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RESEARCH ARTICLE

Collaborative Optimization Control of Torque Unloading and Loading for a Dual-Motor Coupling Drive System

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ABSTRACT Taking a dual-motor planetary two-gear transmission drive system as the research object, this paper analyzes the gear shift process as well as the factors influencing gear shift process and major performance indexes, with emphasis on the effects of torque unloading and loading processes on vehicle impact during gear shift stage. In view of the collaborative operation of main motor, auxiliary motor and gear shift actuator of transmission, this paper proposed a control method and strategy process featuring multiple collaborative gear-shift torque. Then, a simulation platform was built for the vehicle and power system to simulate torque unloading and torque recovery under specified acceleration conditions. Compared with non-collaborative torque controls, the vehicle impact under the multiple collaborative control was reduced by 60% in torque unloading stage and decreased by 80% in torque recovery stage. Furthermore, prototype transmission control units and powertrain systems were developed, and installed on the entire vehicle for validation test. The test results further demonstrated that multiple collaborative gear-shift torque control could effectively reduce the vehicle impact during unloading and loading processes, with gear shift time shortened and gear shift comfort improved.

INDEX TERMS Dual-motor coupling drive system, unloading stage, loading stage, vehicle impact, multiple cooperative torque control.

I. OVERVIEW

Vigorous development of new energy vehicles to accelerate clean energy transformation has become an important way for sustainable development of the global automobile industry [1]. With the increasing complexity in operating conditions of new energy vehicles and the constant innovation of vehicle drive technologies, the centralized drive system configuration of electric buses has gradually shifted from a single motor-matched variable speed drive system to a dual-motor variable speed drive system [2]. Such configuration is characterized by two synergistic driving motors, variable speed device and working modes adaptable to various working conditions [3], [4].

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Relevant studies have put forward a variety of dual-motor variable speed drive system configuration schemes. For instance, Xu et al. [5] proposed a dual-motor drive system configuration for heavy commercial vehicles, which consists of five single motors and four dual-motor operating modes. Wu et al. [6] introduced a dual-motor power system configuration based on planetary gear transmission, whose overall efficiency was superior to that of single-motor variable speed drive system. Tutelea et al. [7] proposed a dual rotor single stator brushless motor system to achieve higher system efficiency. Hu et al. [8] proposed a dual-motor 4-speed drive system, which demonstrated better performance in fuel economy and power owe to a variety of operating modes. Zhang et al. [3] developed a dual-motor planetary rotational speed coupling drive system, which operates in either single or dual-motor mode through clutch action. In single working mode, this system faces certain limitations in performance optimization. Despite of current rich research achievements, most studies mainly focus on system configuration design, parameter matching optimization, energy management and upper control strategy optimization, whereas few researches have been conducted on the process control of coupled variable speed devices. In particular, the system will be more difficult to control under conditions with high system complicity and high coupling-degree. The collaborative control between motor and variable speed device directly affects the system performance.

In the study on gear shift control of dual-motor variable speed drive systems, vehicle impact reduction is a key index in gear shift quality improvement. Nguyen et al. [9] developed a coordinated control strategy to optimize gear shift rules and the corresponding mode switching graphs. Simulation results showed that the coordinated control strategy prevented the torque interruption during gear shift process. Li et al. [10] put forward a collaborative control for main drive motor speed regulation and gear shift motor displacement so as to improve the gear shift quality of integrated motor transmission system. Tian et al. [11] studied the optimal collaborative gear shift control of electric drive system of dual-motor coupled transmission for electric vehicles, in order to improve the gear shift quality in torque stage and inertia stage during the gear shift process. Roozegar et al. [12] developed a gear shift control strategy for the multi-gear variable speed drive system of electric vehicles. Shu et al. [13] proposed a collaborative gear shift control strategy and designed a robust multivariable hierarchical controller to decouple the model. Kim et al. [14] developed an optimal gear shift control strategy via solving polynomials by taking gear shift impact and the change rate of gear shift impact as performance indexes. Sebastian [15] investigated the precise position control of a gear shift actuator, developed fuzzy control method for the gear shift actuator to achieve fast and steady control. Tian et al. [16] developed a double-layer gear shift control strategy to enhance the gear shift performance of the DCT power system. The upper controller determined the optimal torque track of the clutch and engine, while the lower controller controlled the optimal track determined by each actuator. Yu et al. [17] examined clutchless AMT system and inhibited external unknown disturbance by robust control method so as to improve control accuracy of the gear shift actuator and shorten the synchronization time. In view of the synchronization process of clutchless AMT systems, Zhu et al. [18] developed an optimal speed control method based on robust control, and conducted verification through simulations.

Researchers have made extensive studies on the gear shift strategy of electric bus, with fruitful research results achieved. These studies mostly focus on gear shift comfort, fast switching or drive assembly energy loss. However multi-parameter system parameters are strongly coupled, so the collaborative control between the motor and its gear shift device directly restricts the improvement of the gear shift performance.

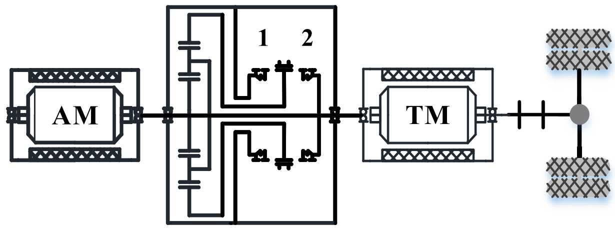

FIGURE 1. Principle configuration of dual-motor AMT system.

In gear shift process, torque unloading and torque recovery exert a relatively major influence on the vehicle impact. Considering control difficulties brought by the high coupling degree of the system, this paper conducts the following studies to improve the gear shift quality:

1) With reference to the currently studied dual-motor coupling drive system configuration, simplified model of the power system, gear shift process and its control diagram are described.

2) The main gear shift performance indexes are analyzed, with the focus on the influence of unloading and loading process on the vehicle impact.

3) In view of the gear shift process of the coupled power system, the multiple collaborative torque control strategy and control process were suggested, and simulation platform was built for the electric bus power system to simulate and analyze the control method.

4) The transmission controller unit was designed to verify the control method, and test prototype was developed to verify the effect on the vehicle.

II. SYSTEM CONFIGURATION AND DYNAMICS MODEL

A. POWER SYSTEM CONFIGURATION

This paper studies a dual-motor coupled drive system (hereinafter referred to as dual-motor AMT system), with its principle configuration shown in figure 1. The drive system optimizes the traditional coupling device into a 2-gear single-planetary transmission. The transmission is matched through auxiliary motor (AM) and coaxially combined with the traction motor (TM) directly connected to the drive axle.

