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Feedforward-feedback shift control with disturbance compensation of a two-speed transmission for electric vehicles

Jianfeng Huang[1], Jianlong Zhang, Wei Huang, Chengliang Yin, Lifang Wang

[1]*School of Mechanical Engineering, Shanghai Jiao Tong University, Shanghai 200240, China,*

Email: 515064@sjtu.edu.cn

Summary

Two-speed transmissions are helpful for improvement of economy and drivability of electric vehicles (EV). This paper discusses gear shift control of such powertrains, with the aim of ensuring constant output torque on the wheel during the whole process. The classic two-phase control scheme (torque phase and inertia phase) is adopted and control strategies are developed for each respectively. To enhance transient responses, the feedforward-feedback controller structure is applied. To improve robustness, a disturbance observer is integrated. Simulations show that the proposed method is able to achieve fast and smooth gear shift robustly while maintaining constant output torque.

Keywords: EV (electric vehicle), HEV (hybrid electric vehicle), powertrain, transmission, control system

1 Introduction

Electric vehicles (EV) are promising alternatives to conventional vehicles which use fossil fuels. Most EVs in the market, especially the passenger ones, do not use transmissions. Rather, they are equipped with reduction gears providing only one fixed speed ratio [1]. However, this does not imply the unnecessity of transmissions. On the contrary, researchers have proved benefits of using transmissions on EV through simulations and experiments [2]. Observed attempts mainly concentrate on automatic manual transmissions (AMT), which use synchronizers for gear shifting and are with power interruption. For high-performance models, intermittent power loss is unacceptable. In such cases, planetary-gear-based transmissions (PGT) may be more promising candidates. Furthermore, for sport-oriented vehicles, constant output torque on the wheel is one of the major requirements.

Researches on gear shifting control of transmissions for electric vehicles have been observed. In [3], optimal control methods are used to fulfill the constant-output-speed and constant-output torque shift control, with the actuator limits considered. This work is further improved in [4] with complement of a backstepping controller which serves as the feedback component. Hybrid Minimum Principle (HMP) is used in [5] to handle state jumps at the switching points of different stages in gear shifting. PID controllers are applied in [6], [7]. Effects of the slip speed reference in the inertia phase are discussed in [8]. Torque observers are adopted in [9] to facilitate shifting control of a two-speed transmission for EV. Novel configurations and control strategies are brought about with the rapid development of EV. However, few literatures have shown concern for the robustness of the controller through dedicated techniques. This paper aims to fill this gap.

Generally speaking, two or three gears are enough for transmissions on EV, due to the high efficiency of modern traction motors. This paper proposes an integrated gear shifting strategy using feedforward-feedback control with disturbance compensation, for both torque phase and inertia phase. The generic dual-clutch model is presented in section 2. The constant-output-torque control strategy is introduced in section 3. Section 4 presents the simulation results. Conclusions are provided in section 5.

2 Generic dual-clutch model

Schematic of the two-speed transmission is shown in Fig1. It comes with two planetary gear sets (PGS) and two brakes (CL1 and CL2).

Its dynamic model is derived using the "autoEQ" function proposed in [10] as:

$$\mathbf{A_s}\dot{\mathbf{X}}_\mathbf{s} = \mathbf{B_s}\mathbf{Y_s} \tag{1}$$

where $\mathbf{A_s}$ is the inertia matrix and $\mathbf{B_s}$ is the input matrix. $\mathbf{X_s}$ is the state variables selected as $\mathbf{X_s} = [\omega_{C1} \quad \omega_{C2}]$, i.e. the rotational speeds. The subscripts C_1 and C_2 denote the nodes "Carrier 1" and "Carrier 2" of the PGS, as is indicated in Fig1. $\mathbf{Y_s}$ represents the external torques which may come from the drive/load or the clutches. For the transmission in Fig1, it is $\mathbf{Y_s} = [T_{in} \quad T_{out} \quad T_{CL1} \quad T_{CL2}]$, where T_{in} and T_{out} are torques exerted on the input and output shaft, T_{CL1} and T_{CL2} are torques transmitted by the two brakes, respectively. Note that T_{out} is by the load on the wheel, which is related to air drag, rolling friction, gradient resistance, etc. and therefore cannot be designed. As for the sign convention, it is assumed that all torques are driving the corresponding shafts (i.e. propelling the vehicle). If this is not the case, then the value would be negative (e.g. for T_{out}). To facilitate controller design and analysis, the dynamic model is further simplified into the generic dual-clutch one, as is shown in Fig2. Here, $K_{T_{in}/T_{CL1}}$ represents the speed ratio between the left side of the low gear clutch (CL1) and the motor:

$$K_{T_{in}/T_{CL1}} = \frac{T_{in}}{T_{CL1}} = \frac{\omega_{CL1}}{\omega_{in}} \tag{2}$$

