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Study of Seamless Gear-Shift Strategy for a Clutchless Automated Manual Transmission

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***Abstract*—A new gear-shift control strategy for an electric bus clutchless automated manual transmission (AMT) is studied. This control method is developed for improving gear-shift efficiency and reducing torque interruption during gear change. The dynamic of the driveline is described and the operation details of synchronizer are studied. A new layout of the transmission is modified from a conventional AMT that enables to control the adjacent gear-shifting actuators independently and simultaneously. This modified AMT could apply the new gear-shift control strategy. Simulation results are demonstrated that synchronous time of the synchronizer is reduced during gear-shift through the new gear-shift control strategy.**

I. Introduction

The transmission is a vehicle principal component that allows the driver to select different gear ratios and transmits the engine or motor torque to the wheels [1]. Recently, advanced transmission technologies improved the driving comfort, fuel efficiency and achieving higher speed and acceleration of vehicles such as Automatic Transmission (AT), Continuously Variable Transmission (CVT), Dual Clutch Transmission (DCT), Automated Manual Transmission (AMT) [2-9]. By using an AT, vehicles can run smoothly during gear-shifting due to AT has a torque converters to transmit engine torque. CVT can be varied continuously while driving within a given range of the ratio. Compared to other automatic transmissions, AMT has higher transmission efficiency which offers more direct connection between the engine/motor and driveline [5]. In addition, AMT has lower cost due to a modification of conventional manual transmission.

On the other hand, DCT often has faster gear shifting with two parallel transmission shafts and clutches structurally compared to the AMT. The two transmission shafts contain the odd gears and even gears so that they able to process gear engagement and release simultaneously [7-8]. Furthermore, the DCT's gear shift quality can be improved without interrupted torque by well-controlled engagement and release between the two clutches. By comparison with DCT, AMT may have shift quality deficiency due to torque interruption during gear selection and engagement. Anyhow, both of DCT and AMT obliged to utilize an intelligence system including gear-shift strategy and control management to control the actuator mechanism through the electric or hydraulic system.

In this paper, a seamless gear-shift strategy is applied the clutchless AMT of electric bus. The structure of EV powertrain system is introduced in Section II. The dynamic analysis and modeling of synchronizer is discussed to realize characteristic of powertrain system and synchronizer system. Section III describes the seamless gear-shifting strategy for an AMT without clutch using a novel gear-shifting actuator control strategy to achieve a seamless gear-shift process. The simulation results of common gear-shift strategy are shown in Section IV in comparison with the proposed method. Finally, a conclusion is drawn in Section V.

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II. The Automated Manual Transmission without Clutch

A. Transmission System

A powertrain layout of the AMT without clutch is shown in Figure 1. A 120 kW power motor connects directly with transmission. The transmission has 3-speed gear ratio with two gearshift actuators and two synchronizers. As shown in Figure 1, the 1st gear and 3rd gear are utilized a gearshift actuator and a synchronizer. Besides, another gearshift actuator and synchronizer serve for the 2nd gear independently. Most of the manual and automated manual transmissions are arranged that one synchronizer may be allowed to lock only one gear at any one time. However, as the transmission layout of the AMT shown in Figure.1, all gearshift actuators could controlled independently at any time which could make the gear-shifting more flexible and unrestrainedly. Moreover, transmission gears are separated into odd gears and even gears for each gearshift actuator and synchronizer is responsible to process gear synchronization and gear release simultaneously during gear change. The modified AMT layout offer chance to achieve a seamless gear-shifting by using a novel shift control strategy.

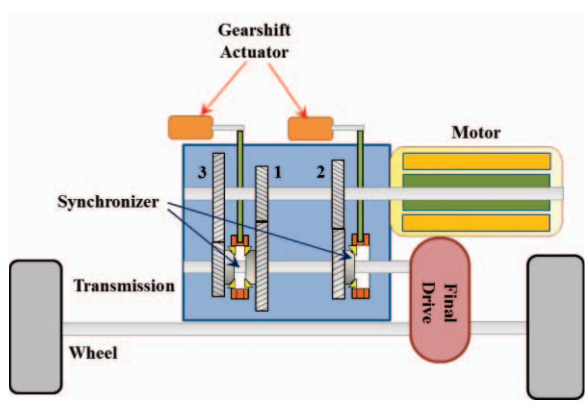

Figure 1. Sketch of a 3-speed AMT without Clutch.

