Research on Continuous Power Shift Process of Hydro-Mechanical Continuously Variable Transmission

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Abstract. Which through the state switch of shifting mechanism and combined with speed regulation of hydraulic transmission system, the shift of Hydro-Mechanical continuously variable transmission (HMCVT) is realized. The traditional method of shift is short intervals or cross, which will easily lead to power interruption and affects the shift quality. In this study, the continuous power-shift is realized in the method of short overlap between segments. The continuous power shift process of HMCVT is studied in theory. The result of theoretical research is verified on the test bench, which provided suggestions for better shift quality and making control strategy.

Keywords: Hydro-Mechanical transmission; Continuous power shift process.

1. Introduction

Hydro-Mechanical Continuously Variable Transmission (HMCVT) consists of a hydraulic transmission, a mechanical transmission, and diffiuence and confluence mechanism, which hydraulic transmission and mechanical transmission are connected in parallel. Power distribution of mechanical and hydraulic circuits is regulated by diffuence and confluence. Continuously variable speed is realized by combining hydraulic transmission with mechanical transmission. It combines the excellent characteristics of hydraulic transmission and mechanical transmission, discards their shortcomings, and has become one of the main development directions of high-power stepless transmission[1-5]. Because of its excellent performance, HMCVT has a broad application prospect in high-power vehicles. HMCVT is equipped with M2 and M3 combat vehicles in the United States and Type 10 main battle tanks of Japan[10,20]. HMCVT manufactured by ZF and Fendt in Germany has been equipped on tractors of famous companies such as Deuta-Fahr, JCB and Steyr[21,22].Caterpillar of USA, Komatsu of Japan, T-130 tractor of former Soviet Union etc. has applied HMCVT in their products[21-23]. The four-range HMCVT developed by German RENK company has been applied to Audi 100 automobiles with good results[23].

Scholars from William Conson University and Purdue University have conducted deep research on modeling, simulation and characterization of HMCVT[23]. Alario Macor and Antonio Rossetti from Italy have done a lot of research on the application of HMCVT in high-power tractors[25-27]. Liu Xiuji and Yuan Shihua from China have put forward relatively complete design methods for military vehicle HMCVT[1,28]. Xu Liyou and others from China developed HMCVT for Dong Fang Hong 1302R tractor[10,21]. Ni Xiangdong and others from China researched characteristics and speed ratio tracking control of HMCVT for tractor[13,14]. Tang Xinxing and others from China designed the HMCVT for construction machinery, which enriched the scheme of HMCVT[29-32].
Limited to the structure and transmission efficiency, the HMCVT single working section has a limited speed range, which is difficult to meet the vehicle requirements. Through the state switch of shift range mechanism, and combined with speed regulation of hydraulic transmission, the ranges are connected to each other and the speed regulation is expanded to form a multi-stage continuously variable transmission. Scholars from China have achieved many research results in design matching, characterization, and speed ratio tracking control of HMCVT. Yuan Shihua pointed out that there were some problems such as speed fluctuation, pressure shock and power interruption in shift range\[^{36,37}\]. Wei Chao analyzed the influencing factors of the shift quality of HMCVT, and put forward the tracking control method of speed ratio in the segment to enable the engine to work in the desired area\[^{3,33-35}\]. Yang Shujun proposed a method to improve the shift quality of HMCVT, but there are still problems such as pressure shock\[^{2,38}\]. Hu Jibin studied the feasibility of braking overlapping in HMCVT, and opened up a new way to improve the shift quality. But there is still a power discontinuity in shift\[^{34}\]. In this paper, based on the two-stage HMCVT, the continuous power-shift is realized in the method of short overlap between segments. Through the analysis of theory and simulation, the continuous power shift process of HMCVT is studied, and the results are verified on the test bench.

2. The Structure Composition and Shift Principle of HMCVT

Figure 1 shows the transmission principle of a two-stage HMCVT. The hydraulic transmission consists of a variable displacement hydraulic component and a fixed displacement hydraulic component. The diffiuence and confluence mechanism is respectively a fixed shaft gear transmission and a k2 planetary row.

