Torque Control Based Speed Synchronization for Two-Speed Gear Electric Vehicle

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ABSTRACT Two-speed gears are used to secure a wide torque-speed range while improving the power efficiency. In electric vehicles (EVs), a simple sleeve mechanism is preferred for gear shifting. This is because, during the speed synchronization process, the propulsion motor can adjust its speed for the new gear ratio. For such a purpose, speed controllers are commonly used. Since the torque control is the normal control mode in the vehicle, the control mode should be switched temporarily to the speed control for gear shift and turned back to the torque control mode. It is very onerous, and requires some setting time for each mode change. In this work, a synchronization method is established in the torque control mode. Furthermore, a recursive least squares (RLS) algorithm is utilized to estimate the vehicle parameters, based on which the vehicle speed variation is predicted during the shifting process. This method works satisfactorily on uphill, downhill, rough or smooth asphalt roads without changing the control mode. Experiments were performed on a wheel dynamo system, along with extensive simulation studies.

INDEX TERMS Motor torque control, two-speed gear, electric vehicle, speed synchronization, PMSM control and shift shock.

NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| $T_m$ | Motor torque. |
| $T_{sh}$ | Transferred wheel shaft torque. |
| $\omega_m$ | Motor shaft angular speed. |
| $\omega_e$ | Electrical angular speed. |
| $\omega_v$ | Wheel angular speed. |
| $v_x$ | Vehicle velocity. |
| $\dot{\omega}_m$ | Motor angular acceleration. |
| $\dot{\omega}_e$ | Electrical angular acceleration. |
| $\theta_e$ | Electrical angle. |
| $J_m$ | Rotor inertia. |
| $B_m$ | Motor damping coefficient. |
| $\sigma$ | Inertia ratio. |
| $J_v$ | Equivalent inertia of vehicle mass. |
| $B_v$ | Equivalent friction coefficient of vehicle. |
| $T_L$ | Road load torque of vehicle. |
| $X$ | Gear state (1 or 2). |
| $J_{vX}$ | Increased inertia ($J_{v1}$ or $J_{v2}$). |
| $B_{vX}$ | Increased fiction coefficient ($B_{v1}$ or $B_{v2}$). |
| $T_{LX}$ | Road load torque ($T_{L1}$ or $T_{L2}$). |
| $i_X$ | Gear ratio ($i_1$ or $i_2$). |
| $r_w$ | Vehicle wheel radius. |
| $s_v$ | Vehicle tire slip. |
| $m_v$ | Vehicle gross weight. |
| $\rho$ | Air density. |
| $C_d$ | Drag coefficient. |
| $A_v$ | Vehicle frontal area. |
| $f_r$ | Rolling resistance. |
| $g$ | Gravity constant. |
| $\alpha$ | Slope grade of road. |
| $\tau_m$ | Motor time constant. |
| $\tau_v$ | Vehicle time constant. |
| $P$ | Number of motor poles. |
| $K$ | Torque control gain. |
| $\Delta t_n$ | Synchronizing time interval. |
| $t_i$ | Initial time of gear shift. |
| $t_i$ | Initial time of neutral state. |
| $\epsilon$ | Buffer time for position control. |
| $K_P$ | Tracking filter P gain. |
| $K_I$ | Tracking filter I gain. |
| $\wedge$ | Estimated value. |
| $*$ | Reference value. |

I. INTRODUCTION

Even in Electric Vehicle (EV), multi-speed gear remains an option to increase the energy efficiency and the range [1].
The multi-speed gearbox allows the motor to change its operation area in the torque-speed plane [2]. In addition, it is possible to increase the motor efficiency while reducing the use of the field weakening operation [3]. However, it has disadvantages in terms of cost, volume, losses and drivetrain mass. Thus, the two-speed gear is studied as a compromise between complexity and efficiency increase [1], [4].

