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Design of hydraulic control system for a new energy vehicle hybrid drive system

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Abstract. An efficient and energy saving hybrid drive system was developed and applied in a new energy vehicle to meet global demands for vehicle energy consumption and exhaust emissions reduction. This new system as a power train is composed of an e-CVT (electric coupled CVT with a motor and a generator built-in), an engine, an Integrated Power battery and a Power Control Unit. The hydraulic control system integrated in the e-CVT system was developed for realizing and supporting the different modes of the hybrid drive system. In this system, a pump driven by the input torque from engine and another pump driven by a small power electric motor as oil source to meet the flow rate demands in each vehicle operating mode. Hydraulic Control Module integrated design, to implement the required flow distribution and pressure control for built-in motor and generator cooling, some needed gears and bearings lubrication, clutch control and clutch cooling. Motor oil cooling system designed to meet the real-time cooling flow requirements of motor. Dynamic simulation and test were done to verify the design is reasonable.

1. Instruction

Automotive energy consumption and exhaust emissions have attracted worldwide attention in recent years, especially in China mainland; the haze has become more frequent in more and more cities. People's lives are shrouded in haze, physical and mental health is jeopardized. Whether travel or home, the quality of life is affected. With the unpleasant citizens blaming the haze of the smog on the exhaust of automobile emissions, and with the country's fuel consumption regulations are becoming increasingly stringent, the auto companies are facing tremendous pressure and their enthusiasm for the development of new energy vehicles has been stimulated. At present, as pure electric vehicles are limited by battery technology, more and more hybrid electric vehicles appear on the auto market. Most of the programs are multi-mode drive hybrid drive system. How to achieve a predominant balance between driving pleasure and low emissions? It is an urgent topic need to be overcome.

Based on this situation, GAC (Guangzhou Automobile Group Co., Ltd.) has developed an e-CVT (electric coupled CVT with a motor and a generator built-in) named G-MC (GAC-Mechatronics Coupling). This kind of HEV scheme is gradually favored by automobile manufactures all over the word because of its high efficiency, energy saving, compact structure and simple design principle. This paper will describe the technology of the new developed hydraulic control system basis on the G-MC system.

2. Hybrid Drive System and E-CVT

Figure 1 shows the block diagram of the new energy vehicle hybrid drive system. The hydraulic control system is integrated in the e-CVT named G-MC system, and controlled by the CCU a kind of transmission controller through a Controller Area Network (CAN). The engine is a 1.5 liter Atkinson cycle engine developed specially for hybrid electric vehicle, combined with e-CVT, has better fuel consumption.

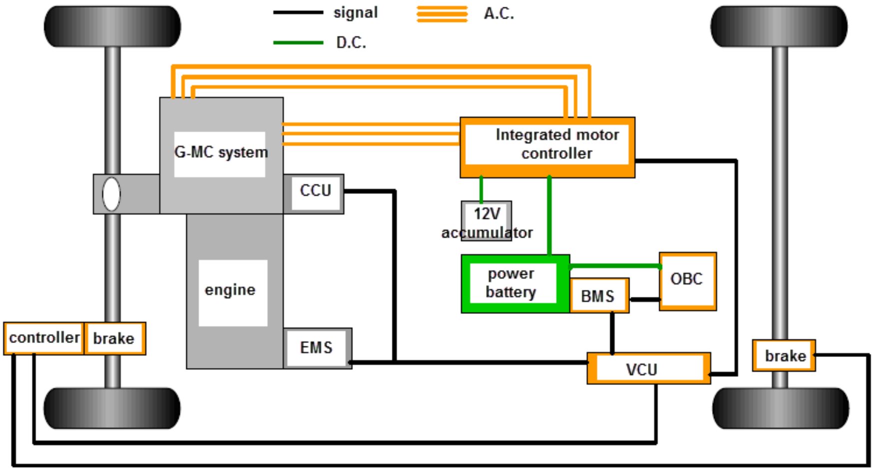
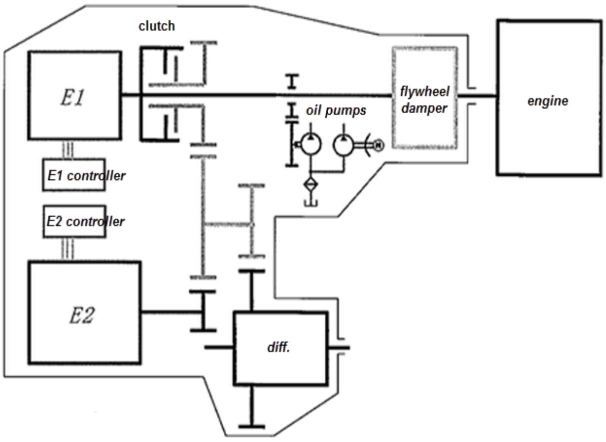


Figure 1. Block diagram of the new energy vehicle hybrid driven system

Figure 2 shows the system structure of the G-MC. The generator E1, driving motor E2, hydraulic control system and transmission box are integrated into one electric coupled CVT, e-CVT. The engine is arranged coaxially with the generator E1, the former transmitting torque through the rigidly connected axial to the latter. A clutch is layout on the input shaft connected to the engine, and the on/off of the engine torque transmission is achieved by the clutch engagement and disengagement control.