The simplification can be done by ignoring the inertias which do not connect directly with the input and output shaft:

$$\dot{\omega}_{S1} = \frac{1}{J_{S1S2}}T_{in} - \frac{1}{J_{S1S2}k_1}T_{CL1} - \frac{1}{J_{S1S2}k_2}T_{CL2} \tag{3}$$

$$\dot{\omega}_{C2} = \frac{1}{J_{C1C2}}T_{out} + \frac{k_1+1}{J_{C1C2}k_1}T_{CL1} + \frac{k_2+1}{J_{C1C2}k_2}T_{CL2} \tag{4}$$

J_{S1S2} is the lumped inertia on the input shaft which may include those of the motor itself and the sun gears. J_{C1C2} is the lumped inertia on the output shaft including those of the carriers, the vehicle body and the wheels.

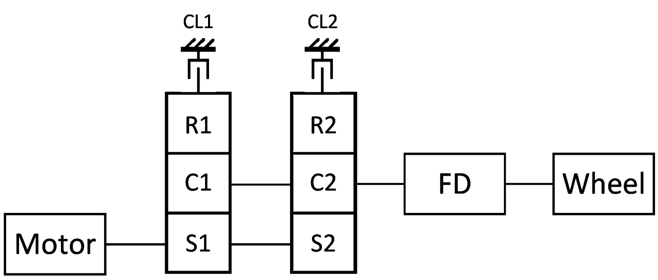

Fig1: Schematic of the two-speed transmission ("FD" is for "Final Drive")

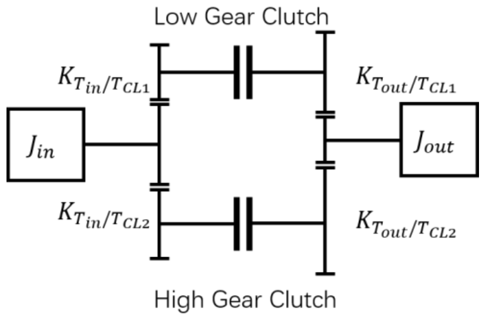

Fig2: Generic dual-clutch model

Gear ratios are shown in Table1. Lever analogy [11] for each gear state is shown in Fig3.

Table1: Gear ratios of the two-speed transmission

Gear state	Gear ratio	Symbolic form
First	4	$1 + k_1$
Second	2.29	$1 + k_2$
Final drive	2.9	/

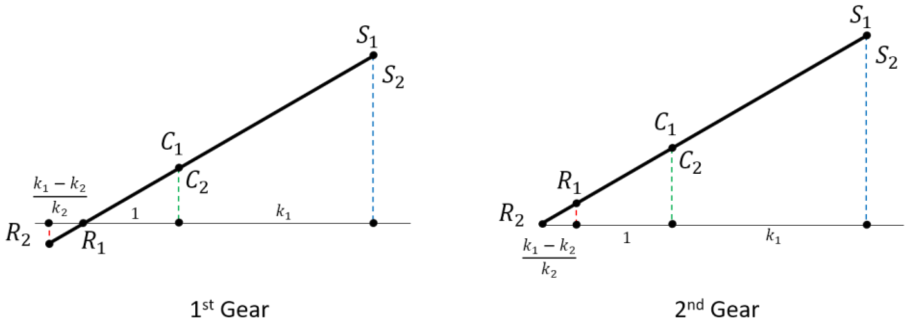

Fig3: Lever analogy of two gear states

3 Constant-output-torque shift control

The shifting process of 1-2 upshift is considered. It is divided into two phases, with different control strategies for each. Before designing the controller, some characteristics of the system should be noted: a) inertia of the motor is much smaller than that of the engine in conventional multi-speed transmissions; b) the motor torque is more stable, more responsive, and more precise than the engine torque, which makes accurate speed regulation possible. These facts should be utilized to simplify controller design and facilitate determination of the control scheme.

General requirements for gear shifting control of the two-speed transmission are as follows:

- Duration of the whole process can be easily manipulated.
- Jerks on both the input and output shaft are small enough, so as to protect the motor shaft and ensure riding comfort.
- Friction work of the two brakes are small enough, so as to protect the two components.
- Output torque of the transmission can be arbitrarily shaped (within the capability and constraint of components) during the shifting process, so as to ensure drivability on some high-performance models.