The power system has a variety of working modes to adapt to different operating conditions. Under medium and low speed and low torque, the vehicle is driven separately by TM motor. When the vehicle operates at high speed, with rapid acceleration or on upward slope, AM motor and variable speed system can be involved in the running to meet the vehicle operation needs for high power, large torque and continuous climbing. Meanwhile, the TM motor is directly connected to the drive axle, making it possible for the vehicle to maintain continuous power during operation, which addresses the problem of power interruption in the gear shift process of the traditional single motor-based gearbox configuration.

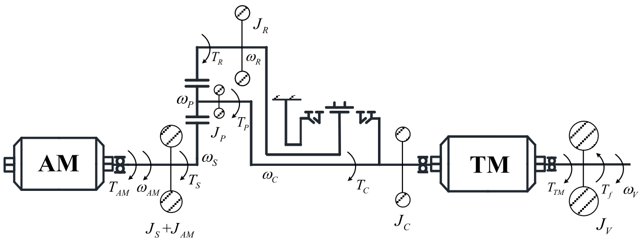

FIGURE 2. Simplified dynamic model of the power system.

B. SIMPLIFIED MODEL OF THE POWER SYSTEM

In the gear ship process, the key research object is the power system, and the vehicle maintains its dynamic characteristics as the controlled object. The following hypotheses are proposed:

(1) Ignore the vibration and damping of the whole transmission chain from the output shaft of the power system to the tire.

(2) Ignore the mechanical clearance on the entire power chain, regard each component as a rigid body.

(3) Except for the power system, the transmission efficiency of other components is presumed to be 100%.

By combining Figure 1, a simplified dynamic model was established for the power system, as shown in figure 2.

In neutral gear state, the planetary transmission is in free state, belonging to the two-degree-of-freedom differential gear train. The revolving speed of AM motor (solar wheel S), gear ring R and TM motor (planetary frame C) displays the following relation:

$$\omega_C = \frac{1}{k_p+1}\cdot\omega_S + \frac{k_p}{k_p+1}\cdot\omega_R \tag{1}$$

where, ω_C, ω_S, ω_R are respectively the absolute angular velocity of the planetary frame, solar wheel and gear ring, k_p is the characteristic parameter of the planetary gear train, and k_p+1 is the first-gear speed ratio of the transmission. In this paper, the characteristic parameter k_p of the planetary transmission is 1.98, so the first-gear speed ratio of the planetary transmission is 2.98.

When the coupling sleeve of synchronizer engages with the coupling gear ring of transmission shell B, the gear ring R and the transmission shell are integrated as one, with speed at 0, and the planetary transmission in gear 1. The speed of AM motor (solar wheel S), gear ring R and TM motor (planetary frame C) exhibits the following relation:

$$\begin{cases} \omega_R = 0 \\ \omega_C = \dfrac{1}{k_p+1}\cdot\omega_S \end{cases} \tag{2}$$

When the coupling sleeve of synchronizer engages with the coupling gear ring of planetary frame C, the gear ring R and planetary frame C are integrated as one, with speed equalized and the planetary transmission in gear 2. The speed of AM motor (solar wheel S), gear ring R and TM motor (planetary frame C) presents the following relation:

$$\begin{cases} \omega_R = \omega_C \\ \omega_C = \dfrac{1}{k_p+1}\cdot\omega_S + \dfrac{k_p}{k_p+1}\cdot\omega_R \end{cases} \tag{3}$$

Further, it can be known that $\omega_R = \omega_C = \omega_S$ and the planetary gearbox has a second-gear speed ratio of 1.

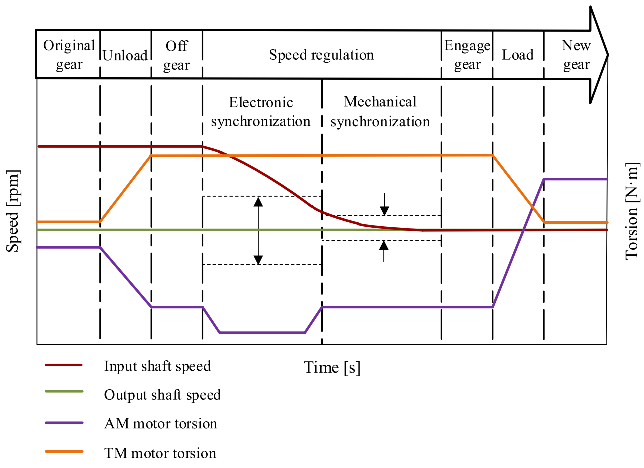

FIGURE 3. Ideal gear shift process of the dual-motor AMT system.

III. GEAR SHIFT INDEX AND PROCESS ANALYSIS

In gear shift, the dual-motor AMT system mainly goes through the stages of unloading stage, off-gear stage, synchronization stage (electronic synchronization, mechanical synchronization), gear-engaging stage and loading stage, as shown in the figure 3.

The gear shift process requires collaborative control of AM motor, TM motor and planetary gearbox. The introduction of TM motor increases the number of controlled objects, which improves the optimization space for gear shift quality to a certain extent, but also adds to control difficulty. In this paper, the major control objects, control objectives and control variables in each gear shift stage of the dual-motor AMT system are summarized in the figure 4.

In the figure, t_i is the gear shift time corresponding to each gear shift stage. j is the impact degree corresponding to each gear shift stage. W is the slipping work in the mechanical synchronization stage. k_{Ploss} is the power loss rate corresponding to each gear shift stage. T_{AM} is AM motor torque. T_{TM} is TM motor torque. $\dot{T}_{\mathrm{AM}}$ is the torque change rate of AM motor. $\dot{T}_{\mathrm{TM}}$ is the torque change rate of TM motor. F is the shifting force of the actuator. $\dot{F}$ is the shifting force change rate of the actuator.

A. GEAR SHIFT TIME

For the single-motor parallel-shaft AMT drive system configuration, complete power interruption is possible in the gear shift process [19]. Nevertheless, the dual-motor AMT system proposed in this study can control the continuous output compensation torque of TM motor during gear shift