Figure 3 shows the G-MC, the driving motor E2 is arranged in parallel with the generator E1. Like the engine, it can transmit power through a gear train and realize continuously variable transmission.



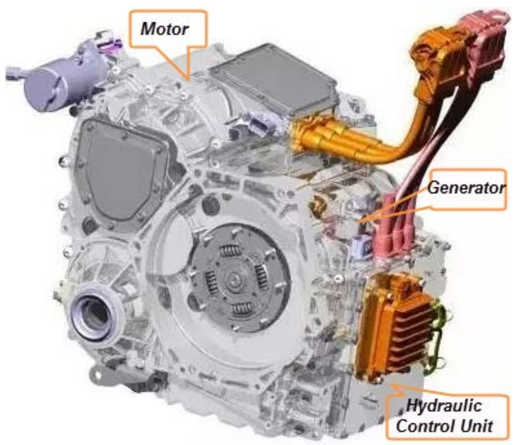


Figure 2. System structure of e-CVT **Figure 3.** Mechatronics Coupling G-MC

The G-MC system has simple and compact structure, and is suitable for A/B platform passage car. Because it has four operating modes, its maximum efficiency can up to 96%, and can be applied to the PHEV/HEV type.

Figure 4 shows the four operating modes of the Hybrid Drive system, namely EV drive mode, extended range mode, Hybrid drive mode and engine drive mode [1]. In the EV drive mode, the energy is provided entirely by the battery, the driving motor E2 drives the vehicle, the engine does not work, and the clutch is disengaged. In the extended range mode, the vehicle is driven by the driving motor E2, and the engine drives generator E1 operating to supplement electric energy. At this mode

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the clutch is still disengaged. In the hybrid drive mode, the clutch is engaged, and the driving motor E2 and the engine simultaneously output power to drive the vehicle. The generator E1 adjusts the power generation torque to achieve engine load regulation, so that the engine can operate in a high efficient economic area and maintains the battery SOC (State of Charge) balance. In the engine drive mode, the clutch is engaged, the engine operates in an efficient area, driving the vehicle directly. The vehicle is in high-speed cruise.

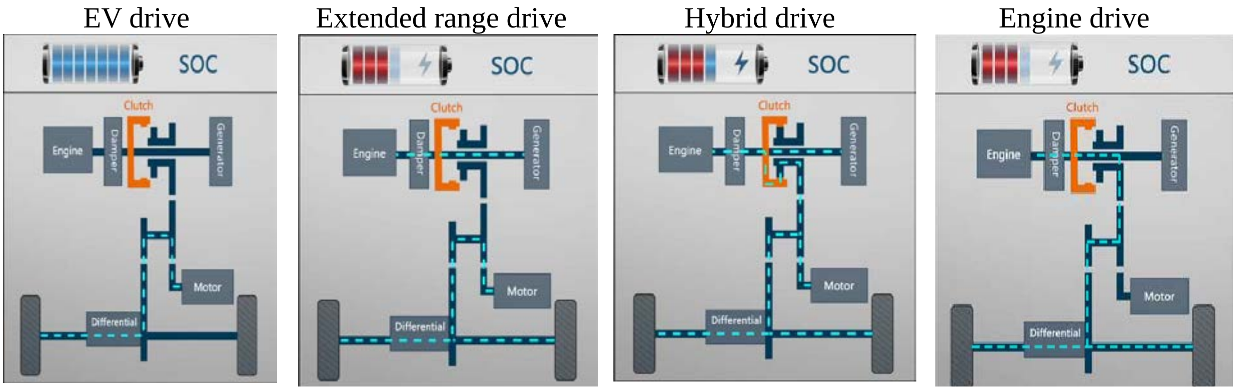

Figure 4. Operating modes

3. Design of Hydraulic Control System

In the G-MC system, the functions that the hydraulic control system needs to realize, or the main work targets include: oil cooling of the generator E1 and the driving motor E2, engaging or disengaging control of the clutch, lubrication of some gears and bearings, and cooling and lubrication of the clutch. The design of the hydraulic control system needs to develop targeted functions according to the needs of the lower and inner users of the G-MC system.

The follow formula (1) is used to calculate oil flow rate [5]:

$$Q = C_q A \sqrt{\frac{2|\Delta P|}{\rho}} \tag{1}$$

where$Q$is the flow rate L/min, C_q is the hydraulic resistance factor, A is the flow aream^2, ΔP is the pressure drop of a control port or a kind of throttle orifice bar, ρ is the oil density Kg/dm^3.

By analyzing the motor cooling, clutch engaging or disengaging and cooling, and gears and bearings lubrication flow requirements of the G-MC under various typical operating conditions, consider the system leakage at the same time to determine the hydraulic system design flow rate of 6.5L/min to 22.5L/min.

The follow formula (2) (3) is used to calculate oil pressure drop [5]:

$$\Delta P = \frac{Q^2 \rho}{2{C_q}^2 A^2} \tag{2}$$

$$P_{main} = \Delta P_1 + \Delta P_2 + \cdots + \Delta P_N \tag{3}$$

Where P_{main} is the main line pressure of the hydraulic control systembar.$\Delta P_N$is a pressure drop of the oil throttle orifice sequence bar.

