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Analysis of Dynamic Characteristics of Clutch-to-Clutch Shifting Control of an Automatic Transmission

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Abstract: The fact that requirements for shift quality in automatic transmissions have been increasing rapidly necessitates the establishment of a suitable shifting control strategy in order to facilitate smoothness of different processes. For this very purpose, this paper introduces a simulation model of an 8-speed automatic transmission for front-drive vehicles with respect to detailed shifting strategies and relative parameters. An impact function can be used to reduce the transmitted torque of the oncoming shift elements before synchronization point in order to damp the impact and thus make the gear shifting process more smooth. This paper makes a systematic introduction of the structure of 8AT, theoretical basis of control strategy, the establishment of the simulation model and the comparison between test results and simulation results. The conclusion shows that with parameters well calibrated, engine torque transferring and speed synchronizing process will be smoother, which helps realizing the ultimate goal of better shift quality with higher efficiency, lower shift loads and improved shifting comfort.*Copyright © 2014 IFSA Publishing, S. L.*

Keywords: Automatic Transmission, Dynamic simulation, Shifting process, Shifting Control, Control strategy.

1. Introduction to Shengrui 8 Speed Transmission

The automatic transmission is currently the most popular type of transmission for its good performance and fuel economy. In general, automatic transmission consists of multi-gear and shifting elements. The worldwide first front-drive 8-speed automatic transmission is developed by Weifang Shengrui Transmission Corporation Limited together with Chemnitz University of Technology, Beihang University and Ricardo UK, Ltd. The 8AT consists of a hydrodynamic torque converter, a 2-shaft gear system, three planetary gears, three transfer gears and five shifting elements (see Fig. 1). It is able to achieve 8 forward gears and 1 reverse gear by engaging three shifting elements each time. In order to improve shift quality and prevent negative feelings of gear shifting or braking, a better control strategy is crucial and worthy of our focus. The shifting logic showed in Fig. 2 indicates that in each gear, only two of the five shifting elements are disengaged and the gear can be shifted with one element being engaged while another being disengaged simultaneously, which is called “clutch-to-clutch shifting” [1,2]. There are mainly four types of shifting strategies: power on upshift, power on downshift (with engine torque being positive), power off upshift and power off downshift (with engine power being negative).

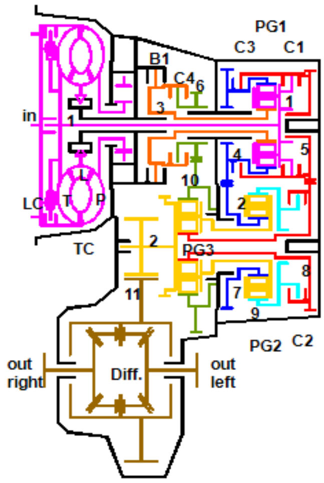


Fig. 1. Structure of 8-speedtransmission.

gear	B1	C1	C2	C3	C4
R	■		■		■
NR			■		■
N1		■			■
N2				■	■
1	■	■			■
2	■			■	■
3		■		■	■
4			■	■	■
5		■	■		■
6		■	■	■	
7	■	■	■		
8	■		■	■	

Fig. 2. Shifting logic of 8AT.

In this paper, we take the power on upshift from the 3rd gear to the 4th gear and the power on downshift from the 8th gear to the 7th gear as examples to analyze control results and the comparison to the simulation results based on the assumption that the torque converter is locked up.

2. Theoretical Basis of the Shifting Process

Fig. 3 shows the simplified model of the vehicle powertrain, where clutch C_2 is the off-going clutch and C_1 is the on-coming one [3].

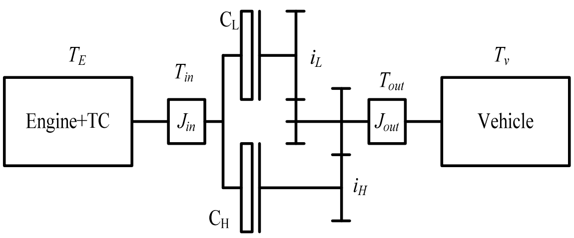


Fig. 3. Simplified model of the vehicle powertrain.

2.1. Torque Transmitted by the Shifting Elements

The torque calculation of the shifting elements depends on whether it is slipping or closed. Take clutch C_1 for example, equation 1 applies when both halves of the clutch are coupled through a friction torque in different rotary speed [3].

$$T_{C1} = \mu_d p_1 A_1 r_1 z_1 \cdot \mathrm{sgn}(\omega_1 - \omega_2) \tag{1}$$

When both halves are locked and rotating in the same speed:

$$\begin{aligned} T_{c1} &= T_e - J_{in} a_{in} - T_{c2} \le \mu_s p_1 A_1 r_1 z_1 \\ &= TorqueCapacity \end{aligned} \tag{2}$$

where μ_d is the dynamic friction coefficient; μ_s is the static friction coefficient; p_1 is the clutch pressure; A_1 is the area of the piston; r_1 is the equivalent friction radius; z_1 is the number of the friction plates; ω_1, ω_2 are the angular speeds of the active side and passive side in the clutch C_1; T_e is the engine torque; J_{in} is the inertia of the transmission input side; a_{in} is the angular acceleration speed of the input shaft of the transmission.