B. Driveline Modeling

In this section, a mathematic model is built concerning a vehicular driveline consists of power motor, transmission,

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final drive, and wheels. The Newton's second law is used to derive the dynamic models of electric bus clutchless AMT.

Figure 2 shows the forces acting on a vehicle in the longitudinal direction. The following relations between friction forces and driving force can be written as

$$F_w = m_v a + (F_a + F_r + F_g) \tag{1}$$

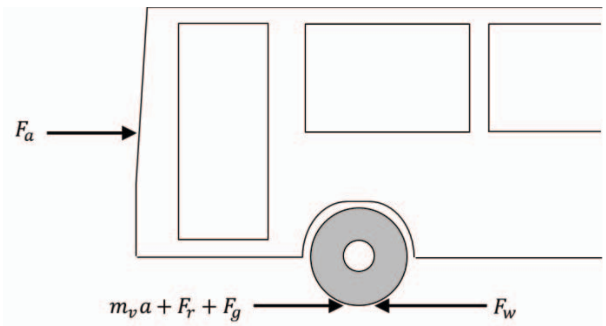

Figure 2. Driving Forces acting on a vehicle

The friction force F_w represented by the sum of the accelerating resistance $m_v a$, aerodynamic drag F_a, rolling resistance F_r, and gradient resistance F_g. The aerodynamic drag is approximated by

$$F_a = \frac{1}{2}\rho A C_d v^2 \tag{2}$$

Where ρ is the air density, A is the vehicle's cross-sectional area, C_d is the drag coefficient, and v is the longitudinal velocity of vehicle. The rolling resistance is approximated by

$$F_r = \mu m_v g \tag{3}$$

Where μ is the rolling resistance coefficient and g is the gravitational acceleration. The gradient resistance is approximated by

$$F_g = m_v g sin\theta \tag{4}$$

By considering the accelerating resistance $m_v a$, the acceleration behavior of vehicle can be obtained by rewritten as

$$m_v a = F_d - (F_a + F_r + F_g) \tag{5}$$

Where m_v is the mass of vehicle, a is vehicle's acceleration, and F_d is the driving torque of wheels. The relation between longitudinal direction and forces acting on the vehicle is shown as above. Without consideration of tire slip, the accelerating resistance can be described by

$$m_v a = m_v R_w \dot{\omega}_w \tag{6}$$

Where R_w is the radius of wheel and $\dot{\omega}_w$ is the angular acceleration of wheel. Therefore, the total of resistance torque F_w can be defined as driving torque F_d.

$$F_w = F_d \tag{7}$$

In order to determine an effective model of transmission, an abbreviated equation can be approximated by the following assumptions:

1) The gears of transmission have no backlash.
2) The torsional flexibility would not be considered.
3) Without considering viscous damping of transmission.

With those assumptions, the driveline model can be simplified as below

$$J_i \dot{\omega}_i = T_m - b_m \omega_i - \frac{T_c}{n_g} \tag{8}$$

$$J_o \dot{\omega}_o = T_c - \frac{F_w R_w}{n_f} \tag{9}$$

The equations (8) and (9) are the dynamic model of input shaft and output shaft, respectively, where J_i is the moment of inertia converted to the transmission input shaft and power motor; J_o is the moment of inertia converted to the output shaft of transmission, $\dot{\omega}_i$and $\dot{\omega}_o$ are the angular acceleration of input shaft and output shaft, respectively; T_m is the power motor output torque; b_m is the viscous damping coefficient of power motor; ω_i is the angular velocity of the input shaft which is connected directly with the power motor; T_c is the synchronizer's cone torque; $n_g$andn_f are the transmission current gear ratio and final drive ratio, respectively; $F_w R_w$ is the resistance torque of vehicle.