3.1 Torque phase

In the torque phase, the load on the motor shaft should be gradually transferred from the off-going clutch to the on-coming one. Requirements for torque phase control are:

- The off-going clutch should be kept engaged during the whole process and no slip occurs.
- The load and the pressure on the off-going clutch should be reduced to zero at the same time

Under the first gear, CL1 is locked, which results in

$$\dot{\omega}_{S1} = (1 + k_1)\dot{\omega}_{C2} \tag{5}$$

$\dot{\omega}_{C2}$ can be expressed with T_{out}, T_{CL2}, and T_{in}, by solving equations (3)(4)(5):

$$\dot{\omega}_{C2} = \frac{k_2^2}{temp}T_{out} - \frac{(k_1 - k_2)k_2}{temp}T_{CL2} + \frac{(k_1 + 1)k_2^2}{temp}T_{in} \tag{6}$$

$$temp = J_{C1C2}k_2^2 + J_{S1S2}(1 + k_1)^2 k_2^2 \tag{7}$$

As has been discussed before, inertia of the motor is rather small compared to that of the engine, and therefore J_{S1S2} can be neglected. Based on this assumption, equation (6) is simplified as

$$\dot{\omega}_{C2} = \frac{1}{J_{C1C2}}\left[T_{out} - \frac{k_1 - k_2}{k_2}T_{CL2} + (k_1 + 1)T_{in}\right] \tag{8}$$

From equation (8) it is obvious that the driving torque on the output shaft is

$$T_{driv} = -\frac{k_1 - k_2}{k_2}T_{CL2} + (k_1 + 1)T_{in} \tag{9}$$

Since all inertias in Fig1 except J_{C1C2} are ignored, equation (9) can be directly derived by using the speed/torque ratios between different nodes of the two PGS, as is illustrated in Fig3. Note that T_{driv} consists two terms regarding T_{in} and T_{CL2}. The former is the motor torque, and the latter is determined by the pressure on CL2 (since it is slipping during the torque phase). Thus, T_{driv} can be manipulated either through controlling the motor or CL2. As an internal torque, T_{CL1} is determined by T_{in} and T_{CL2} as well:

$$T_{CL1} = T_{in}k_1 - T_{CL2}\frac{k_1}{k_2} \tag{10}$$

Equation (10) is also based on the assumption of small motor inertia. For verification, substitute equation (10) into equation (4) and equation (8) is arrived.

During the torque phase, there are totally three components to be controlled, namely the motor and the two brakes. However, according to equations (9) and (10), only one variable can be freely determined. Recalling the requirement on controllability of the shift time, T_{CL1} (transmitted torque of the off-going clutch) should better be selected as the one being manipulated. For example, as the simplest case, T_{CL1} can be commanded to decrease to zero linearly, within a specified duration. Since CL1 is required to be kept engaged along the whole torque phase, its transmitted torque should be no larger than the torque capacity T_{cap1}:

$$T_{CL1} \le T_{cap1} \tag{11}$$

Also recall the second requirement that T_{CL1} and T_{cap1} be reduced to zero at the same time, which suggests the following control law for T_{CL1}:

$$T_{CL1} = \gamma \cdot T_{cap1} \tag{12}$$

Here, γ is a coefficient which satisfies $\gamma < 1$. For practical use, γ is advised to fall between 0.90-0.99. As has been discussed before, trajectory of T_{cap1} in torque phase can be designed as a linear curve:

$$T_{cap1}(t) = T_{cap1}(t_0) - \frac{T_{cap1}(t_0)}{t_1 - t_0}(t - t_0), t_0 \le t \le t_1 \tag{13}$$

where t_0 and t_1 are the start and end point of the torque phase, respectively. Duration of the torque phase can thus be arbitrarily manipulated (within the capacity and constraint of components) through the parameter t_1. Equation (12) ensures that a) T_{CL1} is always smaller than T_{cap1}; b) when T_{cap1} arrives zero, so does T_{CL1}. Thus, both the two requirements mentioned at the beginning of section 3.1 are fulfilled. T_{driv} in equation (9) is specified by the vehicle controller unit (VCU) and related to position of the accelerator pedal. Therefore, it can be treated as known. Now only T_{in} and T_{CL2} remain to be determined. They can be derived by solving equations (9) and (10):

$$T_{CL2} = -\frac{k_2(k_1 + 1)}{k_1(k_2 + 1)}T_{CL1} + \frac{k_2}{k_2 + 1}T_{driv} \tag{14}$$

$$T_{in} = \frac{k_2 - k_1}{k_1(k_2 + 1)}T_{CL1} + \frac{1}{k_2 + 1}T_{driv} \tag{15}$$

Equations (12)-(15) are control laws for the torque phase.

