

Bond Graph Analysis of Automatic Transmission Shifts including Potential of Extra Clutch Control

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ABSTRACT

New generation of torque converter automatic transmissions (AT) include a large number of gears for improved fuel economy. Control requirements for such transmissions become more demanding, which calls for the development of new shift optimization and analysis tools. This paper presents two contributions to the field of transmission dynamics analysis: (i) bond graph method-based shift transient analysis, and (ii) deriving a unique set of conditions for beneficial use of a third (normally-open) clutch for any upshift or downshift, with emphasis on inertia phase. The derived conditions are examined on an example of 10-speed AT based on the clutch torque input trajectory optimization results. The examination results point out that the extra clutch has a potential of significant performance improvement for any single-transition upshift in the inertia phase, in terms of reduced vehicle jerk RMS value due to the suppressed inertia bump effect. The shift quality improvement is possible in the torque phase, as well, but it is of much smaller extent when compared to that related to the inertia phase.

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INTRODUCTION

More recently, the legislative and market pressure for CO₂ reduction and fuel efficient powertrains has led to the development of automatic transmissions that include large number of gears (around 10), [1, 2]. The increased number of gears in ATs, however, increases the number of shift types and shift events progressively, thus making the development of new ATs significantly more time-consuming. Therefore, new approaches for dynamic analysis of vehicles with large number of gears AT have been developed [3].

Still, most of the AT dynamics modeling and control research work published in literature relates to conventional ATs, typically those with four or six gears [4, 5, 6]. The emphasis is on analysis and compensation (typically by means of controls) of distinct effects of AT shift dynamics such as inertia bump, torque hole, engine flare, and transmission tie-up effects [4, 7]. There are, however, several publications that analyze specific effects and related unconventional controls, which are directly or indirectly related to advance ATs with a large number of gears. This relates to double-transition shift [8], where four clutches are active, and single-transition shifts (STS) using an extra, normally-open clutch (three clutches are active) [9]. For better understanding of optimal shift behavior and development of related control strategies, various optimization techniques have been considered in [6, 10, 11].

This paper presents two contributions in the field. Firstly, it analyzes the aforementioned, well-known AT dynamics effects, but by using a new, systematic and illustrative graphical approach based on the bond graph methodology [12]. Although the approach was already applied to ATs [13] (and also to hybrid electric transmissions [14, 15]), it is employed in this paper in a broader sense in terms of covering a wide set of AT single-transition shift dynamics effects. Secondly, the bond graph approach is applied to develop a set of conditions for beneficial use of extra clutch in the inertia phase of single-transition upshifts and downshifts. Compared with [9], where a specific structure of transmission model is used, the extra clutch conditions are derived and analyzed herein for a more general case of any STS in an AT with a large number of gears. The derived condition-based prediction of beneficial use of extra clutch is verified by comparing the prediction results with control trajectory optimization results for a wide range of subsequent and skipping-gear shifts of a 10-speed AT from [16].

POWER TRAIN MODEL

The structure of a power train equipped with a torque converter AT can be depicted by the block scheme shown in Figure 1 [5]. The corresponding model equations are given in Appendix A based on [5, 17].

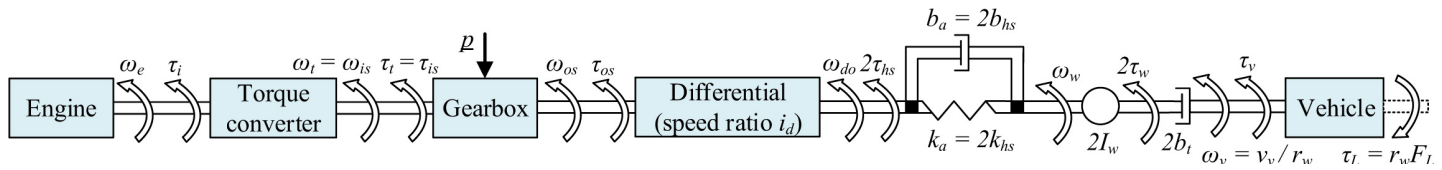


Figure 1. Power train model schematic.