By analyzing the clutch control pressure requirements, the motor cooling system maintains system pressure requirements when the generator E1 and driving motor E2are thermal balance under any oil temperature conditions, the maximum working pressure of the hydraulic system is determined to be 1.35 MPa. And in the hybrid driving mode or engine driving mode, if the hydraulic control module fails, in order to ensure the minimum thermal balance requirements of the driving motor E2, and the vehicle can still run in the other two modes, the working pressure of the hydraulic system is the maximum pressure 1.35MPa. The minimum system pressure is design as 0.32MPa, by considering the design characteristic of Hydraulic Control Unit (HCU) and the minimum demand of energy consumption of driving motor E2 cooling system.

3.1. Principle Design of Hydraulic Control System

Figure 5 is the function program of the hydraulic control system. Based on the functional requirements of the e-CVT for the hydraulic control system, a solution that can implement these functions was developed. This solution mainly includes oil source of system, hydraulic control unit, clutch control function, generator E1 and driving motor E2 oil cooling system.

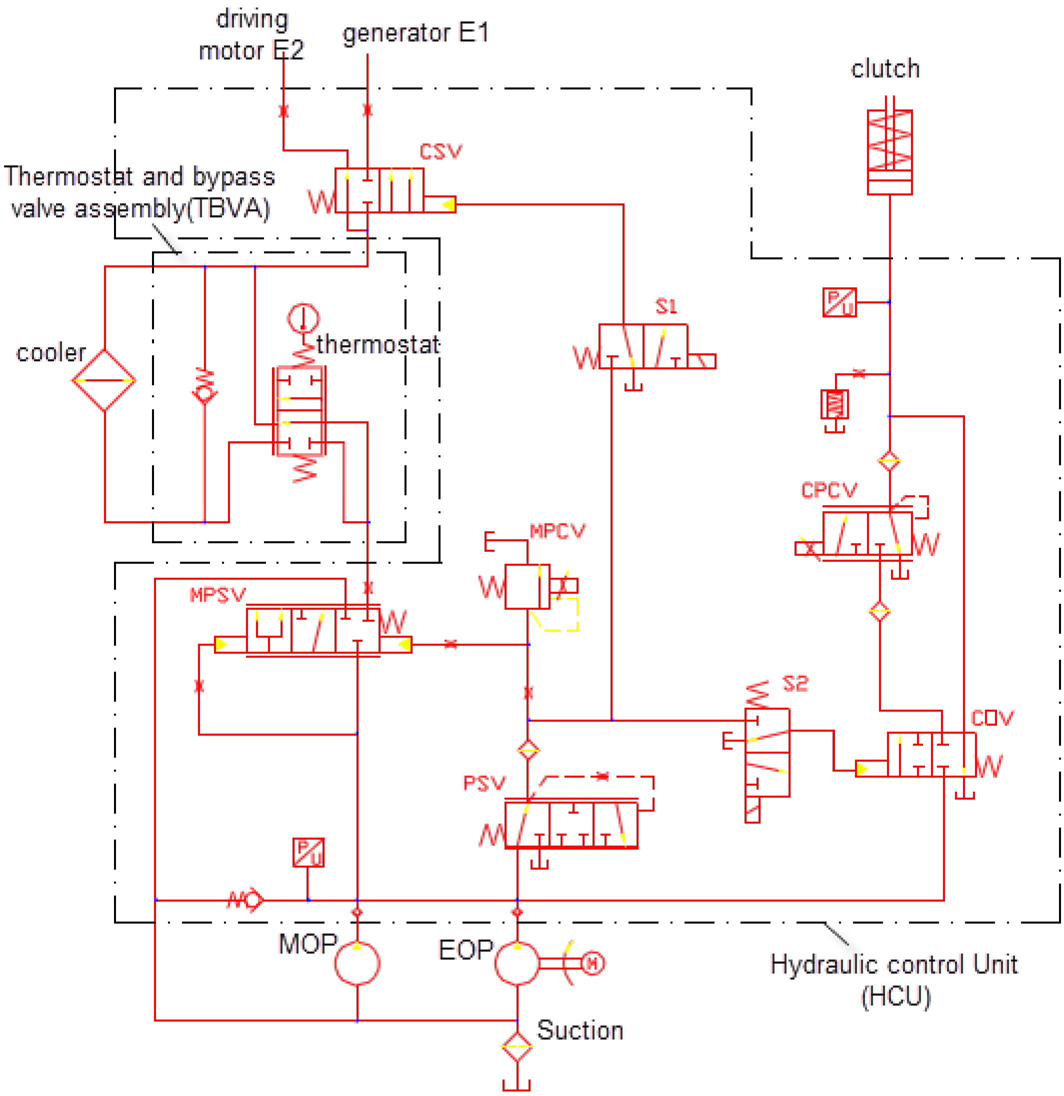

Figure 5. Function program of the hydraulic control system

3.2. Oil Resource of the System

The oil source is a double-pump scheme included a mechanical driven oil pump (MOP) and an electric driven oil pump (EOP)[3]. The displacement of the mechanical oil pump is 10.50 mL/r. It is driven by a pair of gear pairs on the input shaft of the e-CVT connected to the engine. The displacement of the electric oil pump is 4.10 mL/r. The pump is driven by a Direct Current (DC) brushless permanent magnet motor, which controlled by its associated controller. Under different operating conditions, take the different system oil temperature for example, because the generator E1 and driving motor E2 have different thermal balance state, the corresponding needs of cooling oil flow rate of the generator E1 and driving motor E2 are also different. At this time, the speed of the oil pump motor of the EOP can be controlled by the controller, and the output oil flow rate of the EOP can be adjusted in real time to reduce energy consumption and increase efficiency.