$T_{cap1}(t_0)$ in equation (13) should be determined carefully, so as not to cause sudden change of T_{in} and T_{CL2}. According to equation (10), the transmitted torque of CL1 at the beginning of the torque phase is

$$T_{CL1}(t_0) = T_{in}(t_0)k_1 \tag{16}$$

Using equation (12),

$$T_{cap1}(t_0) = \frac{T_{CL1}(t_0)}{\gamma} = \frac{T_{in}(t_0)k_1}{\gamma} \tag{17}$$

This is the exact level that the pressure of CL1 should be reduced to during the filling phase which precedes the torque phase. According to equation (9), T_{driv} at the beginning of the torque phase is

$$T_{driv}(t_0) = (k_1 + 1)T_{in}(t_0) \tag{18}$$

Substitute equations (16) and (18) into (14), and the initial value of T_{CL2} can be determined:

$$T_{CL2}(t_0) = 0 \tag{19}$$

which conforms to common engineering practice.

If the load is transferred seamlessly from the off-going clutch to the on-coming one, then the acceleration of the motor shaft (i.e. the input shaft) is expected to be zero. And suppose that T_{driv} equals to the load on the output shaft, which causes the acceleration of the output shaft to be also zero. Based on assumptions above, solve equations (3) and (4) for T_{in} and T_{CL2} with $\dot{\omega}_{S1}$ and $\dot{\omega}_{C2}$ set to zero, and the solutions would be the same as equations (14) and (15) (except that T_{driv} aced by T_{out}, which is the same under the static assumption). This implies that control laws for the torque phase can be attained by solving the static generic dual-clutch model.

3.2 Inertia phase

At the end of the torque phase, the motor torque and the transmitted torque by CL2 are

$$T_{CL2}(t_1) = \frac{k_2}{k_2 + 1}T_{driv}(t_1) \tag{20}$$

$$T_{in}(t_1) = \frac{1}{k_2 + 1}T_{driv}(t_1) \tag{21}$$

Using the lever analogy in Fig3, it is obvious that both T_{in} and T_{CL2} have reached levels corresponding to static state of the 2nd gear. However, CL2 is still slipping and speeds of its two sides need to be synchronized. Note also that in the inertia phase CL1 is fully disengaged and therefore the driving torque on the output shaft is solely determined by the pressure on CL2. Recalling the constant-output-torque requirement, T_{CL2} should remain constant along the whole inertia phase (assuming that T_{driv} is constant):

$$T_{CL2}(t) = \frac{k_2}{k_2+1} T_{driv}, t_1 \le t \le t_f \tag{22}$$

where t_f denotes the end of inertia phase.

Since CL1 is totally disengaged in this phase, the motor is the sole component to be controlled for speed synchronization. This approach is reasonable in that it fully exploits advantages from the precise output torque as well as fast response of the motor.

The key of inertia phase control is to realize a fast (in terms of the duration) and comfortable (in terms of longitudinal jerk felt by the passengers) speed ratio change [10]. Common practice is to make the slip speed of the on-coming clutch follow a polynomial curve. According to the *GV no-lurch condition*, the derivative of the slip speed at the synchronization point should be zero so as to avoid sudden change of T_{CL2} [12]

$$\Delta\dot{\omega}_{CL2}(t_f) = 0 \tag{23}$$

where $\Delta\omega_{CL2} = \omega_{R2}$. To accomplish this, a 3-4-5 polynomial is selected as the slip speed reference [8], as is shown in Fig4. It is expressed as

$$\omega_{R2}(t) = [\omega_{R2}(t_f) - \omega_{R2}(t_1)]s(\tau) + \omega_{R2}(t_1) \tag{24}$$

$$s(\tau) = 6\tau^5 - 15\tau^4 + 10\tau^3 \tag{25}$$

$$\tau = \frac{t-t_1}{t_f-t_1}, t_1 \le t \le t_f \tag{26}$$

Here, $\omega_{R2}(t_f) = 0$, $\omega_{R2}(t_1)$ can be attained through $\omega_{S1}(t_1)$ and $\omega_{C2}(t_1)$:

$$\omega_{R2}(t_1) = \frac{k_2+1}{k_2}\omega_{C2}(t_1) - \frac{1}{k_2}\omega_{S1}(t_1) \tag{27}$$

Transmissions are typically equipped with speed sensors on both the input and output shaft, and therefore ω_{S1} and ω_{C2} are known. Derivative of ω_{R2} is

$$\dot{\omega}_{R2}(t) = \frac{[\omega_{R2}(t_f) - \omega_{R2}(t_1)]}{t_f - t_1}(30\tau^4 - 60\tau^3 + 30\tau^2) \tag{28}$$

which satisfies $\dot{\omega}_{R2}(t_f) = 0$.

