Research on a Novel Multi-pump and Multi-motor Driving System

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Abstract. With the development of human society, the concept of "carbon neutrality" has gradually attracted people's attention. However, due to the existence of throttling loss, the efficiency of the hydraulic systems widely used is usually lower than 30%. In order to improve the energy conversion rate of the inefficient hydraulic system, a new type of multi-pump and multi-motor (MPMM) driving system is proposed, which can meet the flow and pressure requirements of different working conditions by switching the operation modes. In this paper, a functional symbol representation method of MPMM is specified and a systematic method of how to determine the operation mode of MPMM driving system is presented, which lays the foundation for the design of a new MPMM driving system. The test bench is built up and the correctness of the MPMM principle is verified. In addition, the comparative study of the conventional and the novel system is conducted. The results show that the input power of the hydraulic system adopting MPMM can be significantly reduced.

Keywords: Hydraulic system · Carbon neutrality · Multi-pump and multi-motor · Efficiency · Operation mode

1 Introduction

With the development of human society, the concept of "carbon neutrality" has gradually attracted people's attention [1]. In the field of hydraulic technology, how to improve energy conversion efficiency and reduce carbon emissions has become a hot research topic [2–4].

The hydraulic system has the advantages of high power density and large output force, which has a wide range of applications in construction machinery, agricultural machinery, mobile machinery [5]. However, due to the unavoidable throttling loss and pipeline damping loss caused by the existence of the throttling valve, the working efficiency is usually lower than 30%n [6, 7]. Taking typical hydraulic machinery - excavators and loaders as an example, the average fuel consumption of each heavy excavator

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is 0.0151 m^3/h, and the average fuel consumption of wheel loaders is even 0.0227 m^3/h. Therefore, research on how to reduce the fuel consumption of hydraulic machinery cannot be ignored. The constant power variable pump can not only effectively improve the energy utilization rate of the diesel engine, but also avoid overload, so the constant power variable pump has been widely used in the mechanical field of our country [8]. Then positive flow control systems gradually appeared in people's field of vision, in which the spool stroke of the multi-way valve is proportional to the flow output of the pump, and the displacement of the pump changes with the change of the spool position [9–11]. Due to the serious flow loss and throttling loss in the positive flow system, the negative flow control system comes out, which controls the displacement of the pump by setting a flow detection device on the bypass oil circuit, thereby reducing the loss of the bypass oil circuit and solving the problem of large energy consumption. Subsequently, load-sensing technology appeared and was widely used in excavators [12–14]. The advantage of load-sensing technology is that when the external load changes, the flow of the pump and valve can be adjusted automatically, so that the hydraulic system can provide the flow and pressure to the pipeline to adapt to the external load, thereby improving energy utilization. However, load-sensing system has a big disadvantage that when the valve opening is too large, the oil supply capacity of the pump will not be enough to provide the flow required by the system, and the actuator with higher load will not work properly. At present, reducing energy consumption and improving energy utilization is still an urgent problem to be solved in the field of fluid transmission and control.

Inspired by the previous study, we propose a novel hydraulic transmission configuration: multi-pump and multi-motor (MPMM) driving system to reduce the energy consumption. By switching the working modes of MPMM, a large amount of throttling energy loss can be eliminated through volume control, thereby improving the efficiency of the hydraulic system.

Whereas, as far as known to us, most of the researches on hydraulic energy-saving systems are based on the conventional hydraulic components [15–17]. MPMM are new types of hydraulic components that contains multiple single pumps/motors, which can realize multi-level output, and even realize the differential working mode of the motor. Therefore, the analysis results for the conventional components are not applicable to MPMM. The characteristics of MPMM system are still unknown. There are no systematic researching results can be borrowed directly to analysis the characteristic in the proposed MPMM until now.

Motivated by the above observations, in this paper a functional symbol representation method of MPMM is specified and a systematic method of how to determine the operation mode of MPMM driving system is presented, which lays the foundation for the design of a new MPMM driving system. The comparative study of the conventional and the novel system is conducted. The experiment test bench is built up and the correctness of the MPMM principle is verified.