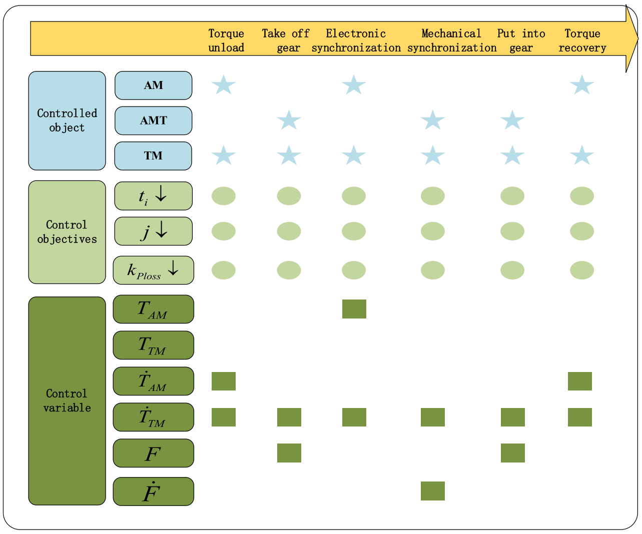

FIGURE 4. **Control diagram of the dual-motor AMT system during gear shift.**

while avoiding complete power interruption. The time of torque unloading stage and torque recovery stage should not be included the time consumption of gear shift time, so the total gear shift time is defined as follows:

$$t_{\sum} = k_{\mathrm{TR}}\left(k_{\mathrm{Ploss}}\right) \cdot t_{\mathrm{TR}} + t_{\mathrm{DE}} + t_{\mathrm{syne}} + t_{\mathrm{synm}} + t_{\mathrm{EG}} + k_{\mathrm{TB}}\left(k_{\mathrm{Ploss}}\right) \cdot t_{\mathrm{TB}} \quad (4)$$

where, $t_{\mathrm{TR}}, t_{\mathrm{DE}}, t_{\mathrm{syne}}, t_{\mathrm{synm}}, t_{\mathrm{EG}}, t_{\mathrm{TB}}, t_{\mathrm{DE}}$ represent the torque unloading time, off-gear time, electronic synchronization time, mechanical synchronization time, gear-engaging time and torque recovery time during the gear shift, respectively. $k_{\mathrm{TR}}\left(k_{\mathrm{Ploss}}\right)$ means the proportion of power loss time at the torque unloading stage to the total time duration of entire unloading process. $k_{\mathrm{TB}}\left(k_{\mathrm{Ploss}}\right)$ is the proportion of power loss time at the torque recovery stage to the duration of the unloading process. These two variables are functions of the power loss rate k_{Ploss} of the vehicle.

B. POWER LOSS RATE

The power loss rate of the vehicle is defined as follows:

$$\begin{cases} k_{\mathrm{Ploss}} = \dfrac{T_{\mathrm{Req}} - T_{\mathrm{out}}^{\mathrm{Shift}}}{T_{\mathrm{Req}}} \times 100\%, \quad k_{\mathrm{Ploss}} \in [0, 100\%] \\ T_{\mathrm{out}}^{\mathrm{Shift}} = T_{\mathrm{AM}} \cdot k_{\mathrm{Shift}} + T_{\mathrm{TM}} \\ k_{\mathrm{Shift}} = \begin{cases} 0, & \mathrm{Shift} = 1 \\ i_{g,j}, & \mathrm{Shift} = 0 \end{cases}, \quad j = 1, 2 \end{cases} \quad (5)$$

where, T_{Req} is the total demand torque of the system. $T_{\mathrm{out}}^{\mathrm{Shift}}$ is the total output torque of the power system during the gear shift. k_{Shift} is the speed ratio corresponding to the current gear. $i_{g,j}$ is the speed ratio of the transmission under different gears. Shift indicates gear shift status. When the value is 1, it means the system is amid gear shift, with the transmission generally engaging in neutral gear or the AM motor outputting zero torque, that is, the system torque output is derived from TM motor alone. When the value is 0, it suggests that the system is amid non-gear shift stage, generally in the torque unloading or torque recovery stage, and the system torque output is derived from both AM motor and TM motor.

The power loss rate can characterize the system power performance during gear shift process. A power loss rate of 100% means the system is in complete power interruption. A zero power loss rate suggests the system has no power loss. For the dual-motor AMT system studied proposed in this work, the system power loss rate k_{Ploss} during gear shift process fluctuates between $(0, \kappa]$, without complete power interruption. The theoretical minimum value of κ is defined as follows:

$$\kappa_{\min} = \frac{T_{\mathrm{Req}} - T_{\mathrm{TMLmt}}}{T_{\mathrm{Req}}} \times 100\% \quad (6)$$

where, T_{TMLmt} is the maximum torque of the TM motor, which is constrained by the external features of the TM motor. In this work, it can be judged that the power loss rate depends on the capacity of the TM motor.

C. VEHICLE IMPACT DEGREE

Vehicle impact is the main factor affecting the comfort. During the gear shift process, torque unloading and torque recovery stage greatly affect the vehicle impact. Moreover, when the capacity of TM motor fails to meet the system torque requirement, the power loss of vehicle will occur. Considering the complementary torque of AM motor and TM motor in the dual-motor AMT system and the absence of power interruption in the system, this paper focuses on the impact of unloading and loading change stage on the vehicle impact.

The entire electric bus is powered by the driving system, which provides partial electric braking force during braking. According to the vehicle driving equation [4], the vehicle dynamics equation of the dual-motor AMT system can be defined as follows:

$$\begin{cases} \left(F_a + F_f + F_w + F_i\right) \cdot r_T = T_{\mathrm{wReq}} \\ T_{\mathrm{wReq}} = T_T + T_{\mathrm{brk}} \\ T_T = \left(T_{\mathrm{TM}} + T_{\mathrm{AM}} \cdot i_{g,j} \cdot \eta_{g,j}\right) \cdot i_0 \cdot \eta_T, \quad j = 1, 2 \\ T_{\mathrm{brk}} = \begin{cases} T_{\mathrm{wReq}} - \max\left(T_T, T_{\mathrm{eMax}}\right), & T_{\mathrm{wReq}} < 0 \\ 0, & T_{\mathrm{wReq}} >= 0 \end{cases} \end{cases} \quad (7)$$

where, F_a is acceleration resistance. F_f is rolling resistance. F_w is wind resistance. F_i is ramp resistance. r_T is wheel rolling radius. T_{wReq} is wheelside drive torque required by the vehicle. T_T is the output torque of the drive system. T_{brk} is the mechanical braking torque of the braking system. T_{TM} is the output torque of TM motor. T_{AM} is the output torque of AM motor. $i_{g,j}$ is the speed ratio of the transmission at different gears. $\eta_{g,j}$ is the efficiency of the transmission at different gears. i_0 is the speed ratio of the main reducer. η_T is the comprehensive efficiency of the transmission system. i_0 is the final drive ratio. T_{eMax} is the maximum allowable electric braking torque in the regenerative energy feedback strategy. The vehicle acceleration resistance is defined as follows:

$$F_a = \delta m\alpha = \delta m\frac{du}{dt} \tag{8}$$

where, δ is the rotational mass conversion coefficient of the vehicle rotating parts starting from the transmission output end. $_m$ is the vehicle mass. $_\alpha$ is the vehicle acceleration. u is the vehicle running speed.

