ANALYSIS AND OPTIMIZATION OF POWERTRAIN JUDDER DURING VEHICLE CREEPING

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In order to solve the powertrain judder and abnormal noise during vehicle creeping, a dynamic model with 4-DOF including transmission, differential, tire and vehicle body is established by taking a hybrid DCT vehicle as the research object. The paper builds the excitation function of clutch friction torque based on the theory of frictional vibration during the process of engagement and the tested result of clutch output torque, and analyzes the powertrain dynamic responses. An equivalent AMESim simulation model is established according to the dynamic model. The torsional mode and the transient response analysis result show that the frequencies coupling between the clutch frictional vibration and the 1\textsuperscript{st} order powertrain torsional mode is the cause of the powertrain judder. The effects of halfshaft torsional stiffness and fluctuation amplitude of clutch friction torque on the judder are studied. The simulation result shows that reduction of the fluctuation amplitude of the clutch friction torque could reduce judder effectively. Meanwhile the analysis results have a good consistent with the test results, which proves that the dynamic model is correct. The vehicle’s test result shows that reduction of the fluctuation amplitude of the clutch friction torque by improving assembly precision of the clutch friction disc can solve the powertrain judder and abnormal noise effectively, which could benefit for other vehicle’s powertrain NVH development.

Keywords: powertrain judder, clutch friction, frequencies coupling, torque fluctuation

1. Introduction

Sometimes, judder and abnormal noise occur inside a powertrain system during vehicle creeping. As a friction component, the clutch may have unstable vibration in the process of engagement. The frictional vibration theory of the clutch disc is complex, which is related to the friction characteristics of the contact interface, the external load and the geometrically accumulated error from manufacture and assembly. The friction force can excite the coupling vibration of the moving components, especially when the natural frequencies of moving components are close to each other, and the modal coupling can result in strong vibration and noise\textsuperscript{1}.

The scholars over the world have carried out related research on the clutch judder. Centea\textsuperscript{2} and Crowther\textsuperscript{3} studied the clutch judder for an automatic transmission which was mainly due to the negative gradient change of the friction coefficient with the relative velocity of the clutch disc. Rabieh et al\textsuperscript{4,5} studied the function relationship among clutch friction coefficient, rotation speed, pressure and surface temperature of the disc and the dynamic response of powertrain during the clutch engagement. Sawanobori et al\textsuperscript{6} studied the theory of forced vibration of clutch, and analyzed that the change of the axial force caused by deviation of the clutch assembly was the main cause of forced vibration. Wickramarachi et al\textsuperscript{7} studied the influence of parameters such as cushion segment
axial stiffness on the powertrain system stability and the dynamic response when the pressure was fluctuated. Meanwhile, some domestic scholars studied the influence of the negative gradient of friction coefficient and the fluctuation of the friction torque on the judder from the point of view of the self-excited vibration and forced vibration of the clutch.

The judder and abnormal noise happened during the vehicle creeping in the electric D block mode, and disappeared after the vehicle was shifted to normal driving condition. Figure 1 is the powertrain layout diagram, and the engine compartment noise, transmission vibration, mount bracket vibration and the VCU and TCU signals are tested synchronously in the process.

![Powertrain layout diagram](image1.png)

![Time domain vibration signals on transmission shell](image2.png)

Fig. 2 Time domain vibration signals on transmission shell

Fig.1 Powertrain layout diagram

The time domain vibration signals on the transmission shell, driving motor rotational speed, transmission input shaft rotational speed and clutch sliding speed are shown in figure 2 and figure 3. A complete process of the creeping happens between 40.5 seconds and 46 seconds, and the vehicle creeps from the idle condition to normal driving condition. Subjectively, the judder and abnormal noise is identified between 43.5 seconds and 45.5 seconds and the several abnormal peaks are found on the vibration signal of the transmission shell in Figure 2, and the interval between the peaks is about 0.1s, which means the frequency is about 10Hz. As shown in Figure 3, the motor speed is almost a constant around 880rpm. The relative sliding speed of the clutch disc has obvious speed fluctuation, and the maximum fluctuation occurs between 43.5 second and 45.5 second when the sliding speed reduces from 580 rpm to 400 rpm.