2 Multi-pump and Multi-motor

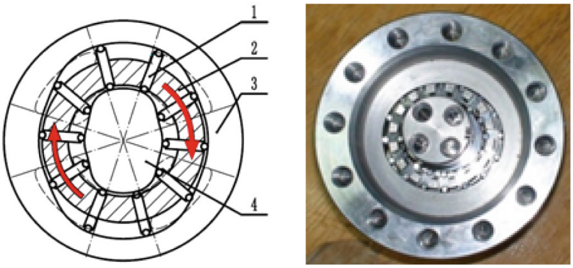
2.1 Principle of MPMM

The MPMM has a unique structure, which can be divided into proportional and parallel types according to the characteristics of the single pump and single motor contained in the casing. When there are several single pumps with different displacements in the same casing, it is a proportional type multi-pump (When used as a motor, it is a proportional multi-motor). The proportional MPMM is shown in Fig. 1.

When there are several single pumps of the same displacement in the same casing, it is a parallel type multi-pump. Internal and external gear pumps are proportional multi-pumps. While external gear pump belongs to parallel multi-pump, in which each group of meshed gears meets the conditions for forming a hydraulic pump, and can form a single pump with the same displacement. The gear type multi-input pumps are shown in Fig. 2.

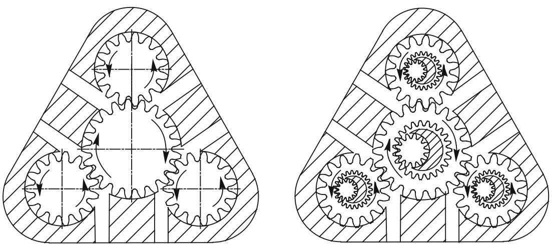
2.2 Differential Operation Mode of Multi-motor

When the input flow is constant, the multi-motor can realize the output of various speeds and torques by switching different connection modes. The differential operation mode of multi-motor is shown in Fig. 3.



1-doubleroller and connecting-bar; 2-rotator; 3-outer stator; 4-inner stator

Fig. 1. Vane type MPMM.



(a) Parallel MPMM (b) Proportional MPMM

Fig. 2. Gear type MPMM

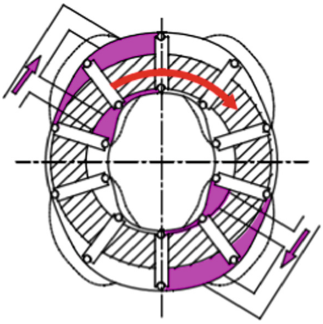


Fig. 3. Differential operation mode of multi-motor

As can be seen in Fig. 3, when a part of single motors in the multi-motor are supplied with pressurized oil in the forward direction while a part of single motors of the multi-motor are supplied with pressurized oil in the reverse direction. The single motors with a small total displacement will rotate in the opposite direction driven by the single motors with a large total displacement, thereby realizing the function of the pump, as shown in Fig. 3. The oil in the "pump" will re-enter the inlet of the single motors that drives the "pump" to rotate, thus completing the differential operation of the multi-motor.

3 Multi-pump and Multi-motor driving system

3.1 Principle of MPMM Driving System

MPMM driving system is based on the novel hydraulic components. We have conducted a series of studies on the performance of components [17–19], such as internal leakage, fluctuation, etc. Due to space limitations, we will not repeat them here.

In the MPMM driving System, the schematic of which is shown in Fig. 4, the multi-pump is served as a dynamic component, and the multi-motor is served as an actuator component. In addition, according to the characteristics of the components used, the MPMM driving system can be classified as proportional, parallel and hybrid types.

Figure 5 shows the experimental platform of MPMM driving system.