When the hybrid drive system is operating in the pure electric mode, the system is completely supplied oil by the EOP, and mainly meets the oil required for cooling the driving motor E2. In the extended range mode, the MOP and the EOP simultaneously supply oil. When the engine speed is lower than a certain value, the output flow rate of the MOP is insufficient to meet the system requirements. At this time, the EOP participates in the work and assisted supplies oil to the system; when the engine speed increases at a certain speed, the output flow of the MOP can meet the system requirements, then the EOP stops working. In this mode, the flow rate provided by the two pumps is mainly used to cool the generator E1 and the driving motor E2. In the hybrid drive mode, the operation way of the two pumps is similar to that in the extended range mode. In this case, except that the cooling of the motor E1 and the driving motor E2 requires a certain amount of flow, the clutch engaging control also requires a certain amount of flow. When the clutch is controlled to engage, the clutch coupling response time is required to be a shorter certain value because of driving comfort requirements. Therefore, when the total volume of the oil conduit is constant, the oil passage is required to have sufficient flow rate while satisfying the control pressure.

Analyze from the angle of safety and reliability, because the hybrid car's electrical system is more complex, it is prone to failure. Once the EOP stops working, the system can rely on the flow provided by the MOP to ensure that the vehicle can be driven in the engine drive mode.

3.3. Design of the Hydraulic Control Unit (HCU)

The function of the hydraulic control Unit is to realize the flow distribution and pressure control of various functions of the hydraulic system [5]. After confirming the system oil function program, select the appropriate solenoid valves and sensors, and then complete the design calculation of the hydraulic spool valves and other hydraulic accessories such as accumulator, check valves, etc., and complete the design calculation of the hydraulic control module valve plate oil channel [4], to avoid hydraulic shocks, and do the hydraulic control system simulation. The following will focus on the design of spool valve.

The key design points of the spool valve mainly include: the form of the valve port, the clamping force, and the flow force. The following analysis of the flow force. Figure 6 is the two forms of the spool valve port.

The spool valve with annular orifice (form a) is shown in the figures below, the spool edge is assumed to have a rounded corner radius rc with dc as clearance on diameter between the spool and the sleeve.

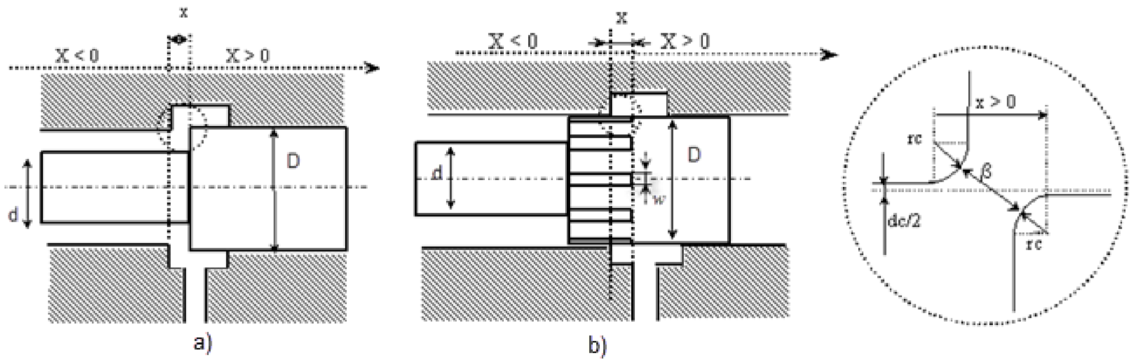

Figure 6. Form of the spool valve port-form a) and b)

the flow area A is:

$$A = \pi \cdot D \cdot x \tag{4}$$

A spool valve with a rectangular orifice (form b) is shown in the figures above, the spool sides have been assumed to have a rounded corner of radius rc with dc as clearance on diameter between the spool and the sleeve valve.

The flow area A is:

$$A = n \cdot w \cdot x \tag{5}$$

The spool valve controls the flow as the spool moves, changes the start of the valve opening or the size of the opening, and also generates hydraulic power at the same time. Flow force has a major impact on the performance of the spool valve. According to the momentum law of the liquid flow, there are two kinds of flow force acting on the spool: static flow force and transient flow force. Static flow force is the force acting on the valve spool due to the change of momentum when the valve spool is moved and the opening is fixed. The effect of the static flow force on the spool valve performance is increasing the force required to operate the spool valve. In the case of high pressure and large flow, this force will be very large, making the manipulation of the spool an outstanding hard problem. At this point, measures must be taken to compensate or eliminate this force. Figure 7 shows the jet angle θ in an annular spool valve with clearance between spool and sleeve. Figure 8 is the curve of hydraulic resistance factor [5].

