure are shown in Figure 11 when GVF = 0.1.

After discharge valve begins to open, these three parameters increase gradually, but the corresponding rotational angles of crank shaft are not identical for the first peak values on the curves. Specifically, cavity pressure begins to decrease at $\theta = 201.6^\circ$, and valve velocity continues to increase until the force on valve plate is negative at $\theta = 202.5^\circ$, while valve lift reaches the maximum at $\theta = 205.1^\circ$.
Figure 9. Pressure distributions on the upper surface of suction plate with different lifts: (a) $h_1 = 0.5\, \text{mm}$, $\theta = 26°$; (b) $h_1 = 2.5\, \text{mm}$, $\theta = 30°$; (c) $h_1 = 4.5\, \text{mm}$, $\theta = 34°$; (d) $h_1 = 4.0\, \text{mm}$, $\theta = 200°$.

Figure 10. Pressure distributions on the lower surface of discharge plate with different lifts: (a) $h_2 = 0.5\, \text{mm}$, $\theta = 242°$; (b) $h_2 = 3\, \text{mm}$, $\theta = 244°$; (c) $h_2 = 4.5\, \text{mm}$, $\theta = 245°$; (d) $h_2 = 4.5\, \text{mm}$, $\theta = 330°$.

Then valve plate strikes lift limiter and rebounds at 0.2 times of initial velocity. With plate’s falling down, the force on valve plate changes from negative to positive so that valve lift increases again. On account of smaller and smaller strike velocity, valve plate vibrates for several times till velocity is attenuated to 0. It occupies $17.5°$ between the two peak values of valve lifts. The strike forms between valve plate and lift limiter are connected with gas-liquid properties, piston motion, striking velocity, valve structures, and so on.

5.3. Suction valve’s dynamic characteristics under different conditions

The effects of gas volume fraction, suction pressure and discharge pressure on dynamic characteristics of suction valve are shown in Figures 12–14.

5.3.1. Effects of gas volume fraction

It can be seen that the opening time of suction valve are generally shorter than the closing time except for GVF = 0.1. That is, the suction plate opens to the maximum within around 0.01 s. It is worth noting that the closing time of GVF = 0.9 is much larger than those of the others. Due to high gas volume fraction, it take up 1/12 time of one rotational cycle of crank shaft.

Meanwhile, the open–close beginning angles both lag behind the corresponding stroke of $\theta = 0°$ and $180°$. The opening angles of suction valve increase from $8°$ to $67.0°$ with gas volume fraction, while the closing angles firstly increase to the maximum and then decrease.

For the changes of suction valve’s velocity with gas volume fraction, the maximum opening velocities increase from 0.68 m/s to 1.0 m/s of GVF = 0.7, and then decrease to 0.63 m/s, while the maximum closing velocities decrease from 1.57 m/s to 0.21 m/s. It could be seen that the closing velocities are larger than the opening velocities when GVF $\leq 0.5$.

However, the striking velocities between suction plate, lift limiter and valve seat are not always same to the maximum velocities, especially for the opening process of GVF = 0.1 and the closing process of all the five conditions. And the velocity curves of GVF = 0.1 and 0.9 have apparent chattering phenomena at different times, which correspond to the variation of lift curves.
5.3.2. Effects of suction pressure

The changes of lift and velocity are more regular with suction pressure ranging from 0.20 MPa to 0.40 Mpa. The lag characteristics of suction valve are weakened to a certain extent, that is, the opening and closing angles of suction valves decrease from 35.4° and 232.5° to 24.0° and 207.8° respectively with suction pressure.

In addition, the strikes between suction plate and lift limiter occur at the time of maximum opening velocities, which increase from 0.62 m/s to 0.97 m/s with suction pressure. In the closing process of the suction valve, the maximum velocities increase from 0.36 m/s to 0.85 m/s, while striking velocities increase from 0.34 m/s to 0.73 m/s.
5.3.3. Effects of discharge pressure
Compared with gas volume fraction and suction pressure, the discharge pressures have little effect on the motion of the suction valve. Particularly, the changes of the closing process can be ignored with discharge pressure. The opening angles of suction valve increase from 19.5° to 24°, and the closing lag angles remain about 27.8° under different discharge pressures.

The striking velocities are same to the maximum opening velocities of suction valve, which increase from 0.84 m/s to 0.97 m/s with discharge pressure. The maximum closing velocity of suction valve and striking velocity between suction plate and valve seat are about 0.85 m/s and 0.73 m/s respectively.

5.4. Discharge valve’s dynamic characteristics under different conditions
The effects of gas volume fraction, suction pressure and discharge pressure on dynamic characteristics of suction valve are shown in Figures 15–17.

5.4.1. Effects of gas volume fraction
It is observed that the motion changes of discharge valve are much more apparent than those of suction valve with gas volume fraction. Due to the comprehensive actions of the fluid’s compression and clearance volume, the opening angle of discharge valve increases from 200.5° to 300° with the growth rate of nearly 50%. So, the lag characteristic is obvious in the opening process of discharge valve as gas volume fraction varies. It provides favorable conditions for concentration and compression in pump cavity.

