Analysis on Synchronizing and Gear Shifting Force of Commercial Vehicle Powertrain

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Abstract. Based on a commercial vehicle transmission as the research object, the paper analyses the influence for commercial vehicle powertrain synchronization and shift performance by the rotational inertia of the clutch driven plate assembly and transmission input speed. According to the synchronization and shifting mechanism of the transmission, transmission shift dynamics model was established. Analyzing the causes of shift impact and the critical shifting force for mis-engagement put forward the shift control strategy to avoid shift impact. We can draw a conclusion that optimizing the clutch driven plate inertia parameter during clutch design can effectively improve the shift performance of the transmission.

1. Introduction

For traditional manual transmission(MT), the driver completes the shift of the target shift position through the shift fork. With the application of hydraulic and electronic control systems in the transmission, automatic mechanical transmission (AMT) has been developed, which has the advantages of simple operation, low cost and reliable operation. In recent years, AMT has been widely used in heavy-duty commercial vehicle.

At present, the proportion of assembling AMT on heavy commercial vehicles in Europe and the United States is gradually increasing. It is predicted that in the next few years, nearly 50% of MT in Europe will be replaced by AMT [1]. The AMT transmission has high reliability, but the shifting performance of the transmission still needs to be studied in depth. For example, during the shifting process, due to the fact that the current gear speed and the target gear speed are not consistent, this will causes mis-engagement problem. Such an operation would increase shift impact and severely damage the synchronizer, affecting the life of the transmission. Literature [2] used ADAMS to simulate and analyse the various stages of the shifting process, and the relationship between shifting force and shifting displacement is obtained, which provides a basis for the transmission shift control. Literature [3] established the clutch dynamics model, analysed and optimized the stiffness and damping of the clutch by ADAMS software to improve the performance of the clutch. Literature [4] analysed the influence of the clutch's output shaft inertia and stiffness on the performance of the wet clutch by bench test and proposed an effective scheme for optimizing the shift performance. Literature [5] studied all factors affecting the gear shifting force by fault tree analysis and found the fundamental reason of the problem is that the capacity of synchronizer torque is not enough and the force of self-
locking spring is too big. To solve the problem of shift impact and synchronizer abnormal abrasion in electronic automatic mechanical transmission (AMT) of hybrid electric vehicle (HEV), the mechanism of shift impact and the causes were analysed and experimented[6]. The existing research results analysed the shifting characteristics of the clutch, self-locking spring and transmission shaft, and there is no analysis of the clutch driven plate inertia. In current paper analysed the influence of clutch driven plate inertia on shifting performance by modelling and analysing the shifting process. Furthermore, using the transmission test bench verified the proposed shift strategy.

2. Analysis of shift process
The equivalent concentrated inertia method is used to simplify the vehicle powertrain system into a linear multi-degree-of-freedom elastic concentration inertia system, and the relevant components exist in the form of concentrated inertia [7].

2.1. Shift performance evaluation index
Shifting force. The main components affecting the shifting force during the shifting process are the fork shaft, the shift fork, the locating sleeve and the synchronizer, and the friction of the surface of each component when in contact [8]. Refer to literature [9], the shifting force can be obtain,

\[ F = \frac{J \Delta \omega \cos \theta}{i R \mu \Delta t} \]  \hspace{1cm} (1)

Where \( J \) is the synchronization inertia; \( \Delta \omega \) is the rotational speed difference; \( \Delta t \) is the synchronization time; \( R \) is the effective radius of the synchronization ring cone; \( \mu \) is the friction coefficient; \( i \) is the leverage ratio; \( \eta \) is the operating system efficiency; \( \theta \) is the synchronization ring cone angle.

Shift impact. The degree of impact is generally used to evaluate shift performance. The mathematical expression of the impact \( j \) is:

\[ j = \frac{d^2 v}{dt^2} \]  \hspace{1cm} (2)

Where \( j \) is the vehicle impact; \( a \) is the vehicle acceleration; \( v \) is the vehicle speed; \( t \) is the time. In general, the upper limit \( j_{\text{max}} \) of the German standard shift impact index is \( 10 \text{m} \cdot \text{s}^{-3} \).

