Research Article

Modeling of Power Transition in Full Power Shift of Hydromechanical Transmission

Yong Bao, Zaimin Zhong, and Shujun Yang

School of Automotive Studies, Tongji University, Shanghai 201804, China
Hebei Key Laboratory of Special Delivery Equipment, Yanshan University, Qinhuangdao 066004, China

Correspondence should be addressed to Shujun Yang; ysj@ysu.edu.cn

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Abstract

This paper shall explore the dynamics of power transition in full power shift of hydromechanical transmission (HMT) and focus on the ideal target displacement ratio. An arithmetic two-range HMT is taken as the research object. A mathematical model of power transition in full power shift is established, including the hydraulic transmission unit model and the brake torque model during the double brakes overlapping. Aiming at the constant output power of HMT in the shift process, a prediction model of the displacement ratio target value is established, and the prediction method is proposed. By combining theoretical analysis and experimental research, it proves that the power transition model can describe the power transition process. And the prediction method of the displacement ratio target value proposed in this paper can complete the power transition when the double brakes overlap. In the power shift process, the output power can be transmitted normally in full power. The power transition model and the prediction method of the displacement ratio target value can provide theoretical and engineering references for the full power shift of HMT.

1. Introduction

HMT is a dual power flow transmission consisting of hydraulic transmission unit and mechanical transmission unit [1, 2]. It can realize high-power stepless speed regulation and is suitable for heavy-duty vehicles [3, 4].

The theory of design and principle of HMT has been basically mature after years of research. Kress [5] studied the performance of four solutions of HMT. Orshansky and Weseloh [6] and Kirejczyk [7] analyzed the operation of three-range split power HMTs. Ross [8] designed an input-coupled dual mode HMT for heavy-duty trucks. Renius and Resch [9] and Xu et al. [10] designed three-range and six-range HMTs for tractors, respectively, and the performances are good. Liu et al. [11] designed a five-range synthesized power split HMT with planetary gears of gear ratio extension, using the modular method. Krauss et al. [12–14] systematically analyzed various HMT schemes and carefully studied the characteristics of HMT. Rossetti and Macor [15, 16] regarded the design of the input-coupled HMT (using three shaft ordinary gear and a simple planetary gear) as an optimization problem and achieved good results. Sun et al. [17–19] designed a power reflux hydromechanical transmission based on a torque converter instead of the traditional hydraulic transmission unit.

In the control of HMT, Savaresi et al. [20, 21] designed a controller for a six-range power split continuous variable transmission. Zhu et al. [22, 23] studied the control strategy for a seven-range HMT based on the shifting time. Cheong et al. [24, 25] studied the control of hydromechanical hydraulic hybrid vehicles. Wei et al. [26, 27] pointed out that there are problems such as speed fluctuation, pressure impact, and power interruption during the power shift of the two-range arithmetic HMT (shown in Figure 1) and analyzed the factors affecting the shift quality. In traditional power shift, the target clutch would be engaged after the current clutch is disengaged, which would cause a transitory output power interruption. There is an inertia phase during the shifting process. The pressure building-up phase in the traditional power shift process is an uncontrollable passive...
process. When HMT shifts to the target range, the pressure building-up process just begins, which would cause a transitory output power interruption.

Based on the compressibility of the hydraulic unit, Hu et al. [28] proposed the feasibility of the two-range arithmetic HMT (shown in Figure 1) working in the stage of the dual range. Based on the double brakes overlapping, Yang et al. [29, 30] proposed a full power shift method and analyzed the characteristics. Because of the double brakes overlapping in the full power shift, there is no inertia phase, and only the torque phase is retained. The pressures in the closed hydraulic circuit can be regulated by actively adjusting the displacement ratio. Interchange between two pressure sides and power transition can be completed when the double brakes overlap. Before the current brake is disengaged, the pressure building-up process is completed. But the mathematical model of power transition in full power shift of HMT, which can describe the principle of power transition, has not been established. The pressure building-up process in the full power shift is an active and controllable process. The pressure building-up process is changed from a passively uncontrollable process to an active controllable one. The power transition can only be completed when the displacement ratio is adjusted to the target value. But the prediction method of the displacement ratio target value has not been proposed.

This paper shall build the power transition model in full power shift to explore the dynamics of power transition and propose the prediction method of the displacement ratio target value to actively control the pressure building-up process. A full power shift approach of HMT is described in Section 2. A mathematical model of power transition in full power shift is established in Section 3, including the hydraulic transmission unit model and the brake torque model during the double brakes overlapping. Aiming at the constant output power of HMT in the shift process, a prediction model of the displacement ratio target value is established in Section 4, and the prediction method is proposed. In Sections 5 and 6, by combining theoretical analysis and experimental research, it proves that the power transition model can describe the power transition process. And the prediction method of the displacement ratio target value proposed in this paper can complete the power transition when the double brakes overlap.