3.2 Classification of MPMM Driving System

3.2.1 Proportional Configuration of MPMM Driving System

The transmission system composed of only proportional MPMM is proportional MPMM driving system. The schematic diagram is shown in Fig. 6.

In the proportional configuration of MPMM driving System, the displacements of the single motors/pumps are different. Therefore, the motor can realize differential operation mode.

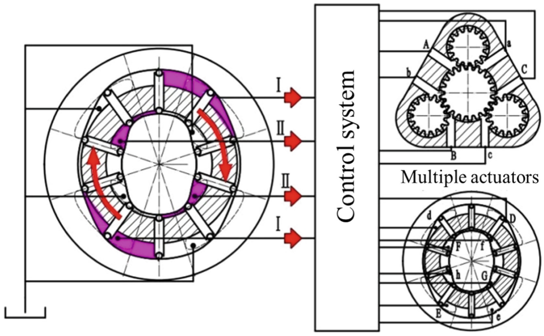


Fig. 4. MPMM driving system

Fig. 5. Experimental platform of MPMM driving system

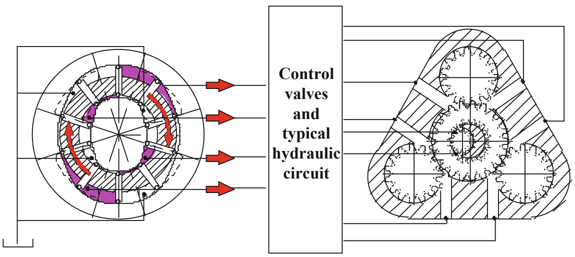


Fig. 6. Proportional Configuration of MPMM Driving System

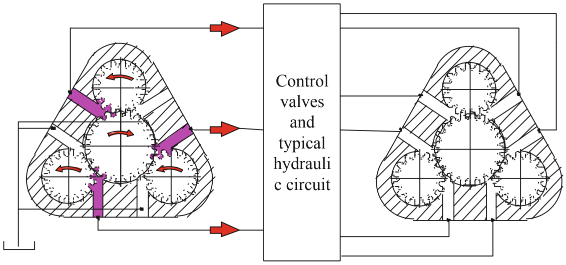


Fig. 7. Parallel Configuration of MPMM Driving System

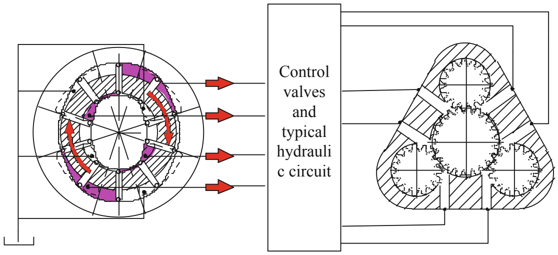


Fig. 8. Hybrid configuration of MPMM driving system

3.2.2 Parallel Configuration of MPMM Driving System

The transmission system composed of only parallel MPMM is called parallel MPMM driving system. The schematic diagram is shown in Fig. 7.

In the parallel configuration of MPMM driving System, the displacements of the single motors/pumps are the same. Therefore, the motor cannot realize differential operation mode.

3.2.3 Hybrid Configuration of MPMM Driving System

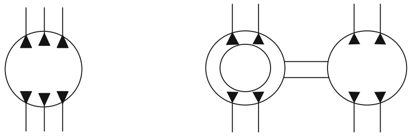
Hybrid type MPMM driving system consists of both proportional and parallel multi-pump and multi-motor. The schematic diagram is shown in Fig. 8.