The relationship between the transmitted torque and the engine torque is shown in Fig. 4. If the actual transmitted toque exceeds the limit of the torque capacity in equation 2, the clutch begins to slip.

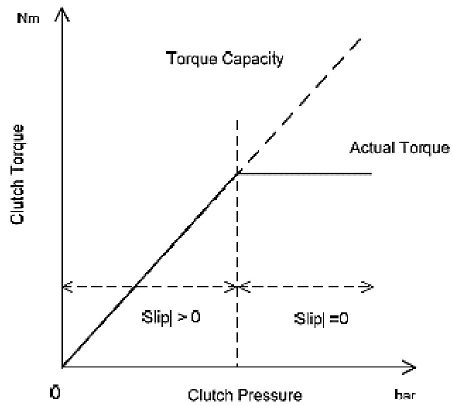
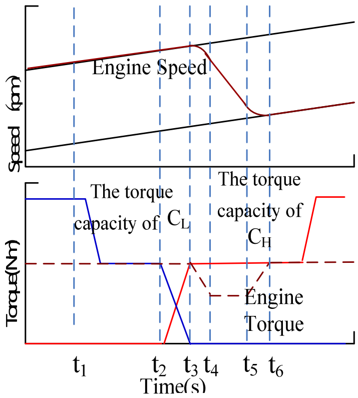


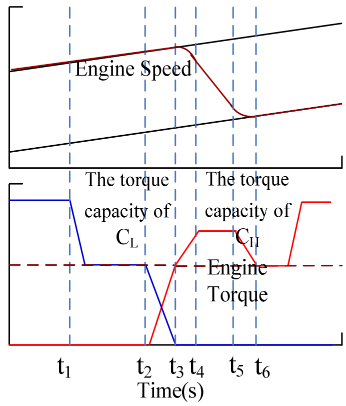
Fig. 4. The relationship between the torque capacity and the actual torque.

2.2. Shifting Process of Power-on Upshift

Power-on upshift refers to the shifting process with output engine torque greater than zero during which the engine speed keeps increasing before it reaches the upshift speed line and causes the transmission to shift up. It is the most common shifting type. The clutch control strategy of power-on upshift consists successively of oil-filling phase, torque phase, speed phase and locking phase (Fig. 5).



(a) with engine torque intervention



(b) without engine torque intervention

Fig. 5. Control strategy for power-on upshift.

1) Oil-filling phase ($t_1 - t_2$).

When control system gives the shifting order at t_1, according to Equation 3, the off-going clutch C_1 starts to reduce its torque capacity at a certain fixed gradient through *PI* control of the oil pressure until the torque capacity becomes smaller than its actual transferred torque and causes a micro-slip. The on-going clutch C_2 reaches the required torque capacity at a very low oil pressure.

$$\begin{cases} T_{C2}(t) = 0^{+} \left(t_1 < t \le t_2\right) \\ T_{C1}(t) = T_E(t) - I_E a_E(t) \left(t_1 < t \le t_2\right) \end{cases}, \tag{3}$$

where $T_{C1}(t)$ is the torque capacity of clutch C_1, $T_{C2}(t)$ is the torque capacity of clutch C_2, $T_E(t)$ is engine torque capacity, I_E is the inertia of the engine, $a_E(t)$ is the angular acceleration of engine output shaft.

2) Torque phase ($t_2 - t_3$).

During this phase, the off-going clutch C_1 transfers its torque to the on-coming clutch C_2, and the transmission still works at the current lower gear ratio. In order to maintain the smoothness and the stability of the engine speed, the torque capacities of $C_1 C_2$ follow Equation 4.

$$\begin{cases} T_{C2}(t) = k_{on}(t)\left(t - t_2\right)\left(t_2 < t \le t_3\right) \\ T_{C1}(t) = T_E(t) - I_E a_E(t) \\ -T_{C2}(t)\left(t_2 < t \le t_3\right) \\ k_{on}(t) = f\left(T_E, t_s\right)\left(t_2 < t \le t_3\right) \end{cases} \tag{4}$$

where $k_{on}(t)$ is the gradient of torque increase of the on-coming clutch C_2.

In torque phase, if the torque capacity of the off-going clutch C_1 decreases at a much faster speed than the torque capacity of the on-going clutch C_2 increases, the transmission will generates a shock because of over-constraint; on the other hand, if the torque capacity of the off-going clutch C_1 decreases too slowly, the engine load will reduce as a result and cause the engine speed to jump. Therefore, a closed-loop slip control over clutch C_1 is needed for a smooth shifting process.

3) Speed phase ($t_3 - t_6$).

During this phase, engine speed decreases from the current gear to the target gear. Clutch C_1 is considered to be disengaged and clutch C_2 transfer all the torque from the input shaft. Equation 5describes the control requirements of the torque capacities of clutch C_1 and C_2 at t_3.

$$\begin{cases} T_{C2}(t) = T_E(t) - I_E \cdot a_E(t) \\ T_{C1}(t) = 0 \end{cases} \tag{5}$$

For the purpose of assisting slip control of the clutch and reducing the impact of shifting, an Engine Management System (EMS) is needed to receive the torque reduction signal as well as the quantitative value from transmission ECU through CAN-bus [10]. The target value of engine torque reduction can be calculated by equation 6.

$$T_{E\_int}(t) = \begin{cases} T_E(t) + I_E a_s \left(1 - f_1\left(\frac{t - t_3}{t_4 - t_3}\right)\right) \\ (t_3 < t \le t_4) \\ T_E(t) + I_E \cdot a_s \\ (t_4 < t \le t_5) \\ T_E(t) + I_E a_s f_1\left(\frac{t - t_5}{t_6 - t_5}\right) \\ (t_5 < t \le t_6) \end{cases} \tag{6}$$

4) Locking phase ($t > t_6$).