The power transition model can describe the principle of power transition, and it can be used to simulate and analyze the dynamics of power transition, which is helpful to understand the principle and develop control strategies. The prediction method of the displacement ratio target value is deployed in the application layer algorithm of the transmission controller, which can actively control the power transition process. The power transition model and the prediction method of the displacement ratio target value can provide theoretical and engineering references for the full power shift of HMT.

2. Full Power Shift Method of HMT

An arithmetic two-range HMT is taken as the research object. Figure 1 shows its principle. The variable-displacement hydraulic component (VDHC) \( P \) and the fixed-displacement hydraulic component (FDHC) \( M \) form the hydraulic transmission unit. The planetary gearsets \( K_1, K_2, \) and \( K_3, \) together with the brakes \( C_{H1} \) and \( C_{H2} \), constitute the mechanical transmission unit and confluence unit. \( n_1 \) and \( n_0 \) represent, respectively, the input and output speed. The ranges and working parts are shown in Table 1. In the H range, the brake \( C_{H3} \) is engaged, and the planetary gearset \( K_1 \) works. In the HM range, the brake \( C_{H2} \) is engaged and both the planetary gearsets \( K_2 \) and \( K_3 \) work.

A full power shift approach of HMT is proposed based on the double brakes overlapping in [30]. The full power shift is divided into 5 stages: current stage, prior stable stage, power transition stage, poststable stage, and target range. For HMT working at the current range, when the ideal power shift timing comes, the target range brake would be engaged with zero speed difference. Then, HMT comes to the prior stable stage, and its torque and power characteristics remain unchanged. According to [30], regulating properly the displacement ratio can make the high- and low-pressure sides of hydraulic circuit interchange. At the power transition stage, the original low-pressure side pressurizes to the target pressure, and HMT comes to the poststable stage with torque and power characteristics remaining identical with those of the target range. The current range brake is disengaged at zero torque, and HMT enters the target range.

In the traditional power shift process, there is a certain time interval between the current range brake disengaging and the target range brake engaging, or there is a short time friction overlap between the two brakes. There is an inertia phase during the shifting process. So, there is the sudden change of speed in the traditional power shift process. But because of the double brakes overlapping in the full power shift, there is no inertia phase, and only the torque phase is retained. So, there is no the sudden change of speed in the full power shift.

In traditional power shift process, after the target brake is engaged, the high-pressure side passively completes the pressure building-up process under the action of
transmission load torque. The pressure building-up process in the traditional power shift process is an uncontrollable passive process. Hence, there is pressure impact in traditional power shift. When HMT shifts to the target range, the pressure building-up process just begins. So, there is power interruption during the power shift.

In the power transition stage of full power shift, the displacement ratio is actively adjusted to the target value. The pressure building-up process in the full power shift is an active and controllable process. Hence, there is no pressure impact in the full power shift. And the pressure building-up process can be accomplished during the double brakes overlapping. Before the current brake is disengaged, the pressure building-up process is completed. In the full power shift process, output power can be transmitted normally in full power. The power transition can be completed during the double brakes overlapping by controlling the high pressure of the target range to the target value actively. In order to improve shift quality, the high pressure of active control needs to be consistent with the pressure of the target range. The displacement ratio target value directly determines the high pressure of active control. The inaccurate prediction value of the displacement ratio will lead to the difference between the high pressure of active control and the pressure of the target range, resulting in shift impact and reducing shift quality. Hence, it is necessary to establish a power transition model and to explore the prediction method of the displacement ratio target value.

### 3. Power Transition Model

According to [29, 30], in the power transition stage, the interchange between two pressure sides of hydraulic transmission unit is completed, and torque can transmit from the current brake to the target brake. Hence, the power transition model should include the hydraulic transmission unit model and the brake torque model.

In the full power shift, if the input speed remains unchanged, the speed of VDH C will be constant. The hydraulic transmission unit completes the power transition process through the pressure releasing and building up of the closed hydraulic circuit. Hence, in the modeling of the hydraulic transmission unit, the mass flow model, pressure model, and hydraulic component model that affect the hydraulic circuit are mainly considered. The composition of the hydraulic transmission unit model is shown in Figure 2. $V_1$ and $V_2$ are the two closed chambers; $n_p$ and $n_m$ are the rotational speeds of the two hydraulic components; $T_p$ and $T_m$ are the torques of the two hydraulic components; $q_{po}$ and $q_{pi}$ are the input flow and output flow of VDH C, respectively; $q_{m}$ and $q_{mo}$ are the input flow and output flow of FDHC, respectively.

