Active damping of transient vibration in dual clutch transmission equipped powertrains: A comparison of conventional and hybrid electric vehicles

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Abstract

The purpose of this paper is to investigate the active damping of automotive powertrains for the suppression of gear shift related transient vibrations. Conventionally, powertrain vibration is usually suppressed passively through the application of torsional dampers in dual clutch transmissions (DCT) and torque converters in planetary automatic transmissions (AT). This paper presents an approach for active suppression of transient responses utilising only the current sensors available in the powertrain. An active control strategy for manipulating engine or electric machine output torque post gear change via a proportional-integral-derivative (PID) controller is developed and implemented. Whilst conventional internal combustion engine (ICE) powertrains require manipulation of the engine throttle, for HEV powertrains the electric machine (EM) output torque is controlled to rapidly suppress powertrain transients. Simulations for both conventional internal combustion engine and parallel hybrid vehicles are performed to evaluate the proposed strategy. Results show that while both the conventional and hybrid powertrains are both capable of successfully suppressing undesirable transients, the EM is more successful in achieving vibration suppression.

Keywords

Hybrid electric vehicle (HEV), powertrain, active damping, vibration control, dynamics

1. Introduction

Major trends in the broader automotive industry are aimed at improving the efficiency of passenger vehicles through new transmission technologies and hybridization of the powertrain. Frequently, this excludes the use of powertrain components such as hydrodynamic torque converters, which possess strong damping properties in addition to respective functional application. As a result, such vehicles are increasingly susceptible to driveline oscillations that are perceived by the driver are poor driving quality, and can be considered a source of noise vibration and harshness (NVH). The purpose of this study is to investigate the application of active damping measures for the suppression of these vibrations in the powertrain for both conventional and hybrid vehicles.

Dual clutch transmission equipped powertrain combine the power-on shifting capabilities of conventional automatics, such as planetary ATs or continuously variable transmission (CVT), with the high efficient components of manual transmissions, such as gears and synchronisers. High quality shift control is required to perform clutch-to-clutch gearshifts without loss of tractive load to the road, while still providing comfort and ride quality of the conventional AT. To increase the powertrain efficiency DCTs eliminate torque converters for the powertrain and consequently loses a significant component of system damping during shifting. To make use of DCTs in both conventional internal combustion engine vehicles as well hybrid electric and full electric vehicles, suppression of transient vibration resulting from gear shift is suggested to improve shift quality.

Extensive research into the control of gear shifts in dual clutch transmissions has been conducted focusing on the study of control during the shift to limit undesirable powertrain response [1-4]. This research is commonly limited by simplification of engine, hydraulic and synchroniser models to provide compact, efficient powertrain models. As a result it is frequently demonstrated that there is significant torsional vibration at the completion of shifting, though powertrain damping is sufficient to reduce vibration after a period of disturbance. The ability to provide rapid and accurate control of the engine and clutches in such research frequently does not consider the contributions of time delay in the engine through ignition of pistons or clutch hydraulics for control of such complex systems. In Walker [5] the integration of detailed hydraulic models are studied in DCT powertrain shift control, these results indicated that accuracy of clutch torque estimation is critical to shift control and the manipulation of engine torque during the final stages of shifting can lead to improvement in shift quality. However, these results still demonstrate many of the traits of a lightly damped system with limited suppression of post shift vibration.

Active powertrain damping in conventional ICE powered vehicles has been under consideration to varying degrees for some time. Berriri [6,7] develops a partial torque compensator for such vehicles that is independent of many vehicle parameters and external variables (i.e. road grade or aerodynamic drag). The compensator modifies the engine torque to suppress oscillations. One of the main limitations for suppression in conventional vehicles is identified as being maintaining vehicle drivability and responsiveness; insofar that extensive variation in of the engine output torque will reduce driving quality, implying that there are practical limits to the rate of suppression which may otherwise result in engine flaring or sluggish response of the vehicle. Fredriksson [8] has performed a similar study, developing PID, pole placement and Linear-Quadratic-Gaussion (LQG) controllers, with LQG being demonstrated as the most successful strategy. Bruce [9] combines feedforward and LQ feedback control and Lefebvre [10] employs H-infinity control successfully to the same issue. Fredriksson [11] and Syed [12] both apply active damping to hybrid vehicles. Syed [12] application is to a power-split HEV utilises active control of motor torque to successfully reduce vibration to imperceptible levels.