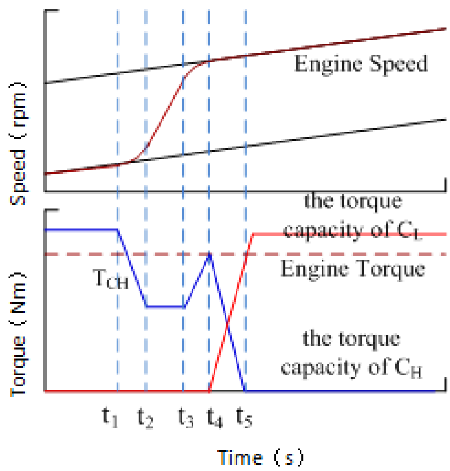
During this phase, clutch C_1 is completely disengaged. Clutch C_2 is engaged to transfer torque given by Equation 7.

$$\begin{cases} T_{C2}(t) = T_E + T_{off}\,(t > t_6) \\ T_{C1}(t) = 0(t > t_4) \end{cases} \tag{7}$$

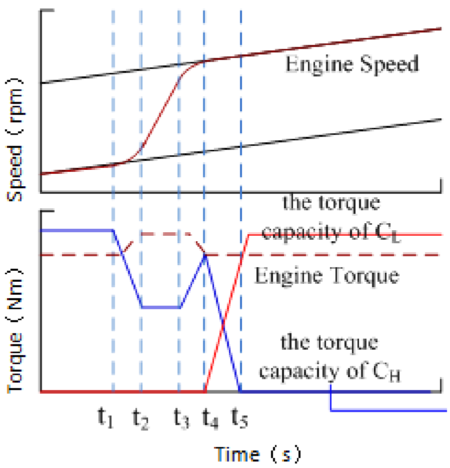
where T_{off} is the increased torque transferred by clutch C_2.

2.3. Shifting Process of Power-on Downshift

The torque swap and speed synchronizing phases of power-on downshift is in the reverse order to that of power-on upshift. The clutch control consists successively of speed phase, torque phase and locking phase. Details are shown in Fig. 6.



(a) without engine torque intervention



(b) with engine torque intervention

Fig. 6. Control strategy of power-on downshift.

1) Speed phase ($t_1 - t_4$).

During this phase, the torque capacity of the off-going clutch C_2 reduces from the engaging value to the value of t_2-t_3 shown in Equation 8, which causes the speed of the input axle to adjust to the speed line of the lower gear at the given acceleration rate. Clutch C_2 starts to micro-slip at t_1 and turns into the sliding state with the slip value slowly decreasing at the acceleration rate $-\alpha_S$. A buffer function is adopted to smoother the speed turning point both in t_1-t_2 period and t_3-t_4 period in order to mitigate the impact of the system.

$$\begin{cases} T_{C1}=0^{+} \\ T_{C2}=\begin{cases} T_E(t)-I_E\cdot a_E(t_1)+ \\ f_1\left(\dfrac{t-t_1}{t_2-t_1}\right)I_E a_s(t_1<t\le t_2) \\ T_E(t)-I_E\cdot a_E(t)(t_2<t\le t_3) \\ T_E(t)-I_E\cdot a_E(t_1)+ \\ \left(1-f_1\left(\dfrac{t-t_3}{t_4-t_3}\right)\right)I_E a_s(t_3<t\le t_4) \end{cases} \end{cases} \tag{8}$$

2) Torque phase (t_4 - t_5).

The speed of the input axle has been increased to the speed of the target gear at the beginning of the torque phase. The torque transferred by clutch C_1 and C_2 is described by Equation 9. This process is similar to that of power-on upshift.

$$\begin{cases} T_{C1}(t)=k_{on}(t)\cdot(t-t_4)(t_4<t\le t_5) \\ T_{C2}(t)=T_E(t)-I_E a_E(t)- \\ T_{CL}(t)(t_4<t\le t_5) \\ k_{on}(t)=f(T_E,t_s)(t_4<t\le t_5) \end{cases} \tag{9}$$

3) Locking phase ($t > t_5$).

In this phase, clutch C_2 doesn't transfer torque while clutch C_1 is completely engaged. This process is similar to that of power-on upshift.

3. Comparison Between Simulation Results and Test Results

The test is carried out in the following way: a dynamic model of the test vehicle powertrain (including control strategies) was established before implementing the closed loop control strategy on the transmission prototype in the test vehicle (Fig. 7).

The simulation model is established on the software platform Mathcad. The real time results of the test vehicle are captured by the calibration software CANape 8.0 where the torque of the shifting elements is calculated. Due to the employment of two spring-damper elements to simulate the torsion damper elasticity and the output shaft, the test results in Fig. 8 and Fig. 9 show more vibration than the simulation results.

Adequate calibration work is required to improve the shifting quality of 8AT, so we need to check the consistency between the simulation results and the test results in the first place. This paper discusses the example of power-on upshift from the 3rd gear to the 4th gear and power-on downshift from the 8th gear to the 7th gear.