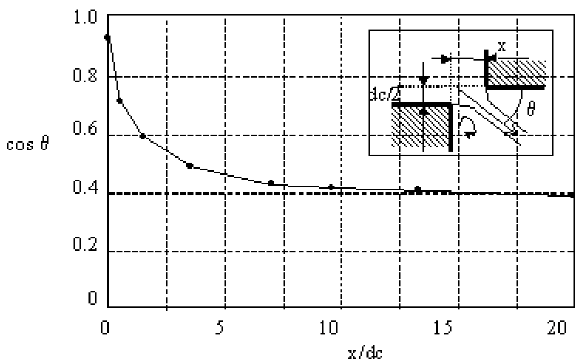

Figure 7. The jet angle θ in an annular spool valve

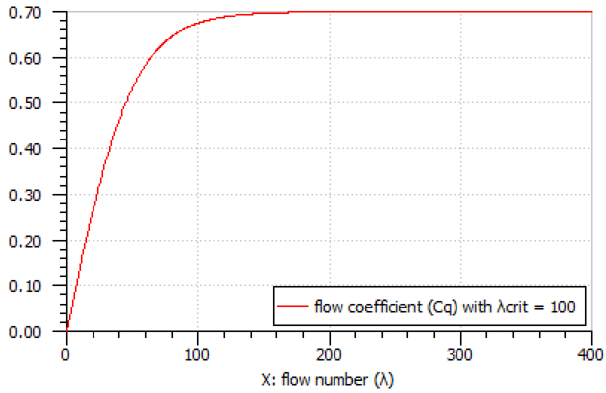

Figure 8. The curve of hydraulic resistance factor.

In figure 7, where x is the valve under lap, dc is the clearance on diameter and θ, the flow jet angle. The value returned is the corresponding angle in radians.

The formula (6) is used for static flow force:

$$F_s = 2 \cdot C_q \cdot A \cdot \Delta P \cdot cos\theta \tag{6}$$

Compensation method for static flow force: using a special-shaped valve chamber; opening oblique holes in the valve sleeve so that the momentum of the liquid flowing out and flowing into the valve chamber can neutralize each other, thereby reducing the axial flow force; changing the neck size of the valve spool, there is a large pressure drop when the liquid flows through the valve spool, so as to generate unbalanced hydraulic pressure on both ends of the valve spool, to offset the axial hydraulic force, etc., which are practical examples used in practice. The static flow force tends to close the valve port, which is equivalent to a restoring force. Therefore, there is another effect on the performance of the spool valve to make the work of the spool valve to be stable.

Transient flow force is the force acting on the valve spool due to acceleration or deceleration in the valve chamber during the movement of the spool valve (when the size of the opening changes). This force is only related to the movement speed of the spool (that is, the rate of change of the valve opening) and has nothing to do with the valve opening itself. When the opening of the valve port changes, the axial velocity of the oil in the valve chamber corresponding to the length l between the

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two valve ports also changes, that is, there is an acceleration or deceleration, and the spool is subjected to an axial reaction F_t, which is the transient flow force, its calculation formula (7):

$$F_t = -m_0 \frac{d_v}{d_t} = -\rho A_s l \frac{d_{(A,v)}}{d_t} = -\rho l \frac{d_q}{d_t} \tag{7}$$

Comprehensive formula (1) (4) (5) (7), when the pressure difference before and after the valve port is the same or little change, transient flow force $F_t$is:

$$\begin{cases} F_t = -\pi C_q D l \sqrt{2\rho\Delta P} \frac{d_{x_v}}{d_t}, & valveportis\ typea \\ F_t = -C_q w l \sqrt{2\rho\Delta P} \frac{d_{x_v}}{d_t}, & valveport\ istypeb \end{cases} \tag{8}$$

When the oil flows out from the valve chamber, the oil, the corresponding length is , is accelerated when the opening of the valve port is increased, and the oil is decelerated when the opening is decreased. In both cases, the direction of the transient flow force is opposite to the direction of movement of the valve spool. It acts as a resistance to resist the spool movement, which is equivalent to a damping force. In this case, $l$takes a positive value, which is called the “positive damping length” of the spool valve. Conversely, when the oil flows into the valve chamber, the change of the flow velocity caused by the opening change of the valve port makes the direction of the transient flow force to be same as the movement of the valve spool. The force acts to help the movement of the valve spool, which is equivalent to a negative damping force. Then $l$takes a negative value, which is called the “negative damping length” of the spool valve. The "negative damping length" on the spool valve is one of the causes of the unstable operation of the spool valve. If there are several valve chambers connected in series, the stability of the valve spool is determined by the combined effects of the damping lengths of the valve chambers.

After the spool valve has been initially designed, the design of the valve body can be carried out. According to the interface information of the hydraulic control unit and the G-MC system (such as the positions of the bolt holes, the connection positions of the oil channels, the installation environment, and the layout space, etc.), the arrangements of the solenoid valves, the spool valves, and the one-way valves and other accessories, and the arrangements of the connection of bolts between the valve body, are determined first. Then discuss the specific layout of the oil circuit, to realize the 3D entity for the function diagram of the Hydraulic Control Unit (HCU).