The shift of HMCVT is realized by the stage switching of shifting mechanism and adjusting the transmission ratio of the hydraulic transmission mechanism to the appropriate value. In order to achieve good shift quality, it is necessary to select the appropriate shift point and control the switching sequence of shifting mechanism properly\[^{1-18}\]. For the HMCVT shown in Figure 1, the shifting mechanism is the brakes B1 and B2, and the switching sequence of them is shown in the table in the upper right corner.

When B1 is combined and B2 is separated, only the hydraulic circuit works, which is the hydraulic section(H). When B2 is combined and B1 is separated, the hydraulic circuit and the mechanical circuit work equally, which is the hydraulic mechanical section(HM).

Without considering the transmission link of \(i_2\), the K1 planetary carriage is used as the output and the shift process is analyzed as follows.

### Figure 1. Schematic diagram of HMCVT.

In H, the output speed of HMCVT:

\[
n_{hi} = n_{p1} = \frac{n_{i1}}{1 + k_1} = \frac{n_{p} \cdot \eta_r}{(1 + k_1)i_p}
\]  

In HM, the output speed of HMCVT:

\[
n_{i2} = \frac{n_{p}}{k_1} \left( 1 + k_2 \right) \frac{\eta_r}{(1 + k_1)i_p}
\]
According to the speed connection condition when shift, equalizing Formulas (1) and (2), the displacement ratio of hydraulic transmission required at the theoretical shift point from H to HM can be solved:

$$\varepsilon_{k-\text{th}} = \frac{i_s(1+k_s)(1+k_p)}{(1+k_s)(1+k_s+k_p)\eta_x}$$

In formula: \(n_e\) -- input speed (r/min); \(\varepsilon\) -- the displacement ratio of hydraulic transmission system: \(\varepsilon = V_p / V_{m} = eV_{\text{max}} / V_{m}\); \(V_{\text{max}}\) -- the max displacement of variable displacement hydraulic component \((cm^3/r)\); \(V_m\) -- the displacement of fixed displacement hydraulic component \((cm^3/r)\); \(e\) -- the rate of displacement of variable displacement hydraulic component (-1~1); \(\eta_x\) -- the volumetric efficiency of hydraulic transmission; \(n\) -- rotating speed (r/min); the \(pc\) in the subscript represents the planetary frame; the \(s\) in the subscript represents the sun wheel; the \(r\) in the subscript represents ring; the \(k\) represents the characteristic parameters of the planetary row corresponding to the numbers; the \(b\) in the subscript represents output; the \(M\) in the subscript represents fixed displacement hydraulic component; the \(p\) in the subscript represents variable displacement hydraulic component; \(i_s\) -- ratio of transmission link \(i\); shown in Figure I; same below.

3. The Working Mechanism for Continuous Power Shift of HMCVT

3.1. Analysis of influencing factors on volume efficiency of hydraulic transmission

From the above analysis of HMCVT shift, it can be known that the theoretical shift point is determined by the structural parameters of HMCVT and the volume efficiency of the hydraulic transmission, and by the torsional vibration characteristics of the mechanical transmission. However, it is very complex, and the influence of the torsional vibration characteristics on shift is much smaller than that of the hydraulic transmission. Limited by length and emphasis, the influence of torsional vibration characteristics on shift is not considered \([2-19]\). The volumetric efficiency of hydraulic transmission can be expressed by formula (4) when considering the rotational speed of variable displacement hydraulic components, oil viscosity, displacement ratio, leakage and pipeline pressure of hydraulic transmission\([1]\):

$$\eta_v = 1 - C_s \frac{\triangle p}{\mu \eta_x} = 1 - C_s \frac{\triangle p}{\mu \eta_x}$$

In formula: \(C_s\) -- leakage coefficient; \(\mu\) -- dynamic viscosity (Pa·s); \(\triangle p = p_{hi} - p_e\) -- pressure difference between inlet and outlet of fixed displacement hydraulic components (Pa).