3.1. Power-on Upshift

Fig. 8 shows the simulation results (left) and test results (right) of 8AT power-on upshift from the 3rd gear to the 4th gear. In the test result (d), from 324.7 s to 325.2 s the torque of the off-going clutch reduces rapidly to 120 Nm and maintains that value. Clutchstarts to micro-slip while clutch enters the oil-filling phase and prepares the clutch to reach the kiss point in advance in order to minimize the lasting time of the shifting process.

Fig. 7. Test vehicle and Shengrui 8-speed transmission prototype.

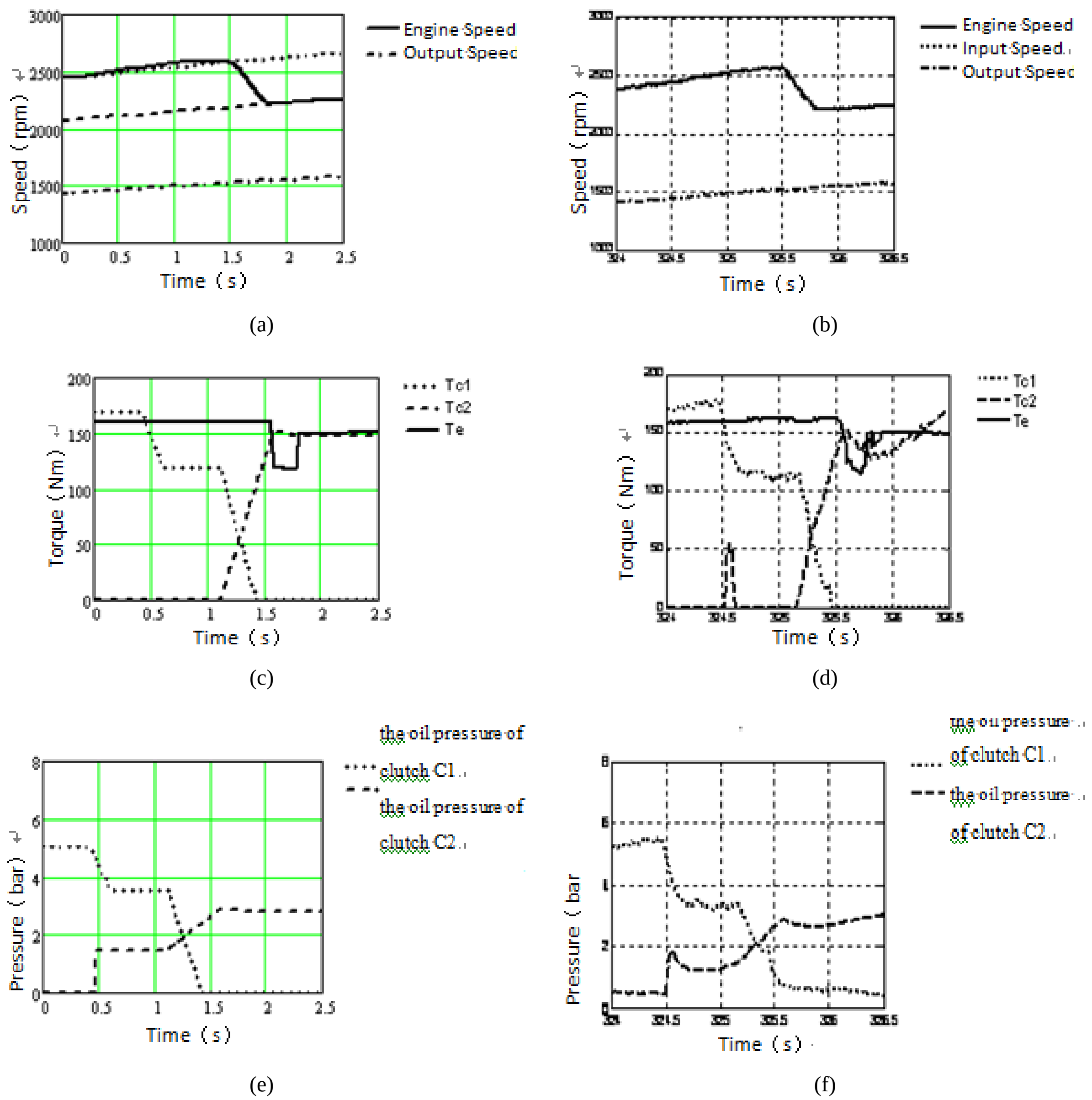


Fig. 8. Simulation results and test results of 8AT power-on upshift.