![Figure 2: Composition of the hydraulic transmission unit model.](image)

In the hydraulic transmission unit model, the on-way loss of oil flow along pipelines is neglected, and the high- and low-pressure circuits are simplified into two closed chambers $V_1$ and $V_2$. The two hydraulic components are energy conversion devices in the hydraulic system, whose input and output are both mechanical power. Considering the causality between input and output, when the double brakes overlap, the inputs are the speeds of two hydraulic components and the outputs are the torques of two hydraulic components.

The schematic diagram of the power transition model is shown in Figure 3. In the figure, the speed model and torque model of hydraulic components constitute the models of hydraulic components, and the mass flow model and pressure model of chambers constitute the models of hydraulic pipes.

#### 3.1. Hydraulic Transmission Unit Model

##### 3.1.1. Principle of Hydraulic Transmission Unit

The hydraulic transmission unit is a volumetric speed regulation system. Its principle is shown in Figure 4. The displacement control mechanism is composed of a proportional valve 9 and a servocylinder 10. The displacement is adjusted by the control current. When the HMT works, one side of the closed hydraulic circuit is high pressure and the other side is low pressure. The high- and low-pressure circuits are separated by the two hydraulic components 1 and 2. The one-way valve (5 or 6) on the high-pressure side is closed. If the pressure on the high-pressure side is too high, the relief valve (7 or 8) will open to release pressure to the low pressure system which uses 4 to limit the maximum pressure in this line. The hydraulic reversing valve 11 and the low pressure relief valve 12 ensure that one of the two sides is low pressure. The boost pump 3 replenishes oil to the low-pressure side through the one-way valve (5 or 6), which can cool the oil in the hydraulic system. The low-pressure relief valve 12 ensures that one of the two sides is low pressure. The boost pump 3 replenishes oil to the low-pressure side through the one-way valve (5 or 6), which can cool the oil in the hydraulic system. The low-pressure relief valve 12 ensures that one of the two sides is low pressure.

At the beginning of the power transmit stage, the displacement control mechanism is adjusted to reduce the original high pressure gradually, but the oil on the low-pressure side can still be leaked through the low-pressure relief valve. When the original high pressure drops to the
same level as the low pressure, the hydraulic reversing valve closes. If the displacement control mechanism is continuously adjusted in the same direction, the original low pressure will start to increase. And the oil on the original high-pressure side will be leaked through the low-pressure relief valve and the original high pressure will become the new low pressure.

3.1.2. Mass Flow Model. According to [29, 30], at the ideal shift timing, the speed difference of to-be-engaged brake is zero. And the speed relationship at the ideal shift timing is as follows:

\[ n_m = \frac{(1 + k_1)(1 + k_2)n_p i_1}{(1 + k_2)(1 + k_1 + k_2)i_2} = \frac{(1 + k_1)(1 + k_2)n_p i_1}{(1 + k_2)(1 + k_1 + k_2)i_2} \]

where \( k_1, k_2, \) and \( k_3 \) are the characteristic parameters of planetary gearsets, respectively. \( n_p \) is the speed of VDHC. \( i_1 \) and \( i_2 \) are the transmission ratios from the input shaft to the hydraulic transmission unit and the mechanical one, respectively.

Before and after the power shift, the functions of two hydraulic components are interchanged and the flow characteristics change. The power flow direction of the hydraulic transmission unit is indicated by “state,” and its value is 0 or 1. When state = 0, the VDHC drives the FDHC. In other words, the outlet of the VDHC and the inlet of the FDHC are connected with the high-pressure side chamber. When state = 1, the FDHC drives the VDHC. In other words, the outlet of FDHC and the inlet of the VDHC are connected with the high-pressure side chamber.

The oil leaks from the high-pressure chamber through the two hydraulic components to the low-pressure chamber.
According to the law of mass conservation, the mass flow on the high side of each hydraulic component is equal to the sum of the mass flow on the low side and the leakage mass flow. According to Section 3.1.1, the mass flow model is established, as shown in equations (2)–(15).

(1) State = 0. When state = 0, the input mass flow of VDHC (the theoretical mass flow) is

\[ q_{m_{pi}} = \frac{q_m V_p \Delta p}{60} \]  

(2)

where \( \rho_2 \) is the oil density in chamber \( V_2, \) \( \text{kg/m}^3; \) \( q_{m_{pi}} \) is the input flow of VDHC, \( \text{m}^3/\text{s}; \) \( \Delta p \) is the displacement ratio; and \( V_g \) is the maximum displacement of VDHC, \( \text{m}^3/\text{r}. \)

Both the hydraulic components are equivalently simplified to the internal leakage mode. Considering the effects of pipeline pressure, oil viscosity, thermal expansion, and oil bulk modulus, the leakage of hydraulic components is

\[ \Delta q = C_s V \Delta p \left( 1 + \frac{\Delta p}{E_1} \right). \]  