The time-frequency analysis of the sliding speed signals of the clutch and the vibration signals on the transmission shell are shown in figure 4 and figure 5, respectively. As shown in figure 4, the sliding rotation frequency decreases from 14 Hz to 7 Hz during the vehicle creeping, and the 1st order fluctuation of the sliding speed is obvious. As shown in figure 5, the vibration energy of the transmission shell has obvious order characteristics, and frequency of the order decreases from 14 Hz to 7 Hz which corresponds to the first rotation frequency of the clutch sliding speed. Moreover, the energy is mostly concentrated around 9.7 Hz that corresponds to the frequency of the judder and abnormal noise between 43 seconds and 45 seconds.

![Colormap of clutch sliding speed](image3.png)

![Colormap of transmission shell vibration](image4.png)

Fig.4 Colormap of clutch sliding speed

Fig.5 Colormap of transmission shell vibration

As shown in figure 6, the tested clutch output torque shows obvious fluctuation during the clutch engagement. The colormap of the clutch torque signals has obvious order characteristics, and frequency of the order decreases from 14 Hz to 7 Hz which corresponds to the 1st order rotation frequency of the clutch sliding speed.
In order to solve the hybrid vehicle’s powertrain judder and abnormal noise during creeping, the paper establishes a 4-DOF dynamic model including transmission, differential, tire and vehicle body, and analyzes the mechanism of the judder. Influence of different parameters on the judder is simulated, and the transmission vibration can be effectively reduced by optimizing the fluctuation range of the clutch friction torque. The powertrain judder and abnormal noise are effectively controlled by optimizing the clutch. The test results prove that the modelling method and analytical method are correct, which could benefit for other vehicle’s powertrain NVH development.

2. Establishment and analysis of dynamic model

During the vehicle creeping, the wet clutch is at the sliding state and the powertrain is not fully connected with the drive motor. For convenience, the powertrain is simplified as a 4-DOF dynamic model as shown in figure 8. \( J_1 \) is moment of inertia of the clutch odd driven disc and DCT, \( J_2, J_3 \) and \( J_4 \) are equivalent moment of inertia of the differential and tire and vehicle body that converts to the transmission input shaft respectively. \( \theta_1, \theta_2, \theta_3 \) and \( \theta_4 \) are the corresponding angular displacement respectively. \( K_1, K_2 \) and \( K_3 \) are equivalent torsional stiffness of the intermediate shaft of transmission and halfshaft and tire respectively. \( C_1, C_2 \) and \( C_3 \) are equivalent equivalent rotational viscous damping respectively. \( T_d \) is the friction torque of the clutch, and \( T_r \) is the equivalent resistance moment of the road resistance that converts to the transmission input shaft.

\[
\begin{align*}
T_r &= \left( \begin{array}{c}
\theta_1 \\
\theta_2 \\
\theta_3 \\
\theta_4
\end{array} \right) \left( \begin{array}{cccc}
J_1 & 0 & 0 & 0 \\
0 & J_2 & 0 & 0 \\
0 & 0 & J_3 & 0 \\
0 & 0 & 0 & J_4
\end{array} \right) \\
&\quad \times \left( \begin{array}{c}
\dot{\theta}_1 \\
\dot{\theta}_2 \\
\dot{\theta}_3 \\
\dot{\theta}_4
\end{array} \right) + \left( \begin{array}{c}
0 \\
C_1(\dot{\theta}_1 - \dot{\theta}_2) + K_1(\theta_1 - \theta_2) \\
C_2(\dot{\theta}_2 - \dot{\theta}_3) + K_2(\theta_2 - \theta_3) + K_3(\theta_2 - \theta_3) \\
C_3(\dot{\theta}_3 - \dot{\theta}_4) + K_3(\theta_3 - \theta_4)
\end{array} \right) + \left( \begin{array}{c}
0 \\
0 \\
0 \\
0
\end{array} \right) \\
&\quad \times \left( \begin{array}{c}
\ddot{\theta}_1 \\
\ddot{\theta}_2 \\
\ddot{\theta}_3 \\
\ddot{\theta}_4
\end{array} \right)
\end{align*}
\]

The dynamic equations of the powertrain in the process of clutch engagement are as follows

\[
\begin{align*}
J_1 \ddot{\theta}_1 + C_1(\dot{\theta}_1 - \dot{\theta}_2) + K_1(\theta_1 - \theta_2) &= T_d \\
J_2 \ddot{\theta}_2 + C_2(\dot{\theta}_2 - \dot{\theta}_3) + K_1(\theta_2 - \theta_1) + K_2(\theta_2 - \theta_3) &= 0 \\
J_3 \ddot{\theta}_3 + C_3(\dot{\theta}_3 - \dot{\theta}_4) + K_2(\theta_3 - \theta_2) + K_3(\theta_3 - \theta_4) &= 0 \\
J_4 \ddot{\theta}_4 + C_3(\dot{\theta}_4 - \dot{\theta}_3) + K_3(\theta_4 - \theta_3) &= T_r
\end{align*}
\]

The friction torque of the clutch is expressed as

\[
T_d = F_f(t) \ast R
\]

where, \( F_f \) is friction force of the clutch on the contacted surface; \( R \) is equivalent friction radius of the clutch.