3.3 Functional Symbols of the Novel Hydraulic Components

MPMM driving system is a new type of hydraulic transmission system. In order to better design the multi-pump multi-motor transmission diagram, the functional symbols of the components should be specified first:

(1) The outer pump is represented by a single circle, while the inner pump is represented by a double circle. Since the inner pump and the outer pump share a rotor, the two circles are connected by a coaxial symbol;

(2) A parallel multi-pump consisting of R single pumps is called an R parallel multi-pump (R is a positive integer). The functional symbol of 3-parallel multi-pump is shown in Fig. 9(a). The number of single pumps contained in the multi-pump is expressed by



(a) 3-parallel multi-pump (b) 2-2 proportional multi-pump

Fig. 9. Functional symbols of multi-pump

drawing arrows within the circle symbol. The direction of the component is represented by the pointing of the triangle;

(3) Proportional multi-pump consisting of single pumps with various displacements, called T1 − T2− … −TR proportional multi-pump. Where T is the number of single pumps with the same displacement. 2-2 proportional multi-pump is shown in Fig. 9(b). Proportional multi-pumps use two circles to represent single pumps with different displacements, and connect them with coaxial symbols.

The function symbol of the multi-motor is similar to that of the multi-pump, the difference is the direction of the arrow. Due to space limitations, we will not repeat them here.

4 Operation Mode of MPMM Driving System

In the multi-pump and multi-motor transmission system, the multi-pump can output various flow rates and the multi-motor can output multi-stage torque through the switching of different connection modes so as to meet different work needs.

4.1 Operation Mode of Multi-pump

The multi-pump can achieve the following functions through different connection methods: (1) One pump supplies multi-stage constant flow to one system; (2) One pump can provide multiple pressures for different hydraulic systems without a pressure reducing valve.

Take the 3-parallel multi-pump as an example, when it supplies oil to a hydraulic system, it can output 3 kinds of flow rate. The connection method is shown in Fig. 10.

For the R-parallel multi-pump, set the displacement of the single pump as q_0 and the motor speed as n, the output flow under each connection mode can be obtained as:

$$Q_1 = nq_0;\ \ Q_2 = 2nq_0\ Q_R = Rnq_0 \tag{1}$$

It can be seen from Eq. 1 that switching the connection mode of the R parallel multi-pump can realize the output of R-level flow, and can supply oil to R systems at most at the same time. In addition, it can also adapt to R different working pressures without a pressure reducing valve. Part of the connection modes of the 2-2 proportional multi-pump are shown in Fig. 11.

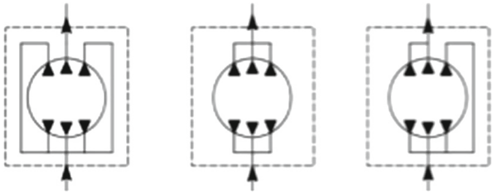


Fig. 10. Operation mode of 3-parallel multi-pump

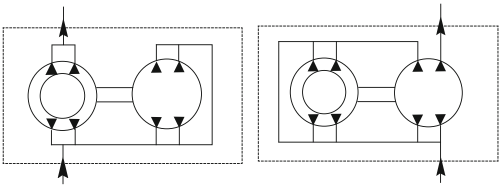


Fig. 11. Operation mode of 2-2 proportional multi-pump

Assuming that the displacements of the inner pump and the outer pump of the 2-2 proportional multi-pump are q1 and q2 respectively, and q1 < q2. When the inner pump and the outer pump supply oil independently, the output flow are:

$$Q_1^* = nq_1;\ Q_2^* = nq_2;\ Q_2^* \big/ Q_1^* = q_2 \big/ q_1 = C_2 \tag{2}$$

Where C_2 is the displacement coefficient of the proportional type multi-pump.

For T1 − T2− … −TR proportional multi-pump, when assuming that q_R is the displacement of the largest single pump, and the displacement decreases in turn with the decrease of the angle scale R. From the Eq. 2, the displacement coefficient C_r of the T1 − T2− … −TR proportional multi-pump can be obtained:

$$Q_R^* \big/ Q_1^* = q_R \big/ q_1 = C_R \tag{3}$$

According to Eq. 1 and Eq. 2, the general expression of the output flow of the multi-pump at a certain motor speed can be obtained:

$$Q = (C_1X_1 + C_2X_2 + \cdots + C_RX_R)Q_1^* \tag{4}$$

where $C_1 = 1$, X_1, X_2…, X_R is the number of working single pumps with different displacements. $(C_1X_1 + \ldots + C_RX_R)$ is the flow coefficient of the multi-pump.

