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Regenerative Braking Torque Estimation and Control Approaches for a Hybrid Electric Truck

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***Abstract*— Regenerative braking torque control problem is an important issue in a hybrid electric vehicle braking system. The braking performance has various influences on the vehicle driving performances such as fuel economy, braking efficiency and drivability. In this paper, a regenerative braking torque estimation approach is proposed which requires the wheel speed measurement only. Based on the estimated regenerative braking torque, a feedback braking torque control scheme is provided to achieve satisfactory control effect in a hybrid electric truck. Finally, simulation results are demonstrated to validate the proposed estimation and control approaches.**

I. INTRODUCTION

The HEVs (hybrid electric Vehicles) provide a promising transport solution to the problems of fossil fuel decreasing and environmental pollution. One of the most contributive systems in the HEVs is regenerative braking system (RBS) which can recover the kinetic energy into electrical energy. In addition to regenerative braking system, a friction brake system (FBS) is still preserved in most HEVs for the sake of braking efficiency. In braking circumstances, the key issues of the braking system control strategy are to determine the braking torque distribution between the two physically isolated systems (RBS and FBS) according to specific control objectives, and to improve the dynamical performance of the brake systems in transient mode. The control objectives are to meet the driver's needs of vehicle braking efficiency, to maximize the recovered energy and simultaneously to satisfy other constrains such as RBS energy storage device state of charge (SOC) and vehicle drivability by managing the output torques of the two braking systems.

In the HEVs, the braking torque forced on the wheel is composed of conventional friction braking torque generated by FBS and negative drive shaft torque introduced by RBS. In order to achieve high precision braking via online control the values of the applied braking torques should be acquirable in realtime. However, the drive shaft torque is difficult and uneconomic to be measured in production vehicles, an effective and feasible RBS braking torque estimation approach is needed. In recent years, several literatures have been proposed for the braking torque control in a HEV. As a model based control approach, the dynamic models of air-over-hydraulic brake system and permanent magnet synchronous motor (PMSM) in a heavy-duty hybrid truck are presented in [1]. The request brake torque for FBS is usually assigned in proportion to the overall braking torque, and the RBS is designated to provide compensation brake torque. In literature [2], an optimal controller is designed to generate the RBS torque command according to the brake pedal position and battery current. Considering the FBS dynamics, the RBS torque command is determined through a function of conventional brake system master cylinder pressure in [3]. Similarly, the wheel cylinder piston position is used to calculate the request regenerative braking torque in [1]. In [4], the integration control of RBS, automatic braking and anti-locked braking system is studied. As for the RBS torque estimation problem, the driven wheel angular speed and transmission output shaft speed can be used to estimate the axle shaft torque[5].

This paper is devoted to solve the RBS torque estimation and control problem in a hybrid truck. The sketch of the hybrid truck is first described. Based on the structure of the braking system the problems for the RBS braking torque estimation and control are formulated. To solve these problems the approaches are developed and the convergence of the RBS torque estimation strategy is analyzed. The simulation results show the effectiveness of the proposed RBS torque estimation and control approaches.

II. SYSTEM STRUCTURE AND PROBLEM FORMULATION

A. System Sketch

The vehicle in our application is a diesel-fueled hybrid electric truck which pairs a traditional diesel engine with an electric motor via a dry friction clutch and a fixed reduction ratio gear set. The overall torque generated by the engine and motor is transmitted to the driven wheels (rear-driven) through automated manual transmission (AMT), propeller shaft, final drive and drive shafts. The FBS is an air-over-hydraulic (AOH) braking system which employs drum brakes to apply friction brake torques on all of the wheels. An AOH braking system combines the use of compressed air and hydraulic pressure for brake operation. It's key components of an AOH brake system are air reserver, proportional valve, brake chamber and wheel oil cylinder[1]. The air reserver is filled with high pressure air and the proportional valve is controlled by a electronic control unit (ECU) to apply compressed air to the brake chamber. A brake chamber contains an air cylinder and a hydraulic cylinder in tandem [6][7]. Each cylinder is fitted with a piston and a common rod. The air piston is of greater diameter than the hydraulic