From formula (4), the relationship between volumetric efficiency and various influencing factors can be obtained:

$$\eta_v = 1 - C_s \frac{\triangle p}{\mu \eta_x} \left(1 + \varepsilon \right)$$

From the formula (5), the main influencing factors of volumetric efficiency are the leakage coefficient, the pressure difference between the inlet and outlet of the fixed displacement hydraulic component, the dynamic viscosity of the hydraulic fluid, the rotational speed of the variable displacement hydraulic component, and the displacement ratio of the hydraulic transmission. When the power of the hydraulic transmission returns, the volumetric efficiency is reciprocal of Formula (5).

3.2. Analysis of power switching process of HMCVT in continuous power shift

When the vehicle is in actual operation, because of the complex and changeable working conditions, it is difficult for the control system to synchronously complete the switching of the separation and combination of the shifting mechanism. When there are short intervals in shift, the power will be interrupted, which brings the shift impact and affects the shift quality\([12-15,18,19]\). At the theoretical shift point, the short overlap between segments before the shift mechanisms to be separated is quickly
separated, continuous power-shift will be realized. However, the shift performance will be affected by the kinematic interference between system components. In severe cases, the system components will be damaged. The velocity and torque characteristics of overlapping are analyzed as follows.

3.2.1. Analysis of velocity characteristics of HMCVT in overlapping

For the HMCVT mentioned above, when the displacement ratio of the hydraulic transmission is near the theoretical shift point, the two brakes are combined. According to the transmission relationship, the speed characteristics of HMCVT can be obtained.

The displacement ratio of theoretical shift point is $\epsilon^*$ without considering the direction of shift. When overlapping and combining near the theoretical shift point, the displacement ratio of the actual shift point is $\epsilon$, then $\epsilon \neq \epsilon^*$ and the states of the planetary frames of K2 and K3 changes, and there are:

$$n_{pc1} = \frac{n_{ij} e n_{p}}{1+k_i (1+k_i) n_{ij}} \neq \frac{n_{ij} e n_{p}}{1+k_i (1+k_i) n_{ij}} = n_{pc1}^* $$

(6)

$$n_{r2} = \frac{n_{r2}}{k_2 \left(1+k_1 (1+k_i) n_{ij} \right) \epsilon + \epsilon} \neq n_{r2} = \frac{n_{r2}}{k_2 \left(1+k_1 (1+k_i) n_{ij} \right) \epsilon + \epsilon} = n_{r2}^* $$

(7)

In the theoretical shift point, there is $n_{pc1}^* = n_{r2}^*$. From formula (6) and (7), we can get $n_{pc1} \neq n_{r2}^*$. According to the structural relationship, the rotational speed of $pc1$ and $r2$ must be equal. If the displacement ratio remains unchanged, only the change of volumetric efficiency makes them equal, let $n_{pc1} = n_{r2}$, the following formula can be obtained:

$$\eta_p = \frac{i_p (1+k_2) (1+k_i) \epsilon}{(1+k_1 (1+k_i) n_{ij} \epsilon) + \epsilon} $$

(8)

Bringing Formula (8) into Formula (6) and (7), we can get:

$$n_{p} = n_{pc1} = n_{r2} = \frac{1+k_2}{(1+k_1 (1+k_i) n_{ij} \epsilon) + \epsilon}$$

(9)

$$i = \frac{(1+k_2) (1+k_i + k_2) n_{ij} \epsilon}{1+k_2} $$

(10)

From Formula (5) and Formula (9) and Formula (10), we can see that: when the displacement ratio remains unchanged and the shifting mechanism is overlapping, the output speed of the system stabilizes at the speed shown in formula (9) after a short time of torque redistribution and speed oscillation of hydraulic circuit and mechanical circuit, which is independent of the displacement ratio and only related to the structural parameters of the system. The transmission ratio of HMCVT in the overlapping process is a certain value, which is determined by the structural parameters of HMCVT. Therefore, the fundamental reason why HMCVT can continuously power shift is that the leakage of hydraulic components and the compressibility of oil in hydraulic pipelines can eliminate the kinematic interference in overlapping. The comprehensive performance is that the volume efficiency of hydraulic circuit changes to adapt to speed equal mentioned above. But it will cause torque redistribution between system components.