In the two-speed gear system, shift shock occurs at the moment of re-engaging due to the speed mismatch. To avoid it, clutches are often used [5]. However, the motor in EV is capable of synchronizing its speed for a next gear ratio, a clutch is not generally used [6], [7]. Thus, the clutchless two-speed gear system is preferred since it is simple in structure, as well as in the shifting process. Usually, the gear shifting process consists of disengagement, motor speed synchronization, and re-engagement [8], [9], and [10]. In the disengagement, the motor torque is controlled to be zero for easy release. Then, the locking device is driven into the neutral zone. In neutral state, the motor speed is adjusted to the synchronizing speed of the next gear. If the speeds are not matched, a slip takes place while causing shift shock. Several studies have been conducted to mitigate potential collisions [9], [12], and [13].

Despite the short shift time (0.5～1 second), the vehicle speed changes during the shifting process. Specifically, it changes in the neutral state while the motor adjusts its speed for synchronization. The amount of vehicle speed change depends on the road load conditions such as slope, road surface condition, cargo weight, etc. However, the target vehicle speed change is not reflected, a speed mismatch between complexity and efficiency increase [1], [4].

The paper is organized as follows: In Section 2, two speed gear structure and a sleeve mechanism are explained. In Section 3, the speed synchronization method is stated with a RLS estimator. Finally, the proposed method is verified through the simulation in Section 4 and the experiment in Section 5.

II. VEHICLE DYNAMICS AND TWO-SPEED GEAR SYSTEM

A point mass vehicle dynamics can be described by

$$J_v\ddot{v} + B_v\dot{v} = T_{sh} - T_L, \tag{1}$$

where $J_v$ is the vehicle inertia translated equivalently from the linear motion, $B_v$ is the viscous friction coefficient, $T_{sh}$ is the wheel shaft torque and $T_L$ is the road load including the slope grade, rolling force, and aerodynamic force. Then, the road load torque is described [6], [16], [20], [21] such that

$$T_L = \left( f_r m_v g \cos \alpha + m_v g \sin \alpha + \frac{1}{2} \rho C_d A_v v_x^2 \right) r_w, \tag{2}$$

where $m_v$ the vehicle gross weight, $\rho$ the air density, $C_d$ the drag coefficient, $A_v$ the vehicle frontal area, $f_r$ the rolling resistance, $g$ the gravity constant, and $\alpha$ the slope angle of road. Let $v_x$ be the vehicle linear speed. Normally, the angular velocity of the tire does not match the expected velocity for linear rolling motion, vice versa. To reflect the realism of the vehicle model, tire-road longitudinal forces were modeled through Pacejka’s Magic Formula. Note that

$$v_x = (1 - s_v) r_w \omega_v, \quad r_w \text{ is the effective wheel radius, } \omega_v \text{ is the angular speed of the wheel, and } s_v \text{ is the slip ratio.}$$

Fig. 1 shows a basic layout of two-speed gear system. Using the torque balance relation [22], [23], The dynamic model of electric motor is described by

$$J_m \ddot{\omega}_m + B_m \omega_m = T_m - \frac{T_{sh}}{i_X}, \tag{3}$$

where $\omega_m$ is the motor shaft speed, $J_m$ is the inertia of rotor assembly, $B_m$ is the motor friction coefficient, and $T_m$ is the motor shaft torque. $\frac{T_{sh}}{i_X}$ is the torque transferred to the wheel. $i_X$ for $X = 1$ or $X = 2$ signifies the reduction gear ratio. The first and second gear ratios are denoted by $i_1$ and $i_2$, respectively. In this work, $i_1 = 12.52$ in the first (low speed) gear, and $i_2 = 6.75$ in the second (high speed) gear.

A. DYNAMICS IN GEAR ENGAGED AND DISENGAGED STATE

The gear has two engaged states:

$$\omega_m = i_X \omega_v. \tag{4}$$

The coupled dynamics in the engaged state are described as from the motor side:

$$T_m - T_{LX} = J_{e,X} \frac{d\omega_m}{dt} + B_{e,X} \omega_m, \tag{5}$$
where \( J_{X} = \left( J + \frac{J}{T_{X}} \right) \), \( B_{X} = \left( B + \frac{B}{T_{X}} \right) \) and \( T_{LX} = \frac{T_{L}}{T_{X}} \). It is an extended motor dynamics which encompasses the load dynamics with increased inertia \( J_{vX} \), increased friction \( B_{vX} \) and an additional road load \( T_{LX} \). Note that \( J_{v1} < J_{v2}, B_{v1} < B_{v2}, \) and \( T_{L1} < T_{L2} \) due to the gear ratio.