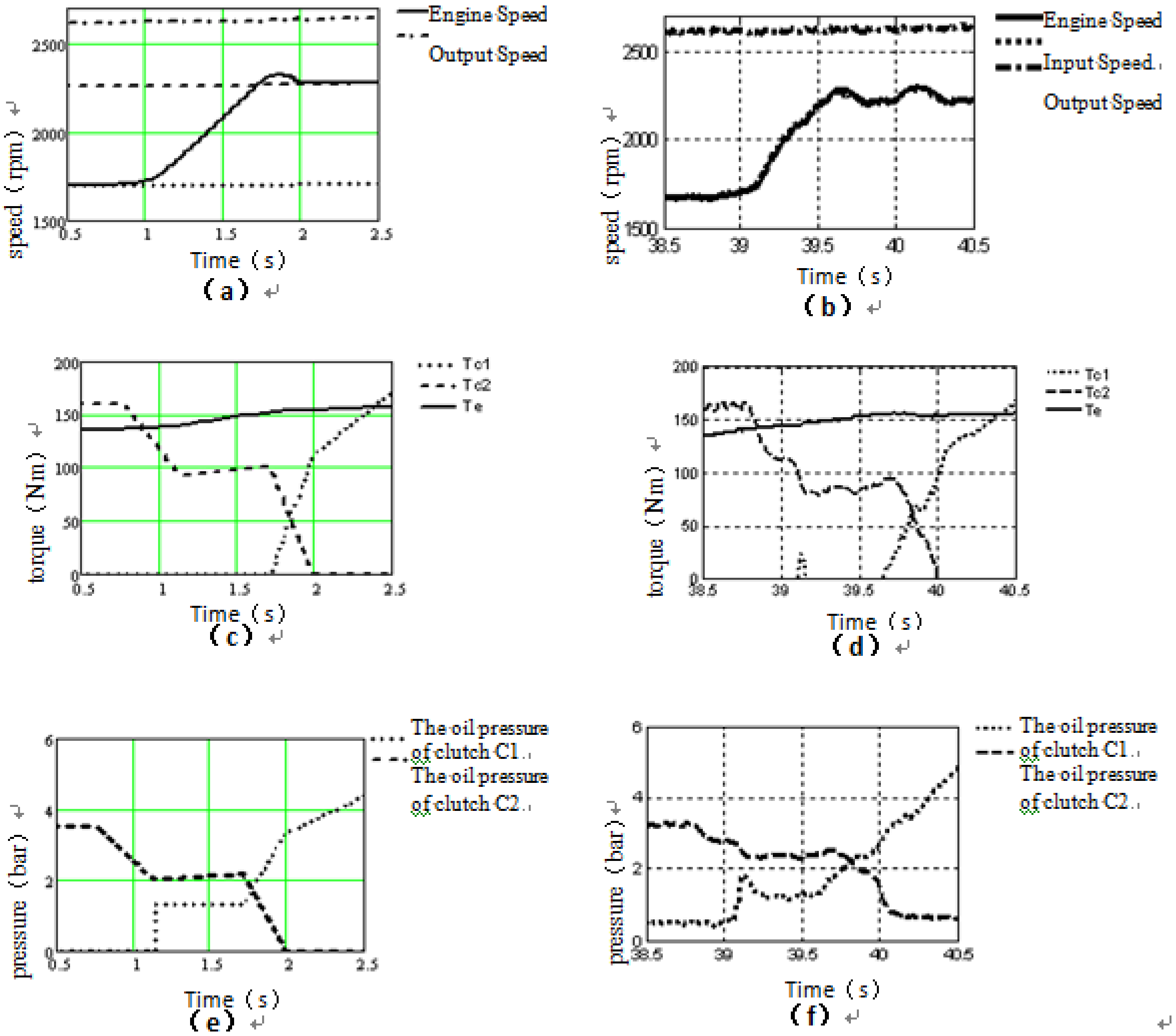


Fig. 9. Simulation results and test results of 8AT power-on downshift.

Fig. 8 shows the simulation results (left) and test results (right) of 8AT power-on upshift from the 3rd gear to the 4th gear. In the test result (d), from 324.7 s to 325.2 s the torque of the off-going clutch C_1 reduces rapidly to 120 Nm and maintains that value. Clutch C_1 starts to micro-slip while clutch C_2 enters the oil-filling phase and prepares the clutch to reach the kiss point in advance in order to minimize the lasting time of the shifting process. Therefore, the oil pressure of clutch C_2 at the end of this period shown in graph (e) is 1.5 bar instead of 0 bar. In order to compensate for the hysteresis of the oil pressure response, the controller gives out a relatively high pulsating pressure. At the same time, due to the stiffness mutation when the clutch piston contacts the plates, clutch C_2 goes through a sharp pressure rise of about 2bar from 324.5 s to 324.6 s in graph (f), thus causes the torque pulse in graph (d). As is shown by the test result in graph (d), the torque transferred by clutch C_1 is gradually taken over by clutch C_2 from 325.2 s to 325.5 s. This torque shifting process is able to remain smooth and steady thanks to FF+PI closed-loop controller. At the end of this period, clutch C_2 transfers all the torque while clutch C_1 is completely disengaged. From 325.5 s to 325.8 s, engine torque reduces from 160 Nm to 120 Nm while the torque transferred by clutch C_2 keeps increasing. Due to the combined effect of the engine torque reduction and the clutch torque increase, the engine speed is pulled down to the 4th gear speed line, see graph (b).

Judging from the graphs above, both simulation results and test results of the power-on upshift from the 3rd gear to the 4th gear of 8AT prove reasonable, which speaks for the reliability and applicability of the control strategy.

3.2. Power-on Downshift

Fig. 9 shows the simulation results and test results of the transmission downshift from the 8th gear to the 7th gear with the torque converter remaining locked. Graph (b) shows that engine speed synchronization period lasts for 700 ms from 39.1 s to 39.8 s. There are two reasons for a longer shifting process. First is for safety. The control of engine torque increase can go far beyond the driver's expectation, which can cause serious danger in some circumstances. Therefore only the control of clutch torque reduction is used here; second, if the engine torque stays high, the speed synchronization process will be adversely affected. In the speed phase, the torque of clutch C_2 reduces to 95 Nm (see graph (d)). According to Equation 6, the speed difference between the engine and the clutch will forces clutch C_2 to slide and pull up the engine speed to reach the 7th gear speed line. The oil filling process of clutch C_1 and the characteristics of this process are similar to those of power-on upshift, see graph (d) and (f).