(3)

where \( E_1 \) is the bulk modulus of oil, MPa; \( C_s \) is the leakage coefficient of hydraulic components; \( V \) is the displacement of hydraulic components, \( \text{m}^3/\text{r}; \) and \( \mu \) is the dynamic viscosity of oil, \( \text{N}\cdot\text{s}/\text{m}^2. \)

The mass of the oil leaking to the low-pressure chamber is the same before and after expansion. Hence, the mass flow rate of the oil leaking from chamber \( V_1, \) through the VDHC is

\[ \Delta q_{m_{pi}} = \rho_1 \Delta q_{p_{pi}}, \]  

(4)

where \( \Delta q_{p_{pi}} \) is the leakage flow rate of VDHC, \( \text{m}^3/\text{s}; \) \( \rho_1 \) is the oil density in chamber \( V_1, \) \( \text{kg/m}^3; \) \( C_{sp} \) is the leakage coefficient of VDHC; and \( \Delta p_1 \) is the pressure difference between the chambers \( V_1 \) and \( V_2, \) MPa.

The output mass flow rates of the two hydraulic components are

\[ q_{m_{po}} = q_{m_{pi}} - \Delta q_{m_{pi}}, \]  

(5)

\[ q_{m_{mmo}} = \rho_2 q_{m_{mo}} = \frac{q_m V_p \Delta p_2}{60}, \]  

(6)

where \( q_{m_{mo}} \) is the output flow rate of FDHC, \( \text{m}^3/\text{s}. \)

The mass flow rate of the oil leaking from chamber \( V_1, \) through the FDHC is

\[ \Delta q_{m_{m}} = \rho_1 \Delta q_{m}, \]  

(7)

where \( \Delta q_{m} \) is the leakage flow rate of FDHC, \( \text{m}^3/\text{s} \) and \( C_{sm} \) is the leakage coefficient of FDHC.

The input mass flow rate of FDHC is

\[ q_{m_{mi}} = q_{m_{mo}} + \Delta q_{m_{m}}. \]  

(8)

(2) State = 1. When state = 1, the output mass flow rate of FDHC is

\[ q_{m_{po}} = \rho_1 q_{po} = \frac{q_m V_p \Delta p_1}{60}, \]  

(9)

The mass flow rate of the oil leaking from chamber \( V_2, \) through the VDHC is

\[ \Delta q_{m_{po}} = \rho_2 \Delta q_{p_{po}} = \frac{C_s q_m V_g \Delta p_2 \rho_2}{2\pi \mu}. \]  

(10)

The input mass flow rates of the two hydraulic components are

\[ q_{m_{pi}} = q_{m_{po}} + \Delta q_{m_{pi}}, \]  

(11)

\[ q_{m_{mi}} = \rho_1 q_{mi} = \frac{V_g q_{m_{pi}} \rho_1}{60}. \]  

(12)

The mass flow rate of the oil leaking from chamber \( V_2, \) through the FDHC is

\[ \Delta q_{m_{m}} = \rho_2 \Delta q_{m} = \frac{C_{sm} V_g \Delta p_2 \rho_2}{2\pi \mu}. \]  

(13)

The output mass flow rate of FDHC is

\[ q_{m_{mmo}} = q_{m_{mo}} - \Delta q_{m_{m}}. \]  

(14)

3.1.3. Pressure Model of Chambers. The pressure change in the chamber is

\[ \frac{dp}{dt} = \frac{1}{V} \frac{\partial p}{\partial \rho} (q_m - q_{m_0} - \rho V) - \frac{\partial p}{\partial \rho} \frac{\partial V}{\partial t}, \]  

(15)

where \( q_m \) and \( q_{m_0} \) are the input and output mass flow rates of the chamber, respectively, \( \text{m}^3/\text{s}; \) \( V \) is the volume of chamber, \( \text{m}^3; \) and \( p \) is the pressure in chamber, MPa.

The shift time is short, so the oil temperature is almost unchanged. The volume change of high- and low-pressure chambers is neglected. Considering the variation of oil bulk modulus \( E_1 = \rho \frac{\partial p}{\partial \rho} \) and density, equation (16) can be written as

\[ \frac{dp}{dt} = \frac{1}{V} \frac{\partial p}{\partial \rho} (q_m - q_{m_0}) = \frac{E_{11}}{\rho_1 V_1} (q_{m_{po}} - q_{m_{mi}}). \]  

(16)

When state = 0, the chamber \( V_1 \) is the high pressure chamber and the pressure model is

\[ \frac{dp_1}{dt} = \frac{1}{V_1} \frac{\partial p_1}{\partial \rho_1} (q_{m_{po}} - q_{m_{mi}}) = \frac{E_{11}}{\rho_1 V_1} (q_{m_{po}} - q_{m_{mi}}). \]  

