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Research on the Hydraulic Pump Test System Based on AMESim

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***Abstract*— A set of hydraulic test system was designed and built to complete the performance test for pumps. Next, the simulation model of the system was built in the hydraulic simulation software AMESim, and the models of tested pump, relief valves and check valves were built by HCD library. According to National Standard JB/T 9714-1999, the necessary test items were simulated and the dynamic responses of the test system were analyzed. Simulation results show that the test system can exactly reflect the performance parameters of tested pump and meet the demand of hydraulic pump performance test. Finally, corresponding test bed was built according to the test system and verified the above simulation results. Experimental results were consistent with simulation results, which proves the feasibility of the test system.**

***Index Terms*— AMESim, hydraulic pump, simulation, test system.**

I. Introduction

Hydraulic pump is the power component of hydraulic system and its performance directly affects the property of whole system. The accurate measurement of pump performance is the important way to improve the property of hydraulic system. Yang Shangxian et al. design a set of test system for high-pressure plunger pump based on CAT technology [1]. Wang Xiangzhou et al. establish a novel mathematical model to predict the performance parameters of hydraulic pump which can save testing time and energy [2]. Zhang Liping et al. develop the hydraulic pump performance measurement system using the LabVIEW software [3]. Whereas common test systems for pump have the problems of low generalization level and measurement accuracy, large occupied area, difficulty in operation and high investment. According to National Standard JB/T 9714-1999 and the requirements of pump performance test, a set of hydraulic test system for pump was designed. The test range is that the pressure test range is from 0-45MPa, the flow test range is from 0-300L/min, the pump rotation speed test range is from 0-6000r/min and the maximum of driving torque of pump is 350Nm.

The simulation model of the hydraulic system was built by the software of AMESim. The simulation tests gave the test data and the results were analyzed.

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II. HYDRAULIC TEST SYSTEM

The principle of the test system is shown in Fig.1. The power unit consists of motor 5 and supply pump 4. Supply pump 4 supplies hydraulic oil for tested pump and the supply is more than the need of the tested pump, and the redundant oil overflows through electromagnetic relief valve 8. Electromagnetic relief valve 8 is used to adjust the oil pressure at the outlet of supply pump 4, electromagnetic relief valve 22 is used to adjust the oil pressure at the outlet of tested pump and load to tested pump. Directional valve 23 is connected in parallel with electromagnetic relief valve 22 to form an impact test unit, which can complete the impact test of tested pump by adjusting the directional valve 23 to switchover the pressure at the outlet of tested pump. Four check valves 9-12 compose the rectifying circuit unit to switchover the oil circuit of tested pump, which can improve the reversing stability, decrease the hydraulic impact and complete bidirectional test of tested pump. The torque and speed of tested pump is measured and exported by torque-speed transducer. Traditional pressure gauge and oval gear flowmeter are also used to make observation on the spot convenient.

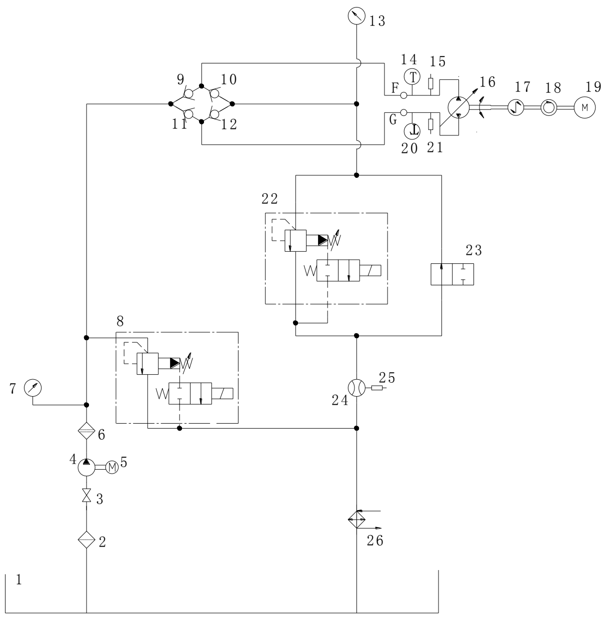


Fig.1 The principle of hydraulic pump test system
1 hydraulic cylinder 2 import oil filter 3 butterfly valve
4 supply pump 5, 19 motor 6 export oil filter
7,13 pressure gauge 8,22 electromagnetic relief valve
9-12 check valve 14,20 temperature sensor
15,21,25 pressure sensor 16 tested pump 17 torque sensor
18 speed sensor 23 directional valve 24 flowmeter
26 cooler

III. SIMULATION MODEL OF THE HYDRAULIC SYSTEM

The hydraulic library of AMESim concludes many general components which can meet demands in most cases. Whereas in order to obtain more accurate dynamics response of the test system, the simulation models in the test system was built using HCD library of AMESim according the parameters of actual selected tested pump and hydraulic valves [8]-[10].