3.2.2. The analysis of torque characteristics of HMCVT in overlapping

The torque equations for the K1 and K2 rows can be listed by the HMCVT component relationship:

$$T_k - (1+k_i) T_{si} + T_{s2} = 0$$

(11)

$$T_M = T_{s1} + \frac{T_{s2}}{k_2}$$

(12)
According to the transmission relationship of HMCVT, only two torque equations mentioned above can be listed, and there are three unknown parameters. Because the leakage of hydraulic transmission system and the compressibility of oil are much better than the flexibility of mechanical components in eliminating kinematic interference, another equation can be listed. Formula (5) is deformed to obtain the characteristics of volume efficiency and system torque. Equal formulas (5) and (8) and the relationship between the torque of fixed displacement hydraulic components and the pressure difference between inlet and outlet, the torque of fixed displacement hydraulic components and other main components can be obtained as follows:

\[
T_m = \frac{\mu mV_M}{20\pi C_i} \left( \frac{\epsilon - i_s (1+k_j)(1+k_j)}{(1+k_j)(1+k_i+k_j)} \right) \quad (13)
\]

\[
T_{i1} = k_1 \frac{(1+k_j)}{(1+k_j)(1+k_i+k_j)} \left[ T_b - (1+k_i)T_m \right] \quad (14)
\]

\[
T_{i3} = \frac{k_1}{1+k_i+k_j} \left[ k_2T_m + T_b \right] \quad (15)
\]

In formula: \(T_m\) -- the torque of fixed displacement hydraulic components (N·m), \(T_{i1}\) -- the torque of K1 ring (N·m), \(T_{i3}\) -- the torque of K3 ring (N·m).

From formula (13)–(15), the relationship curves between torque and displacement ratio, system structure parameters, and rotational speed of variable displacement hydraulic components, can be obtained, as shown in Figure 2. Torque of K1 ring decreases with the decrease of displacement ratio, while that of K3 ring increases with the increase of displacement ratio. Therefore, the torque distribution between the components of the system can be adjusted by adjusting the displacement ratio, so that the torque switching between the brake B1 and the brake B2 can be adjusted, and then the power switching can be realized. When the power switching is completed, the displacement ratio is as shown in \(\epsilon_1\) and \(\epsilon_2\) in Figure 2.

![Figure 2. The relationship between the torque of components and the displacement ratio in overlap.](image)

From H to HM, when the braking torque on the brake B1 changes to zero, the displacement ratio \(\epsilon_1\) and the torque of fixed displacement hydraulic components need to be adjusted as shown formula (16) and (17). At this time, the torque of the fixed displacement hydraulic component is the same as that of HM.

\[
\epsilon_1 = \frac{k_3 \mu mV_M}{20\pi C_i T_b} \frac{\epsilon_0 - 1}{1 + k_3 \mu mV_M} \quad (16)
\]

\[
T_{m,\epsilon_1} = -\frac{1}{k_2} T_b \quad (17)
\]
From HM to H, when the braking torque on the brake B2 changes to zero, the displacement ratio \( \varepsilon_2 \) and the torque of fixed displacement hydraulic component need to be adjusted as shown Formula (18) and (19). At this time, the torques of the fixed displacement hydraulic component is the same as that of H.

\[
\varepsilon_2 = \frac{1 + (1 + k_i) \varepsilon_0 V_{\text{in}} \mu \rho}{20 \pi C T_s} \left( \frac{1 + k_i V_{\text{in}} \mu \rho}{20 \pi C T_s} - 1 \right)
\]

(18)

\[
T_{M3} = \frac{1}{1 + k_i} T_s
\]

(19)

When \( T_M = 0 \), the displacement ratio of variable displacement hydraulic components is the reverse point \( \varepsilon_0 \) of hydraulic circuit power, which can be obtained by formula (12):

\[
\varepsilon_0 = \frac{i_p (1 + k_i) (1 + k_i)}{(1 + k_i) (1 + k_i + k_i) n_i}
\]

(20)