In unloading and loading stages, vehicle impact j_V is mainly the derivative of vehicle acceleration, which is defined as follows:

$$j_V = \frac{d\alpha}{dt} \tag{9}$$

Assuming a vehicle drives on a fair road surface without wheel slippage, the vehicle impact degree can be defined as:

$$j_V = \frac{d\left[\frac{(T_{\mathrm{TM}}+T_{\mathrm{AM}}\cdot i_{g,j}\cdot \eta_{g,j})\cdot i_0\cdot \eta_T+T_{\mathrm{brk}}-(F_f+F_w+F_i)\cdot r_T}{\delta m r_T}\right]}{dt} \tag{10}$$

D. TORQUE UNLOADING PROCESS

Unloading process is the first stage of gear shift process, during which, there are two kinds of torque changes for the AM motor:

(1) The vehicle is in electric braking state before the gear shift. When the AM motor has a negative torque, the torque of AM motor will approach zero with a positive slope during the unloading process. With no requirement for power interruption, the torque of TM motor decreases with a negative slope, which makes up for the vehicle power loss induced by unloading of AM motor.

(2) The vehicle is in the electric drive state before the gear shift process. When AM motor has a positive torque, the torque of AM motor will approach zero with a negative slope during the unloading process. With no requirement for power interruption, the torque of TM motor increases with a positive slope, which makes up for the torque in unloading of AM motor.

It is presumed that the vehicle load torque and mechanical braking torque are constant during the unloading process. When analyzing the vehicle impact, the mechanical efficiency loss of the transmission is not considered i.e., $\eta_{g,j} = 1$. Then, the vehicle impact during unloading process can be defined according to formula (10):

$$j_{\mathrm{VTR}} = \left(\dot{T}_{\mathrm{TM}} + \dot{T}_{\mathrm{AM}}\cdot i_{g,j}\right)\cdot \frac{i_0\eta_T}{\delta m r_T} \tag{11}$$

where, $\dot{T}_{\mathrm{TM}}$, $\dot{T}_{\mathrm{AM}}$ are the torque change slope of TM motor and AM motor, respectively.

The above equation shows the correlation between vehicle impact and torque change slope of AM motor and TM motor, as well as the speed ratio of planetary transmission. In theory, zero-impact unloading is possible by reasonably adjust the sizes of $\dot{T}_{\mathrm{TM}}$ and $\dot{T}_{\mathrm{AM}}$. During unloading process, the torque loss of AM motor can be supplemented by TM motor, with the mathematical expression defined as follows:

$$\begin{cases} \Delta T_{\mathrm{AM}} = T_{\mathrm{AM}}^{\mathrm{t_{AM}^x}} - T_{\mathrm{AM}}^{\mathrm{t_0}} \\ T_{\mathrm{AM}}^{\mathrm{t_{AM}^x}} \to 0 \\ T_{\mathrm{TM}}^{\mathrm{t_{TM}^x}} = T_{\mathrm{TM}}^{\mathrm{t_0}} + \Delta T_{\mathrm{AM}}\cdot i_{g,j} \\ \Delta T_{\mathrm{TM}} = \begin{cases} \Delta T_{\mathrm{AM}}\cdot i_{g,j}, & \left|T_{\mathrm{TM}}^{\mathrm{t_{TM}^x}}\right| \le |T_{\mathrm{TMLmt}}| \\ T_{\mathrm{TMLmt}} - T_{\mathrm{TM}}^{\mathrm{t_0}}, & \left|T_{\mathrm{TM}}^{\mathrm{t_{TM}^x}}\right| > |T_{\mathrm{TMLmt}}| \end{cases} \end{cases} \tag{12}$$

where, ΔT_{AM} is the torque to be unloaded by AM motor. $T_{\mathrm{AM}}^{\mathrm{t_{AM}^x}}$ is the torque of AM motor at the beginning of unloading. $T_{\mathrm{AM}}^{\mathrm{t_0}}$ is the torque of AM motor at the end of AM unloading. $T_{\mathrm{TM}}^{\mathrm{t_0}}$ is the torque of TM motor at the beginning of unloading. ΔT_{AM} is the torque to be compensated for TM motor. $T_{\mathrm{TM}}^{\mathrm{t_{TM}^x}}$ is the torque at the end of unloading, which directly influences the vehicle impact. T_{TMLmt} is external feature limit value of torque when providing torque compensation for TM motor:

(1) When the external features of the motor meet the total torque demand of the system, that is, the TM motor can compensate the torque loss of the AM motor, non-impact unloading is possible by reasonably adjust the torque change rates of the AM motor and the TM motor.

(2) If the external features of the TM motor do not meet the torque demand of the system, that is, the TM motor cannot fully compensate the torque loss of the AM motor. As a consequence unloading of the AM motor will not be finished when the torque of TM motor is loaded to the maximum, thus causing vehicle impact. In view of this problem, the torque of AM motor and TM motor can be subjected to segmented optimization control. Before the torque of TM motor reaches the peak, the AM motor and TM motor are respectively subjected to unloading and loading control at a relatively big slope. If the AM motor has not finished unloading after the torque of TM motor reaches the peak, the torque unloading slope of the AM motor can be adjusted appropriately to reduce the vehicle impact during unloading process. This shows that the dual-motor AMT system can optimize the impact during unloading process by adjusting the torque change rate of the two motors.

From Formula (5), the mathematical expression of the power loss rate in the unloading process can be further defined as follows:

$$\begin{cases} k_{\mathrm{Ploss}}^{\mathrm{TR}} = \dfrac{T_{\mathrm{Req}} - \int_{t_0}^{t_{\mathrm{AM}}^x} \dot{T}_{\mathrm{AM}}\cdot i_{g,j}dt - \int_{t_0}^{t_{\mathrm{TM}}^x} \dot{T}_{\mathrm{TM}}dt}{T_{\mathrm{Req}}} \times 100\% \\ \left|T_{\mathrm{TM}}^{\mathrm{t_{TM}^x}}\right| \le T_{\mathrm{TMLmt}}, \quad \left|T_{\mathrm{TM}}^{\mathrm{t_{TM}^x}}\right| \le T_{\mathrm{Req}} \end{cases} \tag{13}$$

When the torque of TM motor can meet the system requirement, there is no power loss during unloading process and no vehicle impact is caused. When the torque of TM motor cannot meet the system requirement, power loss is inevitable during the unloading process.

E. TORQUE RECOVERY PROCESS

Loading stage is basically the inverse process of unloading stage. By analyzing the vehicle impact during torque unloading process, the expression of vehicle impact at torque recovery stage can be obtained:

$$j_V = \left(\dot{T}_{\mathrm{TM}} + \dot{T}_{\mathrm{AM}} \cdot i_{g,j}\right) \cdot \frac{i_0 \eta_T}{\delta m r_T} \tag{14}$$

Similarly, by appropriately setting the torque change slope of AM motor and TM motor, it is possible to minimize the vehicle impact degree during the loading process. In the torque recovery process, vehicle impact can be analyzed from 2 circumstances:

(1) When system composed of AM motor and AMT has the same torque change rate as TM motor, it is possible to guarantee zero vehicle impact during the torque recovery by reasonably setting the torque change rate of AM motor and TM motor.