In the disengaged (neutral) state, the motor and wheel dynamics are decoupled. Then, \( T_{sh} = 0 \) in (3):

\[
T_{m} = J_{m} \frac{d\omega_{m}}{dt} + B_{m}\omega_{m},
\]

Thus, motor power is not transmitted to the wheel.

### III. GEAR SHIFTING PROCESS AND PROBLEM STATEMENT

Obviously, the motor speed should be changed as the gear ratio changes, i.e., \( \Delta\omega_{m} = (i_{2} - i_{1})\omega_{v} < 0 \) in the upshift (1st \( \rightarrow \) 2nd) or \( \Delta\omega_{m} = (i_{1} - i_{2})\omega_{v} > 0 \) in the downshift (2nd \( \rightarrow \) 1st), where \( \Delta\omega_{m} \) is the amount of speed change for gear shift, and \( i_{1} \) and \( i_{2} \) are the ratios of 1st and 2nd gears. This is called ‘speed synchronization’ or ‘synchronization’.

Fig. 4 shows plots of wheel speed, motor speed, and torque in the gear shifts. The shifting process can be divided into 3-phases: In the first phase, A-B, the propulsion motor torque is reduced to zero for easy disengagement at \( t_{1} \) and the sleeve moves into the neutral zone. Here, time \( \epsilon \) is required for the torque releasing and position control of the shifting motor. In the second phase, B-C, the motor speed changes for synchronization with the next gear to be engaged. The motor torque, \( T_{m}^{1 \rightarrow 2} \) must be negative, when the gear shift is from 1st to 2nd (upshift). On the other hand, positive torque, \( T_{m}^{2 \rightarrow 1} \) should be supplied to increase the motor speed in the transition from 2nd to 1st (downshift). In the final phase, C-D, the motor torque is again reduced to zero, and the sleeve moves into the next gear for the gear locking. Likewise, time \( \epsilon \) is required for final step.

A normal gear shift takes about 0.5–1 second, so it is necessary to synchronize the motor speed to the next gear for a short time. If the speed synchronization is not perfect, shift shock will occur at the moment of re-engagement. For synchronization, speed controllers are commonly used for shifting. However, since torque control is the default of vehicle power control, it is not convenient to change the control mode from torque to speed and then back to torque. In addition, switching control modes at each shift is not only cumbersome, but also requires communication with other control units, which can delay the vehicle’s speed response.

In this work, it is proposed a speed synchronization method in the torque control mode. Another issue treated here is the vehicle speed variation due to the gear shift time required for motor speed synchronization. In general, it is assumed that the vehicle speed remains the same for a short transient time. Nevertheless, the speed changes more or less during coasting, where the amount of speed change...
is \( \omega_v = \omega_v(t_0 + \Delta t_n) - \omega_v(t_0) \) as shown in Fig. 4. It depends on the vehicle load, the road slope, the wind blowing directions, etc. Therefore, it needs to reflect such conditions when determining the target synchronizing speed.

IV. PROPOSED SPEED SYNCHRONIZATION METHOD

When the gear is completely disengaged, the vehicle dynamics turn out to be autonomous. The wheel dynamics is described as

\[
J_v \frac{d}{dt}(i_X \omega_v) + B_v (i_X \omega_v) = -T_L,
\]

Then, the solution is obtained directly

\[
i_X \omega_v(t) = i_X \omega_v(t_0) e^{\frac{t-t_0}{\tau_v}} - (1-e^{\frac{t-t_0}{\tau_v}}) \frac{i_X T_L}{B_v},
\]

where \( t_0 \) is the initial time of completely disengaged state and \( \tau_v = \frac{J_v}{B_v} \).