Further, substituting (1) into (9), the model of output shaft can be determined as following

$$(J_o + \frac{m_v R_w{}^2}{n_f{}^2})\dot{\omega}_o = T_c - \frac{F_a R_w - F_r R_w - F_g R_w}{n_f} \tag{10}$$

Finally, by combining with (8), a simplified driveline model determined as eq. (11). While the target gear was engaged, the dynamics of the entire driveline can be described as

$$(n_g{}^2 J_i + J_o + \frac{m_v R_w{}^2}{n_f{}^2})\dot{\omega}_o = (T_m - n_g b_m \dot{\omega}_o) n_g - T_L \tag{11}$$

Where T_L is the resistance torque consisting of aerodynamic drag torque, rolling resistance torque, and gradient resistance torque.

C. Synchronizer

Synchronizer is one of the key component of transmission. Mostly modern MT, AMT, and DCT vehicle has a synchronizer mechanism. In order to engage target gear and make the gear shafts to reach the same speed, synchronizer provide the friction torque be known as cone torque, which brings the gear to the same speed and engage the new gear. All types of synchronizers mostly adopt the friction clutches due to a high torque transmittal capacity. A double-cone synchronizer is utilized in transmission for this study as shown in Figure 3.

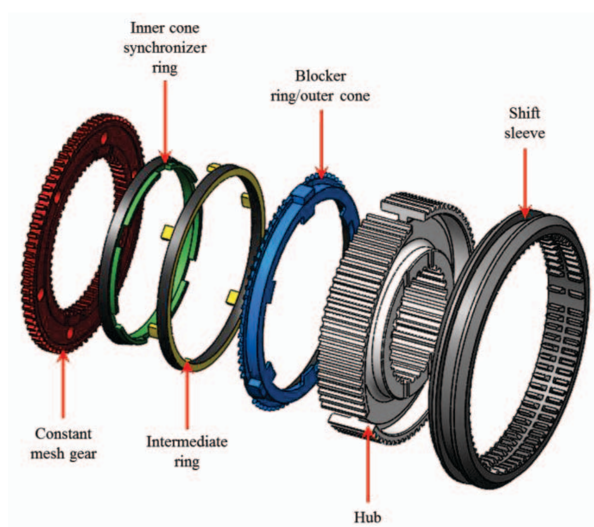

Figure 3. A double-cone synchronizer

Double-cone synchronizer is one of a type of multi-cone synchronizers that can provide more interfaces by using an extra cone surface. It is consists a shift sleeve, hub, blocker ring, intermediate ring, inner ring and mesh gear. A cross-section of a double-cone synchronizer in a neutral position is shown in Figure 4.

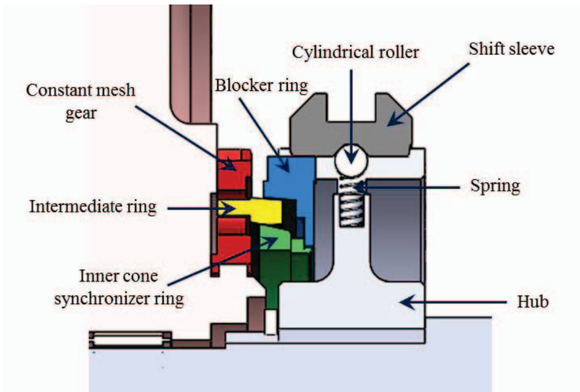

Figure 4. Cross-section of a double-cone synchronizer

A cylindrical roller attached by a spring is used as a strut to provide the initial load for indexing the ring. It arranged by spring-loaded against a groove in the shift sleeve tooth.

A working process of the synchronizer is discussed from the neutral position to the final engagement.

The operation of synchronization is shown in Figure 5. This can be divided into six phases upon to the sleeve axial displacement:

1) First free fly

The shift sleeve moved out of the neutral position by an axial force F_x overcoming the spring-loaded against the roller.

2) Start of the synchronization

Due to the contact between strut and blocker ring, a cone torque T_c build up and pre-synchronization is in progress.

3) Major synchronization

The sleeve move axially through the distance x_2 with the blocker ring in the indexed position and the cone torque provide rapidly.

4) Turning the blocker ring & second free fly

When the angular velocity of mesh gear relative to the blocker ring approaches zero, the cone torque drops to zero, the sleeve is allowed turning the blocker ring splines and move forward due to the index torque T_i which provide by the chamfers of sleeve and blocker ring is now greater than cone torque.

