Journal of Advanced Mechanical Design, Systems, and Manufacturing

Vol. 3, No. 3, 2009

Investigation on the Cross Power Control of Axial Piston Double Pump Based on the Virtual Prototype*

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Abstract

In order to improve the power control flexibility of axial piston double pump and reduce energy consumption, a cross power control mechanism of double pump is developed. Comparing with the traditional summation power control and individual power control methods, the cross power control can distribute the power into two different individual pumps according to the sum of operating pressure of the two pumps. A virtual prototype of displacement adjusting mechanism of the double pump was established to analyze the control flexibility and energy saving ability of the cross power control. With this prototype model, the power distributions were calculated and analyzed at different operating pressures. Besides, the sensing control system of negative load was also simulated by the model as an additional function. And both the experiment results and simulation results were used to prove the performance of summation power control. It is concluded form the simulation and experimental analysis that the cross-power control has advantages in control flexibility, power distribution and energy-saving ability.

Key words: Cross Power Control, Virtual Prototype, Sensing Control of Negative Load, Double Pump, Piston Pump

1. Introduction

Axial piston double pump is widely used in the mobile and industrial machines due to their high working pressure and rotation speed, compact structure and high power density. The pump is a kind of bent axis design, and two axial piston rotary groups share one housing. The electric-motor drives one of the two axes in the rotary groups, the two axes is connected by two meshed gears, as shown in Fig. 1. So the double pump can supply two different pressures. Each output pressure can be determined by the respective load, but it is difficult to distribute the output power according to requirements of loads.

Fig. 1 The axial piston groups in the double pump

There are two kinds of traditional ways to distribute the output power, the summation power control and the individual power control. The summation power control is a pressure related pilot operated control way. The output power in summation power control is limited to a predefined value, so each output power is fixed as half of the predefined power. The

*Received 15 Dec., 2008 (No. 08-0893)
[DOI: 10.1299/jamdsm.3.277]

individual power control is a way that each output power is controlled by a separated constant power control system. The double pump is just like two individual pumps to meet requirements of loads respectively. It is obvious that both the two control ways have their disadvantages. As for the summation power control, the distributed power of each pump is 50% the sum power, when one pump use less than 50%, the rest power can′t be used. This control way is not flexible and always wastes lots of energy. As for the individual control way, two individual control devices make the system complicated, and the total power can′t be limited automatically by the control device.

Compared with the two traditional control ways, the cross power control shows its flexibility, energy saving and easy controllability. The total power is controlled by the sum pressure of the two pumps. When one output power of the pump is less than 50% of the total drive power, the other pump can automatically make use of the remnant power. So the cross power control combines the advantages of the two control ways above. Beside, this control way integrates the sensing control function of negative load.

Virtual prototype is a new engineering technology. Hydraulic virtual prototype technology integrates the advanced CAD modeling way and hydraulic simulation technology to forecast performances and study characteristics of a machine (1). Because of the complicated structure and nonlinear characteristics of the hydraulic-solid coupling, it is time-consuming and expensive with traditional try-and-error design way. And the analyzing results of traditional way are not accurate enough (2). The virtual prototype of piston pump uses such a real model, which is just like a real physics pump. The simulation of virtual prototype even can replace some tests of the physics prototype (3).

With the commercial hydraulic and dynamic software, a virtual prototype of piston pump was made by Atchen Technique University in 2002 (4). The hydraulic characteristics and frictions between the key tribo-pairs were analyzed (5) (6). In order to optimize the incline angle of the swash plate, a virtual prototype of a bent axis piston pump was made in 2003(7). The concept of virtual prototype of piston pump was proposed, the output pressure and flow ripple, the strain and stress of the key parts were all analyzed. It is very useful for the optimization of pump (8). In 2006, the flow ripple of a swash plate type piston pump was studied and optimized with the virtual prototype technology by Zhejiang University (3).

In order to investigate the cross power control, a virtual prototype of double pump was built in this paper. The virtual prototype technology is a new way to investigate performance of machine. As for the cross power control of the double pump, both the hydraulic model and dynamic model are needed. Based on the virtual prototype technology, the pressure, flow and output power of the double pump in different control process were studied. And experimental results were supplied to support the analysis.

