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Optimization analysis on the transmission characteristics of multipurpose power transmission device

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ABSTRACT

A multipurpose power transmission device integrating the hydrostatic, hydro-mechanical and mechanical transmission is designed. This paper mainly discusses the transmission characteristic optimization problem of multipurpose power transmission device from the respective of speed regulation characteristic, shift strategy, and efficiency improvement. In accordance with the design requirements, the kinematic and kinetic analysis for the drive system is performed, the assembly schemes and relevant parameters of power transmission device should be analyzed, and the speed regulation characteristic curve can be obtained. According to the evaluation indexes of shift quality, the shift strategy of power transmission device involving clutches and brakes during the whole speed regulation process, and the best switch time of each component can be found. The hydrostatic system efficiency expression can be derivatized from the efficiency model of pump-control-motor system, and the efficiency of multipurpose power transmission device can be obtained by efficiency definition method; the fitting curves of hydrostatic system efficiency are determined by experimental data, and the efficiency of hydro-mechanical composite power transmission system can be got by conversion mechanism method. The results show that the shift quality of power transmission device can be improved greatly by controlling the switch sequence of clutches and brakes reasonably.

Key words: transmission characteristics; multipurpose power transmission device; speed regulation characteristic; shift strategy; efficiency improvement

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sum F$</td>
<td>total resistance of vehicle operating system (N)</td>
</tr>
<tr>
<td>$F_r$</td>
<td>rolling resistance (N)</td>
</tr>
<tr>
<td>$F_a$</td>
<td>air resistance (N)</td>
</tr>
<tr>
<td>$F_s$</td>
<td>slope resistance (N)</td>
</tr>
<tr>
<td>$F_i$</td>
<td>acceleration resistance (N)</td>
</tr>
<tr>
<td>$F_{tan}$</td>
<td>maximum tangential tractive force (N)</td>
</tr>
<tr>
<td>$G$</td>
<td>vehicle gravity (N)</td>
</tr>
<tr>
<td>$f$</td>
<td>coefficient of rolling resistance</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>road slope angle</td>
</tr>
<tr>
<td>$F_y$</td>
<td>adhesive force (N)</td>
</tr>
<tr>
<td>$\phi$</td>
<td>adhesion coefficient</td>
</tr>
<tr>
<td>$i_g$</td>
<td>transmission ratio of power transmission device</td>
</tr>
<tr>
<td>$n_r$</td>
<td>engine speed (r/min)</td>
</tr>
<tr>
<td>$n_o$</td>
<td>output speed of power transmission device (r/min)</td>
</tr>
<tr>
<td>$e$</td>
<td>displacement ratio</td>
</tr>
<tr>
<td>$D_p$</td>
<td>pump displacement (cm$^3$/r)</td>
</tr>
<tr>
<td>$D_m$</td>
<td>motor displacement (cm$^3$/r)</td>
</tr>
<tr>
<td>$v$</td>
<td>vehicle speed (km/h)</td>
</tr>
<tr>
<td>$r_w$</td>
<td>driving wheel power radius (m)</td>
</tr>
<tr>
<td>$i_m$</td>
<td>transmission ratio of main reducer</td>
</tr>
<tr>
<td>$i_{wa}$</td>
<td>transmission ratio of wheel-side reducer</td>
</tr>
<tr>
<td>$n_p$</td>
<td>pump speed (r/min)</td>
</tr>
<tr>
<td>$T_e$</td>
<td>engine torque (Nm)</td>
</tr>
<tr>
<td>$T_p$</td>
<td>pump torque (Nm)</td>
</tr>
<tr>
<td>$\Delta p_p$</td>
<td>pump system pressure (bar)</td>
</tr>
<tr>
<td>$\eta_{mp}$</td>
<td>pump mechanical efficiency</td>
</tr>
<tr>
<td>$k_i$</td>
<td>transmission ratios of general gears</td>
</tr>
<tr>
<td>$i_{fr1}^{-i_{fr3}}$</td>
<td>transmission ratios of forward ranges</td>
</tr>
<tr>
<td>$i_{rb}$</td>
<td>transmission ratio of reverse range</td>
</tr>
<tr>
<td>$k$</td>
<td>characteristic parameter of planet gear</td>
</tr>
<tr>
<td>$T_m$</td>
<td>motor torque (Nm)</td>
</tr>
<tr>
<td>$T_o$</td>
<td>output torque of power transmission device (Nm)</td>
</tr>
<tr>
<td>$Z$</td>
<td>severity of braking</td>
</tr>
<tr>
<td>$r$</td>
<td>time (s)</td>
</tr>
<tr>
<td>$p_o$</td>
<td>oil pressure of main circuit (bar)</td>
</tr>
<tr>
<td>$Q$</td>
<td>flow rate of speed control valve (L/min)</td>
</tr>
</tbody>
</table>
$j_i$: shift jerk of intermediate shaft  
$j_o$: shift jerk of output shaft  
$\mu$: hydraulic fluid kinetic viscosity ($\text{Pa} \cdot \text{s}$)  
$C_l$: laminar flow leakage coefficient  
$\Delta p_{P_{\text{max}}}$: maximum pressure of pump system ($\text{bar}$)  
$C_r$: laminar flow resistance coefficient  
$\eta_p$: pump efficiency  
$C_m$: mechanical resistance coefficient  
$\eta_m$: motor efficiency  
$\Delta p_{P_{\text{max}}}$: maximum pressure of pump system ($\text{bar}$)  
$\Delta p_{M_{\text{max}}}$: maximum pressure of motor system ($\text{bar}$)  
$v$: laminar flow resistance coefficient  
$f_C$: mechanical resistance coefficient  
$P_\eta$: pump efficiency  
$M_\eta$: motor efficiency  
$P_{\text{max}}$: maximum pressure of pump system ($\text{bar}$)  
$M_{\text{max}}$: maximum pressure of motor system ($\text{bar}$)  
$M_n$: motor speed ($\text{rpm}$)  
$H_\eta$: hydrostatic system efficiency  
$\eta_1$–$\eta_3$: efficiency of forward ranges  
$\eta_4$–$\eta_5$: efficiency of general gears  
$\eta_k$: efficiency of planet gear  
$\delta$: power loss coefficient of planet gear