Table 1. Flow coefficient under different modes

Number of inner pumps	Number of outer pumps	Flow coefficient
0	1	C_2
0	2	$2C_2$
1	0	C_1
1	1	$C_1 + C_2$
1	2	$C_1 + 2C_2$
2	0	$2C_1$
2	1	$2C_1 + C_2$
2	2	$2C_1 + 2C_2$

Equation 4 is applicable to both types of multi-pumps, in the case of parallel multi-pumps $C_R = 1$. The flow coefficient directly affects the output characteristics of the multi-pump. From the analysis of Eq. 4, it can be seen that its value, which is affected by two coupling factors, is determined by the displacement coefficient and the working mode. First of all, the displacement coefficient is an inherent parameter of the multi-pump and depends on the displacement ratio between the single pumps. And from the structural characteristics of the multi-pump and formula (3), it can be known that the limit range of the displacement coefficient is $[1, \infty)$. In order to highlight the advantages of large specific power of hydraulic transmission and maximize the power output per unit volume of components, it is often hoped that the displacement coefficient of the proportional multi-pump is as close to 1 as possible while meeting the strength requirements. Secondly, when the number of optional working modes of multi-pumps is different, such as 1-1 and 2-2 proportional multi-pump, the displacement coefficient is also different.

In summary, in order to reasonably determine the value of the flow coefficient, the actual working requirements and other conditions should be comprehensively considered. It is necessary to first obtain the multi-pump output flow statistics table according to the flow coefficient equation, and then select the working mode and displacement coefficient. According to Eq. 4, the flow coefficient of 2-2 proportional multi-pump in different working modes can be obtained, as shown in Table 1.

According to Table 1, 2-2 proportional multi-pump can realize 8 optional connection modes. After the flow coefficient is determined, the displacement coefficient should be comprehensively considered, and the working mode can be finally selected.

4.2 Operation Mode of Multi-motor

In the hybrid MPMM driving system, under the condition of a certain input flow rate, the multi-motor can realize the output of various speeds and torques by switching the connection mode. For the convenience of calculation, the pressure difference between the inlet and outlet of the motor is assumed to be Δp, and the input flow rate is Q.

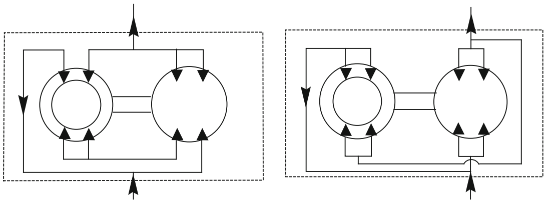


Fig. 12. Two different differential operation mode of 2-2 proportional multi-motor

The multi-motor has the normal working mode and the differential working mode as shown in Fig. 3. When the oil-passing direction of each single motor in the multi-motor is the same, the normal connection mode of the multi-motor can be formed, which is similar to the connection mode when the multi-pump supplies oil to a single system. So it will not be repeated here. If some single motors within the proportional multi-speed motor are supplied with oil in the forward direction, and some are supplied with oil in the reverse direction, as shown in Fig. 12, the single motors with a smaller total displacement are driven by the single motors with a larger total displacement to realize the pump function, forming a differential working mode. It should be emphasized that proportional multi-speed motors have various differential working modes.