(17)

When state = 1, the chamber \( V_2 \) is the high pressure chamber and the pressure model is

\[ \frac{dp_2}{dt} = \frac{1}{V_2} \frac{\partial p_2}{\partial \rho_2} (q_{m_{mo}} - q_{m_{pi}}) = \frac{E_{12}}{\rho_2 V_2} (q_{m_{mo}} - q_{m_{pi}}). \]  

(18)

3.1.4. Model of Hydraulic Components

(1) Torque Model of Hydraulic Components. When state = 0, the VDHC works as a pump and the FDHC works as a motor. Their torque models are
According to equations (25)–(28), the torques of sun gears of \( K_1 \) and \( K_2 \) and brake \( C_{II} \) can be written as

\[
T_{s1} = \frac{(T_s/i_3\eta_3) - k_2 T_m}{(1 + k_1)\eta_{c1} + k_2},
\]

\[
T_{s2} = -\frac{(T_s/i_3\eta_3) + (1 + k_1)T_m\eta_{c1}}{(1 + k_1)\eta_{c1} + k_2},
\]

\[
T_H = \frac{k_1 ((T_s/i_3\eta_3) - k_2 T_m)}{(1 + k_1)\eta_{c1} + k_2}.
\]

The torque relationship of planetary gearset \( K_3 \) is

\[
T_{s3}; T_L; \frac{T_{c3}}{\eta_{c3}} = 1: k_3: - (1 + k_3),
\]

where \( T_{s3} \) is torque of sun gear of \( K_3 \), N-m; \( T_{c3} \) is the torque of planetary carrier of \( K_3 \), N-m; \( T_L \) is the torque transmitted by brake \( C_3 \), N-m; and \( \eta_{c3} \) is the mechanical efficiency of the planetary gearset \( K_3 \).

The torque theoretical relationship between planetary carriers of \( K_2 \) and \( K_3 \) is

\[
T_{c2} + T_{c3} = 0.
\]

The torque transmitted by brake \( C_L \) is

\[
T_L = \frac{(1 + k_2)k_3}{(1 + k_3)\eta_{c2}\eta_{c3}} \cdot \frac{(T_s/i_3\eta_3) + (1 + k_1)T_m\eta_{c1}}{(1 + k_1)\eta_{c1} + k_2}.
\]

The power transition model built in Simulink is shown in Figure 5.

### 4. Prediction Model of Displacement Ratio Target Value

In order to make the ideal working state of the poststable stage the same as that of the target range, the key state parameters such as the hydraulic pressure, the brake torque, and the speed of the FDHC at the poststable stage should be the same as those in the target range. The speeds of hydraulic components in the current range are the same as those in the target range. However, because of the volumetric efficiency of the hydraulic system, the displacement ratios are different. According to the key state parameters of the current range, two key processes are needed to predict the displacement ratio target value. The first one is to predict the hydraulic pressure in the target range based on the current pressure. The second one is to predict the displacement ratio target value based on the flow continuity equation of the closed hydraulic circuit.

#### 4.1. Prediction Model of Target Pressure

In the full power shift, the output power is expected to be transmitted normally in full power. In other words, the aim is to keep the output power unchanged. The hydraulic pressure in the target range is predicted according to the current pressure.

According to equations (25)–(28), the torques of the FDHC in the H range and in HM range are
The output torque before and after power shift is constant, that is, \( T_o/i_2\eta_3 \) is unchanged.

Hence, the torque ratio of the FDHC in the HM range and in H range is

\[
\frac{T_{mHM}}{T_{mH}} = \frac{(1 + k_1)\eta_{cl}}{k_2}.
\]

For the HMT shown in Figure 2, the relation between characteristic parameters of planetary gearsets is given by

\[
1 + k_1 = k_2.
\]

Hence, the torque ratio of the FDHC can theoretically be written as

\[
\frac{T_{mHM}}{T_{mH}} = -1.
\]

However, it is necessary to consider the influence of mechanical efficiency in practical calculation. The relation in H range can be written as

\[
\begin{align*}
T_o/i_2\eta_r &= -(p_{1h} - p_{2h})V_g (1 + k_1)\eta_{mm}\eta_{cl}, \\
p_{2h} &= C,
\end{align*}
\]

where \( p_{1h} \) and \( p_{2h} \) are the high and low pressures in the H range, respectively, MPa, and \( C \) is the low pressure.

The relation in the HM range can be written as

\[
\begin{align*}
T_o/i_3\eta_r &= (p_{1hm} - p_{2hm})V_g k_2, \\
p_{1hm} &= C,
\end{align*}
\]

where \( p_{2hm} \) and \( p_{1hm} \) are the high and low pressures in the HM range, respectively, MPa.