(2) When system composed of AM motor and AMT has a torque change rate different from TM motor, the vehicle has impact during the torque recovery process. The torque recovery process can be controlled in segments. Before the torque of AM motor and TM motor reaches the target value, the torque can be added or reduced according to the relatively bigger torque change slope. After torque of one motor reaches the target value, the other motor can converge to the target torque according to the relatively smaller torque change slope.

According to Formula (5), the mathematical expression of power loss rate in torque recovery process can be further obtained as follows:

$$\begin{cases} k_{\mathrm{Ploss}}^{\mathrm{TB}} = \dfrac{T_{Req} - \int_{t_0}^{t_{\mathrm{AM}}^x} \dot{T}_{\mathrm{AM}} \cdot i_{g,j} dt - \int_{t_0}^{t_{\mathrm{TM}}^x} \dot{T}_{\mathrm{TM}} dt}{T_{\mathrm{Req}}} \times 100\% \\ T_{\mathrm{AM}}^{t_{\mathrm{AM}}^x} = T_{\mathrm{AM}}^u, \quad T_{\mathrm{TM}}^{t_{\mathrm{TM}}^x} = T_{\mathrm{TM}}^u \end{cases} \tag{15}$$

where, $T_{\mathrm{AM}}^u, T_{\mathrm{TM}}^u$ are the target torque command of AM motor and TM motor given by the torque distribution strategy after gear shift process, respectively.

IV. COLLABORATIVE CONTROL OF TORQUE

A. MULTIPLE COLLABORATIVE TORQUE CONTROL

To solve the collaborative control problem of coupled power system, the actuator demanding collaborative control in the transmission control unit is taken as the control "element" to analyze the gear shift process of the dual-motor AMT system proposed in this study. According to the concept of Hadamard product matrix, the following multiple collaborative control method is proposed, including defined elements of controlled object, control objective and control variable.

The mathematical expression of the controlled object set under multiple collaborative control is defined as:

$$\begin{cases} \mathbf{x}_k = \begin{bmatrix} \Theta_{k1} & \Theta_{k2} & \Theta_{k3} \end{bmatrix}^{\mathbf{T}} \circ \mathbf{\Lambda x} \\ \mathbf{\Lambda x} = \begin{bmatrix} \Lambda\mathrm{AM} & \Lambda\mathrm{AMT} & \Lambda\mathrm{TM} \end{bmatrix}^{\mathbf{T}} \\ \Theta_{ki} \in \mathbb{N}^+ \cap [0,1], i = 1,2,3 \\ k \in \mathbb{N}^+ \cap [1,6] \end{cases} \tag{16}$$

where, $\mathbf{x}_k$ represents the controlled object combination in the present control process. $\mathbf{\Lambda x}$ represents the controlled object set of the system, with the controlled object including. AM motor ΛAM, transmission ΛAMT, TM motor ΛTM. Θ_{ki} represents the control command of different controlled objects, 0 means absence of object control, 1 indicates presence of object control. k represents different control processes, and the system has 6 gear shift processes, including torque unloading process, off-gear process, electronic synchronization process, mechanical synchronization process, gear-engaging process and torque recovery process.

The mathematical expression of the control object set under multiple collaborative control is as follows:

$$\begin{cases} \mathbf{J}_k = \begin{bmatrix} \Omega_{k1} & \Omega_{k2} & \Omega_{k3} & \Omega_{k4} \end{bmatrix}^{\mathbf{T}} \circ \begin{bmatrix} \vartheta_k(t) & \vartheta_k(j) & \vartheta_k(k_{\mathrm{ploss}}) \end{bmatrix}^{\mathbf{T}} \\ J_k = \begin{bmatrix} Q_k[\vartheta_k(t)] & Q_k[\vartheta_k(j)] & Q_k[\vartheta_k(k_{\mathrm{ploss}})] \end{bmatrix} \cdot \mathbf{J}_k \\ \Omega_{ki} \in \mathbb{N}^+ \cap [0,1], \quad i = 1,2,3 \\ k \in \mathbb{N}^+ \cap [1,6] \end{cases} \tag{17}$$

where, $\mathbf{J}_k$ is control objective combination. ϑ_{ki} is the control objective after the normalization of the present control process. Ω_{ki} represents the control objective status in the current control process, 0 means the control objective is excluded, 1 suggests that the control objective is included. $Q_k(\vartheta_{ki})$ represents the weight coefficient corresponding to the current control process ϑ_{ki}. The multiple collaborative control aims to find the minimum J_k, namely, the minimum gear shift time $\min\{\vartheta_k(t)\}$, the minimum gear shift impact $\min\{\vartheta_k(j)\}$ and the minimum power loss rate $\min\{\vartheta_k(k_{\mathrm{Ploss}})\}$.

The mathematical expression of the control variable set under multiple collaborative control is as follows:

$$\begin{cases} \mathbf{u}_k = \begin{bmatrix} \mathrm{O}_{k1} & \mathrm{O}_{k2} & \mathrm{O}_{k3} & \mathrm{O}_{k4} & \mathrm{O}_{k5} & \mathrm{O}_{k6} \end{bmatrix}^{\mathbf{T}} \\ \qquad \circ \begin{bmatrix} T_{\mathrm{AM}} & T_{\mathrm{TM}} & \dot{T}_{\mathrm{AM}} & \dot{T}_{\mathrm{TM}} & F & \dot{F} \end{bmatrix}^{\mathbf{T}} \\ \mathrm{O}_{ki} \in \mathbb{N}^+ \cap [0,1], \quad i = 1,2,3,4,5,6 \\ k \in \mathbb{N}^+ \cap [1,6] \end{cases} \tag{18}$$

where, $\mathbf{u}_k$ is the control variable set of the multiple collaborative control process. O_{ki} represents the control state of the control variable in the current control process. 0 indicates the control variable is excluded, and 1 means the control variable is included.