Across each of these studies the dominant trend is to investigate powertrain oscillations resulting from transients initiated in the variation of throttle control, such as tip-in/tip-out studies of the powertrain. This paper goes beyond these studies to integrate control with power-on upshift control of the powertrain. This gear change is chosen as it the most susceptible to undesirable transients [1]), in comparison to down shifts and power-off gear change.

The high efficiencies and flexibility in design of DCTs make for ideal for application to hybrid vehicle systems. One example is for mild hybrid systems such as the ESG presented by Wagner and Wagner [13], where a 10 kW electric machine is used to improve vehicle efficiencies under high demand or low engine efficiency conditions. Alternative hybrid systems have been presented by Joshi [14] for a more complicated hybrid system employing two motors with the DCT used to control power flow of the system, providing as series-parallel type configuration. Such a design is capable of much broader operating modes for hybrid operation. Wang [15] present a hybrid powertrain for application in buses, using various drive cycles to statistically optimise design, comparisons indicate improved efficiency and mobility compared to popular integrated starter/generator designs. Kilian [16] provides the most comprehensive arrangement of hybrid DCT transmissions with electric machines being capable of placed on input shaft, primary shafts, or countershafts. Uses for these electric machines include power supply, power generation, and synchroniser assistance. Holmes [17] provides a simpler hybrid layout for a single electric machine parallel hybrid arrangement with the electric machine between dual clutches and engine, and capable of being isolated from the engine using a third clutch.

The purpose of this paper is to therefore use simulation to investigate active damping of automotive powertrains for conventional and hybrid vehicles, with particular reference to its implementation with gear shift control. Integration of active damping with shift control exceeds many current studies, which focus on tip-in/tip-out throttle control as the reference problem for study. Through application of eight degree of freedom (DOF) powertrain models of various vehicle configurations, the capability to suppress transients resulting from gear shifting is studied. This includes detailed modelling of the ICE to capture speed dependent time delay associated with piston firing [6, 7], and modelling two different parallel HEV configurations.

The remainder of this paper is divided into the following sections. The first section puts forward detailed modelling of the powertrain and sub components, including the multi-body dynamic model, torque models for engine, clutches and vehicle resistance torque. DCT shift control with powertrain vibration suppression is then introduced, and discussed with reference to implementation. Then several simulations are conducted to compare shift control strategies and the impact of vibration suppression on different powertrain configurations. Finally, the paper is summarised and conclusions are made.

2. DCT equipped powertrain models

2.1. The dual clutch transmission equipped powertrain.

Figure 1 presents a basic dual clutch transmission powertrain comprising of engine, coupled clutches, transmission gear train, output drive train including differential, and wheels. The unique aspects of the DCT powertrain are the application of clutches and the arrangement of the gear train. The two clutches have a common drum attached to the input shaft from the engine, and the friction plates are independently connected to odd or even gears. For a full transmission gears 1, 3, and 5 (G1) are driven through the first clutch (C1), while clutch C2 drives gears 2, 4, 6, and R (G2). Synchronisers are denoted as S1 and S2. Thus, the transmission is representative of two half manual transmission, and, in this sense, gear change is realised through the simultaneous shifting between these two half transmissions.