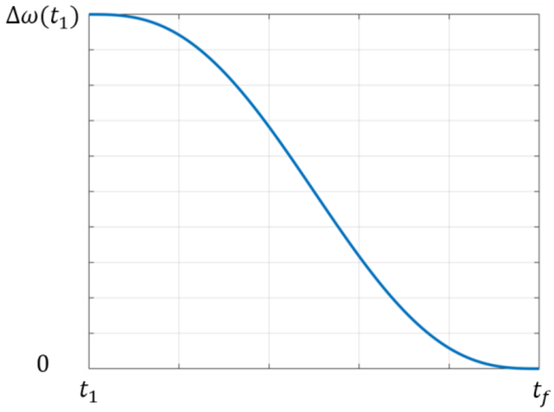

Fig4: Slip speed reference in inertia phase

3.2.1 Feedforward control

The feedforward-feedback control scheme is adopted to solve the tracking problem of the slip speed. Firstly, the inverse kinetic model of the two-speed transmission should be derived. "Inverse" means that the rotational acceleration of the input shaft $\dot{\omega}_{S1}$ is taken as known while the input torque T_{in} is taken as unknown. In the inertia phase, the two-speed transmission is in the neutral gear state, and the forward kinetic model requires typically four quantities known for solution, namely T_{in}, T_{CL1}, T_{CL2}, and T_{out}. In the inverse model, T_{in} and T_{out} are substituted by $\dot{\omega}_{S1}$ and $\dot{\omega}_{R2}$, respectively. T_{CL1} equals to zero along the whole inertia phase as a result of total disengagement of CL1. The aim now is to answer the question "given $\dot{\omega}_{S1}$, $\dot{\omega}_{R2}$ and T_{CL2}, how to determine T_{in}". Among them, $\dot{\omega}_{S1}$ can be attained through an acceleration estimator over the motor speed, which is reported by the motor controller unit (MCU). $\dot{\omega}_{R2}$ is from the designed trajectory in Fig4 and should be apparently known. As has been discussed before, during the inertia phase T_{driv} is determined by T_{CL2} only, and therefore T_{CL2} can be calculated from equation (22), given T_{driv}. Thus, using the autoEQ function [10], T_{in} is determined as

$$T_{in} = \frac{1}{k_2+1}T_{driv} + \dot{\omega}_{S1}\left[J_{R1}\frac{k_2-k_1}{k_1^2(k_2+1)} + J_{S1S2}\right] - \dot{\omega}_{R2}\left[J_{R2}\frac{1}{k_2} + J_{R1}\frac{k_2(k_1+1)}{k_1^2(k_2+1)}\right] \tag{29}$$

Equation (29) is the feedforward control law for T_{in}. Note that according to equation (28), $\dot{\omega}_{R2}(t_1) = 0$. Considering that inertias J_{R1} and J_{S1S_2} are small and negligible, the second term regarding $\dot{\omega}_{S1}$ in equation (29) can be omitted. This also unnecessitates the acceleration observer and simplifies the controller. As a result, $T_{in}(t_1) = \frac{1}{k_2+1}T_{driv}(t_1)$, which is exactly the same as equation (21). This implies that values of T_{in} at the end of the torque phase and at the beginning of the inertia phase are the same. Thus, continuity of the motor torque is ensured. The final feedforward control law is

$$u_{feedforward} = \frac{1}{k_2+1}T_{driv} - \dot{\omega}_{R2}\left[J_{R2}\frac{1}{k_2} + J_{R1}\frac{k_2(k_1+1)}{k_1^2(k_2+1)}\right] \tag{30}$$

3.2.2 Feedback control

To derive the feedback control law, the forward kinetic model should be used. However, as has been discussed before, T_{out} is the load on the wheel and hard to measure. It should be substituted by $\dot{\omega}_{S1}$, which can be attained indirectly. As before, substitute T_{CL2} with T_{driv}, and $\dot{\omega}_{R2}$ can be expressed as

$$\dot{\omega}_{R2} = \frac{1}{J_{R1}k_2^2(k_1+1) + J_{R2}k_1^2(k_2+1)}\left[T_{driv}k_1^2k_2 - T_{in}k_1^2k_2(k_2+1) + \dot{\omega}_{S1}\left[J_{S1S2}k_1^2k_2(k_2+1) + J_{R1}k_2(k_2-k_1)\right]\right] \tag{31}$$

From equation (31) it is obvious that $\dot{\omega}_{R2}$ is related to three quantities, namely T_{driv}, T_{in}, and $\dot{\omega}_{S1}$. If the term regarding $\dot{\omega}_{S1}$ is omitted, equation (31) turns into

$$\dot{\omega}_{R2} = \frac{1}{J_{R1}k_2^2(k_1+1) + J_{R2}k_1^2(k_2+1)}\left[T_{driv}k_1^2k_2 - T_{in}k_1^2k_2(k_2+1)\right] \tag{32}$$