5) Second bump

The sleeve moves forward from the previous position to the mesh gear tooth during second free fly. Another index torque build up while mesh gear tooth block the sleeve which to moves the mesh gear aside to passes through

6) Final free fly

The sleeve completes the lockup and recover the power torque while it is reaches its final position.

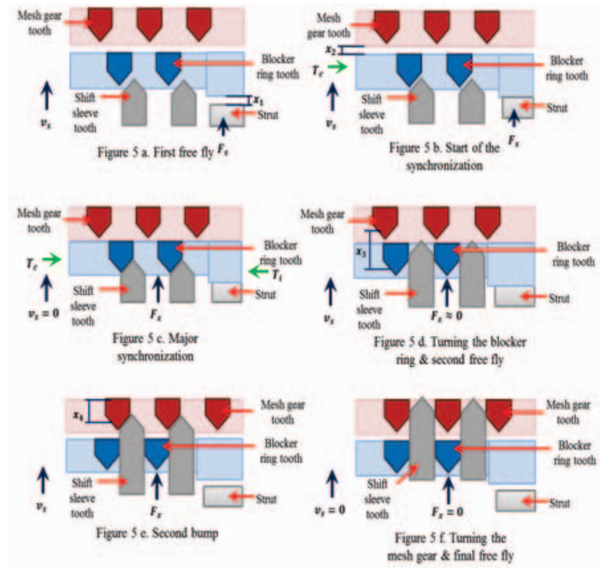

Figure 5. Common synchronization process of a double-cone synchronizer

D. Frictional Torque and Index Torque

A frictional torque can be provided while the sleeve moved out from the neutral position and then contacted to the blocker ring. It will make the speed difference reduction between the gears during synchronization phase.

$$T_f = F_x\left(\frac{R_{c1}\cdot\mu_{c1}}{\sin\alpha_1} + \frac{R_{c2}\cdot\mu_{c2}}{\sin\alpha_2}\right) \quad (12)$$

Where, F_x is the axial force acting on shift sleeve, R_{c1}, R_{c2} respectively the mean cone radius of blocker ring and inner ring, μ_{c1}, μ_{c2}, respectively the dynamic coefficient of friction between blocker ring/inner ring and cone. α_1, α_2, respectively the cone angle of blocker ring and inner ring.

Figure 6 shows that T_f occurs to act on input shaft connecting with mesh gear and output shaft connecting with sleeve and hub. In other word, T_f will bring a driving force on the output shaft and the reacting force can simultaneously make the rotation speed of input shaft slow down.

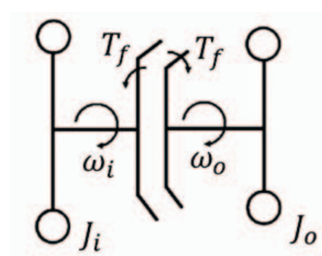

Figure 6. Sketch of synchronizer

Index torque T_i builds up while axial force F_x is acted through the sleeve spline tooth against the blocker ring tooth.

$$T_i = F_x R_{slt}\left(\frac{\cos\frac{\beta}{2} - \mu_{sl}\sin\frac{\beta}{2}}{\sin\frac{\beta}{2} + \mu_{sl}\cos\frac{\beta}{2}}\right) \quad (13)$$

Where, R_t is the pitch radius of mesh gear and blocker ring, β is the angle of tooth at the pitch radius. In order to prevent sleeve pass through the blocker ring before end of synchronization phase, T_f must be greater than index torque at the time.

III. A Seamless Gear-Shift Strategy

In order to formulate a seamless shift control strategy for the transmission system, clarification of the synchronizer's dynamic is essential. The dynamic of synchronizer is discussed in Section II. A common gear-shift control process shown in Figure 7. Mostly, the power motor torque will be cut off while a gear-shift starts, then an axial force of the outgoing gear's actuator F_{x1} acts to disengage the gear. An axial force of oncoming gear's actuator is propelled to start the synchronization and engagement, once the neutral position of outgoing gear is confirmed. The following processes of gear-shifting are described in Section II. The blue frame and red frame of solid line shown in Figure 7 represent the motions of oncoming gear and outgoing gear, respectively. Moreover, the orange dotted lines means the output shaft of transmission is operated in torque interruption state. The purple dotted lines signifies a few driving torque is applied on output shaft by oncoming gear's actuator.