Meanwhile, the occupied angles are less than 4° from the beginning to reaching lift limiter at the first time, and the opening times of the discharge plate are greatly shorter than those of the suction valve. Correspondingly, the maximum and striking velocities both are more than 3 times those of the suction valve in the opening process of the discharge valve, which reach the peak value when GVF = 0.3.

However, in the closing process of the discharge valve, the lag angles and striking velocities are approximately constant values, about 3° and 0.45 m/s respectively, which

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**Figure 15.** Lift and velocity curves of discharge plate as gas volume fraction varies.

**Figure 16.** Lift and velocity curves of discharge plate as suction pressure varies.
are little affected by gas volume fraction. So, the gas and liquid phases could flow out of pump cavity quickly and timely. Even if gas volume fraction fluctuates in a large range, the completion time of mixture’s discharge process are still relatively stable.

It is worth noting that after striking with lift limiter, the discharge plate rebounds and falls down with a certain height when GVF = 0.1 and 0.3, and the vibration times of the discharge plate are shortened rapidly. When GVF > 0.3, once the discharge plate reaches the surface of lift limiter, it would cling to it until beginning to close.

5.4.2. Effects of suction pressure
As suction pressure varies from 0.20 MPa to 0.40 Mpa, the difference of the opening angle is enlarged to 29.7° with a smaller decreasing gradient. The changing rules of maximum opening velocities for the discharge valve is precisely contrary to those for the suction valve. So, the striking velocities between the discharge plate and lift limiter decrease from 3.64 m/s to 2.86 m/s with suction pressure, which account for about 96% of maximum velocities.

As can be noticed, the lift curve has chattering phenomena only when $P_s = 0.25$ MPa. What’s more, its velocity curve appears as two obvious peaks with more than twice the falling velocities.

Similarly with gas volume fraction, suction pressure has little impact on the closing process of discharge valve.

5.4.3. Effects of discharge pressure
The opening process of discharge pressure has small change with discharge pressure. The opening angles of discharge valve increase from 232.2° to 240.6°, which have slightly larger gradients than those of the suction valve. Compared with this, the maximum opening velocities and striking velocities increase rapidly from 1.27 m/s and 1.2 m/s to 2.99 m/s and 2.86 m/s respectively.

It can be seen that the latter half curves coincide with each other under different discharge pressures, that is, the closing process of the discharge valve is nearly independent of the discharge pressure.

6. Conclusions
A mathematical model describing gas-liquid valves’ motion is established, and dynamical simulation of combined valves in a multiphase pump is investigated. The effects of gas volume fraction, suction pressure, and discharge pressure on valves’ motion are obtained by using the commercial CFD code Fluent. Important conclusions have been summarized as follows:

(1) Pressures on the piston end are different from those on valves’ surfaces both in the suction and discharge process. And the high-pressure regions expand from the circular runners to the whole surfaces of valve plates.

(2) After the valve plate strike lift limiter, the plate will stay motionless or rebound at 0.2 times of initial velocity, and vibrate for several times till the velocity is attenuated to 0.

(3) With the increase of gas volume fraction (0.1 ∼ 0.9), suction pressure (0.20 ∼ 0.40 MPa), and discharge pressure (1.0 ∼ 3.0 MPa), the closing lag angle of discharge valve remains 3°, and the striking velocity between discharge plate and valve seat remains 0.45 m/s, while the other angles and velocities change in different ways. Increasing suction pressure properly could help reduce the valves’ lag angles and striking velocities.

Disclosure statement
No potential conflict of interest was reported by the authors.
Funding

The authors gratefully acknowledge the Project 51406183 supported by National Natural Science Foundation of China.

References

Baltussen, M. W., Kuipers, J. A. M., & Deen, N. G. (2017). Direct numerical simulation of effective drag in dense gas-liquid-solid three-phase flows. *Chemical Engineering Science*, 158, 561–568. doi:10.1016/j.ces.2016.11.013

Bassi, F., Crivellini, A., Dossena, V., Franchina, N., & Savini, M. (2014). Investigation of flow phenomena in air-water safety relief valves by means of a discontinuous Galerkin solver. *Computers & Fluids*, 90, 57–64. doi:10.1016/j.compfluid.2013.11.021

Cao, F., Gao, T. Y., Li, S. S., Xing, Z. W., & Shu, P. C. (2011). Experimental analysis of pressure distribution in a twin screw compressor for multiphase duties. *Experimental Thermal and Fluid Science*, 35, 219–225. doi:10.1016/j.expthermflusci.2010.09.004

Jiang, M. L., Wu, J. Y., Wang, R. Z., & Xu, Y. X. (2011). Research on the control laws of the electronic expansion valve for an air source heat pump water heater. *Building and Environment*, 46, 1954–1961. doi:10.1016/j.buildenv.2011.04.003