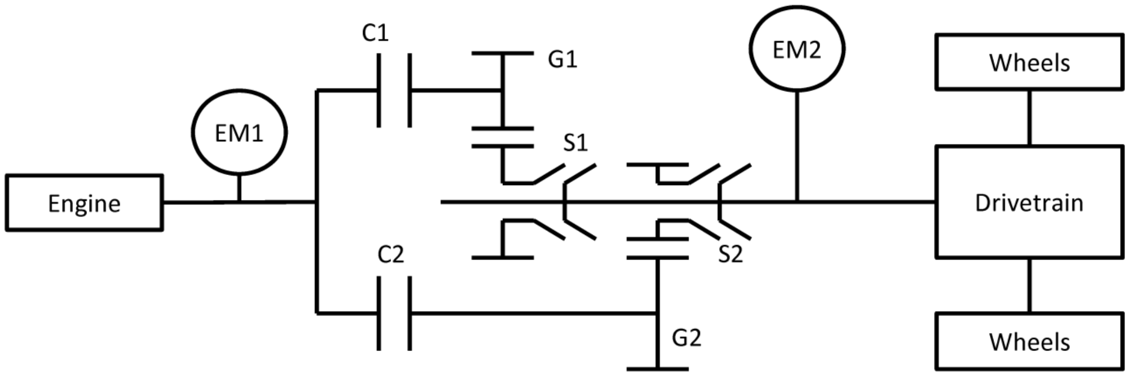

Figure 1: General DCT powertrain layout with different hybridisation variants

Also shown in Figure 1 are the two options considered for the hybridisation of the DCT powertrain based on the location of the electric machine (EM), these are noted as EM1 and EM2. These two configurations provide a parallel type hybrid vehicle powertrain, where the either EM or engine or the two combined can directly drive the wheels. For the configuration with EM1 the motor speed and torque are defined by the engaged gear ratio, and it is isolated from the transmission and wheels when both clutches are open. For EM2 configuration, the motor is downstream of the transmission and therefore has a fixed ratio to the wheels, via the final drive. The limitation to this configuration being that it is not possible to isolate it from the wheels.

The two aspects of gear shifting are representative of manual and automatic transmissions. Prior to shifting the first requirement is to synchronise the target gear. This is realised in an automated process using standard synchronisers that are popular in manual transmissions, having low cost and high reliability. Once the target gear is synchronised clutch-to-clutch shifting can be performed. This aspect of shifting applies similar methods to those performed in automatic transmissions where hydraulically actuated clutches are simultaneously released and engaged, minimising loss of tractive load to the road. The most significant change from AT to DCT clutch control is that there is no longer a hydrodynamic torque converter to dampen any transients developed during shifting. This therefore requires a much more precise application of clutch control to ensure shifting is completed within the minimal time with maximum quality.

2.2. Conventional powertrain modelling

Dual clutch transmission equipped powertrains are similar to conventional automatic powertrains with the exception of no isolation of engine and transmission using the torque converter. Thus powertrain modelling should consider the application of engine models that contain both output torque and engine harmonics. It is therefore necessary to create a reasonably complex engine model to capture torque from piston firing as well as variation in inertia in the moving pistons, connecting rod, and crank shaft. The powertrain itself does not significantly differ from the proposed powertrain in [5] where major powertrain components of flywheel, clutch drum, transmission and vehicle inertias are combined with the engine model to create a compact vehicle powertrain equipped with a DCT. In Fig. 2 below, this model in combined with a four degree of freedom model of the engine to represent an inline 4 cylinder engine. This model is structured to achieve clutch-to-clutch shifts between two gears connected to a single output shaft, presented in Fig. 2, with equations of motion in Eqs. 1-8 in the clutch open condition.

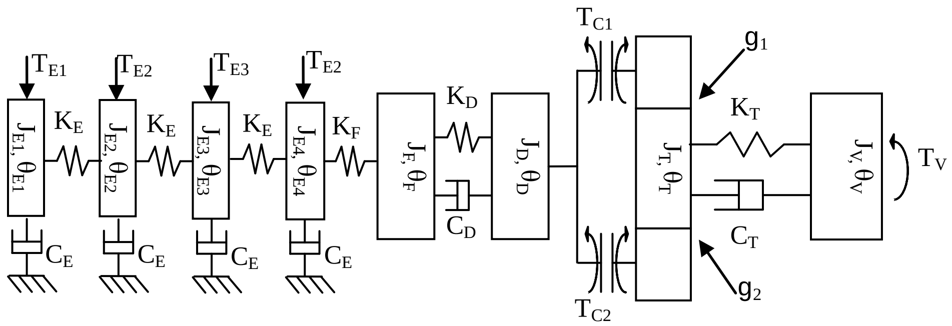

Figure 2: Eight degree of freedom simplified powertrain model

$$J_{E1}\ddot{\theta}_{E1} - C_E\dot{\theta}_{E1} - K_E(\theta_{E1} - \theta_{E2}) = T_{E1} \quad (1)$$

