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Influence of valve's lag characteristic on pressure pulsation and performance of reciprocating multiphase pump

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ABSTRACT

Reciprocating multiphase pump is the key equipment for the oil-gas transport technology in the petroleum industry. The opening and closing motions of suction and discharge valves directly affect the transport performance and stability of pump. In this paper, the whole working cycle of reciprocating multiphase pump is studied by using computational fluid dynamics (CFD) method. The combined actions of piston, suction and discharge valves are dealt with User Defined Functions and dynamic grid technique. Based on the instantaneous motions of valves, the lag characteristics of suction and discharge valves under different working conditions are analyzed. And the influences of lag angles on the pressure pulsation, P-V diagram and input power of the pump are further investigated. The results show that the lag angles have a significant impact on the internal and external characteristics of the reciprocating multiphase pump. Due to the most obvious opening lag angle of discharge valve, the pump pressures and P-V diagram have the sudden fluctuations just after the end of compression process, which appear in the range of $\theta = 240.7^\circ$–273.1° for the initial condition. The gas volume fraction (GVF) plays more important role than the suction and discharge pressures in the working of reciprocating multiphase pump, and the opening lag angles of suction and discharge valves at GVF = 0.9 increase to more than 6 times at GVF = 0.1. By the comparisons of input power, the simulated results for reciprocating multiphase pump show good agreement with the results of prototype test. This study would provide a theoretical basis for the design of high-efficiency multiphase pump and valve.

1. Introduction

With the exploitation and development of offshore oilfields, multiphase transport technology has been increasingly used in the petroleum and natural gas industries (Mohammadzaheri et al., 2016; Kim et al., 2015). It has obvious advantages in reducing the pipelines and investment and enhancing the gas recovery and economic efficiency (Zhang et al., 2015, 2016). Currently various types of multiphase pumps have been developed by the research organizations, oil and gas companies and pump manufacturers, such as axial-flow multiphase pump, screw multiphase pump, rotary multiphase pump, peripheral multiphase pump, etc. And the first two types are the most frequently employed multiphase pumps (Yin et al., 2017; Yang et al., 2017; Pirouzpanah et al., 2017). The high gas content and changing working environments are the huge challenges for the stable operation of multiphase pump (Ganat et al., 2015; Hoehbusch et al., 2015). It is required that the multiphase pump should have dual characteristics of the pump and compressor, and allow the working fluid to be transported under the complex conditions.

Reciprocating multiphase pump is a new type of the internal-compression multiphase pump, which could continuously transport the oil-gas mixture with wide range of gas volume fraction (GVF). The oil-gas pumping is realized by the reciprocating motion of piston and the opening and closing of combined valves. The processes of two-phase flowing into or out of the pump cavity are largely related to the motions of valves. Thus, the combined valves, including the suction and discharge valves, are the key components affecting the transport reliability and efficiency of reciprocating multiphase pump.

Previous researches have provided an insight into the transport characteristics of valve and reciprocating pump or compressor (Morriesen and Deschamps, 2012). Based on the U. Adolph theory or traditional valve theories, the studies of valve' s dynamic motion have been conducted by means of various experiments and numerical simulations (Habing and Peters, 2006; Pei et al., 2016; Barbi et al., 2016). It is believed that the CFD modelling could offer accurate prediction of the

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internal flow in the valves of reciprocating pump and compressor. Through the fluid mechanics and working performance, the lag phenomenon could be generally observed in the reciprocating pump and compressor (Liu et al., 2016). That is, the valve plate is opened or closed at certain angles lagging behind the piston's motion.

The lag characteristic, on the one hand, affects the sensitivity of valve plate, and increases the impact force of valve plate against the limiter or seat, on the other hand, makes the cavity pressure increase and change suddenly (Stiaccini et al., 2014, 2017). It is likely to cause the valve failure, strong vibration, noise and pressure loss of the equipment, which in turn reduces the volumetric efficiency of the pump or compressor, especially for the compressor with gas phase (Farzaneh-Gord and Khoshnazar, 2016). Therefore, the similar lag phenomenon must appear in the reciprocating multiphase pump. However, it is not exactly the same as the pure liquid or gas valve for the oil-gas coupling flow. And the dynamic pressure and performance of multiphase pump are influenced further under different conditions such as the GVF and operation pressures (Rigola et al., 2005). So more detailed lag characteristics and the corresponding performance and stability are required to be studied for the reciprocating multiphase pump.