1. Introduction

The increase of efficiency and reduction of fuel consumption for mobile machinery is one of the key problems in today’s society. Many promising technologies such as multipurpose power transmission devices are the hotspots of researching and investing in the world. Driving system of mobile machinery usually adopts 4 basic transmission forms: mechanical transmission, hydraulic transmission, hydrostatic transmission and electrical transmission. Composite transmission emerging in recent years can play single power flow transmission advantage, and discard its disadvantage. Take hydro-mechanical composite transmission for example, the transmission system composed of hydrostatic transmission device and mechanical transmission device in series can realize speed variation, direction change and overload protection by hydrostatic transmission path, and extend the coverage area of output speed and torque by mechanical transmission path. This transmission mode expands the available efficient zone of system, however, it can not increase peak efficiency of system. When the transmission system is composed of hydrostatic transmission device and mechanical transmission device in parallel, a variable speed drive system with stepless speed regulation performance and a wide range of efficient distribution can be got.

Speed change transmission device with 3 kinds of transmission modes is applicable to the mobile machinery with the great speed variation between operation condition and transportation condition. Hydrostatic transmission is widely used in starting condition, mechanical transmission is widely used in transporting operation, hydro-mechanical composite transmission is widely used in operation condition, and the power output shaft can transfer power to drive other mechanisms.