The friction force between the contacted surfaces of the wet clutch can be expressed as

\[
F_f(t) = \mu(v_{rel}) \ast N(t)
\]

where, \( \mu(v_{rel}) \) is friction coefficient of the clutch, and it is the function of the relative sliding speed; \( N(t) \) is contact pressure of the clutch disc.

Because of the non-uniform contact during the process of the wet clutch sliding, the contact pressure between the clutch discs is constantly changing. As shown in figure 4, the analysis result shows that the fluctuation frequency of the clutch output torque corresponds to the first order rotation frequency of the clutch sliding speed. Therefore, the contact pressure of the clutch is...
\[
N(t) = N_m(t) + N_p \sin(\theta_a - \theta_1)
\]

where, \(N_m(t)\) is the nominal clutch contact pressure; \(N_p\) is the fluctuation amplitude of pressure, which is related to the axial cumulated error of the clutch assembly; \(\theta_a\) is the angular displacement of clutch drive disc.

Based on the LuGre friction model, the friction coefficient of the clutch is expressed as
\[
\mu(\nu_{rel}) = \mu_s + \delta \left( \dot{\theta}_a - \dot{\theta}_1 \right)
\]

where, \(\mu_s\) is the static friction coefficient; \(\delta\) is a constant that affects the slope of the friction coefficient; \(\dot{\theta}_a\) is the clutch driving disc speed; and \(\dot{\theta}_1\) is the clutch driven disc speed.

Substitute equations (3), (4), (5) into (2), obtaining
\[
T_d = \left[ \mu_s + \delta \left( \dot{\theta}_a - \dot{\theta}_1 \right) \right] \cdot [N_m(t) + N_p \sin(\theta_a - \theta_1)] \cdot R
\]

From equation (6), the torque fluctuation is clearly identified during the process of the clutch engagement. The 1st order fluctuation amplitude is defined as \(T_p\),
\[
T_p = \left[ \mu_s + \delta \left( \dot{\theta}_a - \dot{\theta}_1 \right) \right] \cdot N_p \cdot R
\]

According to the test results, the equivalent resistance moment of the road resistance converted to the transmission input shaft can be expressed as
\[
T_r = a \cdot \dot{\theta}_4^2 + b \cdot \dot{\theta}_4 + c
\]

where, \(a\), \(b\) and \(c\) are the constants affecting the change of the road resistance.

Substitute equations (6), (7), (8) into (1), and equation (1) can be written as a matrix
\[
J \dot{\theta} + C \dot{\theta} + K \theta = T
\]

where,
\[
J = \begin{bmatrix}
J_1 & 0 & 0 & 0 \\
0 & J_2 & 0 & 0 \\
0 & 0 & J_3 & 0 \\
0 & 0 & 0 & J_4
\end{bmatrix},
C = \begin{bmatrix}
C_1 & -C_1 & 0 & 0 \\
-C_1 & C_1 + C_2 & -C_2 & 0 \\
0 & -C_2 & C_2 + C_3 & -C_3 \\
0 & 0 & -C_3 & C_3
\end{bmatrix},
K = \begin{bmatrix}
K_1 & -K_1 & 0 & 0 \\
-K_1 & K_1 + K_2 & -K_2 & 0 \\
0 & -K_2 & K_2 + K_3 & -K_3 \\
0 & 0 & -K_3 & K_3
\end{bmatrix},
T = \left[ \left[ \mu_s + \delta \left( \dot{\theta}_a - \dot{\theta}_1 \right) \right] \cdot N_m(t) \cdot R + T_p \cdot \sin(\theta_a - \theta_1) \right]
\]

Substitute the corresponding parameters of the model into matrix (9) and (10), and the angular displacement, angular velocity and angular acceleration responses of the transmission, differential, tire and vehicle are obtained by dynamic time domain response calculations.