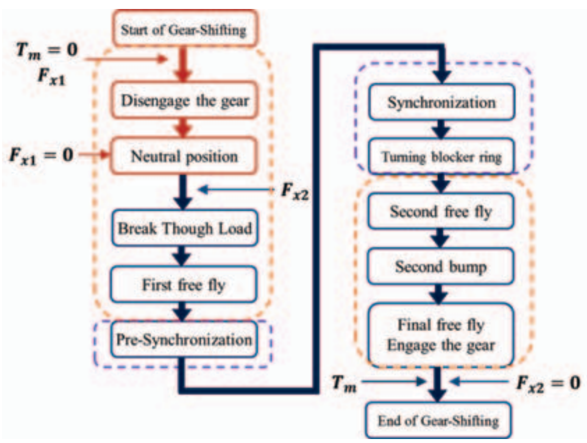

Figure 7. A Common Gear-Shift Control Process

For reducing the torque interruption of driveline during gear shift, a new gear-shift control strategy is formulated as shown in Figure 8. In equipped with independently controllable synchronizer between adjacent gears, the engaged procedure of oncoming gear can be pre-executed before the outgoing gear has not been disengaged. As the engaged procedure of oncoming gear is progressed to the main synchronization phase, the output torque of power motor will be suspended and meantime the disengagement procedure of outgoing gear is accomplished. A few driving torque resulted from cone frictional torque T_f can be supplied to the output shaft of transmission during the main synchronization phase. After the sleeve's teeth pass though the blocker ring, the power torque recovers partially while sleeve is engaging the mesh gear. Finally, the power torque recovers entirely if target gear is engaged completely. Figure 9 shows that the direction of torque acts on the shafts during the gear-shifting.

IV. Simulation and Discussion

This section presents a computer simulation of the proposed gear-shift control scheme. To demonstrate the proposed control scheme's performance, some simulations of the proposed control scheme are compared with those of common gear-shift method. Based on the MATLAB software, the parameters of double-cone synchronizer used in the simulation to analyze the above discussed control strategy are listed in Table I. The configuration of electric bus clutch-less AMT consisted of 8000kg mass bus body, 120kW Power motor with 46kg-m rated torque, and a three-speed MT gear box with 1st gear ratio 3.71, 2nd gear ratio 2.48, 3rd gear ratio 1.353, and final gear ratio 4.777.