A. TORQUE CONTROL BASED SPEED SYNCHRONIZATION

The motor speed \( \omega_m \) should be adjusted so that it equals to \( i_X \omega_v \) for the synchronization. Let \( \Delta t_n \) be the neutral state time. In order to change the speed, the following torque command is applied for \( \Delta t_n \):

\[
T_m^* = -\left( \sigma K \tilde{B}_v - B_m \right) \dot{\omega}_m - \sigma K i_X \tilde{T}_L,
\]

where \( \sigma = (J_m/\tilde{J}_v), \tilde{J}_v, \tilde{B}_v, \) and \( \tilde{T}_L \) are the estimates of \( J_v, B_v, \) and \( T_L \), and \( K \) is a control gain to be determined below. It is assumed here that the motor torque controller is fast enough and precise, so that the same amount of shaft torque is generated as soon as the torque command is applied, i.e., \( T_m^* = T_m \). Applying (9) to (6) it follows that

\[
J_m \frac{d}{dt} \omega_m + \sigma K \tilde{B}_v \omega_m = -\sigma K i_X \tilde{T}_L.
\]

It is depicted in Fig. 5. Then the closed loop of the motor control system is equal to

\[
\frac{\tilde{\tau}_v}{K} \frac{d}{dt} \omega_m + \omega_m = \frac{i_X \tilde{T}_L}{B_v}.
\]

where \( \tilde{\tau}_v = \tilde{J}_v/\tilde{B}_v \). The solution is obtained straightforwardly as

\[
\omega_m(t) = \omega_m(t_0) e^{-\frac{t-t_0}{\tau_v}} - (1-e^{-\frac{t-t_0}{\tau_v}}) \frac{i_X \tilde{T}_L}{\tilde{B}_v},
\]

It remains to set gain \( K \) such that (8) and (12) are the same after \( \Delta t_n \), i.e., \( \omega_m(t_0 + \Delta t_n) = i_1 \omega_v(t_0 + \Delta t_n) \).

\[
\sigma K i_X \tilde{T}_L
\]

FIGURE 5. Torque control based speed synchronization.

1) Gear Upshift from \( i_1 \) to \( i_2 \)

In general, the vehicle speed is not kept constant in the period of neutral state. The vehicle speed before engagement is estimated as

\[
\omega_v(t_0 + \Delta t_n) = \omega_v(t_0) e^{-\frac{\Delta t_n}{\tau_v}} - (1-e^{-\frac{\Delta t_n}{\tau_v}}) \frac{T_L}{\tilde{B}_v}.
\]

Note that the motor speed is equal to \( \omega_m(t_0) = i_1 \omega_v(t_0) \) at the initial state. For synchronization, the motor speed \( \omega_m(t_0 + \Delta t_n) \) must be equal to

\[
i_2 \omega_v(t_0 + \Delta t_n) = i_2 \omega_v(t_0) e^{-\frac{\Delta t_n}{\tau_v}} - (1-e^{-\frac{\Delta t_n}{\tau_v}}) \frac{i_2 T_L}{\tilde{B}_v},
\]

Using (12) and (14), the torque control gain for the synchronization is derived as

\[
K = \frac{\tilde{\tau}_v}{\Delta t_n} \ln \left( \frac{i_1}{i_2} \cdot \frac{\omega_m(t_0) + i_2 \tilde{T}_L/\tilde{B}_v}{\omega_v(t_0)} \right) + 1.
\]

Here, it is assumed that there was no estimation error in vehicle parameters, i.e., \( J_v = \tilde{J}_v, \tilde{B}_v = B_v, \) and \( T_L = \tilde{T}_L \). In summary, the speed synchronization is realized by the torque command (9) with the gain \( K \) for upshift.