$$J_{E2}\ddot{\theta}_{E2} - C_E\dot{\theta}_{E2} + K_E(\theta_{E1} - \theta_{E2}) - K_E(\theta_{E2} - \theta_{E3}) = T_{E2} \quad (2)$$

$$J_{E3}\ddot{\theta}_{E3} - C_E\dot{\theta}_{E3} + K_E(\theta_{E2} - \theta_{E3}) - K_E(\theta_{E3} - \theta_{E4}) = T_{E3} \tag{3}$$

$$J_{E4}\ddot{\theta}_{E4} - C_E\dot{\theta}_{E4} + K_E(\theta_{E3} - \theta_{E4}) - K_F(\theta_{E4} - \theta_F) = T_{E1} \tag{4}$$

$$J_F\ddot{\theta}_F - C_D(\dot{\theta}_F - \dot{\theta}_D) - K_D(\theta_F - \theta_D) + K_F(\theta_{E4} - \theta_F) = 0 \tag{5}$$

$$J_D\ddot{\theta}_D + C_D(\dot{\theta}_E - \dot{\theta}_D) + K_D(\theta_E - \theta_D) = -T_{CL1} - T_{CL2} \tag{6}$$

$$J_T\ddot{\theta}_T - C_T(\dot{\theta}_T - \dot{\theta}_V) - K_T(\theta_T - \theta_V) = \gamma_1 T_{CL1} + \gamma_2 T_{CL2} \tag{7}$$

$$J_V\ddot{\theta}_V + C_T(\dot{\theta}_T - \dot{\theta}_V) + K_T(\theta_T - \theta_V) = -T_V \tag{8}$$

Where J, C, K, γ and T represent inertia, damping, stiffness, gear ratio, and torque, respectively. Also θ represents the angular displacement of each degree of freedom, and is complemented by its time derivatives $\dot{\theta}$ for velocity and $\ddot{\theta}$ for acceleration. For these equations of motion counter-clockwise rotation is taken as the positive direction of rotation in Eqs. 1 to 6. As the direction of rotation is influenced by reduction gears in Eqs. 7 and 8 clockwise is taken as the positive direction of rotation. Subscripts E1-4 represents the four engine elements, E is engine, F is flywheel, D is clutch drum, T is transmission, V is vehicle, and CL refers to clutch 1 or 2. While eqs. 1 to 8 represent the open clutch model, there are two other transmission states, Clutch 1 closed, and Clutch 2 closed. For both of these states eqs. 6 and 7 are replaced with eq. 9 for clutch 1 and eq. 10 for clutch 2 closed. As a result the model reduces by one DOF as clutch drum and transmission elements merge.

$$\left(\gamma_1^2 J_D + J_T\right)\ddot{\theta}_D - K_D\gamma_1(\gamma_1\theta_T - \theta_F) - C_D\gamma_1(\gamma_1\dot{\theta}_T - \dot{\theta}_F) + K_T(\theta_V - \theta_T) + C_T(\dot{\theta}_V - \dot{\theta}_T) = 0 \tag{9}$$

$$\left(\gamma_2^2 J_D + J_T\right)\ddot{\theta}_D - K_D\gamma_2(\gamma_2\theta_T - \theta_F) - C_D\gamma_2(\gamma_2\dot{\theta}_T - \dot{\theta}_F) + K_T(\theta_V - \theta_T) + C_T(\dot{\theta}_V - \dot{\theta}_T) = 0 \tag{10}$$

2.3. Engine torque models.

Piston firing models of the engine that can be rapidly implemented are available in Taylor [18], where variation of crank, piston, and connecting rods are defined as a function of crank angle and crank speed, and by combination with gas torque from piston firing can be modified to the desired configuration. Fig. 3 presents piston head pressure at 0 and 100% throttle, linear interpolation is used to vary piston head pressure for the purpose of throttle control, with the percent throttle determined for each piston at the beginning of the intake stroke. This model introduces a delay in engine control not present in look up table models or other methods for simulating the engine output torque, such as those used in [1,4].