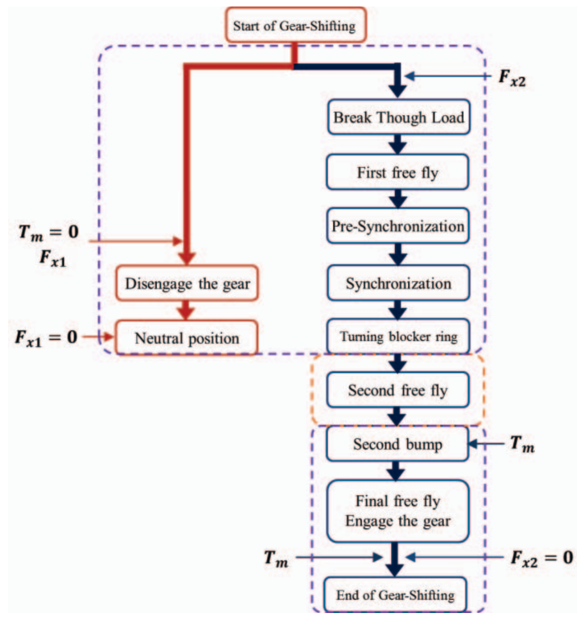

Figure 8. A seamless gear-shift control strategy

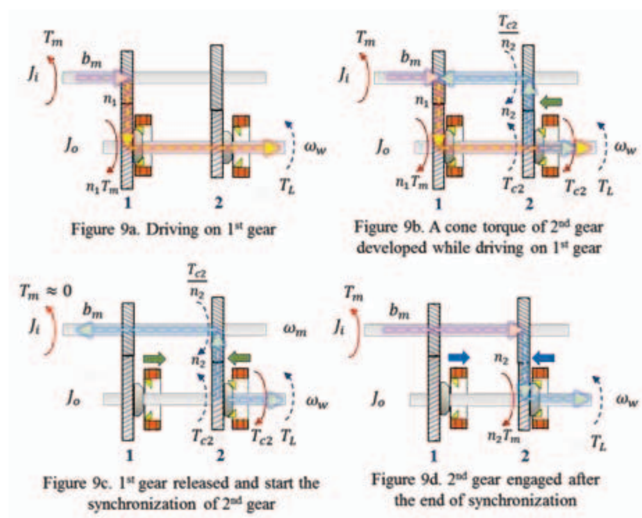

Figure 9. Status of torque transmission during the gear-shifting

As the gear-shifting quality is directly related to the driving comfort of the vehicle, a significant improvement can be attained by reducing the gap of the torque interruption through shortening the gear-changing time. As shown in Figure 10, seamless gear-shift strategy provided a less torque interruption than the common gear-shift control for the upshift from 1st to 3rd gear.

TABLE I. DOUBLE-CONE SYNCHRONIZER PARAMETERS

Parameter	Value
Cone Angle of Blocker Ring (α_1) & Inner Ring (α_2)	6.3°
Mean Cone Radius of Blocker Ring (R_{c1})	0.077 mm
Mean Cone Radius of Inner Ring (R_{c2})	0.07 mm
Mean Chamfer Radius of Sleeve (R_{slt})	0.0875 mm
Chamfer Angle of Sleeve (β)	120°
Coefficient of Friction, Cone Surfaces (μ_c)	0.12
Coefficient of Friction, Sleeve's teeth (μ_{sl})	0.1

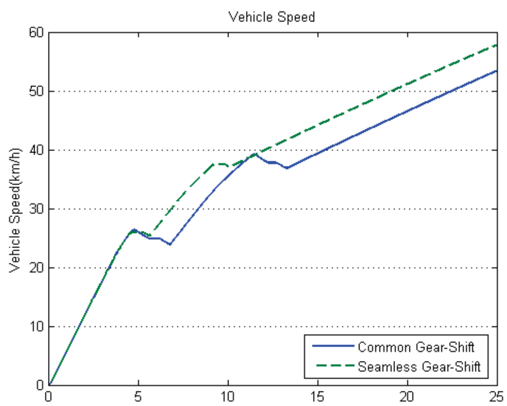

Figure 10. Test result of a upshift from the 1st to 3rd gaers for a comparison bewteen the speed dynamic response of the proposed seamless control scheme and the common control approach

The simulation results of gear position movement and the command of motor duty are shown in Figure 11. The gear shift period of seamless gear-shift control is shorter than the common gear-shift control because the disengagement of outgoing gear is delayed to execute until oncoming gear reaches the synchronization phase.

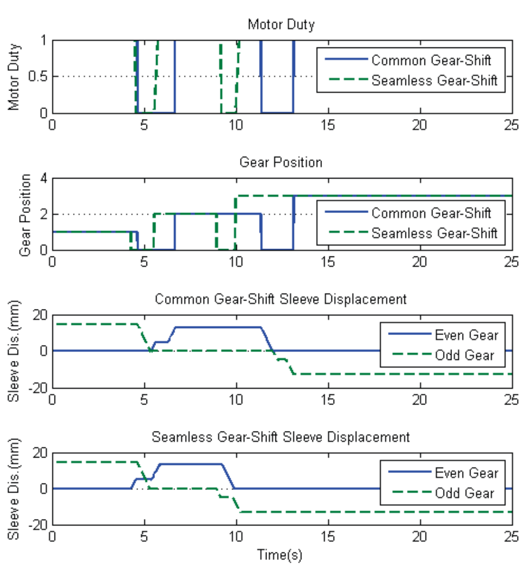

Figure 11. Simulation results of gear operation

A cone torque and index torque act on the shaft and gear to accomplish the synchronization during the gear-shifting. Although the seamless gear-shift control performs a faster gear-shifting than common gear-shift control, it exists some disadvantage events shown in Figure 13. A longer period of cone torque acting on the synchronizer may cause the durability loss. In addition, a high load bump rises greatly due to a torque recovery during the gear engagement on final phase. It may damage the teeth of sleeve and mesh gear.