2.2. Critical shifting force for mis-engagement
At the beginning of the synchronous shifting phase of the mechanical synchronizing state, under normal circumstances, the synchronization ring rotates half a tooth width angle under the action of the friction torque, and the engaging sleeve teeth are brought into contact with the synchronization state; If the engaging sleeve moves too fast, it may move the distance \( d \) before the synchronization ring moves half a tooth width angle, and will directly mesh with the synchronization ring miss the synchronizing phase. At this time, the engaging sleeve and the ring gear still have a large rotational speed difference. When they are mesh, a serious toothing phenomenon will occur, which is called mis-engagement toothing [10].

According to literature [10] the critical value of the shifting force:

\[ F_{\text{shift}}' = f(F_0) \]  \hspace{1cm} (3)

When, \( F_{\text{shift}}' \leq F_{\text{shift}} \), mis-engagement phenomenon does not occur.

3. Theoretical shifting force calculation
The transmission scheme is shown in figure 1. Refer to literature [9], the input part should be converted to the output shaft during the calculation of the shifting force.
3.1. Theoretical shifting force calculation

The parameters of the selected test transmission samples are listed in tables 1 to table 3.

**Table 1.** Moment of inertia of each axis  \( \text{kg} \cdot \text{m}^2 \)

| Input shaft | intermediate shaft | Output shaft | clutch driven plate |
|--------------|-------------------|--------------|---------------------|
| moment of inertia | 4.315 | 12.05 | 1.939 | 37.308 |

**Table 2.** Intermediate axis parameters

| intermediate shaft | Teeth | moment of inertia \( (\text{kg} \cdot \text{m}^2) \) |
|--------------------|-------|----------------|
| 1st                | 11    | 0              |
| 2nd                | 20    | 0              |
| 3rd                | 27    | 0              |
| 4th                | 35    | 0              |
| 5th                | 40    | 0              |
| 6th                | 43    | 0              |
| Rear               | 11    |

**Table 3.** Output axis parameters

| Output shaft | Teeth | moment of inertia \( (\text{kg} \cdot \text{m}^2) \) |
|--------------|-------|----------------|
| 1st          | 43    | 26.8           |
| 2nd          | 47    | 15.8           |
| 3rd          | 38    | 10.1           |
| 4th          | 30    | 5.6            |
| 5th          | 21    | 9.8            |
| 6th          | 40    | 2.9            |
| Rear         | 15.8  |

The theoretical shifting force of the \( 2 \rightarrow 1 \) gears with different speeds and inertias calculated by equation (1) is listed in table 4.

**Table 4.** Calculation of theoretical shifting force/ \( N \)

| Gear position | moment of inertia \( (\text{kg} \cdot \text{m}^2) \) | Effective radius of the ring cone \( / \text{m} \) | Engine speed \( / \text{r} \cdot \text{min}^{-1} \) |
|---------------|-----------------------------------------------|-----------------------------------------------|-----------------------------------------------|
| \( 2^{\text{nd}} \rightarrow 1^{\text{st}} \) gear | 2.793 409 265 | 0.178 2 | 1 500 | 59.61 | 79.48 | 104.34 |
| \( 2^{\text{nd}} \rightarrow 1^{\text{st}} \) gear | 2.796 119 265 | | 2 000 | 62.63 | 82.51 | 109.38 |
| \( 2^{\text{nd}} \rightarrow 1^{\text{st}} \) gear | 2.797 175 265 | | 2 500 | 64.65 | 84.54 | 114.42 |

*Note: The synchronization time is 0.6s.*

The calculation and analysis of the shifting force can be concluded that the shifting force increases with the increase of the engine speed under the same gear, and the shifting force increases as the moment of clutch driven plate inertia increases. It can be seen from the calculation of the theoretical shifting force that the clutch driven plate inertia has an influence on the shifting force.

4. Simulation analyses

4.1. Shifting force simulation analysis
Built shifting process model on the Matlab/Simulink platform and simulated the shift process. Taking 2→1 gears as an example. Using the shifting Simulink model simulate the 2 → 1 gears shifting process, the synchronizer is first engaged with the 2nd gear; at the 5th second, given the shift signal, and the synchronizer disengaged with the 2nd gear and then engaged with 1st gear. In this process, the positional relationship between the engaging sleeve and the locating pin is shown in figure 2.

