Prediction on the operation performance of axial piston pump at low suction pressure

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Abstract: The operation performance of the axial piston pump is affected by external environment pressure, and its conditions at low suction pressure are less studied. The oil model and fluid dynamic model of the axial piston pump are developed and a novel approach for predicting the operation performance of such machines at low suction pressure based on this model is presented. It was found that the inlet pressure of the suction chamber changes linearly with the environment pressure. Compared to the present data from the product manual, this approach is proved to be valid and is used to predict the maximum operating speed and suction pressure at other environment pressure, therefore the low-pressure operating performance map is enriched for the referring of more users of the pump. In future, this study provides a new method for the prediction and the design guide of pump operation at low suction pressures.

Keywords: axial piston pump; low suction pressure; operation performance; maximum operating speed; compressible liquid; cavitation.

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1 Introduction

Axial piston pump is an important energy conversion component of hydraulic system. It also affects the operation performance and noise release of the whole system. Axial piston pump has the advantage of simple structure, compact size, high reliability and high operating pressures, which permitting the wide use in hydraulic machinery. Scholars have studied the modelling of axial piston pump. In the earlier research, Zhai (1978) introduced the modelling method of swash plate axial piston pump. Edge (1988) and Edge and Darling (1986) adopted a lumped parameter approaches based on fluid control volume, and improved the flow characteristic model of single piston by introducing the influence of inertia term of fluid in damping groove. Ivantysyn and Ivantysynova (2000) systematically introduced the change of oil in piston chamber of the axial piston pump. The compressibility of fluid (Jiang and Zhou, 2013) and the dynamic evolution of bubbles (Yuan et al., 2015) are also considered in the later modelling process. More accurate model of axial piston pump is developed by Pelosi and Ivantysynova (2012), which can realise the coupling between 1D model of the whole pump and 3D sub model of important friction pairs of axial piston pump, thus the interaction between Thermal-Fluid-Solid multi-physical fields can be obtained. Ye et al. (2018, 2019) used a novel valve plate utilising damping holes to reduce the noise and established a dynamic model with four masses and 19 degrees of freedom for axial piston pump to study its vibration characteristics.

However, operating conditions of the axial piston pump at low suction pressure are less studied. Chen et al. (2016) studied the operating speed in a range to obtain the pressure drop of discharge pressure. Results indicate that the discharge pressure has little effects on the pressure drop, whereas the speed has great effects on the pressure drop. Wang et al. (2017) obtained the fluctuated output pressure and flow rate characteristic at operating condition of low rotation rate. Too low a rotation speed makes the leakage larger, so that the outlet pressure cannot be stably established, and the pressure and flow fluctuations are significant. But the environment pressure of inlet is not involved. Vacca et al. (2010) established a model considering air release and vapour cavitation of axial piston machines, which can simulate conditions of insufficient flow at the delivery port due to fluid cavitation. The developed model allows the calculation of effective flow rate through the pump at fair and extreme conditions to predict the limitation of the machine.

To study the influence of environment pressure on the operating speed of axial piston pump, the operation performance of the compressible fluid and the motion characteristics of the axial piston pump are analysed theoretically. The fluid model and fluid dynamic model of the axial piston pump are established. A method for predicting the operation performance at low suction pressure is proposed based on this model, which reveals the speed condition of pump at low environment pressure. The reason of pressure drop is analysed theoretically.

2 Description of the simulation model

2.1 Model of the hydraulic oil

The existence of free air and oil vapour will change the characteristics of the fluid, including the density, bulk modulus and viscosity of the two-phase fluid. The fluid pressure is usually higher than the vapour pressure, which means the cavitation of the hydraulic machinery is mainly caused by the released air in the oil when the pressure is lower than the saturation pressure. Assuming that there is no infiltration between the fluid phases, the mathematical formulation for fluid properties are given by equations (1)–(3) according to the literature (Zhou et al., 2013).

$$\frac{1}{\rho}=\frac{x_{\mathrm{g}}}{\rho_{\mathrm{g}}}+\frac{1-x_{\mathrm{g}}}{\rho_{\mathrm{l}}} \tag{1}$$