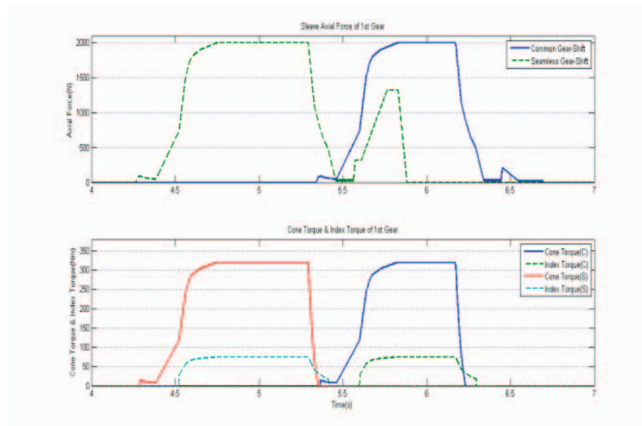

Figure 12. Simulation results of shift axial force and cone torque

V. CONCLUSION

In this paper, simplified mathematical models of driveline and synchronizer are derived and the gear-shifting dynamics are analyzed. A seamless gear-shift strategy for clutch-less AMT was formulated and the simulations verified the torque interrupt period of gear-shifting is reduced significantly if the proposed gear-shift strategy is applied.

The shift quality should take into account the sense of accelerating frustration (i.e. jerk); besides, the efficiency of the synchronizer is also needed to evaluate in the entire shifting procedure. Hence, this is an important topic for the future on analysis with regard to the slipping work, friction characteristics and heat load characteristics of synchronizer during gear shift. In addition, an experimental test rig is required to verify the feasibility of the new gear-shift control strategy.

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REFERENCES

[1] UMESH WAZIR, "Manual Gearbox Synchronizers – An Overview," International Journal of Emerging trends in Engineering and Development, vol. 5, pp. 422–428, 2013.

[2] Aldo Sorniotti, Gabriele Loro Pilone, "A Novel Seamless 2-Speed Transmission System for Electric Vehicles: Principles and Simulation Results," SAE International, 2011-37-0022, 2011.

[3] M.S. Rahimi Mousavi, B. Boulet, “Modeling, simulation and control of a seamless two-speed automated transmission for electric vehicles,” American Control Conference (ACC), IEEE 2014, pp. 3826–3831.

[4] B. Gao, Q. Liang, Y. Xiang, L. Guo, H. Chen, “Gear ratio optimization and shift control of 2-speed I-AMT in electric vehicle,” Mech. Syst. Signal Process., vol. 50, pp.615–631, 2015.

[5] Laszlo LOVAS, Daniel PLAY, “Modeling of Gear Changing Behavior,” PERIODICA POLYTECHNICA SER. TRANSP. ENG. vol. 34, no. 1-2, PP. 35–58, 2005.

[6] Mir Saman Rahimi Mousavi, Ali Pakniyat, Tao Wang, Benoit Boulet, “Seamless dual brake transmission for electric vehicles: Design, control and experiment,” Mechanism and Machine Theory, vol. 94, pp.96-118, 2015.

[7] Manish Kulkarni, Taehyun Shim, “Shift dynamics and control of dual-clutch transmissions,” Mechanism and Machine Theory, vol. 42, no.2, pp.168-182, 2007.

[8] Jiwon J. Oh, Seibum B.Choi, “Driveline modeling and estimation of individual clutch torque during gear shifts for dual clutch transmission,” vol.24, no. 5, pp. 449-463, 2013.

[9] Hui-Long Yu, Jun-Qiang Xi, “Research on Gear Shifting Process without Disengaging Clutch for a Parallel Hybrid Electric Vehicle Equipped with AMT,” Mathematical Problems in Engineering, Vol. 2014, Article ID 985652, pp.1-12, 2014.