The engine is described as a source-of-torque element, where the torque τ_e depends on the accelerator pedal opening p_{th} and the engine speed ω_e . The torque converter is described by nonlinear static curves $R_\tau(R_\omega)$ and $K_i(R_\omega)$, where $R_\tau = \tau/\tau_i$ is the torque ratio, $K_i = \omega_i/\tau_i^{1/2}$ is the capacity factor, and $R_\omega = \omega/\omega_i$ is the speed ratio [18]. The torque converter can be modulated or locked by using the by-pass clutch.

A 10-speed AT from [16] is used as a generic example of an advanced transmission for this study. The transmission/gearbox schematic is shown in Figure 2. The gearbox consists of four planetary gears 1 to 4 and six hydraulic clutches A to F. Its mathematical model is presented in the next subsection.

The transmission output shaft is connected to the differential, which provides reduction of the gearbox output speed ω_{os} by the factor $1/i_d$ and transmits the torque to the left and right compliant half-shafts, connected to the wheels. The half-shafts are modeled in Figure 1 by an equivalent compliant shaft, whose torque $2\tau_{hs}$, stiffness and damping coefficients k_a and b_a , respectively, are two times larger than those of each single half-shaft. Similarly, the lumped wheel inertia is two-fold larger than the single wheel inertia I_w . The powertrain model also includes a linearized tire model and a longitudinal vehicle dynamics model (see Appendix A).

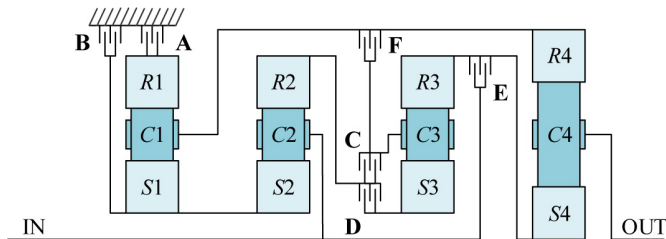


Figure 2. Schematic of AT gearbox [16].

Full Model of Gearbox

The gearbox is modeled by using the bond graph method [12], as shown in Figure 3 (see Appendix B). The main advantages of this method include [5]: direct insight and correlation between the bond graph model structure and the structure of the transmission mechanics (cf. Figures 2 and 3, with the note that the triangular elements of bond graph correspond to planetary gears), ease of determining and handling redundant state variables that are inherent to transmission dynamics, and simplicity of reducing model order (e.g. for particular gears engaged).

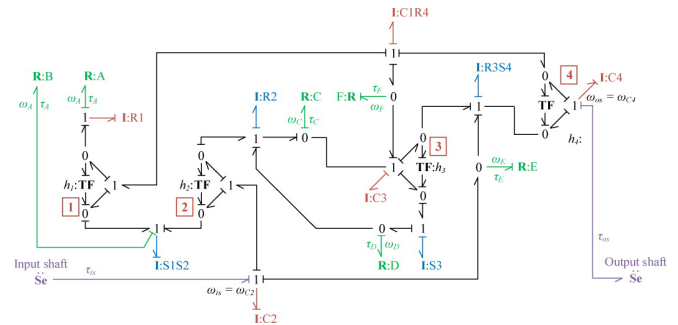


Figure 3. Bond graph model of AT gearbox.

The causality analysis (related to strokes associated with each bond, [5, 12]) shows that five out of nine gearbox inertia elements have integral causality (the elements I_{C2} , I_{R2} , I_{C3R4} , I_{C3} , and I_{C4}), thus revealing that the full gearbox model is of fifth-order. The final model is described by the following state-space equation

$$\mathbf{A} \begin{bmatrix} \dot{\omega}_{is} & \dot{\omega}_{R1} & \dot{\omega}_{C1R4} & \dot{\omega}_{C3} & \dot{\omega}_{os} \end{bmatrix}^T = \mathbf{B} \begin{bmatrix} \tau_A & \tau_B & \tau_C & \tau_D & \tau_E & \tau_F & \tau_{os} & \tau_{is} \end{bmatrix}^T, \quad (1)$$

where $\mathbf{A}$ is the inertia matrix and $\mathbf{B}$ is the input matrix. The clutch slip speeds are obtained from the state variables according to the following algebraic equation:

$$\begin{bmatrix} \omega_A & \omega_B & \omega_C & \omega_D & \omega_E & \omega_F \end{bmatrix}^T = \mathbf{U} \begin{bmatrix} \omega_{is} & \omega_{R1} & \omega_{C1R4} & \omega_{C3} & \omega_{os} \end{bmatrix}^T. \quad (2)$$