2) Gear Downshift from \( i_2 \) to \( i_1 \)

At the initial instant of the neutral state, the motor speed is equal to \( \omega_m(t_0) = i_2 \omega_v(t_0) \). Then, the synchronization condition is

\[
\omega_m(t_0 + \Delta t_n) = i_1 \omega_v(t_0 + \Delta t_n).
\]

Following the same steps in the above, the synchronizing gain \( K \) is obtained such that

\[
K = \frac{\tilde{\tau}_v}{\Delta t_n} \ln \left( \frac{i_2}{i_1} \cdot \frac{\omega_m(t_0) + i_1 \tilde{T}_L/\tilde{B}_v}{\omega_v(t_0)} \right) + 1.
\]

Here, \( K \) is the synchronizing gain for downshift.
The upshift and downshift gains are

\[ K = \begin{cases} \frac{\dot{T}}{\Delta t_n}(A - B) + 1, & \text{upshift} \\ \frac{\dot{T}}{\Delta t_n}(B - A) + 1, & \text{downshift}, \end{cases} \]

where \( A = \ln(\omega_m(t_0) + i_2\hat{T}_L/\hat{B}_e) + \ln i_1 \) and \( B = \ln(\omega_m(t_0) + i_1\hat{T}_L/\hat{B}_e) + \ln i_2 \).

In summary, the estimated parameters and the initial speed at gear release are used to calculate the torque control gain (18). The torque command (9) for synchronization is possible without changing the control mode. \( \Delta t_n \) is the acceleration/deceleration time taken to increase or decrease the speed of the motor. The acceleration/deceleration time is determined by the maximum torque of the motor and the coefficient of inertia of the rotor and gear. \( \Delta t_n \) is determined experimentally in many cases, and in this experiment, 0.3~0.5 seconds was judged to be appropriate.

## B. LOAD PARAMETER ESTIMATION

It is assumed here that no wheel speed sensor is utilized. The wheel speed can be predicted using (13). But, it includes \( J_v, B_v \) and \( T_L \). Note that \( J_v \) is a complex quantity since it includes the translated inertia of linear motion, as well as the inertia of wheels and shafts. Exact value of \( B_v \) is locked by the resolver sensor output \( \theta_e \). Then, \( \dot{\omega}_e \) is obtained from the prior node of the integrator, and the motor acceleration estimate is obtained as

\[ \dot{\omega}_m = \frac{2}{P} \left( K_f(\theta_e - \dot{\theta}_e) + K_p(\omega_e - \dot{\omega}_e) \right). \]

According to [28] and [31], the gain of the tracking filter can be selected through acceleration and rotational inertia. Thus, the bandwidth \( \rho \) is set as 48 and the gains are \( K_p = 2\rho = 96 \) and \( K_f = \rho^2 = 2304 \), respectively.

![FIGURE 6. Tracking filter: Phase locked loop (PLL) to estimate angular acceleration.](image)

In the vehicle, torque is not commonly measured by a torque sensor. It is calculated from the motor currents, for example, such as \( T_m = 3P/4[\lambda_m i_q^e + (\lambda_d - \lambda_q) i_q^e i_q^e] \), where \( P \) is the pole number, \( \lambda_m \) is the back EMF constant, \( \lambda_d, \lambda_q \) are the d and q axis inductances, and \( i_q^e, i_q^e \) are the d and q currents [29]. However the torque equation is not valid due to the flux nonlinearity. Instead, a torque look-up table may be used to strengthen the practicality using \( \{i_q^e, i_q^e\} \) as input.

Since the motor is not linked to the wheel, the RLS algorithm is switched off during the neutral state. Thus, the synchronizing torque control (9) is performed using \( \dot{J}_e(k_i), \dot{B}_e(k_i), \dot{T}_L(k_i) \), where \( k_i \) is the time just before disengaging, or the starting time of gear shift, i.e., \( k_i = kT_s \).

## V. SIMULATION RESULTS

Vehicle parameters for simulation study is summarized in Table 1. The simulation model is established in MATLAB/Simulink and the actual value is obtained from the vehicle simulation tool ADVISOR [30]. Firstly, the performance of the parameter estimator, i.e., the RLS algorithm was tested while changing the speed and road load conditions. Fig. 7 (a) and (b) show the convergence of vehicle parameters when the motor speed changes from 1000 rpm to 2000 rpm and the road grade changes from 20% to -20%, respectively. There
are some errors in the transient state, but the estimates converge to real values within 2 seconds. Vehicle speed change affects the inertia estimate \( J_{v1} \), whereas road slope change has more influence on the road load estimate \( T_{L1} \). All the estimates converge to the correct values in the simulations.

Table 1 lists the experimentally obtained prototype vehicle specifications.