As is shown in graph (d) and (f), the torque transferred by clutch C_2 is gradually taken over by clutch C_1. This torque shifting process is able to remain smooth and steady thanks to FF+PI closed-loop controller. At the end of this period, clutch C_1 transfers all the torque while clutch C_2 is completely disengaged. Both the simulation result in graph (a) and the test result in graph (b) show certain overshoot as engine speed reaches the 7th gear speed line. More parameter calibration is needed to optimize the engine speed line. More parameter calibration is needed to optimize the engine speed line.

Judging from the graphs above, both simulation results and test results of the power-on downshift from the 8th gear to the 7th gear of 8AT prove reasonable, which speaks for the reliability and applicability of the control strategy.

4. Analysis of Simulation Results

4.1. Simulation Results of Power-on Upshift

Fig. 10 shows the simulation results of power-on upshift from 3rd gear to 4th gear with different calibration parameters.

The first column shows optimization of a relatively comfortable driving style, the second column is a basic good shift control compared to others, the third column shows optimization of a relatively sporty driving style and the last column shows the results of inappropriate parameters, i.e. large shifting impact. The operating point of the engine and the vehicle speed remain the same during these simulations.

In the second column, graph (b) shows good calibration of the two clutches' torque gradients in torque phase and the speed phase at the operating point, which result in a quick and smooth speed synchronization in graph (f). Moreover, there only exists a small shifting impact in graph (j) thanks to the impact function.

In terms of optimizing driving comfort which is shown in the first column, it is necessary to reduce the torque change gradients of the two shifting clutches both in torque phase and speed phase. With smaller torque change gradients (see graph (a)), the duration of the two phases extends from 0.36 s to 0.52 s (see graph (e)), increasing the overall shifting time. A more comfortable driving style can be achieved by a smaller shifting impact as is shown in graph (i) (-10.3 m^3/s). However, because the speed phase is prolonged, more heat will be generated from plates friction (see graph (m), friction energy 2050J).

On the contrary, in terms of optimizing sporty drive style, the torque change gradient needs to be increased. As is shown in graph (c), a steeper torque gradient will shorten the duration of both torque phase and speed phase from 0.36 s to 0.24 s (see graph (g)) and realize a relative sporting driving style. Consequently, the maximum shifting impact shown in graph (k) increases to -39.98 m^3/s . The shift load of the friction plates are also reduced to 1432J in graph (o).

In the last column, graph (d) shows the simulation results of the traditional control strategy without the application of the impact function. It shows that the torque of the on-coming clutch does not decrease until the synchronization point, which leads to a big shifting impact in graph (l) (-65.4 m^3/s).

4.2. Simulation Results of Power-on Upshift

Fig. 11 shows the simulation results of a power-on downshift from 8th gear to 7th gear shifting process. The layout of these graphs is the same with power-on upshift discussed above.

In the second column where the calibration parameters are the most appropriate compared to others, the speed synchronization in graph (f) is quick and smooth with the calibration value being 45 Nm for the reduced torque of the off-going clutch at the beginning of the speed phase. In the torque phase, the torque change gradients of the two clutches are also calibrated appropriately to keep a micro-slip before the synchronization with the target gear speed, which helps to avoid engine overshoot and high shift load of the friction plates.

Column one optimizes a comfortable driving style. With a bigger torque value of the off-going clutch being about 60 Nm (see graph (a)), the duration of torque phase and speed phase extends from 0.63 s to 0.85 s (graph (e)). It also shows that the shifting impact is reduced to 2.5 m^3/s and the friction energy is more than 3355J (graph (m)) due to a longer sliding time of the friction plates.

In terms of sporty driving style, unlike the comfortable driving style, the torque value needs to be reduced to a lower level. The simulation results in the third column show that the value of 35 Nm (graph (c)) shortens the duration of torque phase and speed phase from 0.63 s to 0.49 s (graph (g)). The maximum shifting impact increases to 5.3 m^3/s (graph (k)) while the friction energy decreases to 1130J (graph (o)). Good calibration parameters for the torque change gradients of the on-coming clutch and the off-going clutch are necessary in order to keep a required micro-slip in the torque phase. Basically, a bigger gradient of the torque increase of the on-coming clutch helps reduce the slip in torque phase. In the fourth column, it can be observed that with a smaller gradient of the torque increase (graph (d)), the engine speed jumps over the target gear speed much, resulting in heavy shift load of about 3085J (graph (p)) in the friction plates.