2. Design scheme of multipurpose power transmission device

2.1 Structure scheme

A hydro-mechanical composite transmission device was designed (Patent number: ZL201410337988.0). The structure diagram and main components of multipurpose power transmission device are shown in Fig.1, where, ②mechanical transmission assembly mainly includes input clutch for the mechanical transmission system; ③single planetary gear confluence mechanism assembly mainly includes gear ring, sun wheel, planet carrier, brake for gear ring, brake for sun wheel, input shaft for gear ring, output shaft for planet carrier; ④hydrostatic transmission assembly mainly includes input clutch for hydrostatic transmission system, gear pairs for hydrostatic transmission system, variable displacement pump, fixed displacement motor, output clutch and output shaft for hydrostatic transmission system; ⑤shift mechanism assembly mainly includes clutches for high range and low range, and related speed variation gear pairs. The characteristics of multipurpose power transmission device can be described as follows: hydrostatic transmission is used to realize vehicle starting, hydro-mechanical composite transmission is used to realize stepless speed regulation, mechanical transmission is used to realize high efficiency transmission. In general, hydrostatic transmission mode and hydro-mechanical transmission mode are often used in the operation condition, some power is outputted from the power output shaft to drive other mechanisms; mechanical transmission mode is usually used in the transporting
condition. This kind of speed regulation transmission device can switch transmission mode as required, in order to improve the service performance of multipurpose power transmission device\(^{[13]}\).

![Fig.1 Design scheme of multipurpose power transmission device](image)

Main components of multipurpose power transmission device: ① input shaft; ② mechanical transmission assembly; ③ single planetary gear confluence mechanism assembly; ④ hydrostatic transmission assembly; ⑤ shift mechanism assembly; ⑥ output shaft; ⑦ power output shaft.

Components status of multipurpose power transmission device are shown in Tab.1.

<table>
<thead>
<tr>
<th>Components (clutches and brakes) status of multipurpose power transmission device</th>
</tr>
</thead>
<tbody>
<tr>
<td>Range</td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td>F₁</td>
</tr>
<tr>
<td>F₂</td>
</tr>
<tr>
<td>F₃</td>
</tr>
<tr>
<td>R</td>
</tr>
</tbody>
</table>

Note: “●” represents the engaging status of clutches and brakes.

2.2 Parameters design
2.2.1 Kinematic and kinetic analysis for vehicle system

The total resistance of vehicle operating system can be expressed as follows\(^{[14-16]}\):

\[ \sum F_i = F_\alpha + F_\omega + F_\beta \]

(1)

\( F_\alpha \) and \( F_\omega \) can be neglected according to the test data. For the design, We take into account that \( F_{\max} \) should be larger than \( \sum F_i \), that is:

\[ \sum F_i \leq F_{\max} \]

(2)

This paper takes \( G = 58000 N \), \( f \in [0.01, 0.30] \), \( \alpha \in [0, 30^\circ] \). \( \sum F_i \) increases as \( f \) and \( \alpha \) increase in the field of definition. We can deduce that \( \sum F_{\max} = 0.52G \) through calculation.

\( F_{\max} \) is rarely more than \( F_\alpha \), which can be expressed as follows:

\[ F_{\max} \ll F_\alpha = \phi G \]

(3)

Test data of \( f \) and \( \phi \) are shown in Tab.2.

<table>
<thead>
<tr>
<th>Ground</th>
<th>( f )</th>
<th>( \phi )</th>
<th>( F_{\max} )</th>
<th>( F_\alpha )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clay soil</td>
<td>0.02~0.05</td>
<td>0.67~0.72</td>
<td>0.36~0.39</td>
<td>0.67~0.72</td>
</tr>
<tr>
<td>Sandy loam</td>
<td>0.03~0.06</td>
<td>0.48~0.52</td>
<td>0.37~0.40</td>
<td>0.48~0.52</td>
</tr>
<tr>
<td>Grass</td>
<td>0.07~0.08</td>
<td>0.38~0.43</td>
<td>0.41~0.42</td>
<td>0.38~0.43</td>
</tr>
<tr>
<td>Farmland</td>
<td>0.10~0.12</td>
<td>0.68~0.74</td>
<td>0.44~0.46</td>
<td>0.68~0.74</td>
</tr>
</tbody>
</table>