The term regarding T_{driv} can be treated as known disturbances and compensated in a feedforward way. To make it clear, rewrite equation (31) in the state-space form:

$$\dot{x} = ax + bu + b_kd_k + b_ud_u \tag{33}$$

where

$$x = \omega_{R2} \tag{34}$$

$$a = 0 \tag{35}$$

$$u = T_{in} \tag{36}$$

$$b = \frac{-k_1^2 k_2(k_2+1)}{J_{R1}k_2^2(k_1+1) + J_{R2}k_1^2(k_2+1)} \tag{37}$$

$$b_k d_k = \frac{T_{driv}k_1^2 k_2}{J_{R1}k_2^2(k_1+1) + J_{R2}k_1^2(k_2+1)} \tag{38}$$

d_k and d_u denotes known and unknown disturbances, respectively. With $b_u = b$, a time-domain disturbance observer (DO) can be used to estimate d_u [13]:

$$\begin{cases} \dot{z} = -Lb(z+Lx) - L(ax+bu+b_k d_k) \\ \hat{d}_u = z + Lx \end{cases} \tag{39}$$

Note that L should satisfy that $-Lb < 0$. And the component for disturbance compensation in u is

$$u_{du} = -\hat{d}_u \tag{40}$$

Note that $u_{feedforward}$ in equation (30) can be written as two components:

$$u_{feedforward} = u_{dk} + u_{inv} \tag{41}$$

where

$$u_{dk} = -\frac{b_k d_k}{b} = \frac{1}{k_2+1}T_{driv} \tag{42}$$

$$u_{inv} = -\dot{\omega}_{R2}\left[J_{R2}\frac{1}{k_2} + J_{R1}\frac{k_2(k_1+1)}{k_1^2(k_2+1)}\right] \tag{43}$$

u_{dk} is used to compensate the known disturbances $b_k d_k$, while u_{inv} is derived from the nominal model $\dot{\omega}_{R2} = \frac{-T_{in}k_1^2 k_2(k_2+1)}{J_{R1}k_2^2(k_1+1)+J_{R2}k_1^2(k_2+1)}$.

Assuming that both $b_k d_k$ and $b_u d_u$ are completely compensated, the state-space model in equation (33) turns into the nominal one

$$\dot{x}_n = a x_n + bu \tag{44}$$

The feedback control law for the nominal model is

$$u_{feedback} = -F(x-r) \tag{45}$$

F is the feedback gain and can be roughly determined as [14]

$$F = \frac{\ln|\omega_{R2}| - \ln E}{b(t_f - t_1)} \tag{46}$$

where E is the level of residual tracking error when t_f is reached. Note that the sign of F should be determined in accordance with a and b to ensure convergence. This is obvious by substituting equation (45) into (44), whose solution is

$$x_n(t) = \left(x_n(t_0) + \frac{bF}{a-bF}\right)e^{(a-bF)t} - \frac{bF}{a-bF}r \tag{47}$$

For the exponential convergence of $x_n(t)$ to $x_n(\infty) = -\frac{bF}{a-bF}r$, F should satisfy that

$$a - bF < 0 \tag{48}$$

The reference signal r is from the 3-4-5 polynomial shown in Fig4.

Finally, the control law for T_{in} in the inertia phase can be summarized as

$$u = u_{inv} + u_{feedback} + u_{dk} + u_{du} \tag{49}$$

Block diagram of the slip speed controller is shown in Fig5.

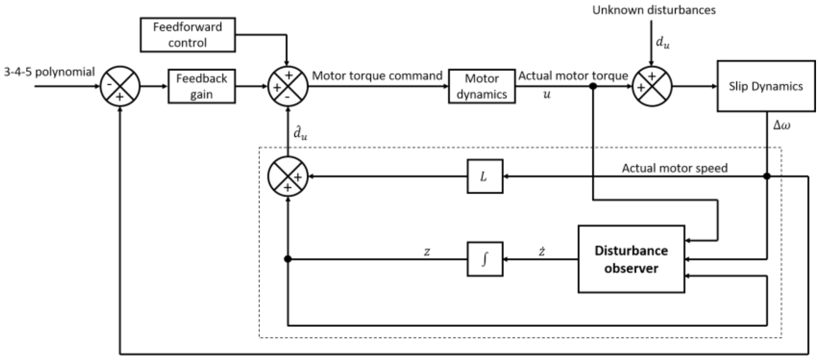

Fig5: Block diagram of slip speed controller

4 Simulation results

A model is developed in MATLAB/SIMULINK (version R2018b) to validate the proposed feedforward-feedback shift controller with disturbance compensation. It mainly consists of four parts: submodels for the motor, the transmission, the vehicle body, and the shift controller. Stiffness and damping on the output shafts are taken into consideration.