Therefore, this paper aims to investigate the whole working cycle of reciprocating multiphase pump by using CFD method and test validation. On the theoretical basis of piston and valves, the lag characteristics of reciprocating multiphase pump are obtained through the opening and closing processes of suction and discharge valves. The dynamic pressure ripples and external performance of reciprocating multiphase pump are analyzed further. Thereafter, the influences of suction/discharge pressure and GVF on the working performances are studied in the reciprocating multiphase pump.

2. Theoretical analysis

The schematic diagram of main hydraulic end is shown in Fig. 1 for the reciprocating multiphase pump. The associated actions of multiple components, especially the motions of piston and valves, affect the working of reciprocating multiphase pump directly.

2.1. Mathematical model for piston and valves

2.1.1. The motion of piston

During the operation of multiphase pump, the reciprocating motion of piston is driven by the rotation of prime mover through the crank link mechanism. The corresponding displacement and velocity of the piston could be expressed as:

$$x = r\left(1 - \cos\theta + \frac{\lambda}{2}\sin^2\theta\right) \tag{1}$$

$$v = \frac{dx}{dt} = \omega r\left(\sin\theta + \frac{\lambda}{2}\sin 2\theta\right) \tag{2}$$

Where x is the piston displacement; r is the crank radius; θ is the crank angle; λ is the link ratio; v is the piston velocity; ω is the crank angular velocity.

According to the structural parameters shown in Table 1, the curves of piston's displacement and velocity could be obtained in the reciprocating multiphase pump, as shown in Fig. 2.

2.1.2. The motions of combined valves

The combined valves mainly include the valve body, upper discharge valve and lower suction valve, and the working diagram of suction and discharge valves are shown in Fig. 3. The valves are located above the pump cavity to gather and pressurize the fluid and realize the high internal compression ratio. And the valve structures of double flow runners are designed with the uniformly-distributed suction and discharge runners in the outer and inner rings of valve body respectively, which could improve the gas-liquid suction behavior and reduce the energy waste.

In the opening-closing process of suction valve or discharge valve, the mass increment of mixture in the pump cavity with the displacement of piston x is equal to the mass of mixture flowing into or out of the pump through the valve clearance per unit time according to the law of conservation of mass. In addition, the valve plate is driven by the total force of inertial force, spring force and pressure difference of upper and lower surfaces on the plate, and the valve lift h changes nonlinealy with the time t. So the flow continuity equations and valves' force equilibrium equations could be established as follows.

For the suction valve,

$$\begin{cases} Ax + V_0 - V_1 d\rho - \rho dV_1 + A\rho dx = \varepsilon_1 C_{v1} A_1 \sqrt{\dfrac{2|P_s - P|}{\rho_{v1}}}\rho_1 dt \\ m_1 \ddot{h}_1 = (P_s - P)A_{f1} + m_1 g - c_1 h_1 - F_1 \end{cases} \tag{3}$$

Table 1
Structural parameters of crank-link mechanism.

Parameters		values
Length of stroke	S (m)	0.09
Crank radius	r(m)	0.045
Link ratio	λ	1/8
Angular velocity of crank	ω (rad/s)	8π

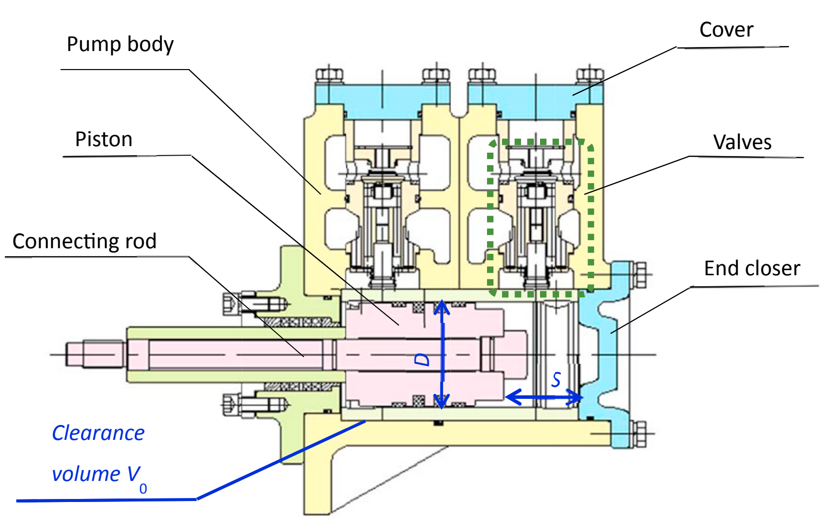

Fig. 1. Schematic diagram of hydraulic end.