According to the above description, each gear shift stage in the gear shift process involves several gear shift performance indexes with mutual coupling and mutual influence. On this

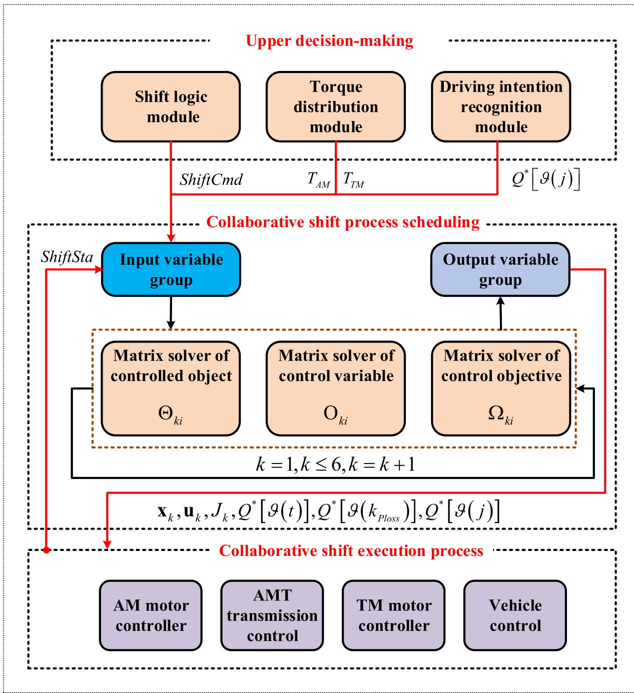

FIGURE 5. Procedure of multiple collaborative gear shift control method.

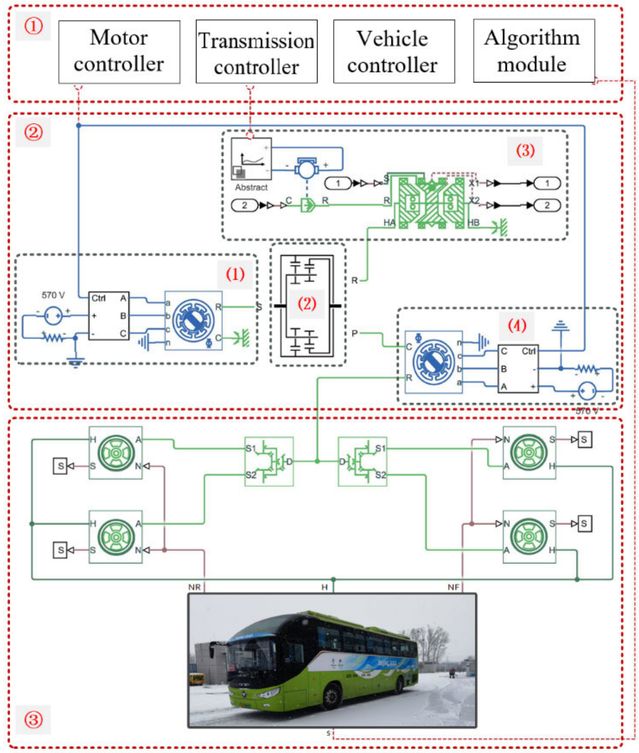

FIGURE 6. Vehicle simulation platform model.

basis, the procedure of the multiple collaborative gear shift control strategy is formulated as shown in figure 5.

In the procedure of multiple collaborative gear shift control method, ShiftCmd is the gear shift command output by the gear logic module of the upper decision-making layer, whose value determines whether it is possible to enter the gear shift process. ShiftSta represents the current gear shift state, whose value indicates whether the current gear shift process is completed or not. If the current gear shift process is over, the next gear shift process will begin. The matrix solver in the procedure is a set of mathematical matrices at each gear shift stage.

B. SIMULATION AND COMPARISON OF UNLOADING AND LOADING PROCESSES

Due to the introduction of the TM motor, a higher requirement is raised for the comfort in the gear shift process compared to the single-motor AMT system. In the gear shift process, vehicle impact is relatively high at the torque unloading and torque recovery stages. In this study, the process of torque unloading and torque recovery is simulated to comparatively analyze the gear shift performance under multiple collaborative gear shift control and non-collaborative gear-shift torque control.

For the proposed dual-motor AMT system configuration scheme, a 12-meter highway bus simulation platform is built based on MATLAB, as shown in figure 6.

In the figure, part ① is the system control module consisting of motor controller model, transmission controller model, vehicle controller model, algorithm module. part ② is the dual-motor coupling drive system module consisting AM motor model (1), planetary transmission model (2), gear shift actuator model (3), TM motor model (4). part ③ is the vehicle dynamics module.

TABLE 1. Parameters of a 12-meter highway bus and its power system.

Item	Parameter Description
Vehicle parameter	Full load mass: 18000 kg, unladen mass: 12500 kg, tire rolling radius: 0.506 m, windward area: 7.2 m², final drive ratio: 5.125, transmission efficiency: 0.92, wind resistance coefficient: 0.6, rolling resistance coefficient: 0.007
Motor parameter	TM Motor: Peak power 175 kW, peak torque 1150 Nm, peak speed 4500 rpm AM motor: Peak power 175 kW, peak torque 1150 Nm, peak speed 4500 rpm
Gearbox parameter	First-gear speed ratio 2.98, second-gear speed ratio 1

The basic parameters of the vehicle and power system are selected as shown in table 1.

Qualitative analysis was made on the process of shifting from the first gear to the second gear under 30% accelerator pedal opening when the vehicle was half-loaded. The simulation control parameters are defined in table 2.

During unloading, the torque of TM motor is constant under non-collaborative torque control method. With the unloading of AM motor, the system output torque decreases. In this circumstance, the power loss rate of the whole

TABLE 2. Control parameters of unloading and loading stages in non-collaborative and collaborative control processes.

Item	parameter					
	Unloading stage			Loading stage		
	AM motor torque change rate (Nm/s)	TM motor torque change rate (Nm/s)	Output shaft torque change rate (Nm/s)	AM motor torque change rate (Nm/s)	TM motor torque change rate (Nm/s)	Output shaft torque change rate (Nm/s)
Non-collaborative control	-1000	0	-2980	3000	0	3000
Multiple collaborative control	-1000	2980	0	3000	-3000	0

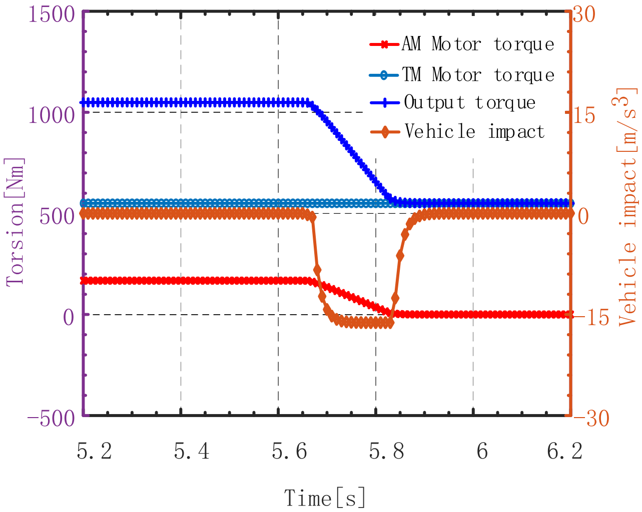

FIGURE 7. Simulation curve of unloading process under non-collaborative control.

vehicle keeps increasing, and the vehicle impact occurs. Under multiple collaborative control method, the torque of TM motor increases proportionally during unloading of AM motor, which supplements the torque loss of AM motor, and the system output torque basically has no loss. Although vehicle impact occurs, the impact degree is about 60% lower compared to the case of non-collaborative torque control.