Khandelwal, V., Dhiman, A., & Baranyi, L. (2015). Laminar flow of non-Newtonian shear-thinning fluids in a T-channel. *Computers & Fluids*, 108, 79–91. doi:10.1016/j.compfluid.2014.11.030

King, C. D., Ölcmen, S. M., Sharif, M. A. R., & Presdorff, T. (2013). Computational analysis of diffuser performance for subsonic aero- dynamic research laboratory wind tunnel. *Engineering Applications of Computational Fluid Mechanics*, 7, 419–432. doi:10.1080/19942060.2013.11015482

Li, B. R., Gao, L. L., & Yang, G. (2013). Evaluation and compensation of steady gas flow force on the high-pressure electro-pneumatic servo valve direct-driven by voice coil motor. *Energy Conversion and Management*, 67, 92–102. doi:10.1016/j.enconman.2012.11.004

Lieu, C. F., Chan, W. K., & Ooi, K. T. (2012). Experimental investigation of the reciprocating ball pump (RBP). *Medical Engineering & Physics*, 34, 1101–1108. doi:10.1016/j.medengphy.2011.10.016

Ma, Y., He, Z. L., Peng, X. Y., & Xing, Z. W. (2012). Experimental investigation of the discharge valve dynamics in a reciprocating compressor for trans-critical CO2 refrigeration cycle. *Applied Thermal Engineering*, 32, 13–21. doi:10.1016/j.applthermaleng.2011.03.022

Masjedian Jazi, A., & Rahimzadeh, H. (2009). Detecting cavitation in globe valves by two methods: Characteristic diagrams and acoustic analysis. *Applied Acoustics*, 70, 1440–1445. doi:10.1016/j.apacoust.2009.04.010

Pei, J. F., He, C., Lv, M. R., Huang, X. R., Shen, K. J., & Bi, K. L. (2016). The valve motion characteristics of a reciprocating pump. *Mechanical Systems and Signal Processing*, 66–67, 657–664. doi:10.1016/j.ymssp.2015.06.013

Pirouzpanah, S., Gadigopuram, S. R., & Morrison, G. L. (2017). Two-phase flow characterization in a split vane impeller electrical submersible pump. *Journal of Petroleum Science and Engineering*, 148, 82–93. doi:10.1016/j.petrol.2016.09.051

Räbiger, K., Maksoud, T. M. A., Ward, J., & Hausmann, G. (2008). Theoretical and experimental analysis of a multiphase screw pump, handling gas-liquid mixtures with very high gas volume fractions. *Experimental Thermal and Fluid Science*, 32, 1694–1701. doi:10.1016/j.expthermflusci.2008.06.009

Shah, A., Chughtai, I. R., & Inayat, M. H. (2013). Experimental study of the characteristics of steam jet pump and effect of mixing section length on direct-contact condensation. *International Journal of Heat and Mass Transfer*, 58, 62–69. doi:10.1016/j.ijheatmasstransfer.2012.11.048

Tanaka, T., & Tsukamoto, H. (1999). Transient behavior of a cavitating centrifugal pump at rapid change in operating conditions—Part I: Transient phenomena at opening/closure of discharge valve. *Journal of Fluids Engineering-Transactions of the ASME*, 121, 841–849. doi:10.1115/1.2823545

Wang, J., Zha, H. B., Mcdonough, J. M., & Zhang, D. H. (2015). Analysis and numerical simulation of a novel gas-liquid multiphase scroll pump. *International Journal of Heat and Mass Transfer*, 91, 27–36. doi:10.1016/j.ijheatmasstransfer.2015.07.086

Wu, D. Z., Wu, P., Li, Z. F., & Wang, L. Q. (2010). The transient flow in a centrifugal pump during the discharge valve rapid opening process. *Nuclear Engineering and Design*, 240, 4061–4068. doi:10.1016/j.nucengdes.2010.08.024

Yang, X., Hu, C. C., Hu, Y., & Qu, Z. C. (2017). Theoretical and experimental study of a synchronal rotary multiphase pump at very high inlet gas volume fractions. *Applied Thermal Engineering*, 110, 710–719. doi:10.1016/j.applthermaleng.2016.08.204

Yu, W., Dianbo, X., Jianmei, F., & Xueyuan, P. (2010). Research on sealing performance and self-acting valve reliability in high-pressure oil-free hydrogen compressors for hydrogen refueling stations. *International Journal of Hydrogen Energy*, 35, 8063–8070. doi:10.1016/j.ijhydene.2010.01.089

Zhang, H. X., Zhang, T. Z., & Wang, W. C. (2009). Influence of valve’s characteristic on total performance of three cylinders internal combustion water pump. *Chinese Journal of Mechanical Engineering*, 22(1), 1–4. doi:10.3901/CJME.2009.01.091

Zhao, Y. L. (2012). *Evaluation and research for Changqing crude oil nature* (Unpublished master’s thesis). Lanzhou University, Lanzhou, China.