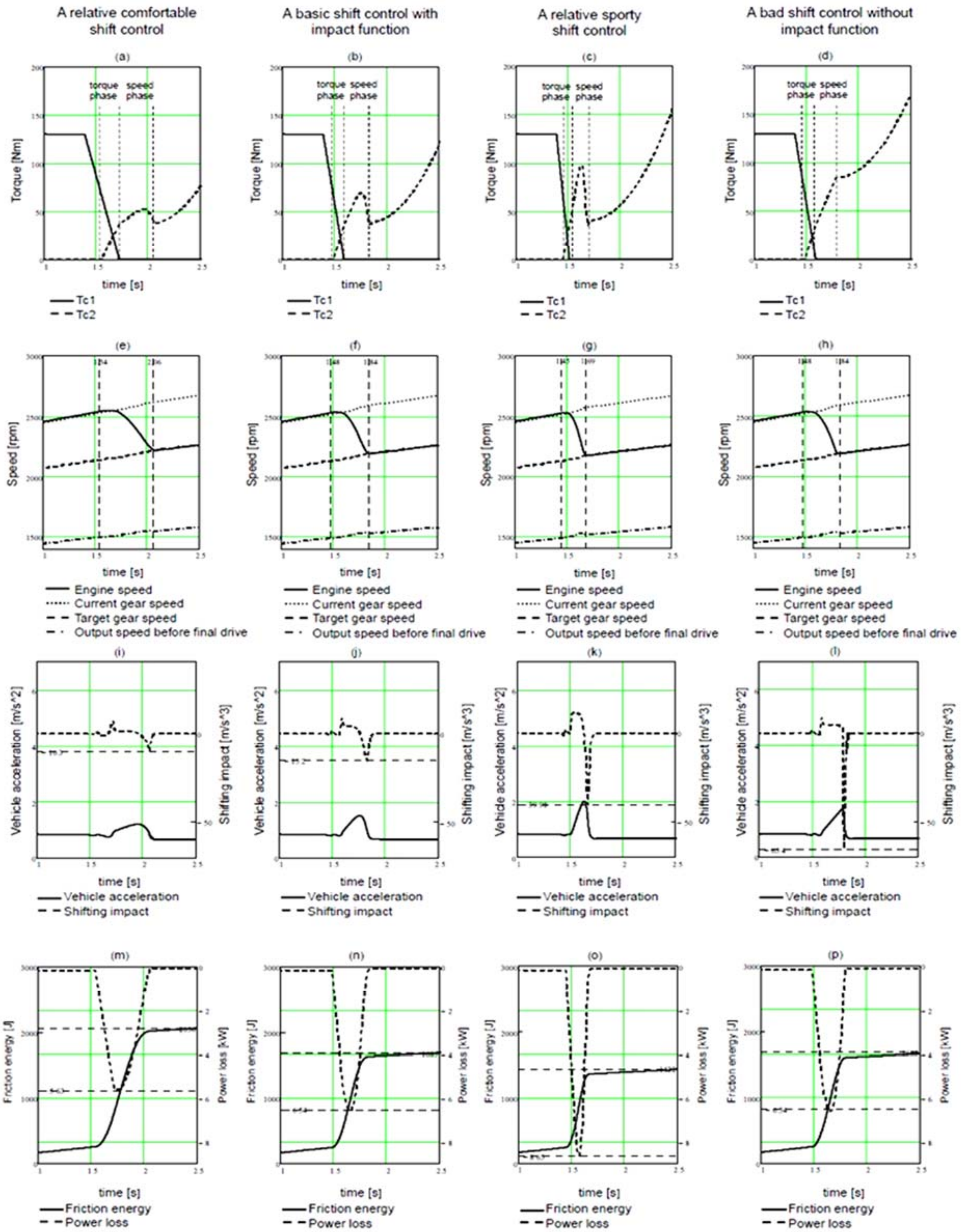


Fig. 10. Simulation results of 3rd gear to 4th gear shift.