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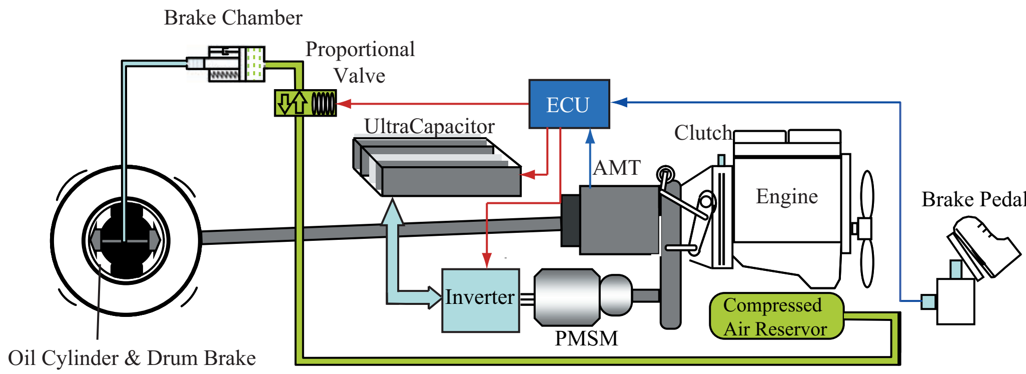

Fig. 1. Vehicle sketch

piston. This difference of the two pistons results in much greater hydraulic pressure than air pressure admitted to the air cylinder. In wheel oil cylinder, the pressed oil forces the piston to push the brake shoes against the drum. In this way, the friction braking torque will be produced to slow down the vehicle.

The RBS use a PMSM as its motor/generator. The sketch of the hybrid truck brake system which neglects the AOH braking system security part is depicted in Fig.1. The electric motor chosen for our application can generate a maximum of 55kw power. A Ultracapacitor is used as the electric energy storage device with maximum output voltage 350v and the inverter is a PWM inverter.

B. Problem Formulation for Torque Estimation

In this part, the regenerative braking torque estimation problem will be formulated. Torque signals are very useful in automotive control applications to achieve better drive feel and increased fuel economy[8]. The most intuitive idea of torque based control is developing torque sensors [9]. However, using torque sensors in production vehicles is difficult and uneconomic. To deal with this problem, several approaches which estimating shaft torques for torque control in traditional vehicles or electric vehicles are developed [5][10][11].

The equation to calculate shaft transmitted torque is described by following equation[12]:

$$T_s = k_s \int (\omega_i - \omega_o)dt + b_s(\omega_i - \omega_o) \qquad (1)$$

where T_s is the shaft torque, k_s the shaft torsional coefficient, b_s the shaft damping coefficient, ω_i the input side speed of shaft and ω_o the output side speed of shaft.

It can be seen that the torque transmitted by shaft can be calculated from Eq.1 with given initial torque value, exactly known k_s and b_s. In practical applications, however, the acquisition of these values is difficult and impractical. This open-loop estimation strategy will lead the estimation error introduced by the initial error all along without medication. The torque estimation problem we faced in this paper is the acquisition of regenerative braking torque value using existing measured signals and make an appropriate medication on the estimated torque.

C. Problem Formulation for Torque Control

In conventional braking system, there is only one braking system and its applied torque is controlled by the movements of brake pedal mechanically. There are no coordination problem in these kind of systems. The braking system in a HEV, on the other hand, consists not only a conventional braking system but also a recreative braking system. These two braking systems should be coordinated properly to achieve the control metrics mentioned before. In our target hybrid truck,the regenerative braking torque T_r acts on the driven wheels and the friction brake torque (front friction torque T_{ff}, rear friction torque T_{fr}) act on all of the wheels. The regenerative braking torque is affected by the motor output torque and the transmission line dynamics.

In this paper, we focus our attention on the braking torque distribution and control problem between the two different braking systems. The effects of the brake torque distribution between front and rear wheels on the vehicle dynamics are neglected. We generalize one equivalent wheel to represent all of the wheels. In this way, the braking torque control problems are simplified as the torque distribution of the overall braking torque and the feedback control of these two systems.