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**FIGURE 8.** Vehicle speed change according to the road grade when coasting for 1 second.

**FIGURE 9.** Shifting performance according to the estimated road load torque \( T_{L1} \) under \( \alpha = 20\% \).

| Parameters | Values | Parameters | Values |
|------------|--------|------------|--------|
| \( C_d \)  | 0.38   | \( \alpha \) | 0 or 3.2 |
| \( s_v \)  | 0.02   | First gear ratio, \( i_1 \) | 12.52 |
| \( f_r \)  | 0.014  | Second gear ratio, \( i_2 \) | 1.225 kg/m³ |
| \( g \)    | 9.81 m/s² | \( m_v \) | 6.75 |
| \( A_v \)  | 1.767 m² | \( \eta_X \) | 2700 kg |
| \( r_w \)  | 0.34 m | \( \eta_X \) | 0.94 |

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Fig. 8 shows the amount of speed change \( \delta \omega_v \) in neutral state for 1 second. This result shows how much the vehicle speed changes while no power is being transmitted. As expected, the speed change is inversely proportional to the slope grade of the road. In the case of a 1 ton truck, the effect of payload capacity looks minor, whereas the slope grade is the most influential.

Fig. 9 shows the synchronization performances in a shift from the 1st to the 2nd gear. The pink dotted line \((i_1 \omega_v)\) and the blue dotted line \((i_2 \omega_v)\) represent the virtual speeds of 1st and 2nd gear, respectively. It is assumed that the slope grade is 20\% and the corresponding load is \( T_{L1} = 148 \) Nm. When \( T_{L1} \) is estimated correctly, the motor speed is controlled to the synchronization speed, 1340 rpm perfectly. There are three cases of error in the estimate, \( \hat{T}_{L1} \). The errors represented by \( \Delta T_{L1}/T_{L1} \) are 10\%, 30\%, and 50\%. And these effects cause speed mismatch and shift shock. For example, if the error is 50\%, it causes 183 rpm speed mismatch at the moment of re-engagement, resulting in shift shock. If the vehicle speed variation is not considered at all during the neutral period, the speed mismatch amounts to 410 rpm.

Fig. 10 shows the comparison of the proposed method with the conventional method using torque-speed-torque control. In the conventional method, a well-established PI speed controller is used for synchronization. Fig. 10 (a) shows the motor speed and the torque for synchronization during the gear shift from 1st to 2nd. The motor speed is 1710 rpm at the moment of disengagement. The pink dotted line \((i_1 \omega_v)\)
FIGURE 10. Comparison of conventional method (torque-speed-torque control) with the proposed method for the motor speed synchronization when the slope grade is 3.2%: (a) upshift (1st → 2nd) and (b) downshift (2nd → 1st).

FIGURE 11. Performance of the proposed speed synchronization method, and the required propulsion motor current. (a) upshift (1st → 2nd) and (b) downshift (2nd → 1st).

FIGURE 12. Motor Torque control block with the proposed gear shifting method.

and the blue dotted line ($i_2\omega_v$) are the virtual speeds of 1st and 2nd gear, respectively. The vehicle is driven on an uphill road of 3.2% grade, so that the vehicle speed $\omega_v$ decreases in the gear disengaged state [1, 2, 2]. The speed trajectory of the proposed method is plotted with green color. Note that $\omega_{m}$ reaches the virtual speed of second gear $i_2\omega_v$ without speed mismatch for the speed synchronization. Therefore, no jerk took place with the proposed method. Meanwhile,
the trajectory of the conventional method is plotted with black color. Because the speed drop of the vehicle was not taken into account, $\omega_{vn}$ leaves a speed difference of 98 rpm without hitting $\omega_n$. Thus, it causes the fluctuation and the shift shock. In the simulation, the jerk torque is modeled as $e(\omega_{vn} - \dot{\omega}_v)$ where $e$ is the jerk coefficient [16]. Fig. 10 (b) shows similar results when the gear was shifted from 2nd to 1st (downshift). The initial speed is 1250 rpm at the moment of disengagement. The same road load condition ($\alpha = 3.2\%$) is assumed. The speed trajectory of the conventional method plotted with black color shows the speed mismatch of 124 rpm. Thus, it results in the fluctuation and the shift shock. Meanwhile, the proposed method plotted with green line shows the stable synchronization.