Transmission ratio is defined as the ratio of input speed to output speed. The transmission ratio of multipurpose power transmission device in this paper can be expressed as follows:

\[ i_s = \frac{n_e}{n_i} \]

(4)

Displacement ratio is defined as the ratio of pump displacement to motor displacement. The volume speed-modulating loop studied in this paper possesses the following characteristic:

\[ e = \frac{D_p}{D_m} = \frac{D_p}{D_{T_{\max}}} \]

(5)

The vehicle speed expression is shown as follows:

\[ v = 0.377 \frac{n_r \gamma}{i_b l_b} \]

(6)

This paper takes \( r_\gamma = 0.400 \), \( i_b = 4.0 \), \( l_b = 5.6 \).

2.2.2 Parameters analysis of hydrostatic system

By comparison, series products of SAUER\_DANFOSS055 can be chosen as hydrostatic system. At the input end of power transmission device, the expression should meet the following conditions\(^{[17-19]}\):

\[ \begin{align*}
\frac{n_\alpha}{l_i} & \leq n_{\alpha \max} \\
T_{\max} & \leq T_{\max}
\end{align*} \]

(7)

The output torque of variable pump can be expressed as follows:

\[ T_p = \frac{D_p n_p D_{\max}}{20} \]

(8)

According to Equation \( (7) \) and \( (8) \), we can
infer that \( i_1 \in [0.59, 1.16] \) . In order to improve utilization of hydrostatic system, this paper takes \( i_1 = 0.67 \).

2.2.3 Transmission ratio of each range

Range F_1 adopts hydrostatic transmission, the transmission ratio of range F_1 can be expressed as follows:

\[
i_{t1} = \frac{(k+1)i_1i_2}{e} \tag{9}
\]

Range F_2 adopts hydro-mechanical composite transmission, the transmission ratio of range F_2 can be expressed as follows:

\[
i_{t2} = \frac{(k+1)i_1i_2}{e} \frac{1}{i_3} \tag{10}
\]

Range F_3 adopts mechanical transmission, the transmission ratio of range F_3 can be expressed as follows:

\[
i_{t3} = \frac{(k+1)i_1i_2}{k} \tag{11}
\]

Range R adopts hydrostatic transmission, the transmission ratio of range R can be expressed as follows:

\[
i_{tR} = \frac{(k+1)i_1i_2}{e} \tag{12}
\]

According to the choice of characteristic parameter of planet gear, and the analysis of speed regulation characteristic, the relevant parameters can be got: \( k = 2.5 \), \( i_1 = 0.67 \), \( i_2 = 1.5 \), \( i_3 = 0.5 \), \( i_4 = 2.0 \), \( i_5 = 3.0 \).

Relationship curves between transmission ratio and displacement ratio are shown in Fig.2.

![Fig.2 Relationship curves between transmission ratio and displacement ratio](image)

Starting, operating and braking check should be carried out at the output end of power transmission device.

In hydrostatic range, the power transferred from transmission device can drive vehicle starting, namely:

\[
T_{M_{max}}(k+1)i_1i_2 \geq T_{max} = \frac{F_{max}f_8}{b_l} \tag{13}
\]

In hydro-mechanical range, the power transferred from transmission device can overcome ground adhesive force, namely:

\[
T_{M_{max}}(k+1)i_1i_2i_3 \geq T_{max} = \frac{F_{max}f_8}{b_l} \tag{14}
\]

Hydrostatic range can afford desired torque when \( Z \geq 0.1 + 0.85(\rho - 0.2) \), namely:

\[
T_{M_{max}}(k+1)i_1i_2 \geq \frac{ZG_{R}}{b_4} \tag{15}
\]

According to Equation (13) \( \sim (15) \), we can calculate that \( T_{M_{max}} \geq 300 \text{Nm} \). For the reason that the displacement of motor is equal to maximum displacement of pump, the hydrostatic components selected in this paper meet the design requirement. A promising power transmission device should have prefect transmission characteristics, including speed regulation characteristic, shift characteristic and efficiency characteristic.