III. REGENERATIVE BRAKING TORQUE ESTIMATION AND CONTROL ALGORITHM

A. Torque Estimation

A feasible RBS torque estimation strategy will be proposed in this section to generate the reliable value of drive shaft torque. In automotive applications, angular speed sensors are reliable and inexpensive. They are widely equipped in production vehicles [5]. The torque estimation strategy proposed here consists two parts: calculated part and medication part. The calculated part use the shaft production model as shown in Eq.1. It utilize the AMT output shaft angular speed and driven wheels angular speed signals to generate the calculated regenerative braking torque. As mentioned before, the regenerative braking torque can be seen the negative torque transmitted by the drive shafts. We assume that the torques transmitted by the two drive shafts are equal. Similar to the equivalent of wheels, an equivalent drive shaft is used to represent the dynamics of two drive shafts. The calculated drive shaft torque T_{rw0} can be formulated as:

$$T_{rw0} = k_{rw0} \int (\omega_r/i_r - \omega_w)dt + b_{rw0}(\omega_r/i_r - \omega_w) \quad (2)$$

with k_{rw0} the equivalent drive shaft torsional coefficient, b_{rw0} the damping coefficient, i_r the main reducer reduction ratio, ω_r the reducer angular speed and ω_w the equivalent wheel angular speed.

The dynamics of the equivalent wheel is affected by several factors, it can be represented by:

$$J_w \dot{\omega}_w = T_{rw} - T_f - T_{load} \quad (3)$$

where J_w is the equivalent wheel inertia, T_{rw} the equivalent drive shaft torque, T_f the friction braking torque, T_{load} the resisting torque acted on the truck. Denote the vehicle mass as m and wheel radius as r, the equivalent wheel inertia can be approximate represented by:

$$J_w = (m + 150)r^2$$

$$Theoverallfrictiontorqueis : T_f = T_{ff} + T_{fr}$$

The resisting torque is formulated as:

$$T_{load} = (\mu m + C_d A/gv^2)r$$

where μ is the inertia moving resistant coefficient, C_d the drag coefficient, A the front projected area, g the acceleration of gravity and v the vehicle speed.

We can see from Eq.3 that the wheel angular speed response contains the drive shaft torque information. Thus, we construct the medication part of the estimated drive shaft torque based on a wheel angular speed observer. It medicates the estimated torque according to the error between the measured speed and the estimated speed of of the wheel.

The medicated drive shaft torque T_{rwm} is constructed using a conventional PI controller as follows:

$$T_{rwm} = k_{pe}\tilde{\omega}_w + k_{ie}\tilde{\theta}_w \quad (4)$$

where k_{pe} and k_{ie} are proportional gain and integral gain respectively, $\tilde{\omega}_w$ is the speed error.

$$\tilde{\omega}_w = \omega_w - \hat{\omega}_w$$

with $\hat{\omega}_w$ the estimated wheel speed.

The estimated driven wheel speed is represented by:

$$J_w \dot{\hat{\omega}}_w = \hat{T}_{rw} - T_f - T_{load} \quad (5)$$

It is worth to note that, the regenerative braking torque estimation strategy should be constructed based on some assumptions. First, the speed sensor is usually mounted on the output side of the AMT rather than on the reducer side. Thus, it is unreasonable for us to use the reducer speed in our estimation strategy. Fortunately, the propeller shaft which connect the AMT and the reducer owns large stiffness so we assume the propeller is stiff. In this way, the measured AMT output speed signals can be used to substitute the reducer speed. Secondly, the vehicle resisting torque and friction braking torque are considered measurable to simplify and highlight our key point.

The structure of the proposed torque estimator is illustrated in Fig.2.

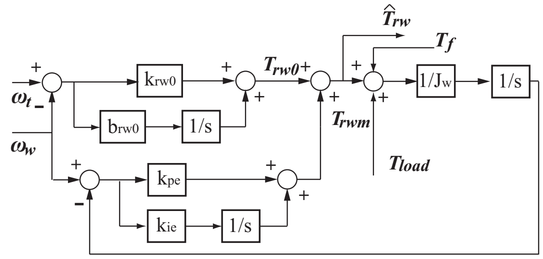

Fig. 2. Estimator Structure

The torque observer is then created and constructed as:

$$\begin{cases} J_w \dot{\hat{\omega}}_w = & \hat{T}_{rw} - T_f - T_{load} \\ \hat{T}_{rw} = & T_{rw0} + k_{pe}\tilde{\omega}_w + k_{ie}\tilde{\theta}_w \end{cases} \quad (6)$$

B. Torque Control

To solve the braking torque control issue, we propose a two-level braking torque control algorithm. The schematic of the strategy is depicted in Fig.3. The Torque Allocator serve as the high-level controller to produce the torque demands for the PMSM controller (T_{md}) and AOH controller (T_{AOHd}) respectively. On the other hand, the PMSM controller and AOH controller as a whole serve as the low-level controller to generate control signals for the actuators.The proposed coordinated controller generate control signals for the PMSM and AOH brake system respectively to satisfy the driver required brake torque. The inputs for the braking torque controller includes ultracapacitor SOC, AMT signals AMT_s(ω_t, ω_{to}, i_t, i_{td}, shift_s), estimated drive shaft torque $\hat{T}_{rw}$, applied AOH system brake torque T_{AOH}, brake pedal position p_k and PMSM signals PMSM_s(ω_m, I_d, I_q).