In the HCU, the pressure of the system main line is controlled by a combination of a solenoid valve MPCV (Main Pressure Control Valve) and a spool valve MPSV (Main Pressure Spool Valve). MPCV is a normally high linear proportional solenoid valve, as pilot valve of the MPSV, along with MPSV, to achieve the oil pressure control via the dynamic balance of the force acting on the spool. The flow control of the motor cooling oil circuit is jointly controlled by a solenoid valve S1 (a kind of on/off solenoid valve) and a spool valve CSV (Cooling Shift Valve). By controlling the control current of valve S1, the cooling flow input of generator E1 is switched on and off. The cooling flow input to the two motors is controlled by the MPCV. The CPCV (Clutch Pressure Control Valve) is a direct-drive, normally-low solenoid valve used for clutch pressure control. According to the real-time requirements of the vehicle's operating conditions for clutch engagement, the control pressure is adjusted in real time. The solenoid valve S2 (a kind of on/off solenoid valve) and spool valve COV (Clutch Opening Valve) combine to realize the safety control of the clutch control circuit. Thermostat and bypass valve assembly (TBVA) integrates a thermostat and check valve to adjust the cooling demand according to the real-time oil temperature status.

3.4. Design of the Motor Cooling system

In the G-MC system, the generator E1 and driving motor E2are integrated in the housing. electric machines(EM) cooling is the primary solution need to be solve for the hydraulic system. The EM cooling adopts the oil cooling scheme, compared with the water cooling scheme, there is no independent water-cooled pipeline, and the entire G-MC system has a compact structure and a high degree of integration. As shown in Figure9, forced cooling with spray pipes is used. According to the distribution of the EM thermal simulation temperature field, a number of oil spray holes are evenly arranged on the oil spray pipe above the line pack on the side of each stator of the EM. The hole diameter of the oil spray hole is about 1mm, and the interval is 20°. It is 2.5mm to 4.5mm away from the stator winding of the EM. There are transitional oil pipes between the oil spray pipes before and after the EM, and several oil spray holes with the same specifications can be arranged on the transition oil pipes to achieve cooling of the stator core of the EM. In addition to the oil spray nozzle for the EM cooling nozzle, there are also individual for some bears and gears lubrication [2].

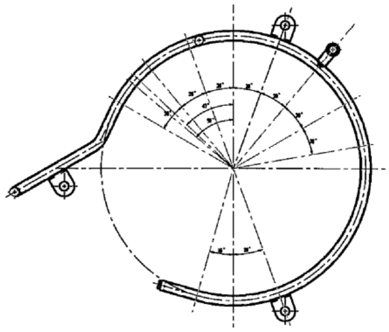

Figure 9. Instance for spray pipe design

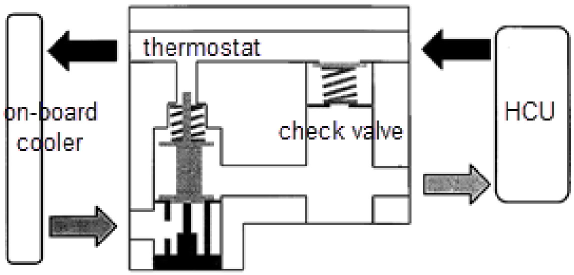

Figure 10. Thermostat and Bypass Valve Assembly(TBVA)

The system oil cooling adopts a radiator with a tube-and-air-cooled structure, and the radiator oil circuit is connected in parallel with a thermostat and bypass valve assembly (TBVA), which integrated with a thermostat and a check valve. As shown in Figure 10, it is the working principle of the TBVA, the thermostat is a temperature-controlled switch valve, when the oil temperature is lower than a certain temperature value, the oil does not need to be cooled, and directly into the oil spray pipes for forced cooling; when the oil temperature rises to a certain value, the thermostat switches the work position, the oil is cooled by the radiator and then enters the oil spray pipes to cool the EM. The check valve is a safety valve and the opening pressure is generally designed to be around 3.0 bars.

At the same time, because of the existence of the valve CSV and valve S1, the system can select the cooling object and adjust the cooling flow in real time according to the real-time operating mode of the G-MC system and the real-time thermal balance requirements of the generator E1 and driving E2 to improve the efficiency and reduce the energy consumption.

3.5. Clutch Control

Wet single clutch specially developed for G-MC system. The function of the clutch in the system: the switch from the pure electric mode or the extended range mode to the hybrid drive mode controls the switching of the output power path of the engine to the differential. When the clutch is engaged, some or all of the engine output torque is transmitted through the clutch to the end of the lower stage reducer and coupled with the torque output from driving motor, to drive the vehicle forward. When the clutch is disconnected, the engine output torque can only drive the generator to generate electricity.