Tested pump is built according to radial piston pump PFR-202 and the model is shown in Fig.2, which rated pressure is 50MPa, rated flow is 2.55L/min and rated speed is 1500r/min. Tested pump concludes three pistons which press on cam with the internal of 120 degrees respectively. Interface A and B are the inlet and outlet of the tested pump, respectively. According to the schematic diagram shown in Fig.1, the simulation model of the test system is built like Fig.3, in which the relief valves and check valves are built by HCD library according to actual structures. The interface A and B between check valves are connected with the interface A and B of tested pump, respectively.

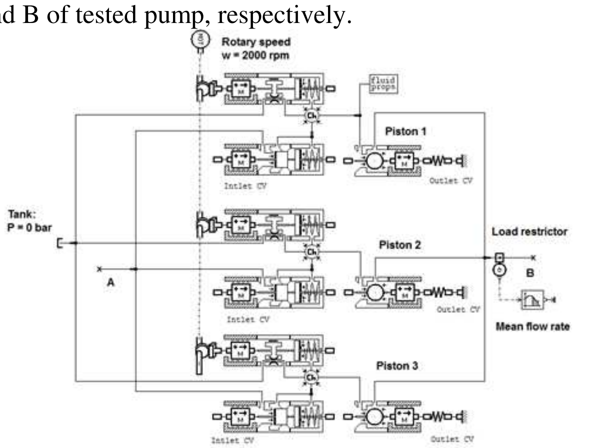


Fig.2 Simulation model of tested pump

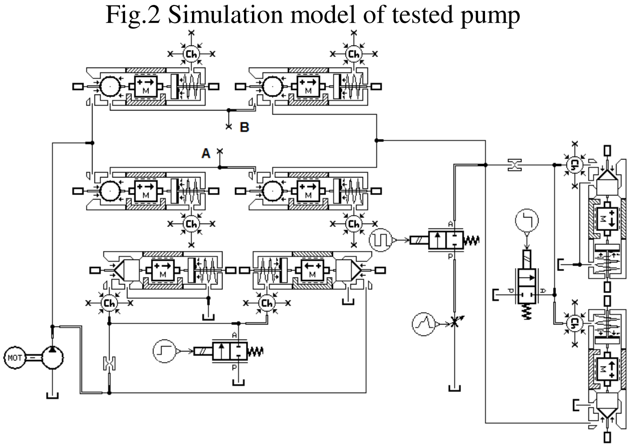


Fig.3 Simulation model of test system

Considering the accuracy of data and the ideality of tested pump, the print interval in simulation was set as 0.002s. The parameters of main components of the test system except for tested pump are listed in Table 1.

Table 1 Main simulation parameters

Items	Parameter setting
Displacement of supply pump/(ml/r)	2
Rated speed of supply pump/(r/min)	1500
The rated pressure of the supply pump/(MPa)	3
The critical pressure of check valves/(MPa)	0.05
The rated pressure of the tested pump /(MPa)	50

IV. SIMULATION RESULTS AND ANALYSES

Though changing related parameters of the system in simulation, the flow rates of tested pump with different speed in no-load condition were obtained and shown in Fig.4. The no-load displacement calculated according to formula (1) is 1.715ml/r, which is 100.9% of the nominal displacement of tested pump and meets the requirements of displacement verification test.

$$V_i = 1000\,\frac{N\left(\sum_{j=1}^{N} n_{e,j}\cdot q_{2,e,j}\right)-\left(\sum_{j=1}^{N} n_{e,j}\right)\cdot\left(\sum_{j=1}^{N} q_{2,e,j}\right)}{N\left(\sum_{j=1}^{N} n_{e,j}^2\right)-\left(\sum_{j=1}^{N} n_{e,j}\right)^2} \quad (1)$$

where V_i is no-load displacement, ml/r; N is the number of measured speeds; n_e is actual speed, r/min; $q_{2,e}$ is actual output flow rate, L/min.