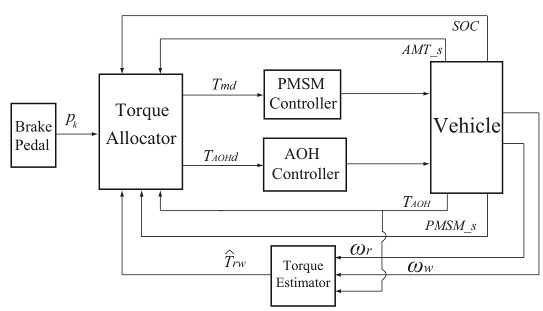

Fig. 3. Braking torque control schematic

1) Torque allocation: In order to maximize the energy recovered in braking situation, the brake torque allocation algorithm is designed to assign as much torque as possible to the regenerative brake system. In addition, the AMT is usually changed to low gear position when the vehicle speed is decelerated and the input shaft will be speed up to keep pace with the new gear ratio. A transmission controller which is not included here will determine the gear shifting sequence and selects the appropriate transmission gear based on the transmission output speed ,the position of acceleration pedal and braking pedal, the state of clutch and current gear position. For the purpose of quick and smooth gear shifting, the generator should withdrawal the resistance torque applied on the input shaft momentarily. In this case, the total brake torque acted on the vehicle will be decreased suddenly. It is difficult to achieve exactly brake torque tracking performance due to the dynamic characteristics of the AOH brake system. One of the feasible solutions is to reduce the synchronization time to minimize the effects of disturbance.

The driver request total brake torque T_B is simply generated from the brake pedal position p_k, represented by:

$$T_B = \eta p_k \tag{7}$$

where η the position-torque conversion coefficient.

There are two phases (Engaged and Gearshifting) for the brake torque allocation according to the sate of the AMT .

In **Engaged** phase, the brake torque allocation algorithm can be represented by:

$$T_{AOHd} = T_B - T_{mmax}(SOC, \omega_m) i_m i_t i_r \tag{8}$$

$$T_{md} = max[T_{md}(T_{rwd}, T_{rw}), \;\; T_{mmax}] \tag{9}$$

where T_{AOHd} is the torque allocated to AOH brake system, T_{mmax} the possible maximum generator brake torque, T_{md} the the torque allocated to generator, T_{rwd} the drive shaft request torque, T_{AOH} the AOH brake system actualized torque.

$$T_{rwd} = T_B - T_{AOH} \tag{10}$$

The torque allocated to generator should be adjusted according to drive shaft torque error and driveline dynamic model. A traditional PI controller is adopted to generate desired torque T_{md}.

$$T_{md}(T_{rwd}, T_{rw}) = k_p(T_{rwd} - T_{rw}) + k_i \int (T_{rwd} - T_{rw}) dt \tag{11}$$

with k_p denote the proportional coefficient, k_i the integral coefficient and T_{rw} drive shaft torque.

The possible maximum generator brake torque is affected by the ultracapacitor SOC and generator angular speed.

$$T_{mmax} = \begin{cases} T_{mmap} & \text{if SOC<1} \\ 0 & \text{else} \end{cases} \tag{12}$$

where T_{mmap} is generated from the motor angular speed, the speed-torque map for the generator is built up according to the machine data.

During the **Gearshifting** phase, the brake torque allocated to the AOH brake system is:

$$T_{AOHd} = T_B - T_{rw} \tag{13}$$

The request torque for the generator should be calculated from the input shaft speed error and applied synchronization torque. Supposing the synchronization should be accomplish within Δ s, the torque for the generator is formulated as:

$$T_{md} = min[J_t \frac{\omega_{td} - \omega_t}{\Delta} + T_s, \;\; -T_{mmax}] \tag{14}$$

$$\omega_{td} = \omega_{to} i_{td} \tag{15}$$

where ω_{td} is input shaft desired angular speed and i_{td} the desired AMT gear ratio, ω_{to} the output shaft angular speed, T_s the synchronization torque.