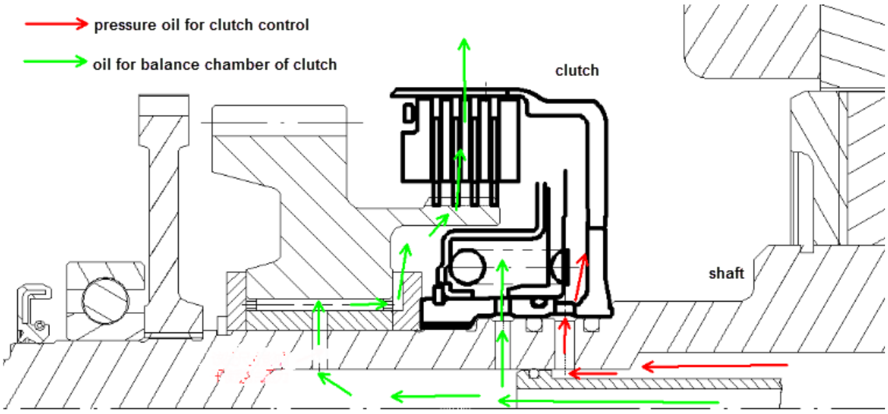

Figure 11. The oil circuit for clutch control

Figure 11 shows the clutch operating oil circuit distribution. The red arrow path indicates the clutch control pressure oil circuit. The oil pressure is set according to the clutch pressure-torque characteristics. The general setting range is 0 to 10 bars. In order to meet the response time requirement of the power switch, the pressure oil needs to meet a certain flow requirement, which is determined by the sum of the pressure chamber volume and the related oil passage volume, and the response time requirement when the clutch is operated. The green arrow indicates the clutch balance oil chamber and the cooling lubricant path.

4. Dynamic Simulation for Hydraulic Control System

In the design process, the dynamical simulation for hydraulic control system should be synchronized. The simulation of the distributed flow and controlled pressure of the hydraulic control system mainly investigate the design rationality of the valve under lap, the overlap and the stroke, and the flow area of the oil passage of the valve body, and the size of the orifices, and the structure of the oil spray pipes. And provide reference for design optimization of HCU and EM cooling system. At present, in the industry, AMESim software is used more often for the simulation of the hydraulic system. As shown in Figure 12, the simulation model of the hydraulic control system established by using AMESim software. In order to make the simulation model looked more concise, the models of the oil spray pipes for cooling generator E1 and driving motor E2 and radiators are done as super-component.

DEXRON-VI is used for the system oil. Enter the volumetric efficiency of the MOP and the EOP. Set the system input parameters.

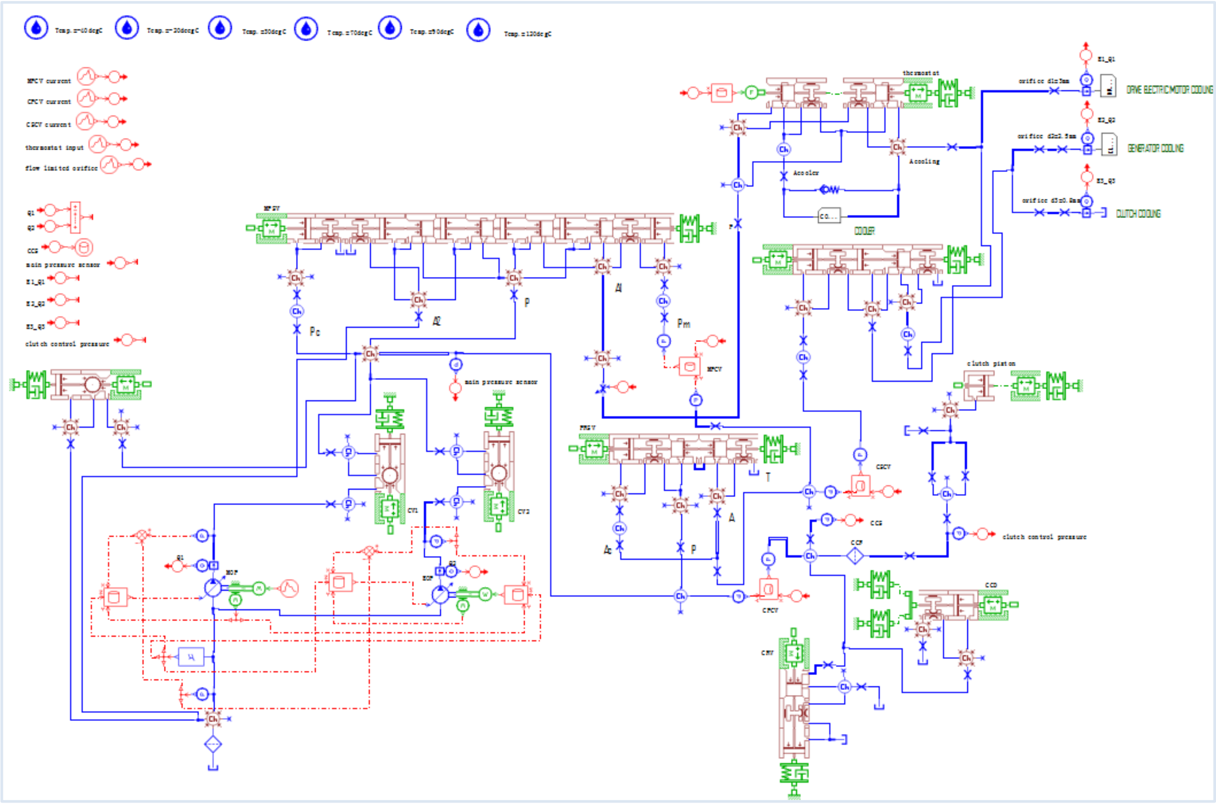

Figure 12. Simulation model of the hydraulic control system

Figure 13 shows the simulation result of the main line pressure of the hydraulic control system. Simulation input conditions: oil temperature is set at 50°C, fluid dynamic viscosity is 17.84cP, MPCV control current is from 0 to 1 Am, then from 1Am to 0, ramp is changed, and the oil source is input with a constant flow rate of 32.5 L/min.