When the inner and outer single motors of the 2-2 proportional multi-speed motor work independently, the output torque and speed are:

$$\left.\begin{aligned} M_1 &= \frac{\Delta p q_1}{2\pi} \\ n_1 &= \frac{Q}{q_1} \end{aligned}\right\}; \quad \left.\begin{aligned} M_2 &= \frac{\Delta p q_2}{2\pi} \\ n_2 &= \frac{Q}{q_2} \end{aligned}\right\} \tag{5}$$

Similarly, according to Eq. 3–5, the general formulas for the output speed and torque of the T1 − T2− …− TR proportional multi-motors can be derived as:

$$\left.\begin{aligned} M &= (C_1X_1 + C_2X_2 + \cdots + C_RX_R)M_1 \\ n &= \frac{n}{(C_1X_1 + C_2X_2 + \cdots + C_RX_R)} \end{aligned}\right\} \tag{6}$$

Where $C_1 = 1$, $(CX_1 + \ldots + CX_R)$ denotes torque coefficient, and $1/(CX_1 + \ldots + CX_R)$ represents speed coefficient.

Equation (7) is applicable to all connection modes of the two types of multi-motors. In normal connection, X is positive; in differential connection, X in Eq. 7 of single motor with reverse connection is negative, and X of single motor with positive connection is positive; when the multi-motor used is parallel type, CR = 1. The selection of torque coefficient and speed coefficient is similar to that of flow coefficient, so only the output table of the motor is listed, as shown in Table 2, and the rest will not be repeated.

Since the displacement of each single motor in the parallel multi-motor is the same, the differential working mode cannot be realized.

Table 2. Torque and speed coefficients under different modes

Number of inner motors	Number of outer motors	Torque coefficient	Speed coefficient
1	1	$C_2 - C_1$	$1/(C_2 - C_1)$
1	2	$2C_2 - C_1$	$1/(2C_2 - C_1)$
2	1	$C_2 - 2C_1$	$1/(C_2 - 2C_1)$
2	2	$2C_2 - 2C_1$	$1/(2C_2 - 2C_1)$
0	1	C_2	$1/C_2$
0	2	$2C_2$	$1/2C_2$
1	0	C_1	$1/C_1$
1	1	$C_1 + C_2$	$1/(C_1 + C_2)$
1	2	$C_1 + 2C_2$	$1/(C_1 + 2C_2)$
2	0	$2C_1$	$1/2C_1$
2	1	$2C_1 + C_2$	$1/(2C_1 + C_2)$
2	2	$2C_1 + 2C_2$	$1/(2C_1 + 2C_2)$

5 Comparative Study of the Conventional and the Novel Systems

The schematic diagram of the throttling speed control system based on 2-2 multi-pumps is shown in Fig. 13. Compared with the conventional throttling speed control system, it is characterized by the use of multi-pumps instead of constant displacement pumps. The multi-pump is controlled by four electromagnetic hydraulic valves, so as to automatically generate the flow required by each stage in the working cycle. At the same time, stepless speed regulation can be realized by adjusting the opening area of the throttle valve. The following will study the power characteristics of the MPMM system under the condition of constant hydraulic cylinder load. Furthermore, a comparative study will be conducted.

When the hydraulic cylinder works under constant load, the working pressure p_1, the oil supply pressure p_p of the pump, and the working pressure Δp_{t1} of the throttle valve are all constant values. The working flow rate q_1 only changes with the flow cross-sectional area of the throttle valve. It is assumed that the leakage of the pipeline and the valve is not considered, and the range of the speed v required by the hydraulic cylinder with the rodless cavity area $A_1 = 1$ is 0–12 m/min. For the multi-pump speed-regulating system, the rated flow rate of 15 L/min with a flow proportional coefficient of 0.5 is appropriate. The rated flow rate of the single inner pump is 2.5 L/min, and the rated flow rate of the single external pump is 5 L/min.

For the convenience of comparison, a constant displacement pump with the same rated flow rate (15 L/min) is selected for the conventional throttling speed control system. Taking the speed v as the abscissa and the power P as the ordinate, the power characteristic curves of the two systems under constant load can be obtained as shown in Fig. 14.