The clutch friction torques are determined from the clutch slip speeds based on a friction model:

$$\begin{bmatrix} \tau_A & \tau_B & \tau_C & \tau_D & \tau_E & \tau_F \end{bmatrix} = \underline{f} \left(\begin{bmatrix} \omega_A & \omega_B & \omega_C & \omega_D & \omega_E & \omega_F \end{bmatrix} \right), \quad (3)$$

and fed back to the state equation (1). The classical friction model shown in Figure 4b is used as a feasible approximation of the "theoretical" friction model from Figure 4a. Here, the threshold Δ_{ω} is set as a trade-off of accuracy (lower Δ_{ω}) and computational efficiency (higher Δ_{ω}). As an alternative to the classical friction model, the Karnopp friction model has been implemented as well, based on [5].

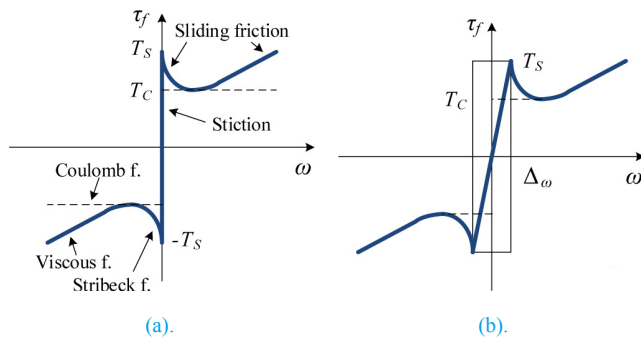


Figure 4. Generalized static friction model (a), and its modification: classical model (b).

Gearbox Model for Single-Transition Shift Analysis

For specific AT analysis and shift control optimization, which relate to single-transition shifts (STS, two active clutches), the AT model order can be reduced by eliminating the state variables related to locked clutches. For example, during 1-2 (and 2-1) shift, the E (C) clutch is to be disengaged, while the clutch C (E) is to be engaged. Clutches A, B and D are locked during the shift, while the clutch F is open (see Table 1).

A model order reduction is based on determination of locked clutch torques by using the condition that their slip speeds are equal to zero.

This results in the following second-order linear state-space model for 1-2 (and 2-1) shift (and similarly for other STS shifts):

$$\mathbf{A}_{\text{red}} \begin{bmatrix} \dot{\omega}_{is} \\ \dot{\omega}_{os} \end{bmatrix} = \mathbf{B}_{\text{red}} [\tau_E \quad \tau_C \quad \tau_{os} \quad \tau_{is}]^T \quad (4)$$

The shift dynamics are usually such that the diagonal elements (a_{11} and a_{22} in Equation (5)) are significantly larger than the cross-diagonal elements (a_{12} and a_{21}), particularly if the input/engine and output/vehicle inertia is lumped to the transmission inertia (i.e. when the influence of torque converter and half-shaft dynamics are neglected, cf. Figure 1). While this assumption usually holds for ATs with lower number of gears, subsequent-gear shifts, and/or conventional shifts without engine torque control, it should generally be taken with reserve for a wide range of shift conditions [19]. For the purpose of simplified analyses conducted in the next section, the cross-diagonal elements of the inertia matrix (i.e. the inertia coupling effect, [13]) are neglected. It can be shown that under this assumption, the model given by Equation (4) reduces to an equivalent dual clutch transmission (DCT) model (see also [13, 20]), whose bond graph is shown in Figure 5. Here, the equivalent gear ratios are obtained from the input matrix elements as: $i_j = -b_{1j}$ and $g_j = 1/b_{2j}$, $j = 1, 2$, while the inertia values I_{in} and I_{out} corresponds to the diagonal elements of inertia matrix $\mathbf{A}$, a_{11} and a_{22} , respectively.

$$\mathbf{A}_{\text{red}} = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix},$$

$$\mathbf{B}_{\text{red}} = \begin{bmatrix} b_{11} & b_{12} & b_{13} & b_{14} \\ b_{21} & b_{22} & b_{23} & b_{24} \end{bmatrix}. \quad (5)$$

Table 1. Clutching table of the gearbox [16] (X means that clutch is locked, thus carrying torque).