Fig. 11 shows the speed trajectory of the proposed method and the $\alpha$-phase current among three phases. The motor speed is compared with the target speed, $\dot{\omega}_v$, where $\omega_v$ is obtained by the wheel speed sensor. The speed differences of upshift is 3 rpm in Fig. 11 (a) and, in the case of downshift, it is -5 rpm as shown in Fig. 11 (b). Since the differences are so small, no fluctuation is observed in the motor speed and the phase current.

VI. EXPERIMENTAL RESULTS

Fig. 12 shows the motor control block diagram for speed synchronization. The speed and position values of resolver are fed into the tracking filter to obtain the acceleration, $\dot{\omega}_m$. Based on $\dot{\omega}_m$ and $\omega_m$, the RLS estimator yields the vehicle parameters $J_c$ and $B_c$ with the road load $T_L$. Using these estimates, a suitable torque command is calculated for synchronization. Thus, it does not require a mode change to the speed control.
Fig. 13 (a) shows an experimental set for gear shift consisting of a 90 kW IPMSM motor, gearbox, disc brakes, and a flywheel. The 200kg flywheel is used to provide an inertia load corresponding to part of the vehicle inertia. A disk brake acts like a road load to the wheel. Fig. 13 (b) shows the shifting device consisting of the gearbox and the shifting motor (actuator). It is also equipped with a permanent magnet linear contactless displacement (PLCD) sensor to monitor the position of the fork-sleeve (locking device). The 100kW traction inverter was used, and the full-bridge circuit for controlling the shifting DC motor was manufactured separately. TI’s TMS320F28377D was used as a microcontroller, and it controls the propulsion motor and the shifting motor. The control and switching period was set to 100us (10kHz).

Fig. 14 and Fig. 15 show experimental results during gear shifts. The blue and green lines show 1-phase current among three phase currents and the speed of the propulsion motor, respectively. The yellow line indicates the gear states; 1st gear, 2nd gear, neutral, and buffer state. The red line shows the current of the shifting DC motor. If the speed mismatch remains at the re-engaging instant, the DC motor current fluctuates sharply to enforce the sleeve into the engagement ring.

Fig. 14 (a) shows a case of the conventional method when the gear shifts from 1st to 2nd. The propulsion motor speed was 1710 rpm at the moment of disengagement, and was decreased to 925 rpm via PI speed control. During the speed reduction in the neutral state, the 1-phase current was 5.4A(peak). Since the wheel speed drop was not considered, speed mismatch occurs. Thereby, current surge is observed in the shifting motor. Incomplete speed synchronization increases the burden of the shifting DC motor. Fig. 14 (b) shows the similar results, but with the proposed control method. In this case, the motor speed dropped to 811 rpm while considering the wheel speed decrease. Since the motor speed was synchronized with the wheel, almost no current fluctuation is observed in the DC motor current.

Fig. 15 (a) shows a case of downshift from 2nd to 1st gear with the conventional PI method. The motor speed increased to 2246 rpm from 1200 rpm. In this case, a large amount of 1-phase current 17A(peak) was applied to speed up the motor. Due to the speed mismatch, a surge current was observed in the shifting motor. The result of the proposed method is shown in Fig. 15 (b). According to the calculated torque, the phase current of the motor increases. In this case, the motor speed was increased to 2108 rpm, and no shift shock was observed.

VII. CONCLUSION
The EV motor can adjust its speed in the neutral state, so that synchronization can be realized without an additional device. In this work, a speed synchronization method is developed in the torque control mode. It results in more accurate synchronization by reflecting vehicle speed changes depending on the road loads. In this study, since wheel sensor is not used, speed synchronization is possible without complex communication with other control units, e.g. transmission control unit (TCU). The synchronization algorithm includes the RLS estimator, the tracking filter, and calculating torque commands. Simulation and experimental studies showed the effectiveness of the proposed method in various cases.

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