As shown in Figure14, it is the flow rate and pressure simulation results of the cooling system when the G-MC system is switched from the pure electric mode to the extended range mode. Simulation input conditions: the system oil temperature is set to 100°C; the absolute viscosity of the oil at this time is 5.21cP.The oil source input a constant flow of 17.5L/min, MPCV control current 1A, the oil temperature has reached a set value, thermostat and bypass valve is open, the valve CSV is opened from the closed state.

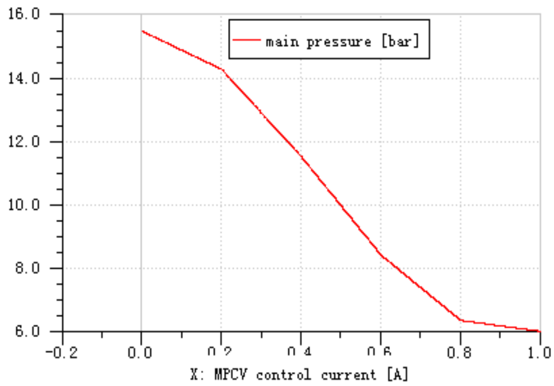

Figure 13. The simulation result of the main line pressure of the hydraulic control system

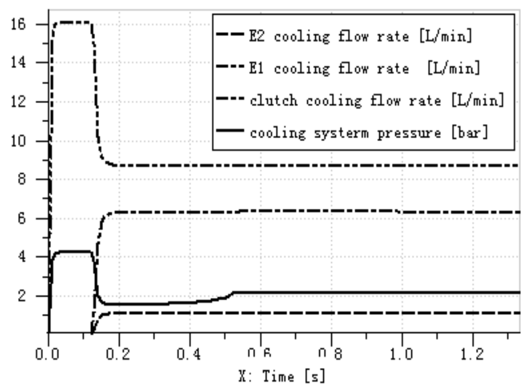

Figure 14. The flow rate and pressure simulation results of the cooling system

5. Test Result

Figure 15 shows the main line pressure test results of the hydraulic control system. Test input conditions: oil temperature is set at 50°C, fluid dynamic viscosity is 17.84cP, MPCV current controlled from 0 to 1 Am, then from 1Am to 0, changed as ramp, and the oil source is input with a constant flow rate 32.5 L/min.

Figure 16 is the cooling system test result. Test input conditions: oil temperature set 100°C, constant input oil flow rate 17.5L/min, MPCV current controlledfrom 0 to 1 Am, changed as step, current step length 0.2Am, hold time 2s for each step. Thermostat and bypass valve is at open state, valve CSV open.

With a same input condition set, the error between the test result and the simulation result is within the acceptable range.

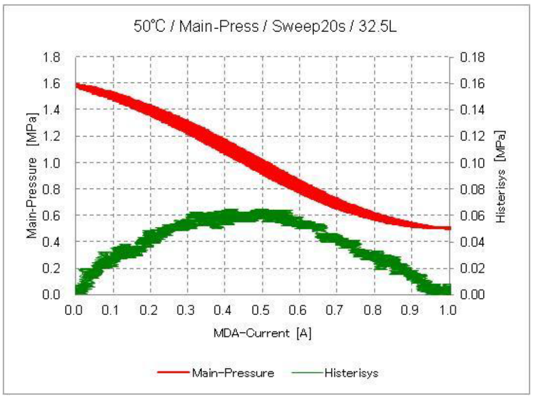

Figure 15. The main line pressure test results of the hydraulic control system

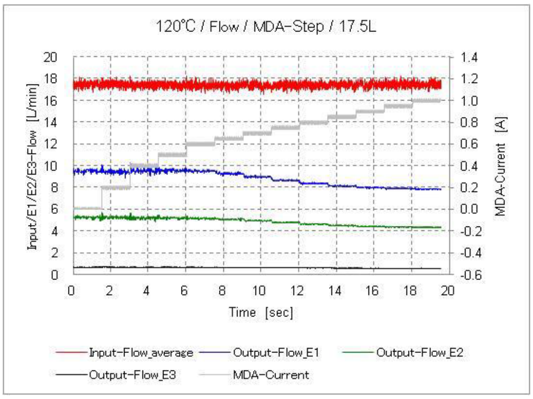

Figure 16. The cooling system test result

6. Conclusion

For the design of the hydraulic control system of G-MC system, first the functions of the sub-level customers of hydraulic control system should be defined. According to functional requirements, formulate hydraulic control system solutions that can implement these functions, and set design goals. Generally, the define of these functions is mainly determined by the control strategy of the vehicle. Before the design of the hydraulic control system, the more detailed information of the vehicle control strategy input, the higher accuracy for the hydraulic system design.

A new double-pump scheme included a mechanical driven oil pump (MOP) and an electric driven oil pump (EOP). The EOP and the MOP cooperate in real time to supply oil, which can efficiently support the flow output of the hydraulic control system, and at the same time reduce the energy loss caused by waste of system flow, and improve efficiency.

The accurate hydraulic control system simulation model can be used to simulate the flow rate, pressure and response of the hydraulic system under the typical operating conditions of the G-MC system, and provide reference for the set and optimization of the design parameters of the system.

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