Gear state	Gear ratio	Ratio step	Torque transmitting elements					
			A	B	C	D	E	F
Rev	-4.622		X	X		X		X
N		-0.98						
1st	4.740		X	X		X	X	
2nd	3.047	1.56	X	X	X	X		
3rd	2.222	1.37	X		X	X	X	
4th	1.821	1.22	X		X	X		X
5th	1.552	1.17	X		X		X	X
6th	1.300	1.30	X			X	X	X
7th	1.000	1.30			X	X	X	X
8th	0.852	1.17		X		X	X	X
9th	0.695	1.23		X	X		X	X
10th	0.643	1.08		X	X	X		X

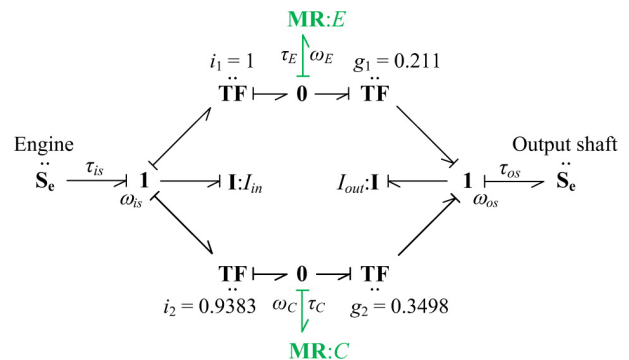


Figure 5. Equivalent DCT bond graph model for 1-2 (and 2-1) shift analysis.

BOND GRAPH ANALYSIS OF TRADITIONAL SINGLE-TRANSITION SHIFT

The analysis is based on the equivalent DCT model shown in Figure 5, and it covers single-transition power-on upshift and downshift events (with the numerical example of 1-2 and 2-1 shifts corresponding to i , g , and I parameters in Figure 5), including various characteristic phenomena that may occur during shifts [20]. The shift analysis also includes the control mechanism related to input torque control (engine torque modulation), as well as unconventional control using off-going clutch in the inertia phase. The analysis will be illustrated by simulation results obtained by using the full power train model from Figure 1 in Matlab/Simulink working environment. For the analysis purposes, the engine torque is considered constant (i.e. the input torque is constant for the torque converter considered to be locked). In the simulation results, the torque converter is open as in the original model.

Power-on Upshift

The following characteristic cases of power-on upshift will be discussed:

- Perfect timing of the off-going clutch disengagement
- Early disengagement of the off-going clutch
- Late disengagement of the off-going clutch

Perfect Timing of Off-going Clutch Disengagement

The power-on upshift can be divided into four phases with two active phases: torque phase and inertia phase. In the first (initial) phase, the first gear is engaged and the input torque is transmitted through the upper path (clutch **E** is locked and the clutch **C** is open), as illustrated in Figure 6a and shown by the simulation results in Figure 7.

The upshift is initiated by engaging the oncoming clutch (clutch **C**). Since the clutch **E** is locked ($\omega_E = 0$), and parameters i_j and $g_j, j = 1, 2$ are such that $i_1 \cong i_2$, and $g_2 > g_1$, the oncoming clutch slip speed ω_E is positive, so that it transfers a positive torque when being engaged (see Figure 7b). Since $\tau_{is} = \tau_e$ is considered to be constant, the reactive torque of the locked off-going clutch falls to satisfy the torque balance equation of the input junction point **1** (the one associated with the speed ω_{is} , note that the input inertia is lumped to the output one for the still-locked off-going clutch). Because the output gear ratios are such that $1/g_1 > 1/g_2$ (with $i_1 \cong i_2$), the **E** clutch torque falls affects the output torque to a larger extent than the **C** clutch torque rise. This results in a fall of the output torque, which is known as **torque hole effect** [17, 20], (see Figure 6b and Figure 7a). Simultaneously with the oncoming clutch torque capacity rise (to keep it locked), the off-going clutch torque is being reduced during the torque phase, so that it should diminish when the off-going clutch torque falls to zero. This is considered as a perfect-timing disengagement of the off-going clutch. At that instant the off-going clutch opens (Figure 7d), the output torque reaches the 2nd gear level (Figure 7a), and the torque phase is finished.