2) Braking Torque Tracking Control: In this part, the actuator controllers for PMSM and AOH system will be developed to track the desired torque values generated from the torque allocator respectively. We focus on the regenerative braking torque control, while the AOH system control signals for the friction brake torque tracking control are generated from a lookup table which is constructed from simulation results of the AOH system. The lookup table use the desired friction torque value as input signal and generate the corresponding control signals for the proportional valve.

The dynamics of PMSM torque production can be represented by the following equations [13]:

$$\begin{cases} \dot{I}_d = & \frac{v_d}{L_d} - \frac{R}{L_d} I_d + \frac{L_q}{L_d} p\omega_m I_q \\ \dot{I}_q = & \frac{v_q}{L_q} - \frac{R}{L_q} I_q - \frac{L_d}{L_q} p\omega_m I_d - \frac{\lambda p \omega_m}{L_q} \\ T_m = & \frac{3}{2} p[\lambda I_q + (L_d - L_q) I_d I_q] \end{cases} \tag{16}$$

where L, I and v are axis inductance, current and voltage respectively, the subscript q and d denote axis q and d; R is resistance of the stator windings; ω_m is angular velocity of the motor; λ is amplitude of the flux induced by the permanent magnets of the rotor in the stator phases; p is number of pole pairs.

It can be seen that the desired torque can be achieved by regulation of currents I_d and I_q in closed loops[14]. The voltages inputs are usually designed to guarantees the convergence of the two currents to their desired trajectory (I_{dd}, I_{qd}). The desired current are formulated as:

$$\begin{cases} I_{dd} = 0 \\ I_{qd} = \frac{2T_{md}}{3p\lambda} \end{cases} \tag{17}$$

Define the tracking errors as

$$e_d = I_d - I_{dd}, \quad e_q = I_q - I_{qd}$$

The regenerative braking torque tracking control inputs for the PMSM can be formulated as:

$$\begin{aligned} v_d &= -k_d L_d e_d + R I_d - L_q p \omega_m I_q \\ v_q &= -k_q L_q e_q + R I_q + L_d p \omega_m I_d + \lambda p \omega_m \end{aligned} \tag{18}$$

C. Convergence Analysis

In this part, the convergence of the proposed regenerative braking torque estimation and tracking control strategies will be analyzed.

1) Torque Estimation: In steady-state, the difference between the final drive speed and the driven wheel speed is zero, therefore, the drive shaft torque T_{rw} and it's estimation calculated part T_{rw0} are constants. Let$v = k_{ie}\tilde{\theta}_w$ and $\tilde{v} = T_{rw} - T_{rw0} - k_{ie}\tilde{\theta}_w$ and with $\tilde{\theta}_w = \int \tilde{\omega}_w dt$, the error system of equation 6 is given by:

$$\begin{cases} J_w\dot{\tilde{\omega}}_w = & \tilde{v} - k_{pe}\tilde{\omega}_w \\ \dot{\tilde{v}} = & -k_{ie}\tilde{\omega}_w \end{cases} \tag{19}$$

Proposition: Considering the error dynamics of Eq.19 with the observer Eq.6, for any positive constant k_{pe} and k_{ie}, the error system Eq.19 is Lyapunov stable, and $\tilde{\omega}_w \to 0$ as $t \to \infty$. Furthermore, $\hat{T}_{rw} \to T_{rw}$ as $t \to \infty$.

Proof: Choose the candidate of Lyapunov function as:

$$V = \frac{1}{2}J_w\tilde{\omega}_w^2 + \frac{1}{2}k_{ie}\tilde{v}^2 \tag{20}$$

Along the trajectory of error system we have:

$$\begin{aligned} \dot{V} = & \ \tilde{\omega}_w\tilde{v} - k_{pe}\tilde{\omega}_w^2 - \tilde{\omega}_w\tilde{v} \\ = & \ -k_{pe}\tilde{\omega}_w^2 \\ \leq & \ 0 \end{aligned} \tag{21}$$