From the Figure II and formula (12) ~ (20), we can see that: When \( \varepsilon = \varepsilon_0 \), the load torque \( T_M \) of fixed displacement hydraulic components is zero, which indicates that \( T_M \) begins to reverse at this point and the direction of power flow in hydraulic circuit changes. When the displacement ratio is less than \( \varepsilon_0 \), the shifting mechanism overlaps and the \( T_M \) is negative, the fixed displacement hydraulic component is driven by the variable displacement hydraulic component, and the hydraulic circuit power is forwardly transmitted. Conversely, the power of the hydraulic circuit is transmitted negatively. \( \varepsilon_0 \) is determined by the structural parameters of HMCVT, and is independent of system load, hydraulic circuit parameters and input speed. As the displacement ratio deviates from \( \varepsilon_0 \), \( T_M \) increases. The higher the input speed of variable displacement hydraulic components is, the greater \( T_M \) is, until the pressure of the hydraulic transmission system reaches the overflow pressure and loses its transmission function, the efficiency of HMCVT is greatly reduced, which should be avoided. The higher the rotational speed of the variable displacement hydraulic component, the smaller the interval for continuous power shift by overlapping is.

3.3. The stage division of power switching process of HMCVT in continuous power shift

According to the above research, the process of continuous power shift can be divided into four stages: speed regulation stage, overlapping combination stage, power switching stage and separation stage. The sequence of displacement regulation and brake operation can be described simply by figure 3. Usually, when shift, the displacement ratio is adjusted to make the speed difference of the mechanisms to be combined almost zero, and then the displacement ratio is basically the theoretical displacement ratio which varies with the load of the system and the working condition of the hydraulic transmission. However, when the HMCVT is designed, the theoretical displacement ratio is designed according to the common working conditions. Therefore, in most cases, it is necessary to calculate the required displacement ratio by measuring the output speed of the hydraulic transmission, while in continuous power shift, the shifting mechanism can be combined without complete synchronization. That is to say, in a certain range of speed difference, the shifting mechanism can be overlapped, and the range of speed difference should be determined by the shifting mechanism and the shift quality.

4. Bench Test

In order to verify the correctness of the theoretical analysis of continuous power-shift process of HMCVT, the test of continuous power-shift process was carried out on the gantry. The test system is shown in Figure 4. The test is divided into three cases, that is, the test is carried out at the theoretical shift point and before and after it. Considering the safety of the test, the output speed of the variable frequency motor is set at 500 r/min, and the inertia of the inertia loading system is set to 50 kg·m². During the test, the measuring position of the output speed of the system is at the output of the K1 planetary carriage in the simulation, so the output speed of the test is comparable with the theoretical analysis. The test results are shown in figures 5-7.
Figures 5-7 shows that at the theoretical shift point, the speed of the transmission is basically unchanged and stable at the theoretical shift point, and there is no impact in the shift process, and the power transmission is continuous; at the non-theoretical shift point, in the overlapping combination stage, the output speed of the system changes rapidly to the output speed of the system when power switching, and the power transmission is continuous, but the fluctuation of speed varies greatly, resulting in greater impact.

**Figure 3.** Schematic diagram of the continuous power-shift of HMCVT. Although the input speed and load torque are different from the actual working conditions, the changing trend and law of the test results for the three cases are consistent, which proves that the theoretical analysis is correct.

**Figure 4.** Test system for HMCVT.

5. Conclusion
(1) Through the analysis of dynamic characteristics of HMCVT in the process of continuous power shift, the working mechanism is studied: within the appropriate range of displacement ratio, the continuous power shift are realized by adjusting displacement ratio while the mechanism to be
separated and combined overlaps. These studies can provide a theoretical basis and reference for the development of shifting strategy, and lay the foundation for improving shift quality.

Figure 5. The test of the continuous power-shift at theoretical shift point.

Figure 6. The test of the continuous power-shift before theoretical shift point.

Figure 7. The test of the continuous power-shift after theoretical shift point.

(2) The process of continuous power shift is divided into four stages: speed synchronization adjustment, overlap combination, power switching and fast separation. In the power switching stage, power switching achieved by adjusting the displacement ratio, the transmission ratio of the HMCVT is constant, which is determined by the parameters of the shunt and confluence mechanism and the structural parameters of the mechanical transmission part, and has nothing to do with the parameters of the hydraulic transmission system.

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