### 3. Simulation analysis and optimization

#### 3.1 Modelling and simulation analysis of judder

As shown in figure 9, the equivalent AMEsim model including drive motor, transmission, differential, tires and vehicle body is established. The model parameters are listed in table 1.

![Fig.9 AMEsim simulation model of the powertrain](image-url)
Table 1: Parameters of the powertrain model

<table>
<thead>
<tr>
<th>Parameters</th>
<th>values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total moment of inertia of odd clutch disc and transmission/kg.m$^2$</td>
<td>0.0098</td>
</tr>
<tr>
<td>Moment of inertia of differential/kg.m$^2$</td>
<td>0.0498</td>
</tr>
<tr>
<td>Moment of inertia of tire /kg.m$^2$</td>
<td>1.224</td>
</tr>
<tr>
<td>Moment of inertia of vehicle body/kg.m$^2$</td>
<td>136</td>
</tr>
<tr>
<td>Torsional stiffness of intermediate shaft of transmission/N.m/°</td>
<td>350</td>
</tr>
<tr>
<td>Torsional stiffness of halfshaft/N.m/°</td>
<td>354</td>
</tr>
<tr>
<td>Torsional stiffness of tire/N.m/°</td>
<td>1180</td>
</tr>
<tr>
<td>Rotational viscous damping /N.m/rev/min</td>
<td>2</td>
</tr>
<tr>
<td>Clutch input torque/N.m</td>
<td>15</td>
</tr>
<tr>
<td>Mean road resistance torque /N.m</td>
<td>21</td>
</tr>
<tr>
<td>Motor speed/rev/min</td>
<td>880</td>
</tr>
<tr>
<td>Transmission ratio of the first gear</td>
<td>59/14</td>
</tr>
<tr>
<td>Transmission ratio of the differential</td>
<td>73/19</td>
</tr>
</tbody>
</table>

The calculation results of the first three order torsional modes of the powertrain during vehicle creeping are listed in table 2. The frequency of the 1st order torsional mode is 9.4Hz, which is close to the frequency of the judder and abnormal noise, 9.7Hz, in the test.

Table 2: Torsional mode of the powertrain during creeping

<table>
<thead>
<tr>
<th>Modal order</th>
<th>Modal frequency/Hz</th>
<th>Modal shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>First</td>
<td>9.4</td>
<td>Vibration of transmission and differential</td>
</tr>
<tr>
<td>Second</td>
<td>39.7</td>
<td>Vibration of differential and tire</td>
</tr>
<tr>
<td>Third</td>
<td>144.5</td>
<td>Vibration of differential</td>
</tr>
</tbody>
</table>

The speed and acceleration responses of each component are obtained by the simulation. As shown in figure 10, the 1st order fluctuation of the clutch sliding speed between 580 and 400 rpm is huge. In figure 11, the abnormal peak on the transmission vibration signal is found in the above speed range, and the corresponding frequency is consistent with the 1st order modal frequency of the powertrain. In figure 12, obvious order characteristics are identified on the colormap of the angular acceleration of transmission, and the order frequency decreases from 14Hz to 7Hz that corresponds to the first rotation frequency of the clutch sliding speed, and the energy is mostly concentrated around 9.4Hz. The simulation results have a good consistent with the vehicle’s test results of the judder in figure 2, figure 3 and figure 5, which verifies that the theoretical model is correct.
### 3.2 Analysis of influence factors

#### 3.2.1 Influence of torsional stiffness of halfshaft

The analysis result in figure 13 is the peak curve of the transmission vibration response varying with the torsional stiffness of the halfshaft. As shown in figure 13, the peak value of the transmission vibration response decreases gradually with the increase of the torsional stiffness of the halfshaft, and its decrease trend is not obvious after the torsional stiffness increases to 700 N.m/degree. As shown in figures 14 and 15, the fluctuation of the clutch sliding speed and peak value of the transmission vibration decreases slightly when the torsional stiffness increases from 354N.m to 500N.m/degree. The simulation results show that increase of torsional stiffness of the halfshaft reduces the amplitude of transfer function of the transmission vibration, which has a little benefit to reduce the powertrain judder.