In order to reduce the input speed ω_{is} to the 2nd gear level and lock the oncoming clutch **C** (for the constant τ_{is}), an extra torque capacity needs to be applied to clutch **C** (see Figure 7b). The increased torque of clutch **C** is at the same time transmitted through the ratio $1/g_2$ to the output junction point **1** (the one associated with the speed ω_{os}), as illustrated in Figure 6c. Since the off-going clutch torque is now equal to zero, this results in the output torque rise above the 2nd gear level, which is known as **inertia bump effect** [20], (see Figure 7a). When clutch **C** closes, its torque (and, consequently, the output torque) drops from the reached clutch torque capacity level back to the 2nd gear torque level. This is an additional torque disturbance event, which completes the torque bump shape of the overall torque disturbance.

In the fourth (final) phase, Figure 6d, the second gear is engaged and the input power is transmitted through the lower power path (clutch **C** is locked, and the clutch **E** is open).

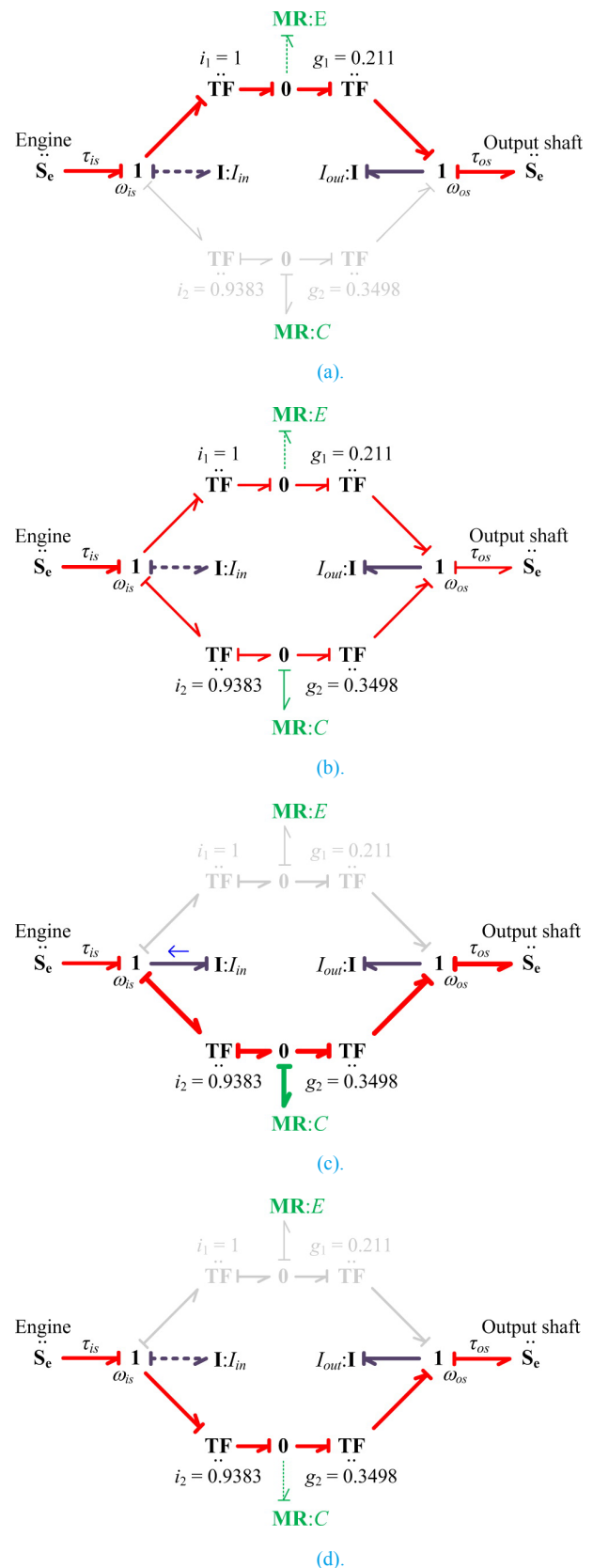


Figure 6. Bond graph analysis for power-on upshift and perfect timing of off-going clutch disengagement: (a) 1st gear engaged; (b) torque phase; (c) inertia phase; (d) 2nd gear engaged.