3. Transmission characteristics of multipurpose power transmission device

3.1 Speed regulation characteristic

Main parameters of multipurpose power transmission device can be set as follows:

\( n_s = 1600 \text{r/min} \quad T_e = 400 \text{Nm} \quad \rho_s = 40 \text{bar} \quad Q_s = 4 \text{L/min} \). Displacement ratios of hydrostatic system and ranges at different time periods are shown in Tab.3[20-22].

<table>
<thead>
<tr>
<th>t</th>
<th>0-5</th>
<th>5-10</th>
<th>10-15</th>
<th>15-20</th>
<th>20-25</th>
<th>25-30</th>
</tr>
</thead>
<tbody>
<tr>
<td>e</td>
<td>0</td>
<td>0.5</td>
<td>0.5</td>
<td>1.0</td>
<td>1.0</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Intermediate shaft (shaft 3) and output shaft (shaft 4) speed curves of multifunctional power transmission device are shown in Fig.3.
Fig.3 Intermediate shaft and output shaft speed curves of power transmission device

Fig.3 shows that output shaft speed produce fluctuating phenomenon at the switching points (10s and 20s), and the general speed regulation characteristic is prefect. So the optimization for the shift point characteristic becomes the research focus. The minimum speed of output shaft is 123.14r/min at the time point of 11.14s, which means that the speed drop amplitude in position I is 73.03%; the minimum speed of output shaft is 414.64r/min at the time point of 20.63s, which means that the speed drop amplitude in position II is 22.12%. The solution of shift switching optimization problem in the 2 positions plays a decisive role for improving shift quality.

Jerk is defined as the second-order differential equation of vehicle longitudinal speed. $j_i$ can be regarded as the jerk of rear axle caused by the intermediate shaft. Similarly, $j_i$ can be regarded as the jerk of rear axle caused by the output shaft. Intermediate shaft and output shaft shift jerks are generated by the engagement and disengagement of switch components, which have close relationship with the related shafts. Intermediate shaft and output shaft shift jerks of power transmission device are shown in Fig.4.

Fig.4 Intermediate shaft and output shaft shift jerks of power transmission device

Relatively large jerks have been heavily focused on the 7 positions in Fig.4. The jerks in position I and IV are generated by the change of hydrostatic system displacement ratio. Although the change rate of displacement ratio is large, it has little influence on the shift jerks. The jerks in position II are generated by the engagement of Clutch C5 at the time point of 11.15s. At this moment, $j_{\text{max}}=-7.76$, $j_{\text{max}}=18.66$, the negative sign means both are opposition directions. The jerk in position III is generated by the engagement of Clutch C2 at the time point of 12.03s. At this moment, $j_{\text{max}}=-9.38$, $j_{\text{max}}=-8.85$, thereafter Clutch C4 and Brake B1 complete the switching process quickly. The jerk in position V is generated by the engagement of Brake B2 at the time point of 20.55s. At this moment, $j_{\text{max}}=15.82$, for the reason that the hydrostatic system can absorb braking energy, there is no obvious impact at shaft 4. The jerk in position VI is generated by the engagement of Clutch C4 at the time point of 22.28s. At this moment, $j_{\text{max}}=11.13$, $j_{\text{max}}=-6.28$. The jerk in position VII is generated by the engagement of Clutch C5 at the time point of 23.15s. At this moment, $j_{\text{max}}=-3.17$, $j_{\text{max}}=8.33$. General speaking, the jerk caused by shift actuator disengagement is larger than that caused by shift actuator engagement. Clutch C1 complete the disengaging process between the engaging process of Brake B2 and Clutch C4. Obviously, the diagram of jerk can well reflect the shift process and quality.