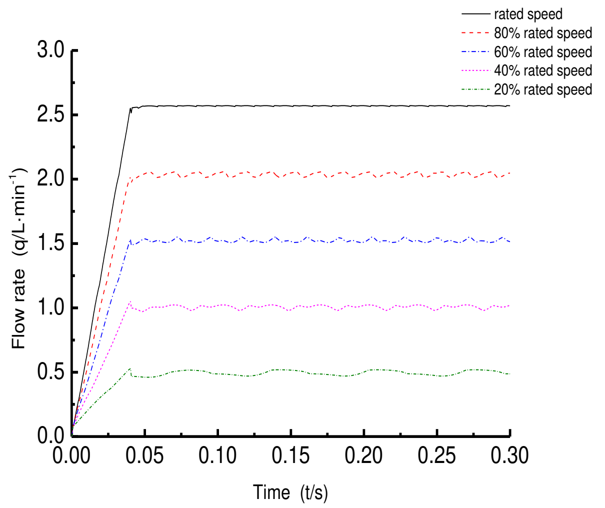


Fig.4 Result of displacement verification test

With the maximum displacement of tested pump, the speed of tested pump was set as 100%, 85%, 70%, 55%, and 40% of rated speed, the outlet pressure of tested pump was set from no load to 25%, 40%, 55%, 70%, 80%, and 100% of rated pressure, respectively. The tested pump data at each pressure point such as input torque, output flow rate were obtained through simulation. The power and efficiency of the points were obtained by calculating through following formulas:

Output hydraulic power

$$P_{2,h} = -\frac{p_{1,e}\cdot q_{2,e}}{60\,000} \quad (2)$$

Input mechanical power

$$P_{1,m} = -\frac{2\pi n_e T_1}{60000} \quad (3)$$

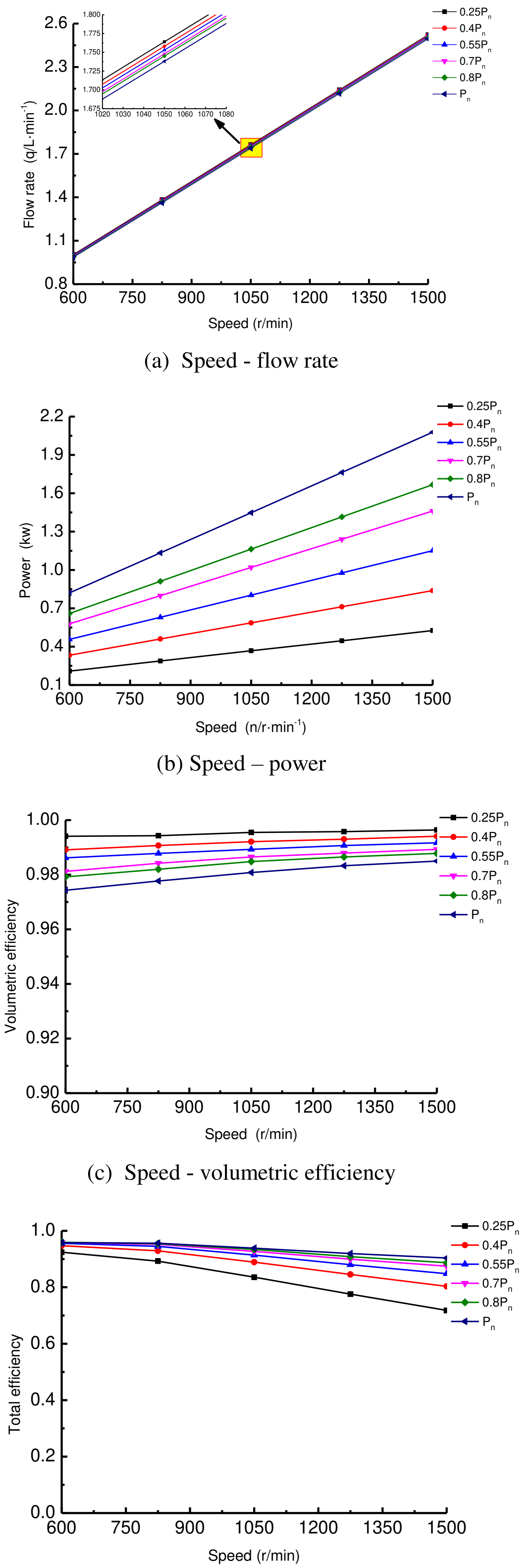
Volumetric efficiency

$$\eta_v = \frac{V_e}{V_i} = \frac{q_{2,e}/n_e}{q_{2,i}/n_i} \times 100\% \quad (4)$$