Firstly, the constant-output-torque control (COTC) is compared to the conventional constant-input-torque control (CITC), which keeps the input shaft torque constant along the whole shifting process. Simulation results are shown in Fig7. In COTC, during the torque phase the motor torque is increased to compensate the reduction of output torque due to the change of gear ratio. In contrast, CITC simply maintains the motor torque. As a result, the output shaft torque decreases during the upshift. The pressure level of the on-coming clutch in COTC is higher than that of CITC to keep the output torque constant. In the inertia phase, COTC relies on the motor torque for slip speed regulation, whereas CITC changes the pressure of the on-coming clutch to accomplish this.

Advantages of the feedforward-feedback scheme is validated through comparison with pure feedforward and feedback ones, as are shown in Fig6. Both pure feedback and feedforward control fail to follow the slip speed reference. Under the pure feedback one, the slip speed falls to zero quickly and its derivative fails to meet the *GV no-lurch condition* at the synchronization point. As a result, significant torque fluctuations are observed on the output shaft. For the pure feedforward control, since model simplification is applied in derivation of the control law, the tracking error is quite large at the end of the interval and there are also output torque fluctuations. On the other hand, if the feedforward and the feedback control are combined, the slip speed reference is then well followed and the output torque is successfully kept constant and smooth.

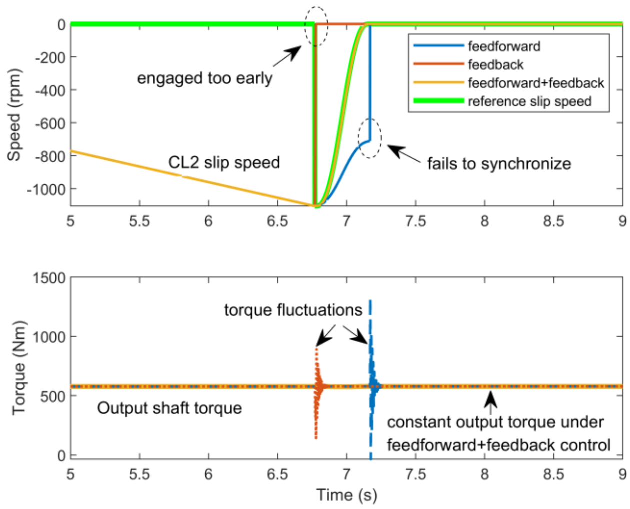

Fig6: Comparison with pure feedforward and feedback controller

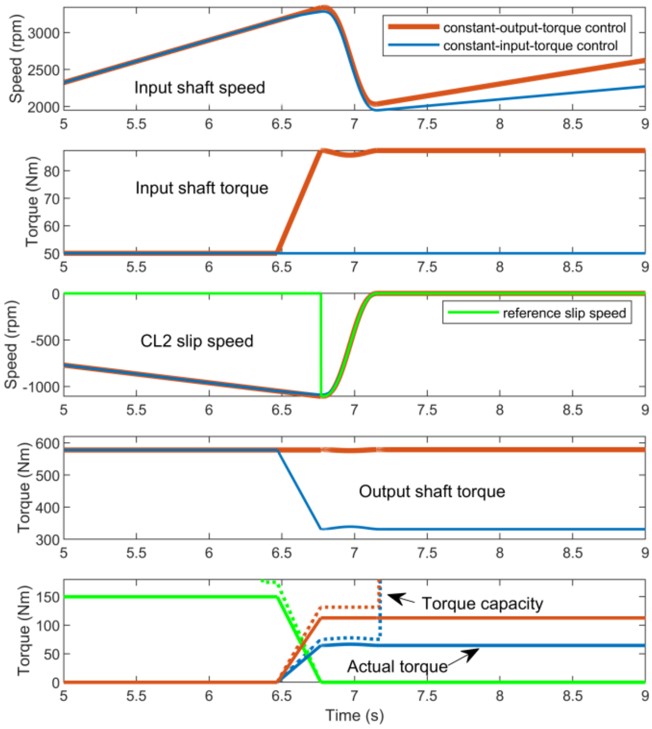

Fig7: Comparison with constant-input-shaft torque control

Simulations above are based on the nominal model, i.e. there are no disturbances. Necessity of the disturbance observer is evaluated by adding a friction torque (1.5 Nm) on the input shaft. Performances of the three controllers are compared, namely the conventional proportional-integral control (PI), the feedforward-feedback control, and the feedforward-feedback control with disturbance compensation. Results are shown in Fig8. As can be seen, all the three controllers follow the slip speed reference generally well. However, at the synchronization point (when the rotational speed of the on-coming clutch decreases to zero), slip speed from the feedforward-feedback controller with DO comes with a more "gentle landing" (derivative of the slip speed has a smaller magnitude) than those of the other two. Consequently, the driving torque on the output shaft is smoother and without a sudden change (though it is small for PI and feedforward-feedback without DO). Therefore, for practical engineering use where there is bound to be unknown disturbances and uncertainties, DO should always be integrated in the controller.