This implies the Lyapunov stability of the error system. From Eq.21 and Eq.19, we can conclude:

$$\dot{V} = 0 \Leftrightarrow \tilde{\omega}_w = 0 \Rightarrow \tilde{v} = 0$$

According to Eq.6:

$$\tilde{v} = 0 \Rightarrow \hat{T}_{rw} = T_{rw0} + k_{ie}\tilde{\theta}_w \tag{22}$$

Due to $\tilde{v} = T_{rw} - T_{rw0} - k_{ie}\tilde{\theta}_w$, we have:

$$T_{rw} = T_{rw0} + k_{ie}\tilde{\theta}_w \tag{23}$$

Denote $\tilde{T}_{rw} = T_{rw} - \hat{T}_{rw}$, according to Eq.22 and 23 we have $\tilde{T}_{rw} = 0$.

Thus, $\hat{T}_{rw} \to T_{rw}$ as $t \to \infty$.

2) Torque Control: As for the PMSM torque tracking control convergence analysis. The error system derived from Eq.16 is formulated as:

$$\begin{cases} \dot{e}_d = \dfrac{v_d}{L_d} - \dfrac{R}{L_d}I_d + \dfrac{L_q}{L_d}p\omega_m I_q \\ \dot{e}_q = \dfrac{v_q}{L_q} - \dfrac{R}{L_q}I_q - \dfrac{L_d}{L_q}p\omega_m I_d - \dfrac{\lambda p\omega_m}{L_q} \end{cases} \tag{24}$$

Proposition. For the considered error system Eq.24, control inputs Eq.18 and controller gains satisfy $k_d > 0$ $k_p > 0$, the tracking errors will converge to zeros in finite time.

Proof. The following Lyapunov function candidate is chosen:

$$V = \frac{1}{2}e_d^2 + \frac{1}{2}e_q^2 \tag{25}$$

Along any trajectory of system Eq.24, we have

$$\begin{aligned} \dot{V} = & \ e_d[\frac{v_d}{L_d} - \frac{R}{L_d}I_d + \frac{L_q}{L_d}p\omega_m I_q] \\ & +e_q[\frac{v_q}{L_q} - \frac{R}{L_q}I_q - \frac{L_d}{L_q}p\omega_m I_d - \frac{\lambda p\omega_m}{L_q}] \\ = & \ -k_d e_d^2 - k_p e_p^2 \leq 0 \end{aligned} \tag{26}$$

which implies the stability of the system.

IV. SIMULATION RESULTS

In this section, the regenerative braking torque estimation and control strategy are tested in a complete powertrain model. In this model, the dynamics of AOH system is using the model described in literature [1]. The effectiveness of the proposed brake system coordinated control strategy is studied by simulation on the constructed system model. The simulation is performed in braking circumstances with gear shifting maneuver. The initial states are set as: 86km/h vehicle initial speed, 5th AMT gear position and ultracapacitor $SOC = 0$. The AMT gear shift is commanded when the AMT output shaft angular speed $\omega_r \leq 200rad/s$. The AMT will change from 5th gear position to 4th gear when the condition is satisfied.

The effectiveness of the proposed drive shaft torque estimation strategy is shown in Fig.4, the parameters used in the estimator with 40 percent error and the intitional torque is set as -350Nm. Fig.5 shows example of comparison between

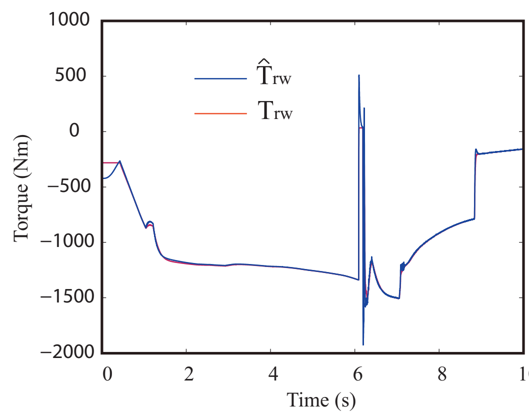

Fig. 4. Regenerative braking torque estimation

the equivalent AMT input shaft angular speed ω_{te} and AMT output shaft angular speed ω_{to}. We can see from this figure the synchronization period is limited to 0.3s. The brake torque distribution between the PMSM and AOH system is demonstrated in Fig.6, the tracking performance of the PMSM torque controller is illustrated in Fig.7,and the overall braking torque tracking performance is depicted in Fig.8. Under this braking torque, the vehicle speed response is shown in Fig.9.