3.2 Shift characteristic

The multipurpose power transmission device studied in this paper includes 7 shift actuators (5
clutches and 2 brakes, and there are 4 shift actuators (3 clutches and 1 brake) to complete shift switch each time. The orthogonal tests (4 factors and 3 levels) can be used to solve the optimization problem of shift components sequence, and find out the best and worst working conditions. According to analysis, 3 typical working conditions can be got as follows:

Condition 1 (the best condition): Hydrostatic range → Hydro-mechanical range. Clutch C₂ is engaged at the time point of 10.0s, Clutch C₄ is disengaged at the time point of 10.0s, Clutch C₅ is engaged at the time point of 10.5s, and Brake B₁ is disengaged at the time point of 10.0s.

Hydro-mechanical range → Mechanical range. Clutch C₁ is disengaged at the time point of 20.5s, Clutch C₄ is engaged at the time point of 19.5s, Clutch C₅ is disengaged at the time point of 19.5s, and Brake B₂ is engaged at the time point of 20.5s.

Condition 2 (the common condition): Hydrostatic range → Hydro-mechanical range. Clutch C₂, C₄, C₅, B₁ are switched at the time point of 10.0s.

Hydro-mechanical range → Mechanical range. Clutch C₁, C₄, C₅, B₂ are switched at the time point of 20.0s.

Condition 3 (the worst condition): Hydrostatic range → Hydro-mechanical range. Clutch C₂ is engaged at the time point of 9.5s, Clutch C₄ is disengaged at the time point of 10.5s, and Brake B₁ is disengaged at the time point of 10.5s.

Hydro-mechanical range → Mechanical range. Clutch C₁ is disengaged at the time point of 19.5s, Clutch C₄ is engaged at the time point of 20.5s, Clutch C₅ is disengaged at the time point of 20.5s, and Brake B₂ is engaged at the time point of 19.5s.

Output shaft speed and shift jerk curves of the common condition (Condition 2) are shown in Fig.3 and Fig.4. Output shaft speed and shift jerk curves of the best condition (Condition 1) and the worst condition (Condition 3) are shown in Fig.5.
mechanical range, the maximum jerk of output shaft of condition 1 is 7.20 at the time point of 22.44s, the maximum jerk of output shaft of condition 3 is 9.16 at the time point of 23.56s. The maximum jerk of output shaft of condition 1 is smaller than that of condition 3, and the impact interval of condition 1 is also smaller than that of condition 3, which means shift quality is greatly improved.

The friction work of each component of multifunctional power transmission device can be shown in Table 3.

**Tab.3 The friction work of each component of multifunctional power transmission device**

<table>
<thead>
<tr>
<th>Shift components</th>
<th>C₁ (J)</th>
<th>C₂ (J)</th>
<th>C₃ (J)</th>
<th>C₄ (J)</th>
<th>B₁ (J)</th>
<th>B₂ (J)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condition 1</td>
<td>5216</td>
<td>9453</td>
<td>5332</td>
<td>8373</td>
<td>1140</td>
<td>7678</td>
</tr>
<tr>
<td>Condition 2</td>
<td>5366</td>
<td>4510</td>
<td>5261</td>
<td>5011</td>
<td>1034</td>
<td>6346</td>
</tr>
<tr>
<td>Condition 3</td>
<td>6095</td>
<td>4470</td>
<td>5704</td>
<td>3082</td>
<td>1011</td>
<td>1773</td>
</tr>
</tbody>
</table>

As shown in Tab.3, the total friction work of Condition 1, Condition 2 and Condition 3 is respectively 162470J, 115473J and 113712J. Obviously, Condition 2 saves 29% energy loss than Condition 1. Condition 3 only saves 1.5% energy loss than Condition 2, however, the steady shift characteristic of Condition 3 gets remarkable improvement. The result shows that the proportion of friction work generated by Clutch C₂ and C₄ accounts for more than 85% of the total friction work.