During the torque recovery process, the torque of TM motor remains constant under non-collaborative torque control method. With the torque recovery of AM motor, the system output torque restores the target value, resulting in a great vehicle impact. Under multiple collaborative control method, as the torque of AM motor restores the target value, the torque of TM motor is unloaded to the target value, and the system output torque fluctuates insignificantly, with vehicle impact reduced by about 80% compared with the case under non-collaborative torque control.

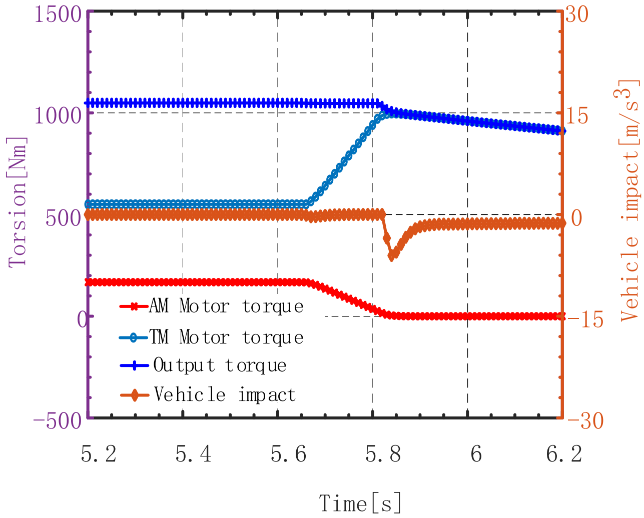

FIGURE 8. Simulation curve of unloading process under collaborative control.

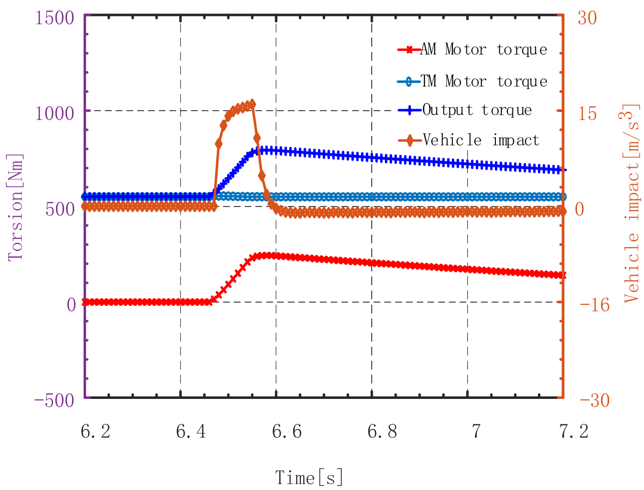

FIGURE 9. Simulation curve of loading process under non-collaborative control.

V. COMPARISON OF LOADING TEST

To verify the above control methods, the transmission controller hardware and the corresponding algorithm program were developed. The microcontroller unit adopts Infineon 32-bit TC234 platform, as shown in figure 11.

The main functions of TCU are as follows:

(1) System communication: TCU receives the information from the vehicle controller and AM/TM motor controller, calculates and processes the information for strategic decision and fault diagnosis. It also sends status information of the electric drive system to the vehicle controller as well as control command information to the motor controller.

(2) Information collection: TCU collects relevant digital and analog variables and processes the collected information for decision-making of gear shift strategy.

(3) Strategic decision covers dual-motor torque distribution decision, gear shift timing decision, collaborative gear shift process control, etc.

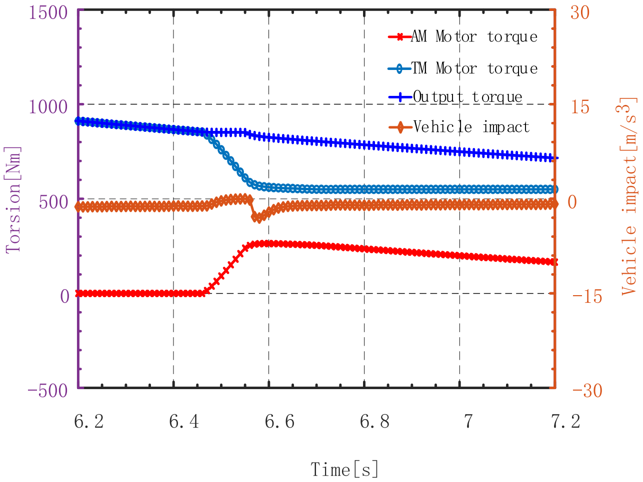

FIGURE 10. Simulation curve of loading process under collaborative control.

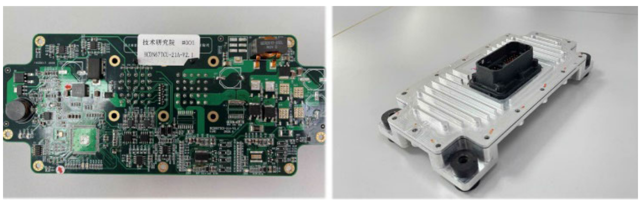

FIGURE 11. Transmission controller TCU hardware.

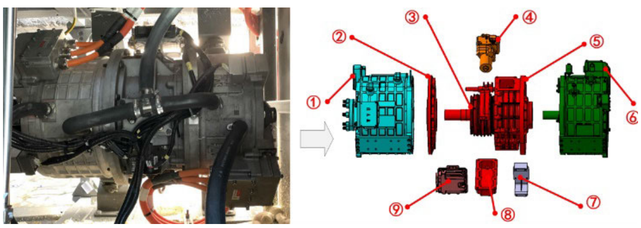

FIGURE 12. Prototype and composition of dual-motor AMT system.

(4) Fault handling: By monitoring the power system status, occurred faults are subjected to hierarchical processing.

Next, the following dual-motor coupling drive power system prototype was developed, which was then loaded with the test sample vehicle to verify the actual result of the above research.

In the figure, ① is the TM motor, ② is the rear housing of the planetary transmission, ③ is the gear pair of the planetary transmission, ④ is the gear shift actuator, ⑤ is the front housing of the planetary transmission, ⑥ is the AM motor, ⑦ is the TCU holder, ⑧ is the oil pan of the planetary transmission, and ⑨ is the controller of the planetary transmission (TCU).

The figure shows the sample vehicle on the rotary hub test bench, which verifies the influence of collaborative torque control of AM motor and TM motor on the vehicle impact degree during the unloading and loading stages when 1 gear is shifted to the 2nd gear. The transmission is in the 1st gear during unloading process and in the 2nd gear during

FIGURE 13. Rotary hub test of sample vehicle.