**Figure 2.** positional relationship between the engaging sleeve and the locating pin

The equivalent moment of inertia $J_{rd}$ of the input shaft was changed to 2.793 409 265, 2.796 119 265 $\text{kg} \cdot \text{m}^2$, respectively. The simulation results of the shifting force are shown in figure 3. It can be conclude from the simulation results that the shifting force increases with the increase of the equivalent moment of inertia $J_{rd}$ at the same engine speed.

**Figure 3.** Shifting force under different inertia (N)

The engine speed was changed to 1 500, 2 000 $r \cdot \text{min}^{-1}$ for simulation. The simulation results are shown in figure 4. It can be conclude from the simulation results that the shifting force increases with the increase of the engine speed under the same equivalent moment of inertia $J_{rd}$.

**Figure 4.** Shifting force (N) at different engine speeds
4.2. Shift impact simulation
The shift impact simulation was performed at the same engine speed, equivalent moment of inertia $J_{rd}$ of the input shaft was changed to 2.793 409 265, 2.796 119 265 kg · m$^2$, respectively, and the simulation results are shown in the figure 5.

![Figure 5. Shifting impact under different inertia](image)

It can be seen from the simulation results of the shifting impact of figure 5 that the maximum shift impact is within the range of the upper limit $J_{max} = 10m · s^{-3}$ of the German standard. The increase of the clutch driven plate inertia will increase the shifting impact and affect the shifting performance. Therefore, in the design of the clutch, the influence of the clutch driven plate inertia on the manipulability of the transmission should be fully considered. At the same time, the shift control strategy should be optimized to avoid shifting impact.

4.3. Studies on Avoiding Shift Impact Control Strategy
Avoiding mis-engagement phenomenon can reduce the shift impact. From the previous analysis, the critical shifting force $F_{shift}$ to avoid mis-engagement teething is:

$$F_{shift} \leq F_{shift}^{'},$$

For AMT, the shifting force can be controlled by controlling the cylinder chamber pressure of the shift actuator cylinder at different stages [11]. A flow chart of the control strategy for avoiding shift impact is shown in figure 6.

![Figure 6. Shift synchronization process control flow chart](image)
First, the position of the shift actuator is detected by the shift stroke displacement sensor. Then, the shifting actuator eliminates the shifting clearance. After the shifting clearance is eliminated, the synchronizer starts to enter the synchronization phase, and the electronic control unit controls the pressure of the shifting actuator cylinder to increase the force acting on the engaging sleeve, so that the synchronizer quickly enters the synchronization phase. In this process, the electronic control unit continuously adjusts the pressure of the cylinder according to the difference between the actuator position, the current gear position and the target gear speed. When detecting that the shift shaft speed is tighter than the target speed, the cylinder will adjusted again and quickly complete the shift process [12-13].

5. Bench test verification

5.1. Transmission test bench

For the above shift control strategy, an experiment was conducted on a test bench as shown in figure 7. The test bench can simulate the process of transferring the whole vehicle power from the clutch driven plate to the wheel during the shifting process. The main components are an electrical system, a pneumatic shifting actuator, a data acquisition system, and a simulated road resistance mechanism. The drag motor is connected with the inertia flywheel to simulate the inertia of the vehicle during driving; the transmission is connected to the clutch driven plate at the input end to simulate the inertia of the input end of the vehicle when shifting. During the test, the data acquisition system collects the sensor's signal in real time, and controls the movement of the shifting mechanism, and records the shifting data [14-15].

![Figure 7. Transmission test bench](image)

5.2. Bench test results

Take the test data of 2→1 gears for analysis. The data for the bench test are listed in table 5.

| Equivalent moment of inertia $J_{rd}$ | Engine speed/(r·min$^{-1}$) |
|--------------------------------------|-------------------------------|
|                                      | 1 500 | 2 000 | 2 500 |
| $2^{\text{nd}}$→$1^{\text{st}}$      | 2.793 409 265                  | 64.12  | 90.60  | 108.26 |
|                                      | 2.796 119 265                  | 68.28  | 91.47  | 112.91 |
|                                      | 2.797 175 265                  | 73.14  | 96.76  | 114.79 |

Note: The synchronization time is 0.6s.
5.3. Analysis of test results

Take the test simulation and test data of 2→1 gears for analysis. Through the bench test, the transmission shifting force data under different conditions is obtained. According to the experimental data, the relation curve between the actual shift force and synchronization time was obtained. As shown in figure 8 and figure 9.