5 Conclusions

A feedforward-feedback shift controller with disturbance compensation for a two-speed transmission for electric vehicles has been developed, with the aim of ensuring constant output torque.

Control strategy for the torque phase is derived using the generic dual-clutch model. For the inertia phase, the slip speed is commanded to follow a 3-4-5 polynomial, with the feedforward-feedback control. The feedforward component is attained through the inverse dynamic model. The feedback part is the proportional control. To improve robustness, a disturbance observer is integrated.

Effectiveness and performances of the controller are validated through simulations. Comparison with the conventional constant-input-torque control shows that the proposed method comes with advantages of keeping the driving torque on the output shaft constant. Robustness of the controller is confirmed through comparative simulations involving other non-disturbance-compensation methods. Thus, the controller is suitable for practical engineering use, especially on high-performance electric vehicles.

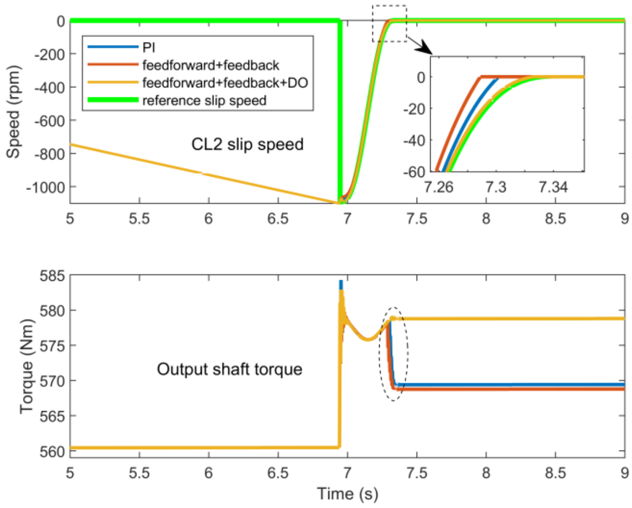

Fig8: Comparison with non-DO-based controllers

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Authors

Jianfeng Huang is a PhD candidate in Shanghai Jiao Tong University. His research interest includes configuration design, parameter optimization, and gear shift control of different kinds of transmissions for electric vehicles and hybrid electric vehicles.

Jianlong Zhang is an associate professor of vehicle engineering, Shanghai Jiao Tong University. His major research interests are powertrain control, coordinated control of electric vehicles and automotive software engineering. He participated in and conducted tens of natural funded projects, ranging from the national sci-tech support plan, the 863 plans, to the 985 plans. He is also an active contributor to several international programs. He has received a number of awards including one title for the Provincial and Ministerial Scientific Achievements, two for the Science and Technology Progress. He has published more than twenty papers and one monograph.

Wei Huang was born in Anhui, China, in 1993. He received the B.S. degree in vehicle engineering from Xi'an Jiao Tong University, Xi'an, China, in 2014. He is currently working toward the Ph.D. degree with the School of Mechanical Engineering, Shanghai Jiao Tong University, Shanghai, China. His research interests include dynamics and control for advanced powertrain systems.

Chengliang Yin was born in Inner Mongolia, China, in 1965. He received the Master and Ph.D. degree in vehicle engineering from Jilin University, China, in 1996 and 2000, respectively. He is currently a Full Professor of Shanghai Jiao Tong University, where he is also the Vice Dean of Institute of Automotive Engineering as well as the Vice Director of National Engineering Laboratory for Automotive Electronic Control Technology. His research interests include control of automotive electronics, electric vehicles, especially the research and development of hybrid electric vehicles.

Lifang Wang received her Bachelor, Master and Ph.D. degree from Jilin University, China, in 1992, 1994 and 2007, respectively. She is currently a Full Professor of the Institute of Electrical Engineering of Chinese Academy of Sciences, where she is also the Vice Dean of key Laboratory of power electronics and transmission. Her research interests include electric vehicle control, lithium-ion power battery system, wireless charging and electromagnetic compatibility technology.