#### 3.2.2 Influence of fluctuation amplitude of clutch friction torque

Figure 16 shows the peaks of the transmission vibration responses varying with the fluctuation amplitude of the clutch friction torque. The peak values increase gradually with the increase of the fluctuation of clutch friction torque, and they show obvious linear relationship. As shown in figures 17 and 18, the 1st order fluctuation of clutch sliding speed and transmission vibration decreases obviously when the fluctuation of friction torque decreases from 0.8N.m/15N.m to 0.4N.m/15N.m, and the peak value of the transmission vibration reduces from 450 rad/s to 200 rad/s. The simulation results show that the reduction of fluctuation amplitude of the clutch friction torque reduces the amplitude of source excitation, which effectively reduces the powertrain judder.

### 4. Test research

Since the clutch geometrically accumulated error from manufacture and assembly has a great influence on the fluctuation of clutch friction torque, the fluctuation amplitude of the clutch friction...
torque can be reduced by improving assembly precision of the clutch friction disc. The optimized clutch with the 1st order fluctuation amplitude $T_p \leq 0.4 \text{N.m}/15\text{N.m}$ is picked out, and the fluctuation amplitude $T_p$ of friction torque of the wet clutch with judder and abnormal noise problem is detected, and the test results are listed in table 3. After the original problem clutch $T_p$ was detected and its value is $0.83 \text{N.m}/15\text{N.m}$, which show that its fluctuation amplitude is too high, while the optimized clutch $T_p$ value is $0.05 \text{N.m}/15\text{N.m}$, which is almost close to the acceptable limited level.

Table 3: Tested value of fluctuation amplitude $T_p$ of clutch friction torque

<table>
<thead>
<tr>
<th>The state of clutch</th>
<th>$T_p$ (N.m/15N.m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>original state</td>
<td>0.83</td>
</tr>
<tr>
<td>optimized state</td>
<td>0.05</td>
</tr>
</tbody>
</table>

The optimized clutch is installed on the vehicle, and the test results are shown in figures 19 and 20. The abnormal vibration on the transmission shell disappears, and the judder and abnormal noise cannot be identified by engineers’ subjective evaluation. Therefore, reduction of the fluctuation amplitude of the clutch friction torque $T_p$ by improving assembly precision of the clutch friction disc can effectively diminish the powertrain judder.

Fig.19 Optimization effect of transmission vibration by reducing fluctuation of clutch friction torque

Fig.20 Optimization effect of sliding speed by reducing fluctuation of clutch friction torque

It is an effective way to reduce the fluctuation amplitude of the clutch friction torque by more stringent manufacture and assembly control of the clutch disc. However, over stringent control will impact the clutch production efficiency, so a practical standard $T_p \leq 0.4 \text{N.m}/15\text{N.m}$ is proposed for the disk mass production. Table 4 lists the $T_p$ values of the mass-produced clutches after the stringent control process is executed. All the clutches are installed on the vehicles, and subjectively, the judder and abnormal noise cannot be perceived.

Table 4: Tested value of fluctuation amplitude $T_p$ of clutch friction torque

<table>
<thead>
<tr>
<th>Clutch number</th>
<th>$T_p$ (N.m/15N.m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>H280279</td>
<td>0.06</td>
</tr>
<tr>
<td>H280280</td>
<td>0.10</td>
</tr>
<tr>
<td>H280281</td>
<td>0.18</td>
</tr>
<tr>
<td>H280282</td>
<td>0.35</td>
</tr>
<tr>
<td>H280284</td>
<td>0.20</td>
</tr>
<tr>
<td>H280288</td>
<td>0.22</td>
</tr>
<tr>
<td>H280293</td>
<td>0.14</td>
</tr>
<tr>
<td>H280294</td>
<td>0.32</td>
</tr>
<tr>
<td>H280315</td>
<td>0.08</td>
</tr>
</tbody>
</table>
5. Conclusions

1) The paper establishes a 4-DOF dynamic model including transmission, differential, tire and vehicle body, builds the excitation function for the clutch friction torque based on the theory of frictional vibration during engagement phase of the wet disc and the tested result of clutch output torque, and analyzes the powertrain dynamic response.

2) An equivalent AMEsim simulation model is established according to the dynamic model. The torsional mode and the transient response analysis result show that the frequency coupling between the clutch friction-vibration and the 1st order powertrain torsional mode is the cause of the powertrain judder. Influences of the torsional stiffness of the halfshaft and fluctuation amplitude of clutch friction torque on the judder are studied.

3) The test results and subjective evaluation prove that the dynamic model is correct. The test results show that reduction of the fluctuation amplitude of the clutch friction torque by improving assembly precision of the clutch friction disc can effectively diminish the powertrain judder and abnormal noise.

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