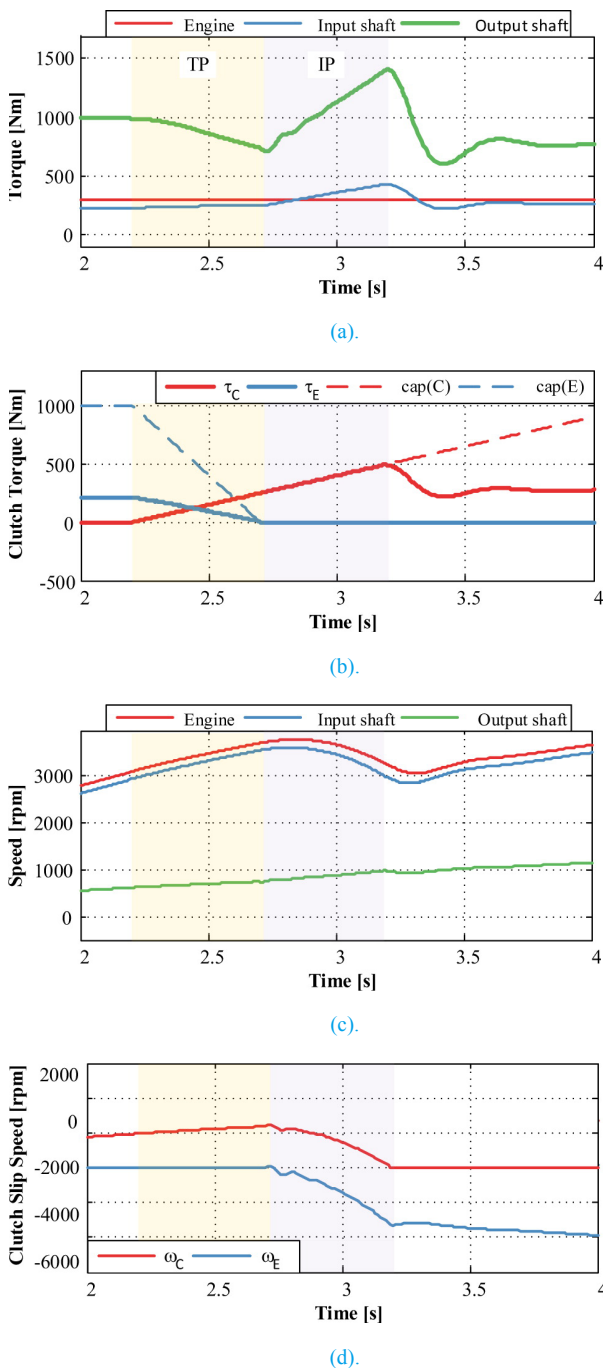


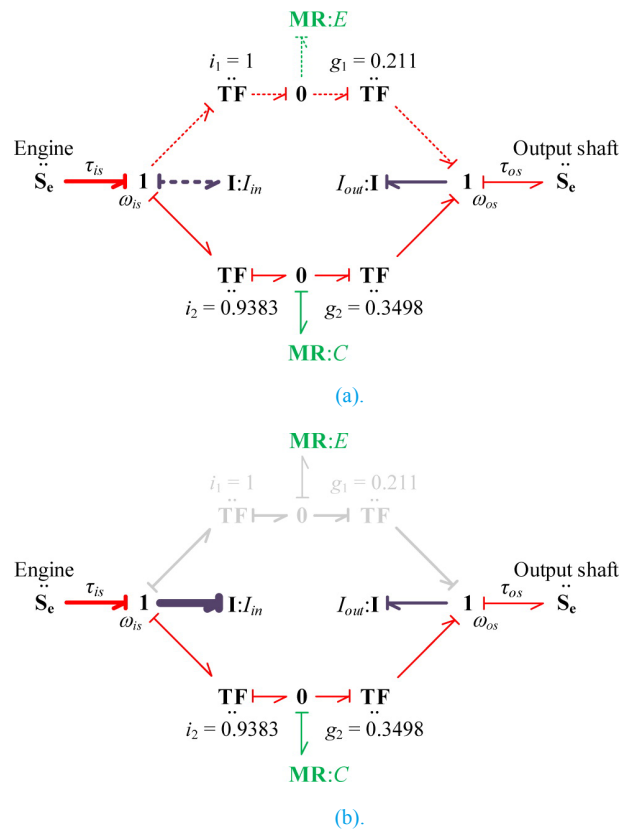
Figure 7. Simulation results of 1-2 power-on upshift with perfect timing of off-going clutch disengagement.