It can be seen that the input power of the system after using the multi-pump is significantly reduced. This phenomenon is because the input power is equal to the product

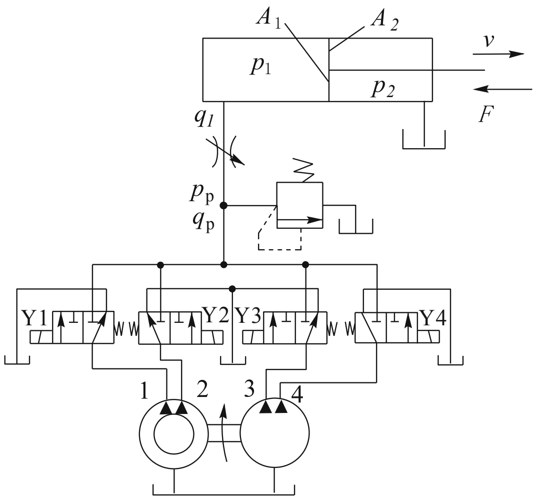
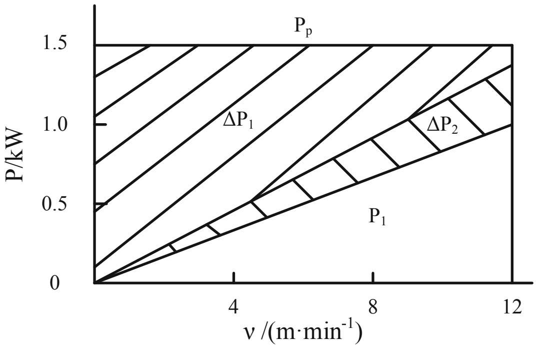
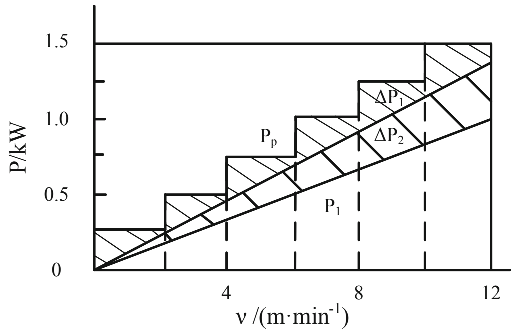


Fig. 13. The MPMM test bench



(a) Conventional throttling speed control circuit



(b) MPMM speed control circuit

Fig. 14. Power characteristics of throttling speed control loop under constant load

Table 3. Operation modes of multi-pump

v (m/min)	q_1 (L/min)	Y1	Y2	Y3	Y4
0–2	0–2	+	+	–	–
2–4	2–4	+	+	–	–
4–6	4–6	+	–	+	–
6–8	6–8	–	–	+	+
8–10	8–10	+	–	+	+
10–12	10–12	+	+	+	+

of the operation pressure and the flow, switching the different working modes of the multi-pump can convert different displacements without throttling loss. According to the working conditions, the operation modes of multi-pump is shown in Table 3.

In addition, it can be found that the widely used single pump single motor hydraulic transmission is merely a special case of MPMM driving system.

6 Experimental Verification of the Novel System

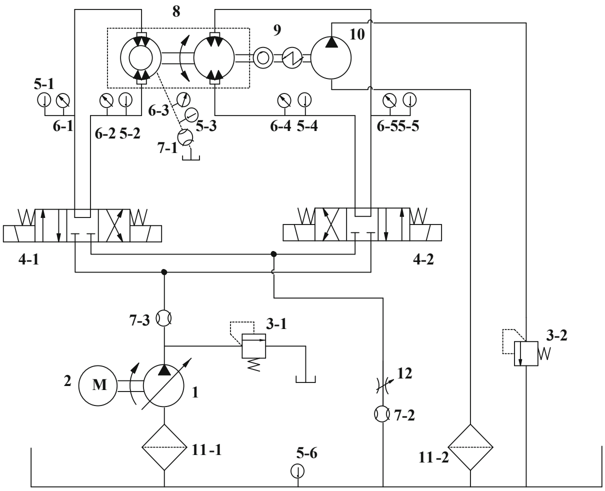
The prototype of 1-1 proportional multi-motor is shown in Fig. 2. The parameters are as follows: the displacement of the outer motor is 152.16 mL, and the displacement of the inner motor is 29.93 mL. When the external motor is working together, the displacement is 180.09 mL. When the internal and external motors are working in differential connection, the displacement is 124.23 mL, the roller radius is 3 mm, and the connecting rod groove radius is 3.3 mm.