ρ_1 is the density of pure oil, ρ_g is the density of free air in the oil, x_g is the mass fraction of air.

$$E=\frac{1}{\dfrac{\alpha_g}{\lambda p}+\dfrac{1-\alpha_g}{E_l}} \tag{2}$$

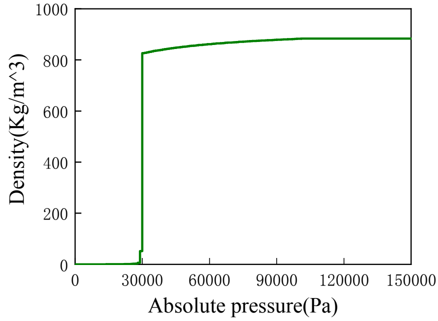
α_g is the volume fraction of air.

$$\mu=\alpha_g\mu_g+\left(1-\alpha_g\right)\mu_1 \tag{3}$$

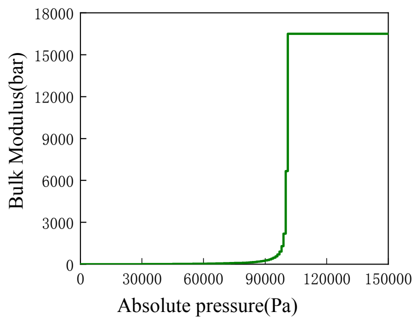
μ_g is the viscosity value of air, μ_1 is the viscosity value of pure oil.

There is usually a certain amount of air in liquid which cannot be ignored. According to Figure 1, when the environment pressure reaches vapour pressure, the fluid starts to liquefy from the vapour, resulting in a sharp increase in density and viscosity. When it reaches separation pressure, the released air from liquid results in a great change in bulk modulus.

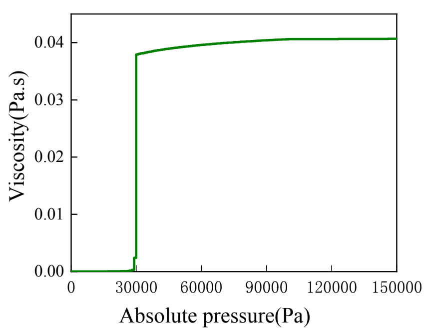
Figure 1 The change rule of the properties of no. 46 hydraulic oil: (a) density, (b) bulk modulus and (c) viscosity (see online version for colours)



(a)



(b)



(c)

2.2 *Fluid dynamic model*

Figure 2 illustrates the framework of fluid dynamic model based on lumped parameter approaches. The reciprocating motion of the piston in the displacement chamber makes the fluid flow from the suction chamber to the discharge chamber to complete the operating cycle of suction and discharge.

The periodical connection between the displacement chambers and the high pressure port (HP) and the low pressure port (LP) results in the control volume of the displacement chamber changes periodically. The volume of displacement chamber and the HP and LP are equivalent to control volume, the number of control volume is N + 2 in the fluid dynamic model.

The pressure in the control volume is:

$$\frac{dp}{dt} = \frac{E}{\rho V}\left(\sum \dot{m}_{in} - \sum \dot{m}_{out} - \sum \dot{m}_{leak} - \rho \frac{dV}{dt}\right) \tag{3}$$

Mass flow from the outside into the control volume, $\dot{m}_{in}$, mass flow from the control volume to the outside, $\dot{m}_{out}$, and leakage mass flow, $\dot{m}_{leak}$, are taken into account by the mathematical formulation.