2. Nomenclature

p_1 Pressure of port 1 at the double piston pump (Pa)
p_2 Pressure of port 2 at the double piston pump (Pa)
x Compressing displacement of spring (mm)
x_0 Pre-compressing displacement of spring 1 (mm)
x_1 Compressing displacement of spring 1 while spring 2 starts working (mm)
x_{pre} Pre-compress of the spring
x_{max} Maximum displacement of spring (mm)
k_1 Compound stiffness of spring 1
k_2 Compound stiffness of spring 2
C_1 Coefficient of piston area
S_1, S_2 Effective area of the piston in control cylinder (mm^2)
x_{set} Defined displacement of control spring
S_3 Acting area of the chamber 3 or 3′ (mm^2)

S_p Acting area of the output pressure (Pa)
q_{max} Maximum output flow (L/min)
S_4 Effective area in chanmber 4 or 4′ (mm^2)
S_5 Effective area in chanmber 5 or 5′ (mm^2)
S_0 When S_4=S_5, setting S_4=S_5=S_0 (mm^2)
p_{pre} Pre-defined pressure according to power requirement (Pa)
p_{xi} Feedback pressure from x_i , i=1,2 (Pa)
S_x Area of control spool in chamber (mm^2)
d_p Diameter of piston (mm)
h_{r1} Gap height between piston and cylinder (μm)
l_r Contact length between cylinder and piston (mm)
μ Dynamic viscosity of the fluid (Pa·s)
ε Eccentricity ratio of piston
p_f Pressure inside piston chamber (Pa)
p_0 Environmental pressure inside pump chamber (Pa)
λ Pressure ratio
h_{r2} Gap height between valve plate and cylinder block (μm)
r_1, r_2 Structural parameters of slipper (mm)
φ_1, φ_2 Angle structural parameters of valve plate (°)
R Distribution radius of piston (mm)
R_1, R_2, R_3, R_4 Radius structural parameters of valve plate (mm)
m_p Mass of the piston (kg)
B_p Viscous coefficient
k Compound stiffness of compound spring
W_{set} Defined control power (kW)
W_{sum} Sum of power (kW)
W_{output} Output power (kW)
s Displacement of piston (mm)
v Velocity of piston (mm/s)
a Acceleration of piston (mm^2/s)
t Time (s)
ω Rotation speed (r/min)
α Changing of angle of piston (°)
λ Pressure ratio

3. Principle of cross power control

The cross power control is designed for distributing the output power of double-pump. Its control principle is shown in Fig. 2. The two individual pump-units are driven by two meshed gears. Both the output pressure p_1 and p_2 act on the same area of two ladder spools. The hollow piston is driven by spring, which is used to build a relationship between the output flow and pressure.

The curve of the relationship between the output flow and pressure is shown in Fig. 3. When the compressing displacement x of spring is between the pre-compressing displacement x_0 of spring 1 and the inflexion displacement x_1+ x_0 of the curve, the spring 1 drives the control piston, so the output flow and pressure changes according to Eq. (1). When the control piston passes the point x_1+x_0, two springs are parallel connected to acts on the control piston. So the compound stiffness k of parallel springs is greater than the stiffness k_1 of spring 1, and the slope of the curve between x_1+x_0 and x_{max} is bigger than that between x_1+x_0 and x_0 in Fig. 3. These two connected beelines are approximate with the hyperbola, which is used to describe the constant power characteristics.

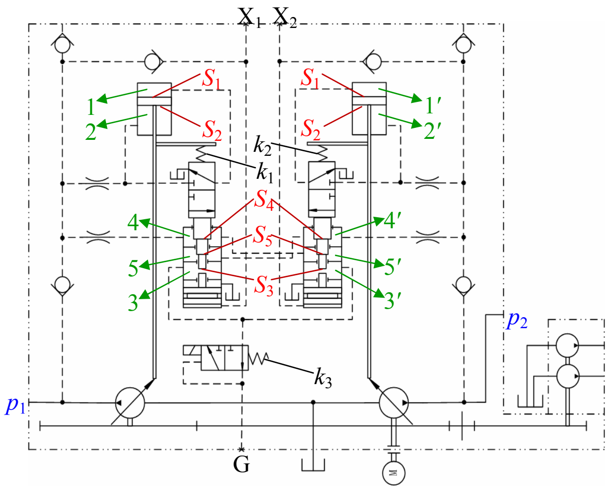


Fig. 2 Principle of cross power control system

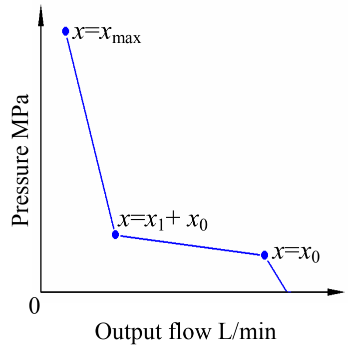


Fig. 3 The relationship between the output flow and pressure of individual pump

$$p_1(C_1S_1 - S_2) = k_1x_0 + \frac{x_{max} - x}{x_{max}}k_1 \quad (x_1{+}x_0{>}x{>}x_0) \tag{1}$$

$$p_1(C_1S_1 - S_2) = k_1(x_0 + x_1) + \frac{x_{max} - x}{x_{max}}k_2 \quad (x_{max}{>}x{>}x_1{+}x_0) \tag{2}$$