V. CONCLUSION

This paper focus on the regenerative braking torque estimation and control problem for a hybrid electric truck. This paper starts with the description of the studied truck. Aiming at the problem introduced by this new kind of braking system in hybrid electric vehicle, the regenerative braking torque estimation and control problems are formulated. A useful regenerative braking torque estimation strategy is proposed to facilitate the brake torque tracking control. A two-level braking torque control strategy is then constructed. The PMSM torque tracking controller is constructed to produce control signals for PMSM based on the instructions from

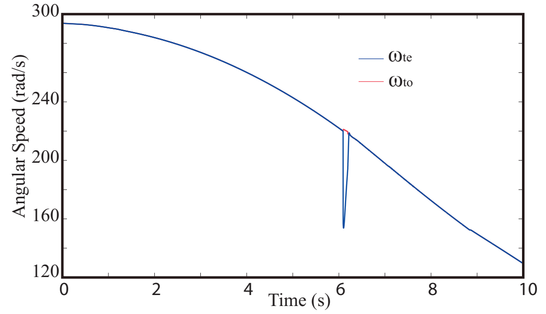

Fig. 5. AMT equivalent input shaft angular speed & output shaft angular speed

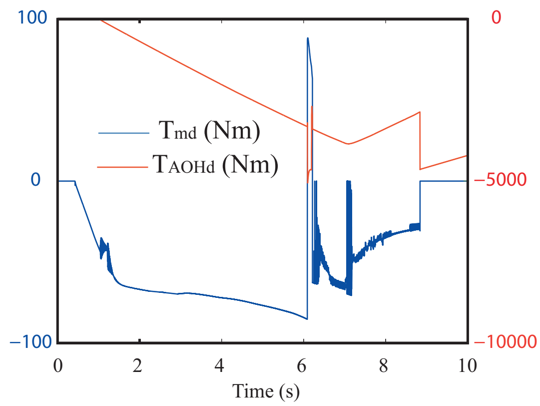

Fig. 6. Braking torque distribution

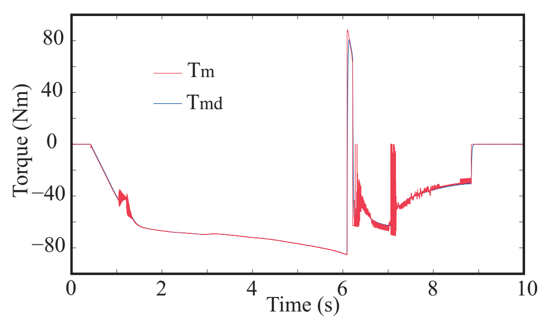

Fig. 7. PMSM output torque tracking

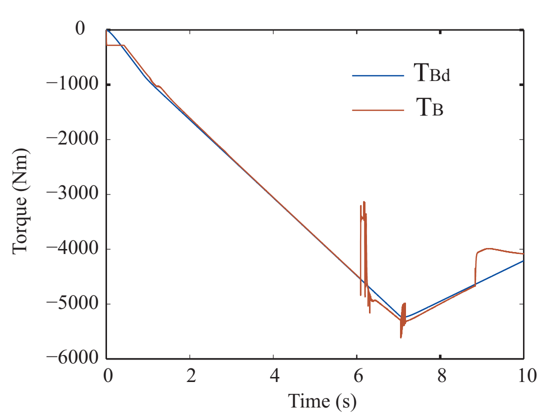

Fig. 8. Overall braking torque tracking

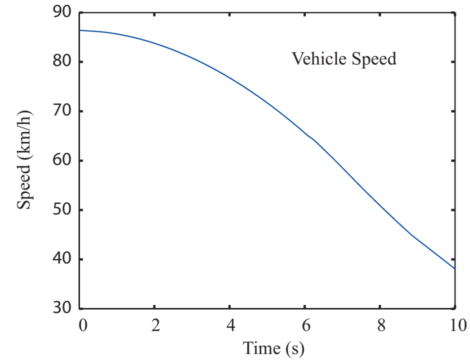

Fig. 9. Vehicle speed response

high-level controller. The stability of the proposed torque estimation and control strategies are tested and verified. The simulation results demonstrate the proposed method can make good compromise between the brake torque tracking performance, vehicle fuel economy and ride comfort.

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