The friction work generated by Clutch C₂ occurs mainly in the shift process from hydrostatic range to hydro-mechanical range, and the friction work generated by Clutch C₄ occurs mainly in the shift process from hydro-mechanical range to mechanical range.

As shown in Fig.1, power output shaft can output power to drive other mechanisms. This paper assumes that the power output shaft can output power only in the hydro-mechanical range. Speed regulation characteristic curves of multifunctional power transmission device at different speeds are shown in Fig.6.

**Fig.6 Speed regulation characteristic curves of multifunctional power transmission device**

Power output leads to the time extension of speed drop, and forms a pit. Namely, time for driving disk to turn driven disk is extended, however, no significant change in the lowest point of speed drop is found, which shows that power output has a much smaller effect on system shift characteristic than shift shock. That is to say, the dynamic characteristic of transient condition is largely decided by the shift characteristic.

### 3.3 Efficiency characteristic

#### 3.3.1 Efficiency characteristic analysis based on empirical formulas

Multifunctional power transmission device is composed of hydrostatic transmission, hydro-mechanical transmission, and mechanical transmission, so the efficiency of hydrostatic system determines the efficiency of whole transmission system to a large extent. The efficiency of hydrostatic system can be calculated by empirical formulas, and the correlation coefficients of empirical formulas can be determined by test.

This paper assumes the interstitial fluid of hydrostatic system is the Newton liquid with the steady laminar motion, and ignores the gap change and compressibility of fluid[23].

The expression of pump efficiency is:

$$\eta_p = \frac{1 - C_1 10^{-7} \Delta p_r}{1 + C_1 \frac{\mu \nu}{10^{-3} \Delta p_r} + \frac{C_r}{|\nu|}} \quad (16)$$

The expression of motor efficiency is:
Hydrostatic system efficiency is affected by many factors, and there are also differences in \( C_f \), \( C_i \), and \( C_s \) with the different types and models of pump and motor. \( \mu \) is related to temperature and working condition.

According to test data, we can infer that:
\( C_f = 0.01 \), \( C_i = 0.8 \times 10^{-6} \), \( C_s = 0.2 \times 10^{-6} \).

Efficiency expressions of hydrostatic system and power transmission device can be given as follows:

\[
\eta_H = \eta_H' \eta_H'' \eta_H''' \tag{18}
\]

\[
\eta_1 = \eta_1' \eta_1'' \eta_1''' \tag{19}
\]

\[
\eta_2 = \frac{\epsilon + k_{ij}}{\epsilon + k_{ij1}} \eta_1(\eta_1'' \eta_1''' \eta_1'), \epsilon < 0 \tag{20}
\]

\[
\eta_2 = \frac{\epsilon + k_{ij1}}{\epsilon + k_{ij2}} \eta_1(\eta_1'' \eta_1''' \eta_1'), \epsilon > 0 \tag{21}
\]

According to Equation (16) ~ (21), efficiency curves of multipurpose power transmission device are shown in Fig.7.

![Fig.7 Efficiency curves of multipurpose power transmission device](image)

**Fig.7 Efficiency curves of multipurpose power transmission device**

The curves in Fig.7 show that the hydrostatic system efficiency \( \eta_H \) increases as \( \frac{\Delta p}{\mu n} \), but the change scope is small. The efficiency curves of each range can be plotted out when \( \frac{\Delta p}{\mu n} = 75 \) , we can infer that \( \eta_1 < \eta_2 < \eta_3 \). Reverse range and start range adopt hydrostatic transmission, the reason why the efficiency of reverse range and start range is smaller than that of hydrostatic system is the efficiency loss of gear transmission. Adopting large displacement ratio for hydrostatic range can improve system efficiency effectively. The total efficiency of hydro-mechanical composite transmission can reach above 85%. When \( \epsilon = 0 \), the hydro-mechanical composite transmission can be regarded as a lower range mechanical transmission. The mechanical transmission possesses the high efficiency, but the flexibility of system is lower, so it demands much of the road conditions.