As shown in Fig. 2, the control chambers of 3 and 3′, 4 and 5′, 5 and 4′ are connected respectively, so each control piston are driven by the sum pressure of the three chambers. The pressure p_3 of chamber 3 and 3′ is a pilot control pressure, which is controlled by the proportional valve to change the constant power point. So the power point can be defined by the pilot pressure p_3 and a control spring k_3. The defined control power W_{set} is

$$W_{set} = \frac{(k_3x_{set} - p_3S_3)}{S_p}q_{max} \tag{3}$$

The control pressure p_1 in chamber 4 and 4′, p_2 in chamber 5 and 5′ are the output pressures of the two pump units. In stable condition of constant power, the control pressures are

$$p_1S_4 + p_2S_5 = k_3x_{set} - p_3S_3 \tag{4}$$

When S_4=S_5=S_0,

$$p_1 + p_2 = \frac{k_3x_{set} - p_3S_3}{S_0} = p_{pre} \tag{5}$$

There are two similar pump units in the double pump, so the sum of power W_{sum} of the double pump is

$$W_{sum} = (p_2 + p_2)q_{max} = p_{pre}q_{max} \tag{6}$$

It is obviously that the sum power of the double pump is distributed to two pump units (p_1q_{max} and p_2q_{max}) according to the output pressure (p_1+p_2) of each pump. Pressure p_1 and p_2 are determined by loads requirements. So when one pump unit use part of the total power to drive one load, the other pump unit can make full use of the rest power to drive another load automatically. So the drive power can be used effectively and reduce power wasting.

The pilot port X_1 and X_2 of flow feedback can be connected to respective pressure valves, which can supply a pressure signal respected to the flow changing. When the flow rate is greater than the load required, the pressure difference acts on the spool area in chamber 2, and the output power is reduced to

$$W_{\text{output}} = \frac{k_3 x_{\text{set}} - p_3 S_3 - p_{xi} S_x}{S_0} q_{\max} \tag{7}$$

As shown from Eq. (7), the output power determined by the pre-compress of the spring 3, the pilot pressure p_3 and the sensing pressure p_{xi}. The sum power can be distributed into a pair of matched loads according to the sum pressure (p_1+p_2). In sensing condition of neglected loads, the feedback pressure p_{xi} helps to save remnant energy.

4. Modeling of the virtual prototype

4.1 Dynamic model

The virtual prototype of axial piston double pump was built with multi-models. And dynamic model is the basic part of virtual prototype. The real time simulation data transfer with other models using corresponding interfaces in dynamic model. So the kinetic relationships and motion parameters of double pump was analyzed. There are several hypotheses as follows.

(1) The rotation of middle shaft is stable and the speed is defined as constant.

(2) The incline angle of bent axis changes in a defined work range by rotation drive.

(3) The oil film between piston and cylinder, cylinder and valve plate is stable, so the friction coefficient is constant.

As shown in Fig. 4, the middle shaft of piston pump rotating around its axis drives the pistons and correspond slippers rotating at the same speed[9] [10].

Figure 4 shows the calssic bent axis type pump motion principle. The incline angle between main axis and center line of the cylinder is β. Because of the spherical joint between the piston and the rod, there is an angle α between the rod and the center line of cylinder, which is different with the swash type piston pump. When the rotation group is rotating, the changing of angle α is small about 1°~ 2°, so it is always ignored. Then the displacement of the piston can be described as follows

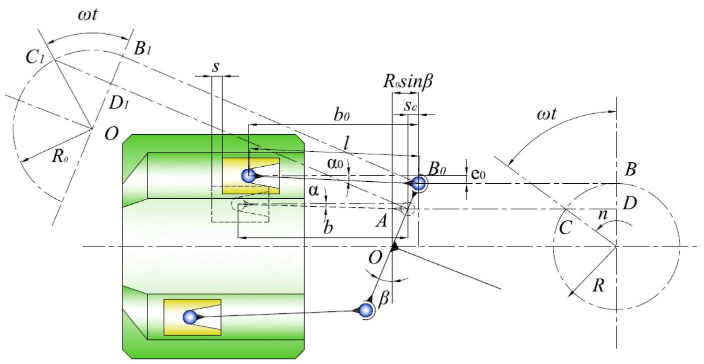


Fig. 4 The motion parameter of piston pump

$$s = R\sin\beta - AO\sin\beta \tag{8}$$

From Fig.4, the distance of AO is

$$AO = R\cos(\omega t) \tag{9}$$

Based on Eqs. (8) and (9), the displacement of piston is

$$s = R\sin\beta[1-\cos(\omega t)] \tag{10}$$

The velocity of the piston is

$$v = \frac{\mathrm{d}s}{\mathrm{d}t} = R\omega\sin\beta\sin(\omega t) \tag{11}$$