Total efficiency

$$\eta_t = \frac{p_{2,e}\cdot q_{2,e} - p_{1,e}\cdot q_{1,e}}{2\pi n_e T_1} \times 100\% \quad (5)$$

where $p_{2,e}$ is output pressure, kPa; T_1 is actual input torque, Nm; V_e is actual displacement, ml/r; $q_{2,i}$ is actual output rate with no load, L/min; n_i is the speed with no load, r/min; $p_{1,e}$ is input pressure(which is positive when higher than atmospheric pressure and negative when lower than atmospheric pressure), kPa; $q_{1,e}$ is output flow rate, L/min. The curves obtained by calculating are shown in Fig.5.



(a) Speed - flow rate

(b) Speed – power

(c) Speed - volumetric efficiency

(d) Speed - total speed

Fig. 5 the curves of tested pump parameters changing with speed

Fig.5 (a) shows the curves of tested pump flow rate changing with speed under different pressures. The output flow rate under any pressure increases with the rise of speed. But the increasing range decreases because the leakage increases with the rise of pressure. Fig. 5 (b) shows the curve of tested pump output power changing with speed under different pressures. According to formula (2), output power is the product of outlet pressure and flow rate. With the constant pressure, flow rate increases with the rise of speed, which makes the output power increasing. With the constant speed, the output flow rate is almost invariant while the power increases with the rise of outlet pressure. Fig.5 (c) shows the trends of volumetric efficiencies changing with speed under different pressures. According to formula (6),

$$\eta_v = 1 - \frac{\Delta q}{q_t} \tag{6}$$

where Δq is flow loss and qt is theoretical flow rate, Δq is proportional to outlet pressure and unrelated to speed theoretically and q_t increases with the rise of speed, which causes the volumetric efficiencies η_v under different pressures decrease with the rise of speed. Fig.5 (d) shows the trends of total efficiencies changing with speed under different pressures. Although volumetric efficiency increases with the rise of speed, the friction loss caused by relative sliding on piston inner surface increases with the rise of speed, which causes that mechanical efficiency decreases markedly, therefore the total efficiencies under different pressures decrease with the rise of speed.

The above characteristic curves show that the trends of flow rate, power and efficiency of tested pump obtained from the tested pump are constant with those given by test standard.

The tests of other items on hydraulic pump could also be operated by the test system built in AMESim. Other tests don't be described here due to limited space. Above simulation results and analyses show that the test system meets the requirements of pump performance test.

V.COMPARISON BETWEEN SIMULATION AND TEST RESULTS

Corresponding test bed was built based on the simulation and analysis of the designed hydraulic pump test system. The test system was divided into testing part and information collecting and processing part. Testing part includes pump test bed (shown in Fig.6) and valves control bed (shown in Fig.7).