According to equations (34) and (35), based on the pressures in the H range, the high pressure in the HM range can be predicted by

\[
p_{2hm} = \frac{(p_{1h} - p_{2h})(1 + k_1)\eta_{mm}\eta_{cl}}{k_2} + p_{1hm}
\]

Based on the pressures in the HM range, the high pressure in the H range can be predicted by

\[
p_{1h} = \frac{(p_{2hm} - p_{1hm})k_2}{(1 + k_1)\eta_{mm}\eta_{cl}} + p_{2h} = \frac{p_{2hm} - p_{1hm}}{\eta_{mm}\eta_{cl}} + p_{2h}.
\]

As can be seen from equations (36) and (37), the pressures in the target range can be predicted according to the pressures in the current range and the mechanical efficiency of the FDHC. The prediction model of target pressure is mainly based on equations (36) and (37) to predict the target pressure. The high pressure has little difference between the H range and HM range. Due to the influence of mechanical efficiency, the high pressure in the range is slightly higher than that in the HM range.

The two efficiency parameters \( \eta_{mm} \) and \( \eta_{cl} \) are the most important parameters, which should be calibrated. And only the two parameters need to be calibrated. These two parameters can be calibrated by analyzing the relationship between the output torque and the pressures in the closed hydraulic circuit.

4.2. Prediction Model of the Displacement Ratio Target Value.

When predicting the displacement ratio target value, the relationship between the volumetric efficiency of the hydraulic system in the two steady state conditions of the prior stable stage and the poststable stage is mainly studied. The high pressure has little difference between the H range and HM range. Therefore, the influence of pressure on viscosity and density can be neglected. At the same time, considering the oil temperature is basically unchanged before and after the power shift, it can be assumed that the oil viscosity remains unchanged during the power shift.

According to the flow continuity equation, the outlet flow rate of the VDHC is equal to the inlet flow rate of the FDHC at the ideal shift timing from the H range to HM range:

\[
q_{po} = q_{mi}.
\]

That is,

\[
\frac{\varepsilon_{h-hm}^* V_g n_p}{60} - \frac{C_p \varepsilon_{h-hm}^* V_g (p_{1h} - p_{2h})}{2\pi \mu}
\]

\[
= \frac{V_g n_m}{60} + \frac{C_m V_g (p_{1h} - p_{2h})}{2\pi \mu},
\]

where \( \varepsilon_{h-hm}^* \) is the displacement ratio at the ideal shift timing from the H range to HM range and \( \mu \) is the dynamic viscosity of oil during the power shift, N·s/m².

The following equation can be obtained:

\[
C_p \varepsilon_{h-hm}^* + C_m = \frac{2\pi \mu (\varepsilon_{h-hm}^* - n_p)}{60(p_{1h} - p_{2h})} = \frac{2\pi \mu (\varepsilon_{h-hm}^* - n_p)}{60(p_{1h} - p_{2h})}.
\]

\[
C_p \varepsilon_{h-hm}^* + C_m = \frac{2\pi \mu (\varepsilon_{h-hm}^* - n_p)}{60(p_{1h} - p_{2h})}.
\]
where \( \varepsilon_0 \) is the theoretical displacement ratio in the power shift when the leak is ignored. According to [29, 30], \( \varepsilon_0 \) equals to the speed ratio of the two hydraulic components when the double brakes overlap and \( \varepsilon_0 = n_m/n_p \).

Considering that \( C_{sp} \) is much smaller than 1 and \( \varepsilon_{h-hm}^* \) is close to \( \varepsilon_0 \), it is assumed that \( C_{sp} \varepsilon_{h-hm}^* = C_{sp} \varepsilon_0 \). The following equation can be obtained:

\[
C_{sp} \varepsilon_{h-hm}^* + C_{sm} = C_{sp} \varepsilon_0 + C_{sm} = \frac{2\pi \mu (\varepsilon_{h-hm}^* - \varepsilon_0) n_p}{60 (P_{th} - P_{ph})}.
\] (41)

The inlet flow rate of the VDHC is equal to the outlet flow rate of the FDHC at the ideal shift timing from the HM range to H range:

\[
q_{pi} = q_{mo}.
\] (42)