The acceleration of the piston is

$$a = \frac{\mathrm{d}^2 s}{\mathrm{d}t^2} = R\omega^2\sin\beta\cos(\omega t) \tag{12}$$

Based on the equations above, the motion of basic parts can be defined.

According to the real dynamic relationship between different parts, the proper joints and constraints are shown in Table 1. In the dynamic software, all these joints and motions were added to corresponding parts. Then the basic dynamic model was finished with a 3-D structural model adding dynamic relationships.

Table 1 The joints and motions of pump parts

	Ground (shell)	Middle shaft	Cylinder	Piston
Middle shaft				Spherical joint
Cylinder				Cylindrical joint
Piston		Spherical joint	Cylindrical joint	
Valve plate	Fixed joint		Planer joint	
Bearing	Fixed joint	Revolute joint		

Besides, the rotation speed of the middle shaft should be added, then the basic model of piston pump can be driven and all parts can move just like a real pump without oil.

4.2 Hydraulic model

As there is no fluid force and motion in the basic dynamic model, so the dynamic model can only simulate the motion of piston pump and doesn't have the function of sucking oil and charging oil to drive load. In order to build the hydraulic model, there are some hypotheses as follows,

(1) The piston pump works stably and the rotation speed of middle shaft is defined constant.

(2) The oil film in the gap between piston and cylinder, cylinder and valve plate is stable. And there is only leakage of laminar flow.

(3) There is hydrostatic balance in the gap between cylinder and valve plate, and the pressure ratio λ is constant (λ=0.9).

(4) The viscosity of the fluid oil is invariable.

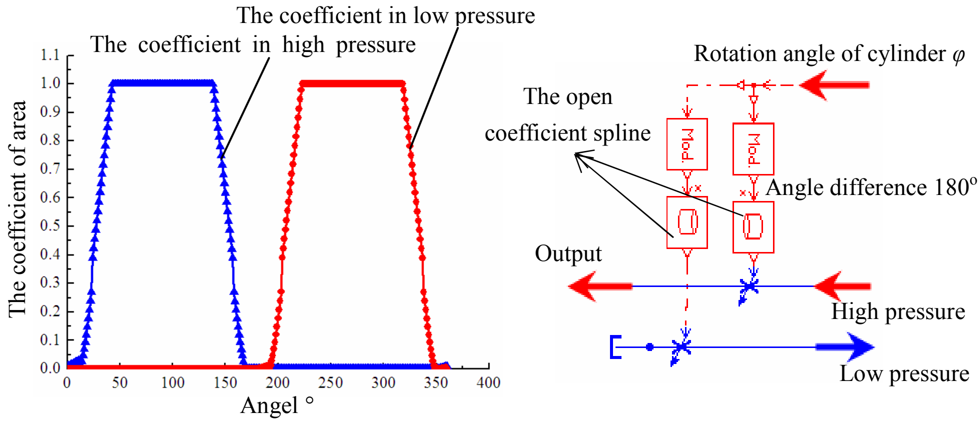
Based on the motion relationship between the pump parts and the flow influence on the pump, the hydraulic model of the piston pump starts with the basic flow model.

As shown in Fig. 4, the low pressure oil is sucked to the cylinder bore when the piston moves to the right. While the piston moves to the left, the piston charges the oil out to drive loads. This model is the basic unit of one piston. And this unit includes two important leakages, q_{v1} between piston and cylinder, q_{v2} between cylinder and valve plate, which are described in Eqs. (13) and (14).

$$q_{v1} = \frac{\pi d_p h_{r1}^3}{12\mu l_r}(1+1.5\varepsilon^2)(p_1 - p_0) - \frac{\pi d_p h_{r1} v_p}{2} \tag{13}$$

$$q_{v2} = \lambda \frac{\pi h_{r3}^3}{6\mu}\left[\frac{1}{\ln(R_2/R_1)} + \frac{1}{\ln(R_4/R_3)}\right](\varphi_2 - \varphi_1)(p_1 - p_0) \tag{14}$$

The input oil and output oil in the cylinder bore is supplied through the valve plate, so the open area of the kidney bore in valve plate is used to control oil sucking and charging. Figure 5 (a) shows the open area coefficient curves with changing of rotation angle. The red curve shows the sucking bore area coefficient in low pressure. The blue curve shows the charging bore area coefficient in high pressure. Because the angle difference of the two different bore is 180°, so it can be modeled with a throttle valve controlled by the open area size. The distributed flow is controlled by the cylinder bore open area size, which is showed in Fig. 5 (b).