Figure 2 Fluid dynamic model for simulation (see online version for colours)

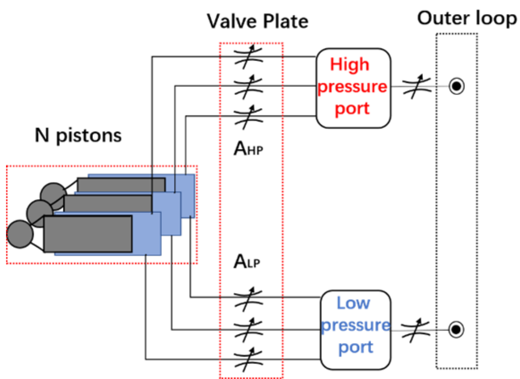
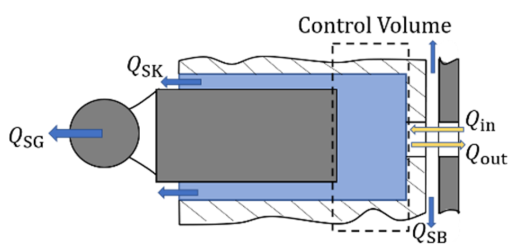


Figure 3 Displacement chamber pressure control volume (see online version for colours)



Flows enter and exit a control volume are shown in Figure 3. The leakage flow occurs between valve plate and cylinder block, Q_{SK}, piston and cylinder bore, Q_{SB}, slipper and swash plate, Q_{SG}.

The flow rate between control volumes can be calculated by orifice flow equation.

$$\dot{m} = C_q A \sqrt{2\rho \Delta p} \tag{5}$$

C_q is the flow coefficient, A is the flow area.

The lumped parameter model of axial piston pump is solved based on MATLAB. The pressure-flow rate characteristics of the pump can be obtained by solving N + 2 control volume pressure equations simultaneously.

2.3 Initialisation of simulation

Parameters of ISO VG46 hydraulic oil shown in Table 1 are used in the oil model. The fluid parameters and the structural parameters of the axial piston pump are shown in Table 2.

Table 1 Oil parameters

Density at standard atmospheric pressure (kg.m^3)	883
Bulk modulus at standard atmospheric pressure (bar)	16500
Viscosity at standard atmospheric pressure (Pa.s)	0.040

Table 2 Structural parameters of the axial piston pumps

Parameters	*Unit*	*HPR75*	*HPR135*	*HPR210*
Number of pistons		9	9	9
Angle of port plate	°	20.8	20.8	20
Piston diameter	Mm	19	23	27
Diameter of distribution circle	Mm	81.05	99.10	115.60
Suction port size	Mm	38	50	76.2
Volume of suction port cavity				389614.7
Pressure port size	Mm	19	32	38.1
Volume of outlet port cavity				192524.5

The number of pistons of axial piston pump, operating speed, environment pressure and load pressure are taken into account by the boundary conditions. The load pressure is 42 MPa.