\[
J_{rd} = 2.793 \times 10^6 \text{kg m}^2
\]

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Figure 8. Actual shifting force and theoretical shifting force curve under different inertia

Figure 9. Actual shifting force and theoretical shifting force curve at different engine speeds

It can include from the shifting force curves of figure 8 and figure 9 that the change trend of the shifting force increases with the increase of the inertia of the clutch driven plate assembly at the same input shaft speed during the 2→1 gears; Under the same equivalent moment of inertia \(J_{rd}\), as the input shaft speed increases, the change trend of the shifting force increases with the increase of the input speed. Comparing the test value and theoretical value and simulation value of the shifting force, it can be seen that the changing trend of the shifting force is basically the same, the average error is about 8%, and the overall error is basically within 10%. The error is mainly caused by the acquisition of the shift process data and the accuracy of the test bench.

6. Conclude

Through research, we can draw the following conclusions:

(1) By establishing the power transfer model of the shifting process and analyzing the factors affecting the shifting force simplified the theoretical calculation formula of the shifting force.

(2) Analyzed the shifting process of the synchronizer and the mechanism of the shift impact, and obtain the optimal shifting force that achieves the shifting requirement without generating the toothing phenomenon.

(3) The dynamic model of the shifting process is established, and the model is simulated by Matlab/Simulink. The simulation results show that the inertia of the clutch driven plate assembly affect the shift performance.

(4) The test results show that the moment of inertia of the clutch driven plate assembly affect the shifting force of the different gears.

From the above analysis, it is shown that the moment of inertia of the clutch driven plate assembly affect the shift performance. Therefore, optimizing the clutch driven plate inertia parameter during clutch design can improve the shifting performance of the transmission.
References

[1] Lu X T and Hou G Z 2004 Introduction of the AMT control system structure and main foreign AMT products 5 Automobile Technology 19-22

[2] Chen Y X, Zang M Y and Chen Y 2012 Shift force analysis of manual transmission based on virtual prototyping technology 23 (8) China Mechanical Engineering 996-1000

[3] Yuan D and Pan Y X 2009 Optimal design for stiffness and damping of automobile friction clutch International Conference on Natural Computation. IEEE 369-373.

[4] Fatima N, Marklund P and Larsson R 2013 Influence of clutch output shaft inertia and stiffness on the performance of the wet clutch 56 (2) Tribology Transactions 310-319.

[5] Yue L J, Si Z M and Chen J 2012 Analysis of causes of large gear shifting force on a light bus and the improvement 35 (11) Journal of Hefei University of Technology (Natural Science) 1465-68

[6] Liu C W and Liu K L 2011 Research and control on AMT shift torque of hybrid electric bus 25 (1) Journal of HuBei Automotive Industries Institute 22-24

[7] Wang X F 2010 Automotive chassis design Beijing: Tsinghua University Press 127-133

[8] Cheng X X, Chen H X and Tian G Y 2014 Investigation on influence of shift force on shift impact of integrated motor-transmission system (4) Automotive Technology 1-5

[9] Wang Y, Xi J Q and Chen H Y 2009 A study on the mechanism and countermeasures for shift-impact in AMT 31 (3) Automotive Engineering 253-257.

[10] Tian J Y 2009 Construction principle and design method of vehicle automatic transmission Beijing: Peking University Press 154-186

[11] Liu Y G 2010 Study on integrated control of passenger vehicles equipped with Dual Clutch Transmissions PhD thesis College of Mechanical Engineering of Chongqing University, Chongqing, China

[12] Zhu X F 2014 Research on shift control strategy of Dual Clutch Transmission M.A. thesis Xi’an University of Science and Technology, China

[13] Hoshino H 1999 Analysis on Synchronization Mechanism of Transmission International Congress & Exposition.

[14] Kim J, Sung D and Seok C 2002 Development of shift feeling simulator for a manual transmission International Body Engineering Conference & Exhibition and Automotive & Transportation Technology Congress 502–513.

[15] Hu J J, Li G H and Wu G Q 2008 Accurate calculation of clutch torque transmission during vehicle starting 30 (12) Automotive engineering 1083-86