(a) Opening area coefficient curves (b) The valve plate model

Fig. 5 The model of valve plate

To simplify the modeling, the flow model and the valve plate model were combined to an integrated unit. As one pump has nine pistons, so the whole pump model includes nine integrated piston models, shown in Fig. 6. The whole model (left in Fig. 6) can be integrated to the pump icon (right in Fig. 6). The pump icon include the hydraulic model, dynamic model and their interface, and it can be used in a system as a hydraulic component.

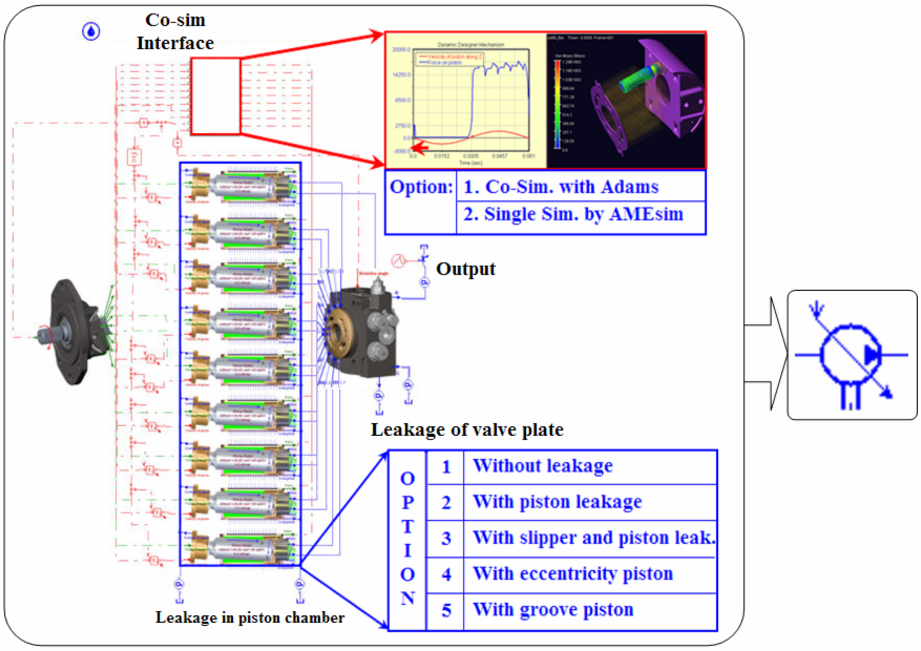


Fig. 6 The whole hydraulic model of the pump

4.3 Modeling of cross power control

As for the double pump with cross power control, it is focused on the displacement adjusting mechanism, as shown in Fig. 7. The ladder spool drives the main spool, and then the pilot oil drives the piston to change the incline angle of the swash plate and cylinder. It is a complicated movement, and it is necessary to make the control mechanism model. The model mainly includes three parts: piston control part (part I), valve control part (part II) and standard pump unit (part III), shown in Fig. 8. In part I, the piston is driven by the control pressure and two power springs, the force balance can be termed as

$$p_1(C_1S_1 - S_2) = m_p \frac{d^2x}{dt^2} + B_p \frac{dx}{dt} + kx + F_v \tag{15}$$