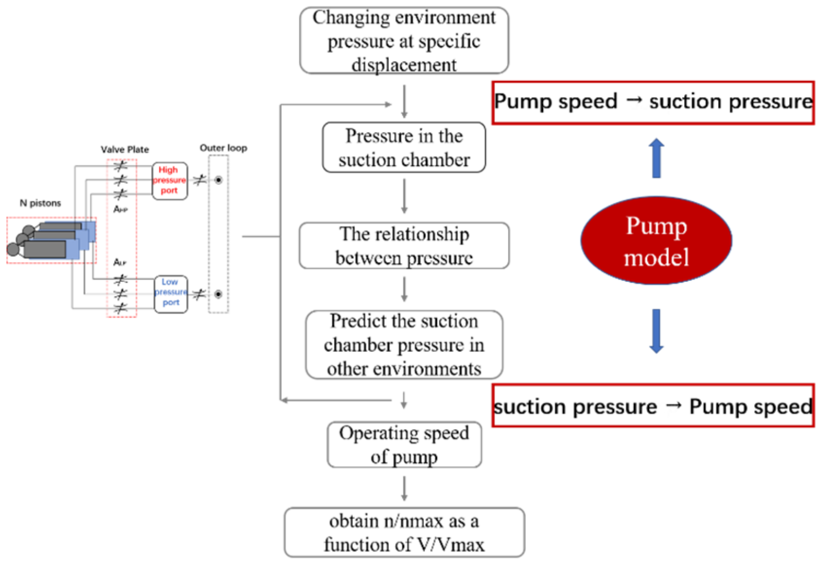
3 Method and validation

3.1 Research method

The pressure-flow rate characteristics of axial piston pump can be obtained by fluid dynamic model. The change of environment pressure directly affects the inlet pressure of suction chamber without considering the external circuit of hydraulic machinery. The low suction pressure can cause critical cavitation and affect the performance of hydraulic machinery. A novel approach is found for predicting the operation performance of machines at low suction pressure. According to the operating speed of the product manual of Linde Hydraulics at different environment pressure, the suction chamber pressure at specific displacement is calculated. Thus, the suction chamber pressure as a function of environment pressure can be obtained. The function can predict the suction chamber pressure at other environment pressures. According to the fluid dynamic model and the predicted suction pressure, the operating speed is calculated in reverse.

The principle of predicting the operation performance of axial piston pump at low suction pressure is shown in Figure 4.

Figure 4 Principle of performance prediction at low suction pressure (see online version for colours)

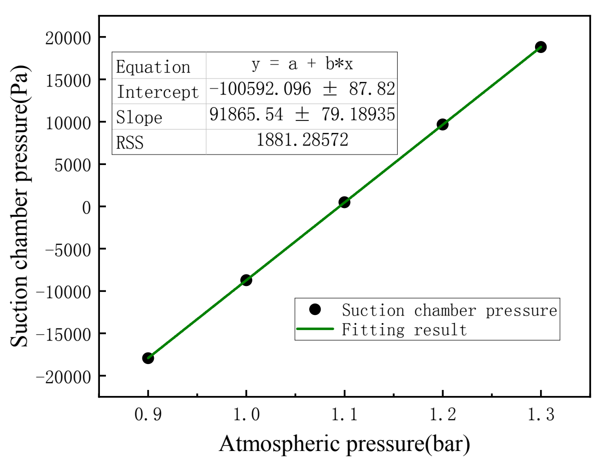


3.2 Model validation

Figure 5 illustrates the pressure in the suction chamber of three types of Linde hydraulic pump at environmental pressures of 0.9, 1.0, 1.1, 1.2, and 1.3 bar. It was found that the inlet pressure of the suction chamber changes linearly with the environment pressure.

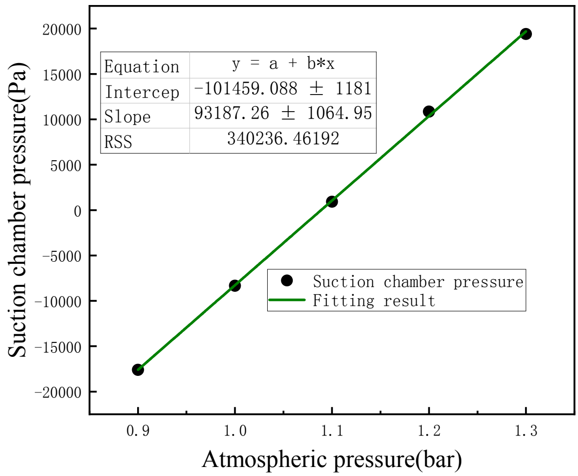
The suction chamber pressure as a function of environment pressure can be used to predict the suction chamber pressure at other environment pressure. According to the predicted pressure, the operating speed of the pump can be obtained.

Figure 5 Change rule of suction pressure: (a) HPR75; (b) HPR135 and (c) HPR210 (see online version for colours)



(a)

Figure 5 Change rule of suction pressure: (a) HPR75; (b) HPR135 and (c) HPR210 (see online version for colours) (continued)



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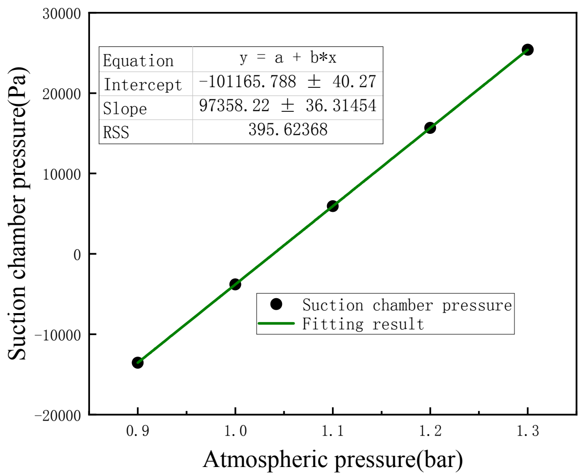