That is,

\[
\frac{C_{sm} V_g (P_{th} - P_{thm})}{2\pi \mu} + \frac{C_{sp} \varepsilon_{h-hm}^* V_g P_{ph} - P_{thm}}{60} = \frac{V_g n_m}{60}.
\] (43)

where \( \varepsilon_{h-hm}^* \) is the displacement ratio at the ideal shift timing from the HM range to H range. The following equation can be obtained:

\[
C_{sp} \varepsilon_{h-hm}^* + C_{sm} = \frac{2\pi \mu (n_{m} - \varepsilon_{h-hm}^* n_p)}{60 (P_{ph} - P_{thm})} = \frac{2\pi \mu (\varepsilon_0 - \varepsilon_{h-hm}^*) n_p}{60 (P_{ph} - P_{thm})}.
\] (44)

Considering that \( C_{sp} \) is much smaller than 1 and \( \varepsilon_{h-hm}^* \) is close to \( \varepsilon_0 \), it is assumed that \( C_{sp} \varepsilon_{h-hm}^* = C_{sp} \varepsilon_0 \). The following equation can be obtained:

\[
C_{sp} \varepsilon_{h-hm}^* + C_{sm} = C_{sp} \varepsilon_0 + C_{sm} = \frac{2\pi \mu (\varepsilon_0 - \varepsilon_{h-hm}^*) n_p}{60 (P_{ph} - P_{thm})}.
\] (45)

The leakage coefficient of hydraulic components and the oil viscosity are unchanged during the power shift. Hence, the following can be obtained from (37) and (41):

\[
\frac{\varepsilon_{h-hm}^* - \varepsilon_0}{\varepsilon_0 - \varepsilon_{h-hm}^*} = \frac{P_{th} - P_{ph}}{P_{ph} - P_{thm}}.
\] (46)

5. Simulation Analysis

The input speed of HMT is \( n_t = 1000 \) r/min, and the load torque \( T_{o} = 300 \) N·m remains unchanged. The low pressure in the closed hydraulic circuit is \( C = 2.6 \) MPa. Based on the power transition model established in this paper, the power transition processes are simulated and analyzed. During the simulation, the two brakes are reversed. From 0 s to 0.3 s, the displacement ratio is the same as the value at ideal shift timing. So, from 0 s to 0.3 s, HMT is at the prior stable stage. At 0.3 s, the displacement ratio changes to the target value, and then it remains unchanged.

According to [29, 30], \( \varepsilon_{h-hm}^* \) is the displacement ratio at the ideal shift timing from the H range to HM range and is also the optimal target value of the displacement ratio in the power shift from the HM range to H range. \( \varepsilon^*_{h-hm} \) is the displacement ratio at the ideal shift timing from the HM range to H range and is also the optimal target value of the displacement ratio in the power shift from the H range to HM range.

During the power shift from the H range to HM range, the displacement ratio is changed from \( \varepsilon^*_{h-hm} \) step to \( \varepsilon^*_{h-hm} \) at 0.3 s and then remains unchanged, as shown in Figure 6. In the simulation condition, according to (42), \( \varepsilon^*_{h-hm} \) is equal to 0.92 and \( \varepsilon^*_{h-hm} \) is equal to 0.84. The results are shown in Figure 7.

As shown in Figure 7, during the power shift from the H range to HM range, the displacement ratio is changed from \( \varepsilon^*_{h-hm} \) step to \( \varepsilon^*_{h-hm} \) at 0.3 s, and the original high-pressure side completes the pressure release process at 0.48 s. And the original low-pressure side completes the pressure building-up process at 2.2 s, and the high and low pressures are interchanged. After 2.2 s, the torque of \( C_{th} \) is 0 and the load torque transmits from \( C_{H} \) to \( C_{l} \). The power transition is completed when the double brakes overlap, and HMT shifts to the poststable stage.

During the power shift from the HM range to H range, the displacement ratio is changed from \( \varepsilon^*_{h-hm} \) step to \( \varepsilon^*_{h-hm} \) at 0.3 s and then remains unchanged, as shown in Figure 8. In the simulation condition, according to (42), \( \varepsilon^*_{h-hm} \) is equal to 0.92 and \( \varepsilon^*_{h-hm} \) is equal to 0.84. The results are shown in Figure 9.

As shown in Figure 9, during the power shift from the HM range to H range, the displacement ratio is changed from \( \varepsilon^*_{h-hm} \) step to \( \varepsilon^*_{h-hm} \) at 0.3 s, and the high-pressure side completes the pressure release process at 0.42 s. And the original low-pressure side completes the pressure building-up process at 2.5 s, and the high and low pressures are interchanged. After 2.5 s, the torque of \( C_{l} \) is 0, and the load torque transmits from \( C_{l} \) to \( C_{H} \). The power transition is completed when the double brakes overlap, and HMT shifts to the poststable stage.
completed when the double brakes overlap, and HMT shifts to the poststable stage.