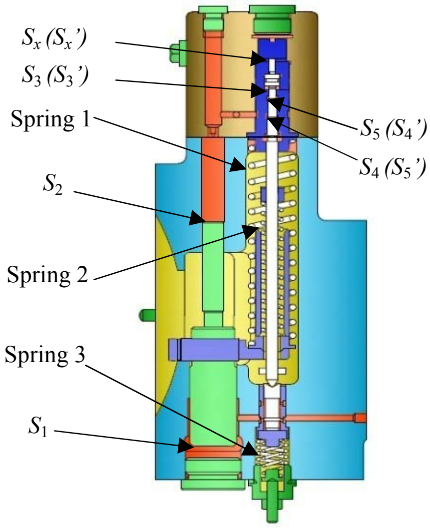


Fig. 7 The displacement changing mechanism

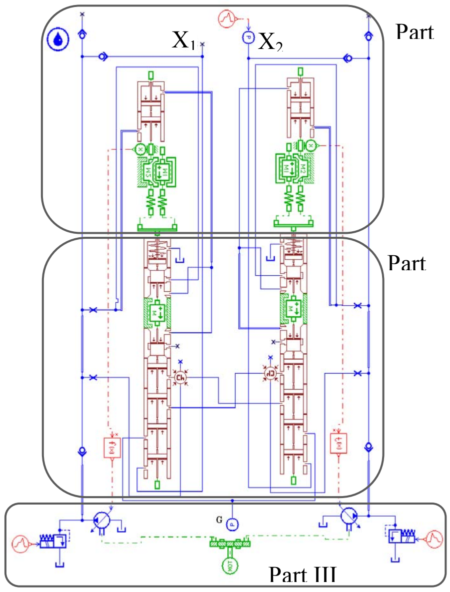


Fig. 8 Model of the double pump with cross power control

In part II, the ladder pilot spool and the main valve are connected by the support rod, and the ladder pilot spool need overcome the pre-stressing force of the big power spring. The relationships between the output power and the parameters of the spools and springs are explained by Eqs. (3) ~ (6).

Combining the dynamic relationship above and hydraulic principle, the whole model of the virtual prototype was made (Fig. 8) and it can describe most of important characteristics.

Before the analysis of the virtual prototype, some initialization conditions need to be defined. The parameters of three springs are shown in Table 2.

Table 2 The important parameters of springs in the pump

Spring	Stiffness N/mm	Spring diameter mm	Rope diameter mm	Free length mm
1	5.03	23.6	3.4	168
2	4.5	12	2	90
3	37	10.4	2.4	33.5

The pilot pressure for the ladder spool can change the point of output power based on Eq. (3), and the pressure is controlled by a proportional valve. In order to simplify the model, a pressure source G in Fig. 8 is used to replace of the proportional valve. The relationship between the pressure of the source G and the current of the valve is shown in Fig. 9 according to experiment results.

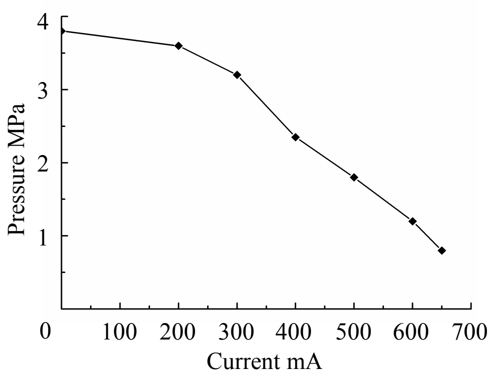


Fig. 9 The relations hip between the pilot pressure and current of proportional valve

With the different definition of the

pilot pressure and the pre-stressing of the pressure adjustment spring (spring 3), the virtual prototype can simulate the working condition at different power points. If the feedback of the flow required by load is considered, the pilot pressure port X_1 or X_2 is usable, and then the respective pressures need to be defined. For example, the pressure of the port X_2 in Fig. 8 is defined by a signal unit.

5. Results and discussion

Based on the simulation of the virtual prototype and experimental results, the cross power control function is proved and the control performance is analyzed.

In order to get the more effective conclusion, the simulation and experiment have same initial condition (Table 3). Several pressure points considered in the simulation and respective current are shown in Table 4.

Table 3 The initialization condition of the experiment.

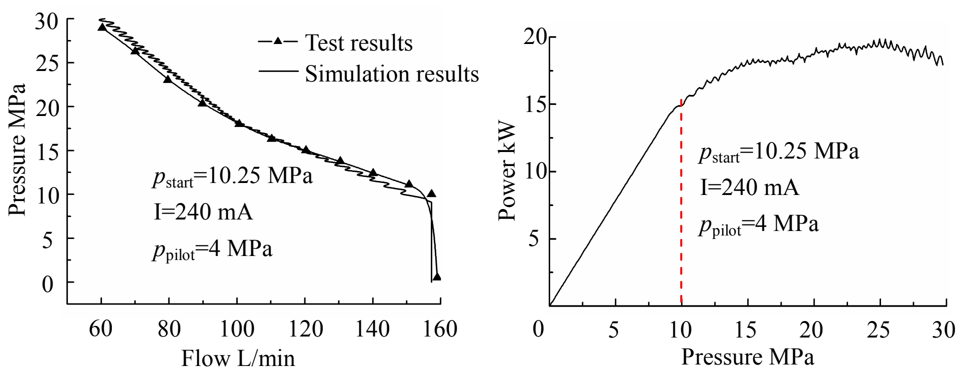
Condition	Data
Displacement cm^3/r	107×2
Temperature °C	50±5
Rotary speed r/min	2000
Pilot pressure MPa	4
Oil	MR20S

Table 4 the pilot pressure points considered in the simulation.