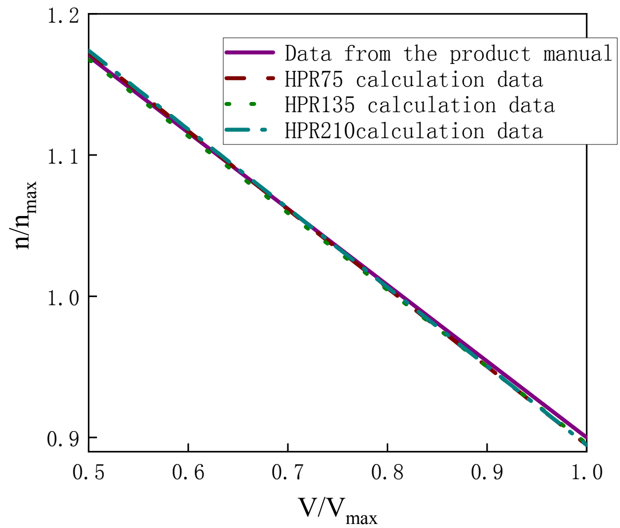
(c)

Figure 6 illustrates the comparison between the calculated operating speed and the speed of product manual at environment pressure of 0.8 bar.

The curve of normalised speed, $n / n_{\max}$, as a function of normalised displacement, $V / V_{\max}$ by the simulation model is consistent with the existing curve, thus proving the correctness of the simulation model and the research method.

Therefore, this model can further expand the prediction of the $n / n_{\max} - V / V_{\max}$ curve at other environment pressures. Enrich the low-pressure operating performance map for the referring of more users of the pump.

Figure 6 Comparison between data of simulation and data from the product manual (see online version for colours)



4 Application and discussion

4.1 Prediction of other suction pressures

Based on the models and the method, the $n/n_{\max} - V/V_{\max}$ curve at other environment pressure is predicted. The enriched low-pressure operating performance map is finally completed.

Figure 7 illustrates that when the environment pressure is reduced, the curve basically shows a parallel downward trend, so the diagram is still a set of parallel straight lines. But as the environment pressure decreases, its maximum speed decrease rapidly, and the decrease rate increases, which shows that the absorption capacity of the open pump rapidly deteriorate with low suction pressure.

4.2 The lowest environment pressure for normal operation

Environment pressure has restrictions on the use of hydraulic pump. When the environment pressure is too low, the hydraulic pump cannot reach the maximum continuous operating speed. The normal operating environment pressure of the hydraulic pump can be estimated according to the standard of maximum continuous operating speed, after predicting the environment and operating conditions of the hydraulic pump by the above-mentioned methods.

Figure 8 illustrates a comparison of Linde hydraulic pump HPR210 operating condition and maximum continuous operating speed at 0.8 bar and 0.6 bar environment pressure. The results show that at the environment pressure of 0.8 bar, the maximum continuous operating speed can be reached in the displacement range of 0–0.815. The application range of displacement is about 82%. And at the environment pressure of 0.6 bar, the displacement can reach the maximum continuous operating speed only in the range of 0–0.62. The application range of displacement is about 62%. Therefore, it is considered that 0.8 bar is more suitable than 0.6 bar as the lowest environment pressure that can be used normally without booster pump.

We can obtain the environment pressure and displacement range of the hydraulic pump in normal use by calculating the use capacity of the hydraulic pump, which avoid the damage to the hydraulic pump caused by unreasonable use conditions.