### 3.3.2 Efficiency characteristic analysis based on test data

Closed stepless speed change system is mainly composed of variable displacement axial piston pump and fixed displacement axial piston motor, and the efficiency of system is largely decided by speed, pressure, displacement, and so on\[24\]. This paper carries out test by the design schemes of hydrostatic system test bench and power transmission device test bench\[25\]~\[27\]. The efficiency formulas of pump and motor can be got according to the test data, which can be shown as follows\[28\]~\[30\]:

\[
\eta_p = 0.87 \left( \frac{n}{n_{max}} \right)^{10} + 0.035 \sin \left( \frac{4n}{n_{max}} \right) \exp \left( -33 \frac{p}{p_{max}} \right)
- \exp \left( -50 \frac{p}{p_{max}} \right) + \exp \left( 0.5 \frac{p}{p_{max}} \right) \tag{22}
\]

\[
\eta_m = 0.87 \left( \frac{n}{n_{max}} \right)^{10} + 0.035 \sin \left( \frac{4n}{n_{max}} \right) \exp \left( -33 \frac{p}{p_{max}} \right)
- \exp \left( -50 \frac{p}{p_{max}} \right) + \exp \left( 0.5 \frac{p}{p_{max}} \right) \tag{23}
\]

Hydrostatic system efficiency can be determined according to the fitting equations:

\[
\eta_H = \eta_H' \eta_H'' \eta_H''' \tag{24}
\]

Generally speaking, there are 3 methods to calculate transmission efficiency of planetary gear mechanism: transmission ratio method, force migration method and conversion mechanism method, and this paper chooses the last one. Conversion mechanism method assumes that the frictional loss power of planetary gear transmission is equal to that of conversion mechanism, the transmission efficiency of planetary gear mechanism should associate the transmission efficiency of conversion mechanism with the relationship expression of the frictional power loss.
of conversion mechanism, and finally get the transmission efficiency of planetary gear mechanism.

The main ranges efficiency of multipurpose power transmission device can be got by conversion mechanism method:

\[
\eta_{p2} = \eta_k \eta_2 \eta_3 \eta_1 \eta_{ik}
\]

By comparing efficiency curves in Fig.7 and Fig.8, there are some differences between the both, however, the test results agree well with the theoretical analysis. That means hydrostatic system possesses preferable efficiency characteristic under the working condition of larger displacement, higher speed and medium pressure. Mechanical transmission efficiency is higher than hydro-mechanical transmission efficiency, and hydro-mechanical transmission efficiency is higher than hydrostatic transmission.

4. Conclusion

(1) Introduce a multipurpose power transmission device, which can realize the switch among hydrostatic, hydro-mechanical and mechanical transmission by clutches and brakes, and the relevant parameters can be got by kinematic and kinetic analysis for vehicle system. This paper also analyses the transmission characteristics of multipurpose power transmission device, including speed regulation characteristic, shift characteristic and efficiency characteristic.

(2) Speed regulation characteristic shows that multipurpose power transmission device can realize flexible start by hydrostatic transmission, stepless speed change by hydro-mechanical transmission, efficiency transportation by mechanical transmission. Power output shaft also can output power to drive other mechanisms, which reflects prefect transmission and output characteristic.

(3) Shift characteristic shows that shift quality can be improved effectively by controlling the switch sequence of actuators by orthogonal analysis method. Optimal shift strategy under different conditions should be recorded in the controller to ensure prefect shift quality and excellent transmission performance.

(4) Efficiency characteristic shows that hydrostatic system possesses preferable efficiency characteristic under the working condition of larger displacement, higher speed and medium pressure. Although hydrostatic transmission efficiency is relatively low, it can realize flexible operation; the mechanical transmission efficiency is relatively high, so it demands much of road conditions; the hydro-mechanical composite transmission can realize efficiency improvement easily in the scope of higher speed and whole displacement ratio.

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Data availability

The datasets used and/or analysed during the current study available from the corresponding author on reasonable request.

Reference


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