	1	2	3	4	5	6	7
Current mA	0	200	300	400	500	600	650
Pressure MPa	3.8	3.6	3.2	2.35	1.8	1.2	0.8

5.1 Constant power control with the same output pressure

The cross power control is one kind of constant power control. When the two output pressure of the double pump is equal, each pump can work in constant power principle. Both the start pressure p_{start} of the output port and the control current of pilot pressure in G port determine the constant power point. In order to analyze the control performance of the double pump, two different work conditions in two power points are shown in Fig. 10 and Fig. 11.

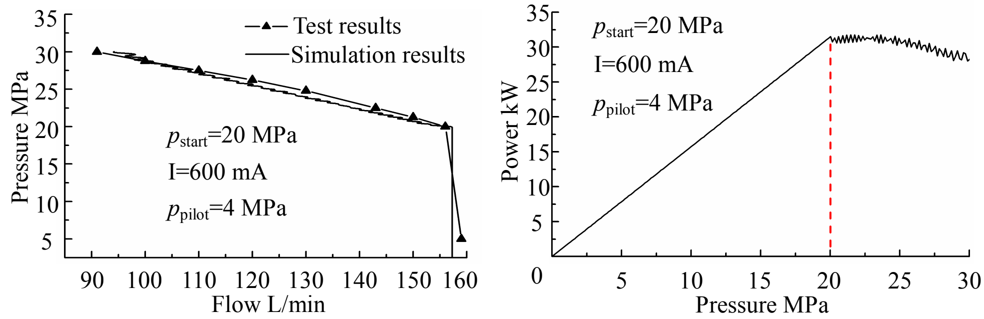


(a) The relationship between flow and pressure

(b) The power curve with the pressure changing

Fig. 10 Constant power characteristics at p_{start}=10.25MPa

When p_{start}=10.25 MPa and I=240 mA, the pressure required by load reaches up to 10.25 MPa. The relationship curve of pressure along with flow runs according to a hyperbola, shown in Fig. 10 (a), and the pressure rises with flow dropping. In Fig. 10 (b), it is obvious that the output power of the pump is approximate constant after the output pressure up to about 10.25 MPa



(a) The relationship between flow and pressure

(b) The power curve with the pressure changing

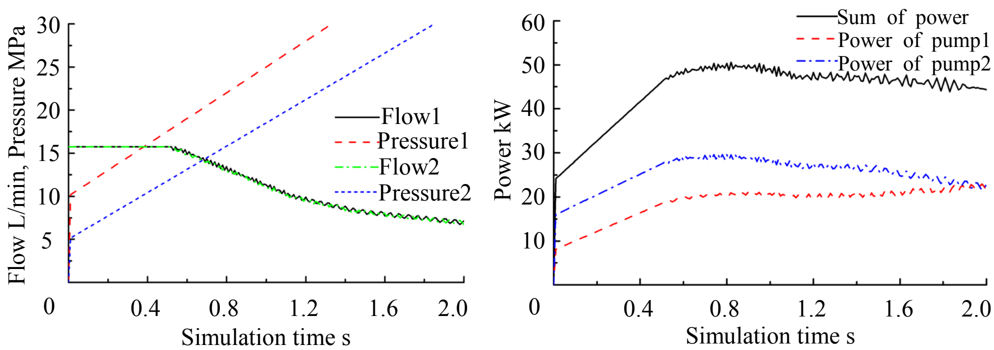
Fig. 11 Constant power characteristics at p_{start}=20MPa

As shown in Fig. 11 (p_{start}=20 MPa and I=600 mA), the curves in picture Fig. 11 (a) is away from the hyperbola (constant power curve), especially at the turning point. So when the constant power point is too high, the output power changes. But the output power is still in an acceptable range of constant power from the curve in Fig. 11 (b).

Besides, according to Fig. 10 (a) and Fig. 11 (a), the simulation result is very close to the test result. So it can be concluded that the virtual prototype is reliable.

5.2 Cross power control with different output pressure

With the same output pressure, each pump works in constant power condition and exports the same power to different loads. But in real working conditions the double pump usually need to supply different pressures and power required by different loads. The cross power control can distribute the input power properly according to the pressures required by different loads.