Figure 7 The enriched operating performance map (see online version for colours)

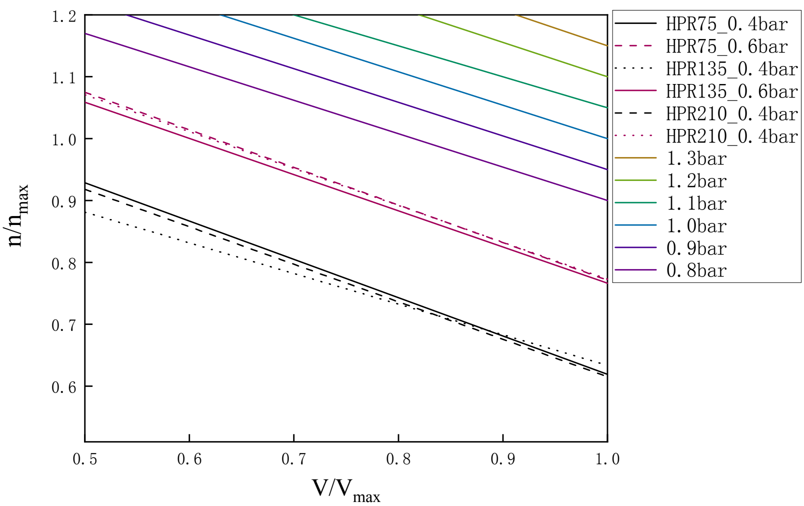
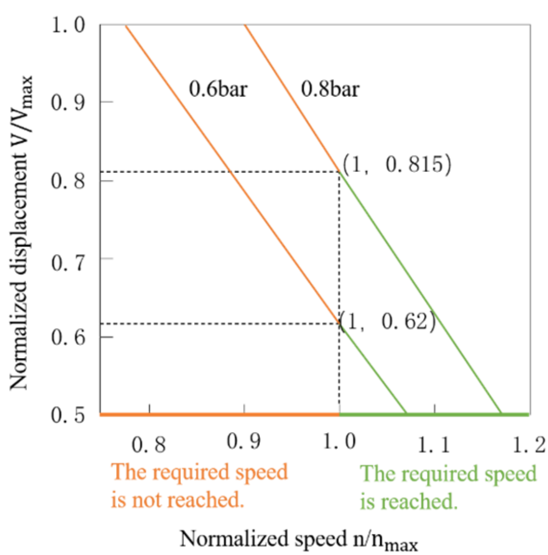


Figure 8 Comparison between operating conditions and maximum continuous operating speeds (see online version for colours)

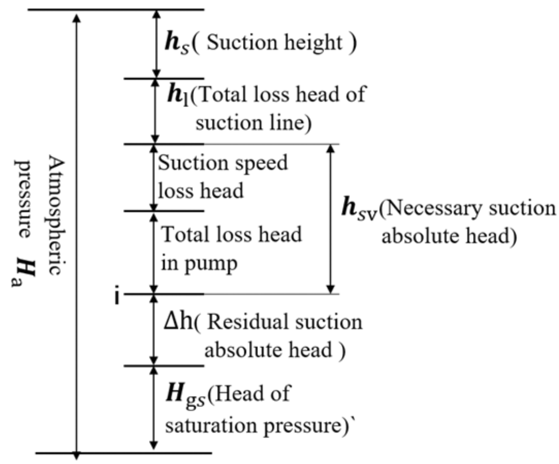


4.3 Theoretical explanation

In addition to the expansion of the enclosed volume caused by the barrier during the transition of the piston discharge stroke to the suction stroke, the pump's suction system can also cause cavitation (Zhai, 1978).

Figure 9 illustrates the pressure distribution of pump's suction system. Only when $\Delta H > 0$ is satisfied, the hydraulic machinery will not cause critical cavitation.

Figure 9 Distribution of the suction pressure of hydraulic pump



The pressure used in the paper is given by:

$$h_{min} = H_a - H_{gs} \tag{6}$$

Or

$$h_{min} = \Delta h + \frac{\omega^2 R_0^2}{2g}\left[\xi\left(\frac{\frac{\pi}{4}d^2}{A_1}\frac{R}{R_0}tg\beta\right)^2 + \zeta_0\frac{\frac{\pi}{4}d^2}{A_1}\frac{R}{R_0}tg\beta\right] \tag{7}$$

The saturation pressure is basically unchanged, and formula (6) shows that with the decrease of environment pressure, $h_{\min}$ decreased linearly. This can be attributed to Figure 5, where the suction pressure varies linearly with the environment pressure.