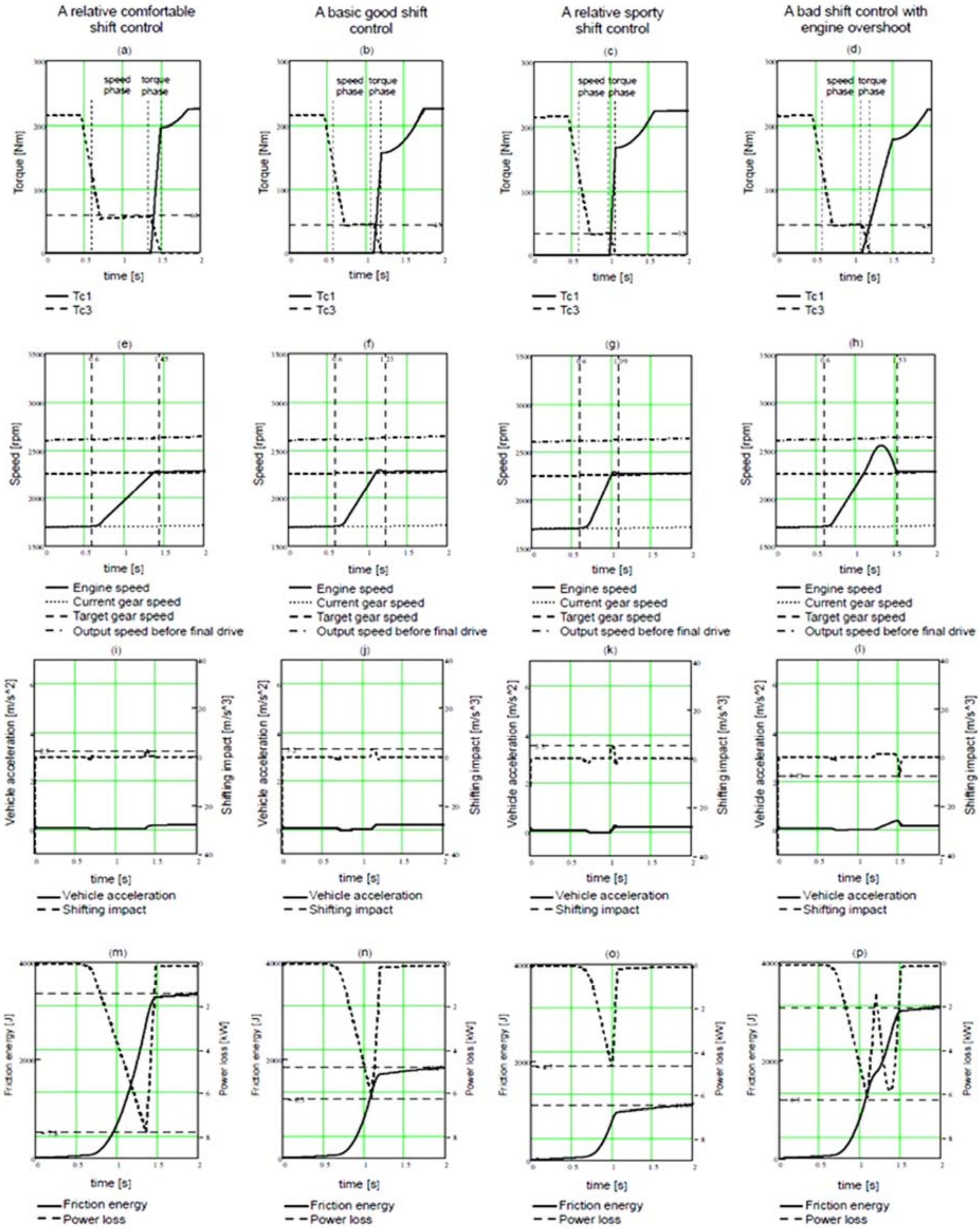


Fig. 11. Simulation results of 8th gear to 7th gear shift.

5. Conclusions

This paper establishes the dynamic model of a new type of FF 8-speed transmission and mainly discusses the dynamic characteristics of the torque, speed, impact etc. of the oil-filling phase, torque phase, speed phase and locking phase during the shifting processes of the power-on upshift and power-on downshift. It also simulates and calculates the dynamic characteristics of the clutch during each phase and clarifies the control requirements of the clutch.

Based on the results above, this paper deals with the analysis of the clutch control strategies and proposes a slip control algorithm. Together with the feed-forward PID control algorithm, the control over the shifting clutch is achievable. Influences of different calibration parameters are analyzed using the simulation model built in Mathcad. Through the comparison of the simulation results and the test results, we can tell the control strategies realize the precise, steady and smooth control over the clutch. This paper proves the applicability of the control strategies and contributes to the further development of the clutch control strategy.

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References

[1]. Minowa T., Ochi T., Kuroiwa H., et al., Smooth Gear Shift Control Technology for Clutch-to-Clutch Shifting. *SAE Technical Paper*, 1999, 1999-01-1054.

[2]. Tenberge P., Gläser K., Xu X., et al., 8-Gang-Automatikgetriebe für Frontqueranwendungen, AT-Getriebestrukturen mit Planetenradstufen und Stimradstufen, in *Proceedings of 8th International CTI Symposium, Innovative Automotive Transmissions*, Berlin, 2009 (in German).

[3]. Goetz M., Levesley M. C., Crolla D. A., Integrated Powertrain Control of Gearshifts on Twin Clutch Transmissions, *SAE Technical Paper*, 2004, 2004-01-1637.

[4]. John E. Marano, Steven P. Moorman, Clutch-to-Clutch Transmission Control Strategy, *SAE Technical Paper Series*, 2007, 2007-01-1313.

[5]. Xingyong Song, Zongxuan Sun, Pressure-Based Clutch Control for Automotive Transmissions Using a Sliding-Mode Controller, *IEEE/ASME Transactions on Mechatronics*, 17, 3, 2012, pp. 534-546.

[6]. Bingyang Luo, Sining Liu, Yimin Mo, Automatic clutch control strategy research based on multi-mode control, in *Proceedings of the International Conference on Systems and Informatics (ICSAI)*, 2012, pp. 90-94.

[7]. Karnopp, D., Computer Simulation of Stick-Slip Friction in Mechanical Dynamic Systems, *ASME J. Dyn. Syst., Meas., Control*, 107, 1, 1985, pp. 100–103.

[8]. Song, X., Sun, Z., Yang, X., and Zhu, G., Modeling, Control and Hardware-in-the-Loop Simulation of an Automated Manual Transmission, *Proc. Inst. Mech. Eng., Part D (J. Automob. Eng.)*, 224, 2, 2010, pp. 143–160.

[9]. Han-Sang Jo, Yeong-II Park, Jang-Moo Lee, et al., A Study on the Improvement of the Shift Characteristics for the Passenger Car Automatic Transmission, *Vehicle Design*, Vol. 23, Issue 3-4, 2000, pp. 307-328.

[10]. Megli T. W., Haghgooie, M., Colvin D. S., Shift Characteristics of a 4-Speed Automatic Transmission, in *SAE. International Congress & Exposition*, 1999, 1999-01-1060.