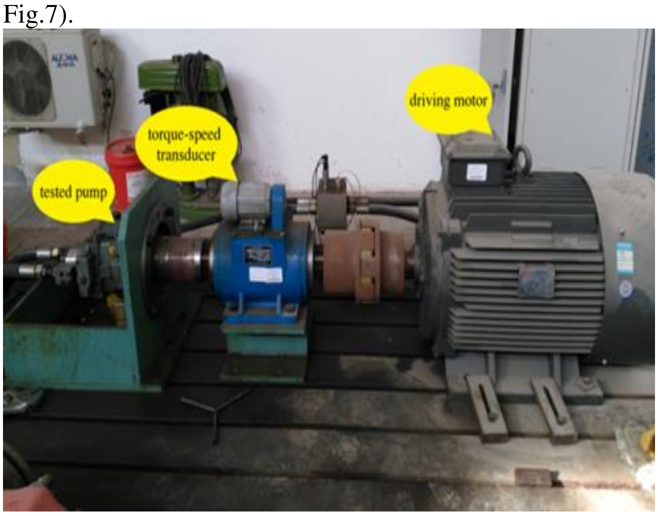


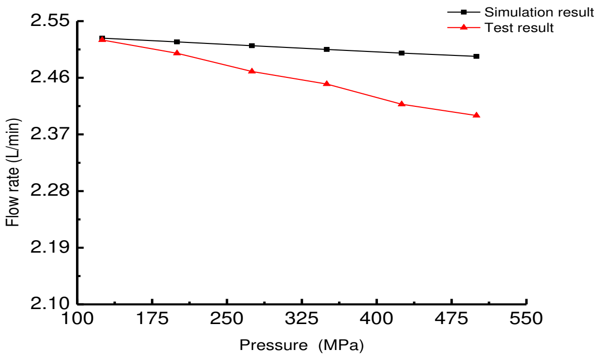
Fig.6 Pump test bed

Fig.7 valves control bed

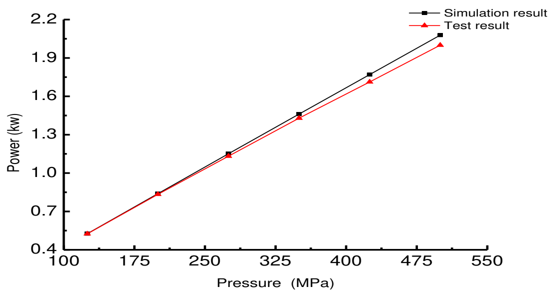
Information collecting and processing part is shown in Fig.8. The performance test was completed by operating the control cabinet. The pressure, flow rate, torque and other information produced in the testing process was collected by the signal acquisition cabinet which can show the information on the screen in real time and transmit it to the software in computer to calculate and process.

Fig.8 Information collecting and processing part

Only the data related to efficiency test are shown here because of limited length. Under rated speed, the outlet pressure of tested pump was adjusted at 6 bisectrices in the range from no-load pressure to rated pressure respectively. The flow rate, power, efficiency and other data were obtained by measuring and calculating and shown in Fig.9 and 10.



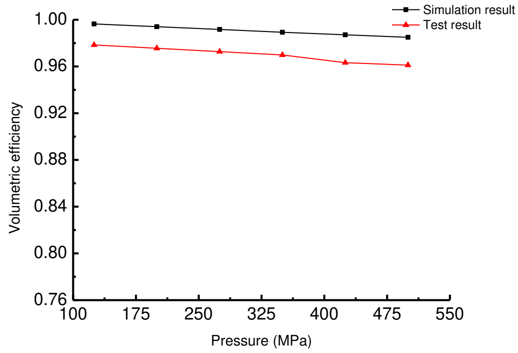
(a) Pressure - flow rate



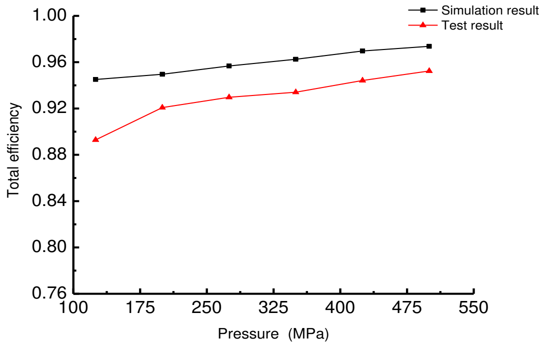
(b) Pressure- power

Fig. 9 Pressure - flow rate, Pressure - power

Fig.9 shows the curves of output flow rate and output power changing with pressure under rated speed, respectively. The leakage of tested pump is proportional to outlet pressure, therefore the actual output flow rate decreases with the increase of pressure under rated speed. But the change is small and the output power increases with the rise of outlet pressure. The variation of test result is the same as that of simulation result and the error is allowable, which proves the feasibility of the test.



(a) Pressure - volumetric efficiency



(b) Pressure - total efficiency

Fig. 10 Pressure - efficiency

Fig.10 is the curves of volumetric efficiency and total efficiency of tested pump changing with pressure under rated speed. Because of leakage loss and residual volumetric loss, the volumetric efficiency of radial piston pump decreases with the increase of outlet pressure [11]. Under low-pressure working condition, the proportion of mechanical friction loss is large and mechanical efficiency is low. With the increase of working pressure, the mechanical efficiency increases gradually and the change is larger than that of the decrease of volumetric efficiency. And it can be also known from formula (5) that the total efficiency increases with the rise of outlet pressure. The variation of test result is confirmed to that of simulation result and the error is allowable, which proves the feasibility of the test.

IV. CONCLUSION

The analyses of simulation results show that the test system can complete no-load displacement test, efficiency test and other tests of the pump and can obtain the performance parameters of tested pump accurately, which meets the requirements of pump performance test. The comparison between the actual test result from test bed and the simulation result from AMESim shows that the designed system has feasibility and can be applied to the test in actual working condition.

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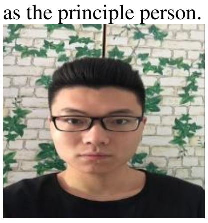

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