Early or Late Disengagement of Off-going Clutch

The too fast disengagement of clutch **E** results in opening this clutch before the oncoming clutch torque is built up to the 2nd gear level. Therefore, when the off-going clutch gets open (and does not support the reactive torque anymore), the output torque suddenly drops to a low level related to the insufficient torque of oncoming clutch (see [Figure 8a](#)). As shown in [Figure 9a](#), this results in a **strong torque hole** including notable excitation of the shuffle mode vibrations. For the constant input torque, the early-ceased off-going clutch torque, and still insufficient oncoming clutch torque, the input speed ω_{is} increase becomes more emphasized (see [Figure 8b](#)). This results in

the **engine speed flare effect** (see [Figure 9a](#)) [20], and as a consequence the inertia bump becomes more emphasized afterwards ([Figure 9a](#)). The oncoming clutch torque continues to increase and eventually becomes sufficiently large to decelerate the input speed and finally lock the oncoming clutch. Because the oncoming clutch needs more time to build up the torque and overcome the input inertia torque, the shift is prolonged ([Figure 9b](#)).

The second characteristic case relates to late disengagement of the off-going clutch. Under these conditions, the off-going clutch does not open when its reactive torque diminishes (Figure 9b). To satisfy the torque balance at the input junction point **1** in the presence of growing oncoming clutch torque (and for the constant input torque), the still-locked off-going clutch starts transferring negative reactive torque (Figure 8c; note that the solid blue arrows added to the bond graph indicate the change of torque direction i.e. power direction, see Appendix B). Since $1/g_1 > 1/g_2$ holds, this results in the continuing **fall of the output torque** τ_{os} in the torque phase, i.e. τ_{os} will dip down below the 2nd gear level (Figure 9a). Since both clutches transfer torque in the torque phase, the transmission is **tied up** [20], with some recirculation of power present (Figure 8c). Because the oncoming clutch now needs to overcome additional torque (coming from the off-going clutch), the shift is prolonged and inertia bump effect is more emphasized (cf. Figures 6c and 8d and see Figure 9a).



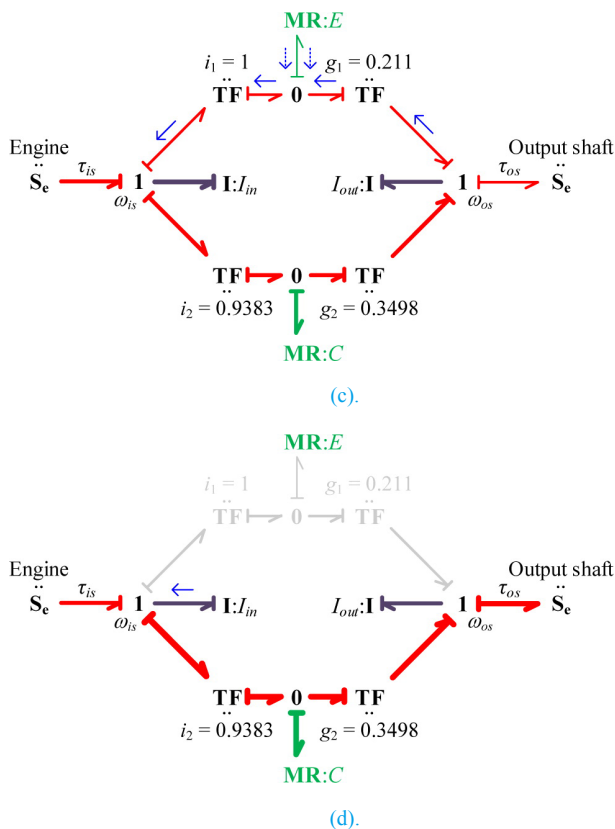


Figure 8 (cont.). Bond graph analysis for torque phase (a) and inertia phase (b) of early disengagement of off-going clutch, and for torque phase (c) and inertia phase (d) of late disengagement of off-going clutch.