Because of the difference between the maximum operating speed (n) and the average radius of the distribution window of the valve plate (R_0) among three types of pumps, there are different suction pressures.

5 Conclusions

In this paper, the oil model and fluid dynamic model of the axial piston pump are developed and a novel approach for the predicting the operation performance at low suction pressure is presented. It was found the pressure in the suction chamber of the axial piston pump changes linearly with the environment pressure. As the suction pressure decreases, the operating speed decreases and the rate of decreasing increases, which indicates the suction performance deteriorates. The lowest environment pressure for normal operation is obtained. The method in the paper permit the prediction of operating conditions of piston pumps at different environment pressures. The method is versatile, which is instructive for the operation of piston pumps at low suction pressures.

References

Chen, L., Ye, S., Chang, Z., Gou, X. and Sun, K. (2016) 'Simulation research on the revolution-pressure performance of an aeronautical hydraulic axial piston pump', *2016: Proceedings of 2016 IEEE/CSAA International Conference on Aircraft Utility Systems (AUS)*, China, Beijing, pp.1195–1199.

Edge, K.A. (1988) 'A theoretical model of axial piston pump flow ripple', *1988: Proceedings of the First Bath International Fluid Power Workshop*, UK, Bath, pp.113–136.

Edge, K.A. and Darling, J. (1986) 'The pumping dynamics of swash plate piston pumps', *ASME Journal of Dynamic System Measurement and Control*, Vol. 111, No. 2, pp.307–312.

Ivantysyn, J. and Ivantysynova, M. (2000) *Hydrostatic Pumps and Motors, Principles, Designs. Performance, Modelling, Analysis, Control and Testing*, Academia Books International, New Delhi.

Jiang, D.I. and Zhou, H. (2013) 'Pressure and flow characteristic modeling of water hydraulic axial piston pump based on variable fluid properties', *Journal of the China Coal Society*, Vol. 38, No. 1, pp.171–176.

Pelosi, M. and Ivantysynova, M. (2012) 'Heat transfer and thermal elastic deformation analysis on the piston/cylinder interface of axial piston machines', *Journal of Tribology*, Vol. 134, No. 4, p.041101.

Vacca, A., Klop, R. and Ivantysynova, M. (2010) 'A numerical approach for the evaluation of the effects of air release and vapour cavitation on effective flow rate of axial piston machines', *International Journal of Fluid Power*, Vol. 11, No. 1, pp.33–45.

Wang, H., Zhu, K., Shi, G., Tanida, M., Hayashi, M. and Shinagawa, M. (2017) 'Performance of swash-plate type axial piston pump with different rotation rates', *Chinese Hydraulic & Pneumatics*, Vol. 41, No. 8, pp.100–106.

Ye, S., Zhang, J., Xu, B., Zhu, S., Xiang, J. and Tang, H. (2019) 'Theoretical investigation of the contributions of the excitation forces to the vibration of an axial piston pump', *Mechanical Systems and Signal Processing*, Vol. 129, pp.201–217.

Ye, S.G., Zhang, J.H. and Xu, B. (2018) 'Noise reduction of an axial piston pump by valve plate optimization', *Chinese Journal of Mechanical Engineering*, Vol. 31, No. 1, UNSP 57.

Yuan, S., Zhou, J., Luo, X. and Jing, C. (2015) 'Dynamic evolution and effects of gas phase in cavitation of axial piston pump', *Acta Armamentarii*, Vol. 36, No. 3, pp.559–565.

Zhai, P. (1978) *Design of Swashplate Axial Piston Pump*, 1st ed., Coal Industry Press, Beijing.

Zhou, J.J., Vacca, A. and Manhartsgruber, B. (2013) 'A novel approach for the prediction of dynamic features of air release and absorption in hydraulic oils', *Journal of Fluids Engineering*, Vol. 135, No. 9, p.091305.