TABLE 3. Control parameters (motor torque change rate) in different control tests.

Item	parameter					
	Unloading process			loading process		
	AM motor torque change rate (Nm/s)	TM motor torque change rate (Nm/s)	Output shaft torque change rate (Nm/s)	AM motor torque change rate (Nm/s)	TM motor torque change rate (Nm/s)	Output shaft torque change rate (Nm/s)
Control test 1	-1600	0	-4768	1800	0	1800
Control test 2	-1400	4172	0	4172	-4172	0

loading process. The control test group was arranged, with basic parameters shown in table 3.

Control test 1 was carried out amid non-collaborative gear shift control process, and control test 2 was carried out in collaborative gear shift control process. Under non-collaborative gear shift control, the torque change rate of output shaft was not 0 in unloading and loading stages, so the unloading process had greater vehicle impact than the loading process. Under collaborative gear shift control, the torque change rate of the output shaft was 0 in unloading and loading stages. Theoretically speaking, vehicle impact will not occur during the unloading and loading stages, so gear shift without impact is possible.

The test data curve is shown in figure 14. In the figure, ① is the unloading process, ② is the loading process. In the unloading stage of the non-collaborative gear shift control process, the torque of AM motor was unloaded from about 500 ms to 0, while the torque of TM motor remained constant. The vehicle impact degree reached -25 m/s^3 at the maximum, with obvious vehicle impact. In the loading stage, the torque of AM motor was loaded from 450 ms until the target torque, and the vehicle impact reached about 8 m/s^3 at the maximum. Although the torque change slope of AM motor was smaller

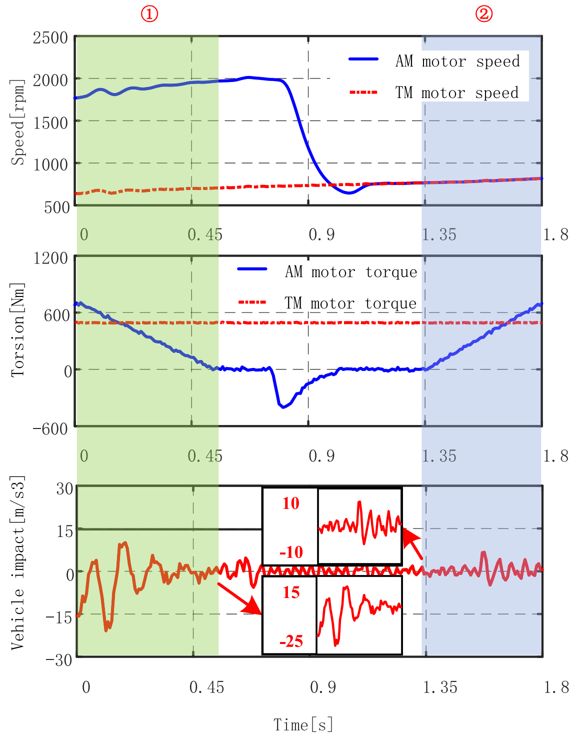

FIGURE 14. Data acquisition curve of non-collaborative gear shift control process.

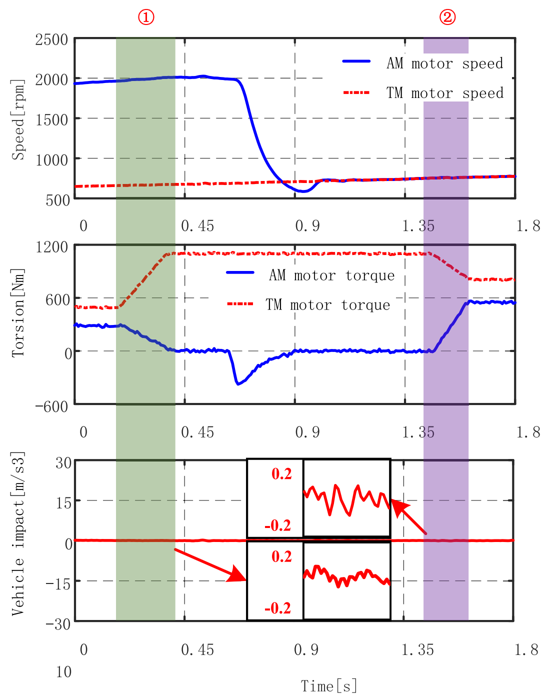

FIGURE 15. Data acquisition curve of collaborative gear shift control process.

in unloading stage than in loading stage, the speed ratio of transmission in unloading stage was about 3 times of that in loading stage, which is the primary reason for the significant difference in vehicle impact between unloading stage and loading stage.

In the unloading stage of the collaborative gear shift control process, the torque of the AM motor was unloaded from 170 ms to 0 with a slope of −1400 Nm/s, while the torque of the TM motor was supplemented from 170 ms with a slope of 4172 Nm/s. The torque change rate of the output shaft was 0 throughout the process, and the vehicle impact degree did not exceed 0.2 m/s^3 at the maximum, so gear shift without impact was basically achieved. In the loading stage, the torque of AM motor was loaded from 120 ms to the target torque with a slope of 4172 Nm/s, while the torque of TM motor was unloaded from 120 ms to the target torque with a slope of −4172 Nm/s. The torque change rate of output shaft was 0 throughout the process, and the vehicle impact degree did not exceed 0.2 m/s^3 at the maximum.

The test results show that compared with non-collaborative gear shift control, multiple collaborative gear shift control can significantly reduce the vehicle impact in the unloading and loading stages of the gear shift process, with comfort improved. Gear shift without impact is possible under some working conditions. In addition, by torque coordination between AM motor and TM motor in unloading and loading stage during gear shift, a larger torque change slope can be set to shorten the unloading and loading time, shortening the duration of the gear shift process.

VI. CONCLUSION

Based on the configuration of a dual-motor coupling drive system proposed in this study, the factors influencing the gear shift process and main performance indexes were analyzed, and performance indexes of the power loss rate were introduced considering non-power interruption characteristics of the system. Then, the author proposed a control method and strategy process featuring multiple collaborative gear-shift torque based on the gear shift process. The coupling control method and ideas are applicable to the cooperative control of different control objects, control objectives and control variables in each stage of gear shift process.

Focusing on the impact of torque unloading and loading process on the vehicle impact during gear shift, a simulation platform was built to simulate the control method of the dual-motor coupling power system of electric buses. The simulation results showed that, compared with non-collaborative torque control, the vehicle impact under multiple collaborative control was reduced by 60% in the torque unloading stage and reduced by 80% in the torque recovery stage. Next, the prototype was developed for transmission control unit and power system, and the vehicle was tested to verify the effectiveness of the proposed method. The test results showed that multiple collaborative gear-shift torque control could effectively reduce the vehicle impact during unloading and loading processes, with gear shifting time shortened and gear shifting comfort improved.

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