In order to verify the principle correctness of the multi-motor, the MPMM test bench was built, as shown in Figs. 15 and 16, in which the measurement accuracy is grade B according to GB/T7936-2012. Since multi-motor is a new type of hydraulic component, the derived capacity V_i was acquired through experiments according to GB/T 7936-2012 and JB/T1089-2008.

$$V_i = 1000 \frac{N(\sum_{j-1}^{N} n_j q_{vj}) - (\sum_{j-1}^{N} n_j)(\sum_{j-1}^{N} q_{vj})}{N(\sum_{j-1}^{N} n_j^2) - (\sum_{j-1}^{N} n_j)^2} \tag{7}$$

Where V_i denotes the derived capacity of the test motor, N is the number of measurements, q_v represent input flow (L/min).

The test results of the derived capacity of the multi-motor is shown in Fig. 17. It can be seen that the test result is slightly lower than the theoretical value, which is caused by the internal leakage of the component. The derived capacity is the largest when the internal and external motors work together, while the derived capacity is the smallest when the internal motor is working alone, which shows that conversion of different displacements of 1-1 proportional multi-motor can be realized through switching the working mode.



1-Hydraulic pump 2-motor 3-1, 3-2-relieving valve 4-1,4-2-directional valve 5-1~5-6-thermometer 6-1~6-5-pressure gauge 7- 1~7-3-flow meter 8-test motor 9-speed torque tester 10-load pump 11-1, 11-2-oil filter 12-throttle valve

Fig. 15. Schematic diagram of experiment system

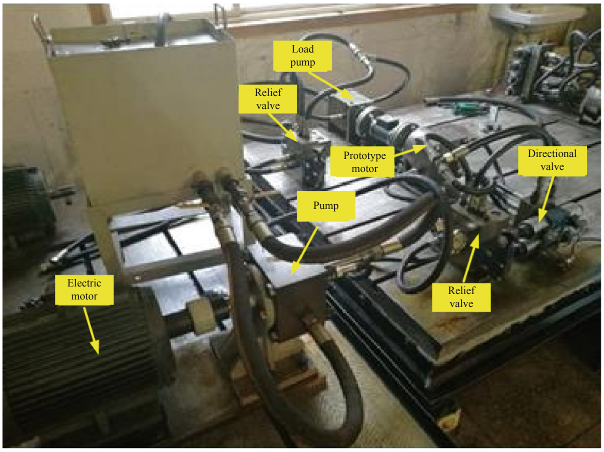


Fig. 16. Throttling speed control system based on multi-pump

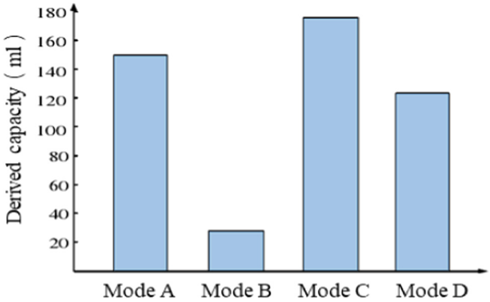


Fig. 17. Derived capacity of different operation mode

7 Conclusions

MPMM system is a new type of hydraulic transmission method, in which proportional and parallel MPMM are the component basis. In this paper, a functional symbol representation method is specified and a systematic method of how to determine the operation mode is presented, which lays the foundation for the design of a new MPMM driving system. The results of the comparative study of the conventional and the novel system show that the input power of the hydraulic system adopting MPMM can be significantly reduced.

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