(a) Relationships between flow and pressure of pump1 and pump2

(b) Sum power and power of each pump

Fig. 12 Characteristics of the cross power control with different output pressure (p_{1start}=10 MPa, p_{2start}=20 MPa)

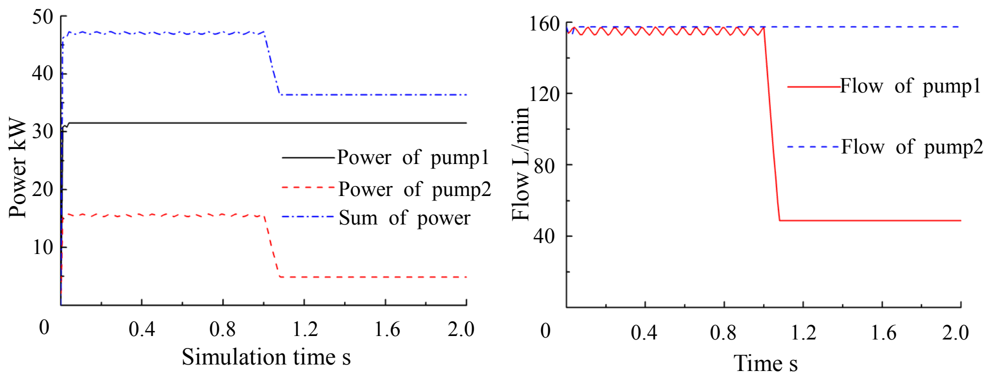
In the cross power control, a start-adjusting pressure is set to define the power point of the double pump. Figure 12 shows the double pump driving different loads in the defined power points (p_{1start}=10 MPa, p_{2start}=20 MPa). Two output pressures of the double pump are different, as shown in Fig. 12 (a). Because two different pressures act on the same area of the ladder spool of the control mechanism for each pump, two output flows are approximately same. The power condition is shown in Fig. 12 (b). Before the sum of the two pressures required by loads reaches the sum of p_{1start} and p_{2start}, the total output power isn't reached constant stage, so the output power increases with the pressure rising. In constant power stage, the total power maintains stably at a small range. As for each pump unit, the power of pump 1 is higher with output pressure increasing. With the pressure rising, the power of pump 1 reduces a little and the power 2 increase, but the changes are in an acceptable range, and the total power is stable from start-adjusting pressure to the end.

Based on the analysis above, it is obvious that the cross power control properly distribute the total power to meet the need of different loads. One pump can make full use of the remnant energy of another pump. So the cross power control is energy-saving comparing with traditional individual control. However, the pump can only supply same flow rate with different pressure. If two loads need different flow rate, there will be a cause of energy waste. However, the sensing control of negative load can reduce the energy-wasting.

5.3 Sensing control system of negative load in cross power control

With the principle of cross power control, the output flows from the two ports are the same in constant power control. In order to meet the needs of different flow rates, two sensing control ports X_1 and X_2 are designed for each pump. If flow rate of one load in the system need decreased, a flow-test valve in the system can export a control pressure, which is proportional with the flow rate of the load. The control pressure is used to reduce the displacement and save the remnant energy of the respective pump by the port X_1 and X_2.

With the virtual prototype, the sensing control of the negative load is researched. When the loads pressures are 10 MPa and 20 MPa, and the control pressure 5 MPa is supplied to port X_2 for the pump 2 after one second. As shown in Fig. 13, the flow rate of pump 2 reduces quickly, and parts of the power are saved from the power curve in Fig. 13 (b).



(a) Flow changes with control pressure of negative flow

(b) Power changes with control pressure of negative flow

Fig. 13 Results of sensing control of negative flow in cross power control (p_{1start}=10MPa, p_{2start}=20MPa)

It is very useful that the displacement can be changed based on the need of loads, and the energy consuming is reduced. The sensing control of the negative loads is a better choice for the mobile machine with different loads.

6. Conclusion

(1) Compared with traditional control way, the cross power control distributes the power into different loads more flexibly. The double pump can not only work in constant power condition, but also output power according to the loads needing. So the power from

the motor is used effectively.

(2) When the two output pressures are equal, each pump can works in constant power condition and have same energy consuming. If the start-adjusting pressure is too high, the constant power curve changes a little far from the hyperbola, which is an ideal constant power curve.

(3) When the double pump output different pressure required by loads, the sum power can be distributed automatically. Because the two pressures act on each control spool, so the output flows are still same. In order to meet the different flow demand, the sensing pressure of negative flow is used to reduce the displacement of the pump, so remnant power is saved by the sensing control of negative loads.

Acknowledgement

The authors would like to express their gratitude towards the National Key Technology R&D Program of the Eleventh Five-year Plan of China for providing necessary funding of this project (the grants No. 2006BAF01B01, No. 2006BAF01B04) and the Technological Research and Development Programs of the Ministry of Chinese Railways (the grant No. 2009J006-L).

Appendix

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