In summary, the speed ratio of hydraulic components in the target range is the same as that in the current range, but the displacement ratios are different due to the volumetric efficiency of the hydraulic system. Based on the pressures of the hydraulic system in the current range, the pressures in the target range can be predicted, and then the displacement ratio target value can be predicted. If the displacement ratio target value at the poststable stage equals the ideal shift displacement ratio in the target range, the original high pressure at the poststable stage can be maintained at $C$, the original low pressure at the poststable stage can be equal to the high pressure in the target range, and the torque of the brake to be disengaged is 0. The power transition is completed when the double brakes overlap, and HMT shifts to the poststable stage. The working state of the poststable stage is the same as that of the target range.

6. Full Power Shift Test

The full power shift test of the HMT whose configuration is shown in Figure 1 is carried out on the test bench, as shown in Figures 10 and 11. It is a typical closed-electric transmission test bench.

The input speed remains at 1000 r/min, and the load torque is 300 N-m. The power shifts from the H range to HM range and then shifts from the HM range to H range.

The test results of power shift from the H range to HM range are shown in Figure 12. The displacement control...
Figure 8: Displacement ratio.

Figure 9: Simulation results of the shift from HM range to H range, when the displacement ratio target value is $\varepsilon_{\text{target}}^{\text{hms}}$: (a) pressures in closed hydraulic circuit and (b) brake torques.

Figure 10: The schematic diagram of test bench.
current is reduced from 549 mA to 512 mA (shown in Figure 12(a)), corresponding to the change of the displacement ratio from 0.92 to 0.83 (shown in Figure 12(b)). The original high pressure in the H range is 8.3 MPa. After the power shift, the test value of the high pressure in the HM range is 6.8 MPa, and the simulation value is 6.7 MPa. And
the simulation result of the pressure building-up process is basically consistent with the test result. The speed of the FDHC is basically maintained at 837 r/min, the output speed is basically maintained at 236 r/min, and the output torque is basically maintained at 300 N·m.

According to (42), the predicted value of the displacement ratio is 0.84. The actual and theoretical values of the displacement ratio are shown in Figure 12(b). And the predicted value is larger than the actual value by 0.01. The error of the predicted value is acceptable.

The original high pressure in the HM range is 6.6 MPa. After the power shift, the test value of the high pressure in the H range is 8.0 MPa and the simulation value is 8.1 MPa. And the simulation result of the pressure building-up process is basically consistent with the test result. The speed of the FDHC is basically maintained at 837 r/min, the output speed is basically maintained at 236 r/min, and the output torque is basically maintained at 300 N·m.

According to (42), the predicted value of the displacement ratio is 0.93. The actual and theoretical values of the displacement ratio are shown in Figure 13(b). And it is larger than the actual value by 0.01. The error of the predicted value is acceptable.

According to Figures 12(a) and 12(c), the disengaged time of C_H is 1.43 s and the completion time of pressure
Conclusions could be drawn: the target displacement ratio are studied. The following concepts of hydraulics established a power transition model in full power shift process.

The errors of the predicted values of the displacement ratio shown in Figures 12(b) and 13(b) are both 0.01. The prediction process is divided into two steps. The first step is to predict the target pressure based on the current pressure and the mechanical efficiency of the fixed-displacement hydraulic component. The second step is to predict the displacement ratio target value based on the current displacement ratio and the current pressures and the target pressures prediction value. The error of the target pressure predicted in the first step will lead to the error of the displacement ratio predicted value. When deriving (42) in the second step, we assume that $C_{sp}\Delta p_{h\rightarrow hm} = C_{sp}\Delta p_0$ and $C_{sp}\Delta p_{h\rightarrow hm} = C_{sp}\Delta p_0$, which will also cause errors. When the displacement control current is equivalent to the displacement ratio, the hysteresis effect of the solenoid valve is ignored, and the current and the displacement ratio are simplified to a linear relationship. Taking these factors into consideration, the errors of the predicted values are acceptable, and the prediction method of displacement ratio target value is reasonable.

7. Conclusions

This paper establishes a power transition model in full power shift of HMT, which is based on the double brakes overlapping. The dynamics of the power transition and the ideal target displacement ratio are studied. The following conclusions could be drawn:

1. An arithmetic two-range HMT is taken as the research object. Based on the new full power shift method, the power transition model is established, including the hydraulic transmission unit model and the brake torque model during the double brakes overlapping. Under the premise of neglecting the dynamic reversing process of the hydraulic reversing valve, the power transition model can describe the principle of power transition and the interchange between two pressure sides of the hydraulic transmission unit.

2. Aiming at the constant output power of HMT in the shift process, a prediction model of the displacement ratio target value is established and the prediction method is proposed. The bench test proves that the prediction method of the displacement ratio target value is reasonable, which can make the high-pressure side of the target range complete the pressure building-up process when the double brakes overlap. And the power transition is completed, and the output power can be transmitted normally in full power.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

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