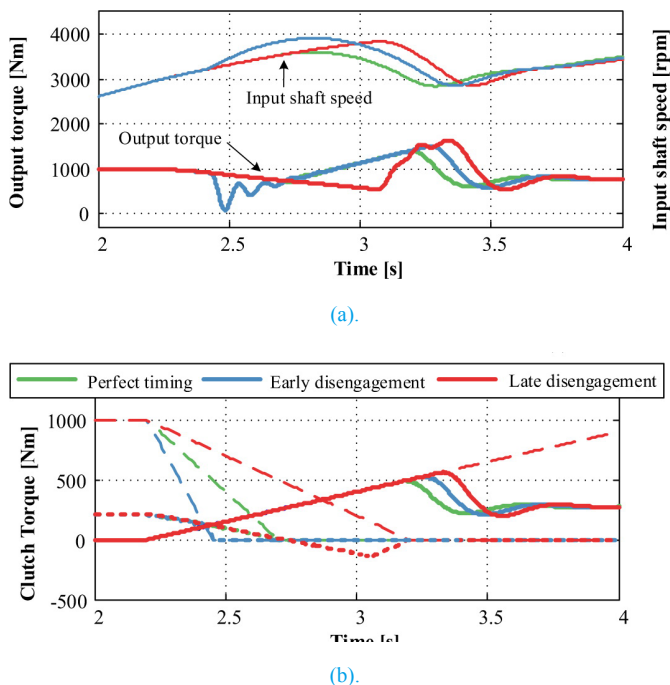


Figure 9. Simulation results of 1-2 upshift with early/late off-going clutch disengagement and comparison with perfect timing synchronization.

Negative Off-going Clutch Power Flow during Inertia Phase

The off-going clutch, which is conventionally open in the inertia phase, has a negative slip speed in that phase (because the input speed reduces). This means that, if the off-going clutch would be modulated in the inertia phase, it would transfer a negative torque (Figure 10), thus providing a **potential for reduction of the inertia bump**. In order to prevent the negative torque τ_E from disturbing the torque τ_C in reducing the input speed, the oncoming clutch torque growth should be reinforced, thus resulting in a negative power recirculation through the two torque paths and related (clutch) power loss increase (Figure 10). Because $1/g_1 > 1/g_2$ and $i_1 \approx i_2$, the opposite actions of torque capacity increase of the two clutches are such that they finally results in inertia bump reduction (the contribution of torque reduction is stronger than that of input speed reduction, $i_1 g_1 < i_2 g_2$). The above analysis will be illustrated by the optimization results given in [11].

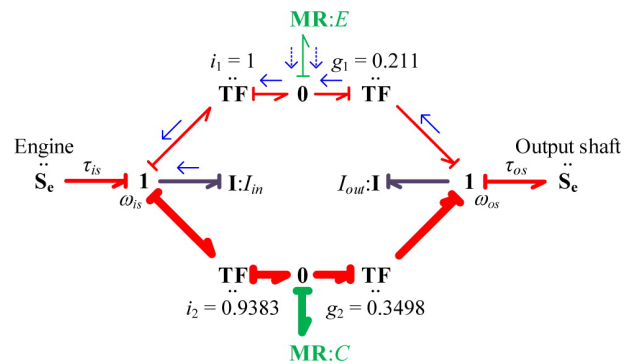


Figure 10. Bond graph analysis for inertia phase of 1-2 power-on upshift in case of negative off-going clutch power flow imposed by off-going clutch torque modulation.

Power-on Downshift

For the downshift analysis, the same BGM from Figure 5 is used. The analysis is conducted for the cases of perfect-timing, early, and late engagement of the oncoming clutch.

Perfect-Timing Engagement of Oncoming Clutch

The power-on downshift is again divided into four phases. In first phase, the 2nd gear is engaged, i.e. the input torque is transmitted through the lower path of the bond graph model in Figure 6d. Since the clutch C is locked ($\omega_C = 0$), and $i_1 \approx i_2$, $g_2 > g_1$, the slip speed of oncoming clutch E is negative.

The downshift is initiated by reducing the torque capacity of the off-going clutch C, in order to increase the input speed to the 1st gear level (for the constant input torque and the oncoming clutch torque equal to zero at the input junction point 1). Simultaneously with the off-going clutch torque fall, the output torque is being reduced during