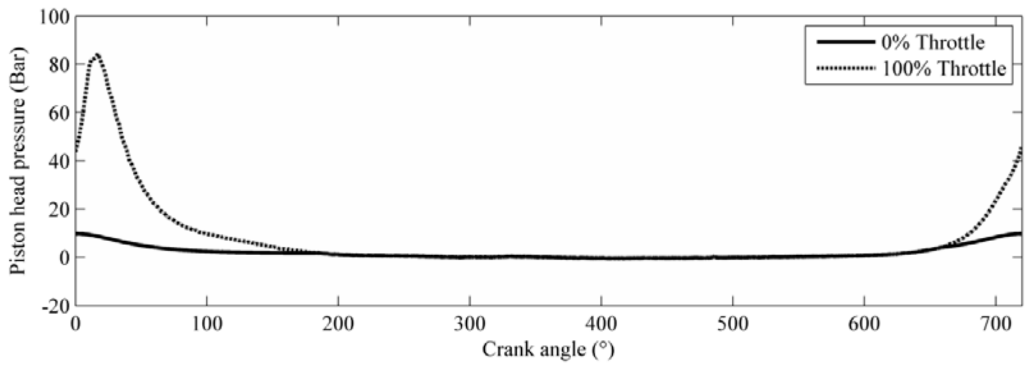

Figure 3: Piston head pressure distribution

2.4. Clutch torque model

Clutch torque is defined using the piecewise clutch model presented in [5]. This determines clutch state and is based on the stick-slip algorithm with four elements relating clutch piston displacement, slip speed and average torque in the clutch. The piecewise clutch model is:

$$T_C = \begin{cases} 0 & X < X_0 \\ n\mu_D \dfrac{r_0^3 - r_I^3}{r_0^2 - r_I^2} \times F_A & X \geq X_0, \left|\Delta\dot{\theta}\right| \geq 0* \\ T_{avg} & X \geq X_0, \left|\Delta\dot{\theta}\right| < 0*, T_{avg} < T_{C,S} \\ n\mu_S \dfrac{r_0^3 - r_I^3}{r_0^2 - r_I^2} \times F_A & X \geq X_0, \left|\Delta\dot{\theta}\right| < 0*, T_{avg} \geq T_{C,S} \end{cases} \tag{11}$$

Where, n is the number of friction plates, X is piston displacement and X_0 is the minimum displacement required for contact between friction plates, μ_D is dynamic friction, μ_S is static friction, r_0 and r_I are the outside and inside diameters of the clutch plates, and F_A is the pressure load on the clutch. T_{avg} is the average clutch torque derived from the model dynamics using the equations of motion (1-8), as follows

$$T_{avg} = \frac{T_{C1a} + T_{C1b}}{2} \tag{15}$$

$$T_{C1a} = -J_D\ddot{\theta}_D - K_D(\theta_D - \theta_E) - C_D(\dot{\theta}_D - \dot{\theta}_E) - T_{C2,1} \tag{16}$$

$$T_{C1b} = \left(-J_T\ddot{\theta}_T + K_T(\theta_V - \theta_T) + C_T(\dot{\theta}_V - \dot{\theta}_T) + \gamma_{2,1}T_{C2,1}\right)/\gamma_{1,2} \tag{17}$$

2.5. Vehicle resistance torque model

Vehicle torque models are well established, comprising of rolling friction losses, aerodynamic drag, and the impact of road incline. The vehicle torque is derived as:

$$T_V = \left(M_v g \sin(\varphi) + \frac{1}{2} \rho_{air} A_v C_D V^2 + M_v g f_T \right) \times R_W \tag{18}$$

Where M_v is vehicle mass, g is gravity, φ is angle of inclination, A_v is vehicle area, C_D is drag coefficient, V is vehicle speed, ρ_{air} is air density, and f_T rolling resistance coefficient.

2.6. Hybrid configuration and modelling

The flexibility of hybrid vehicles with many choices in engine/electric machine configuration enables the study of different electric machine locations on vehicle performance and the impact on transient vibration suppression. In this section two configurations are considered. The electric machine will be positioned at the flywheel in Case 1, see Fig. 4 (a), and in Case 2 it is located on the output shaft, see Fig. 4 (b). This enables different methods for evaluating torsional vibration suppression where the electric machine will have variable torque multiplication through the transmission when located at the flywheel, and constant torque multiplication if located at the output shaft.

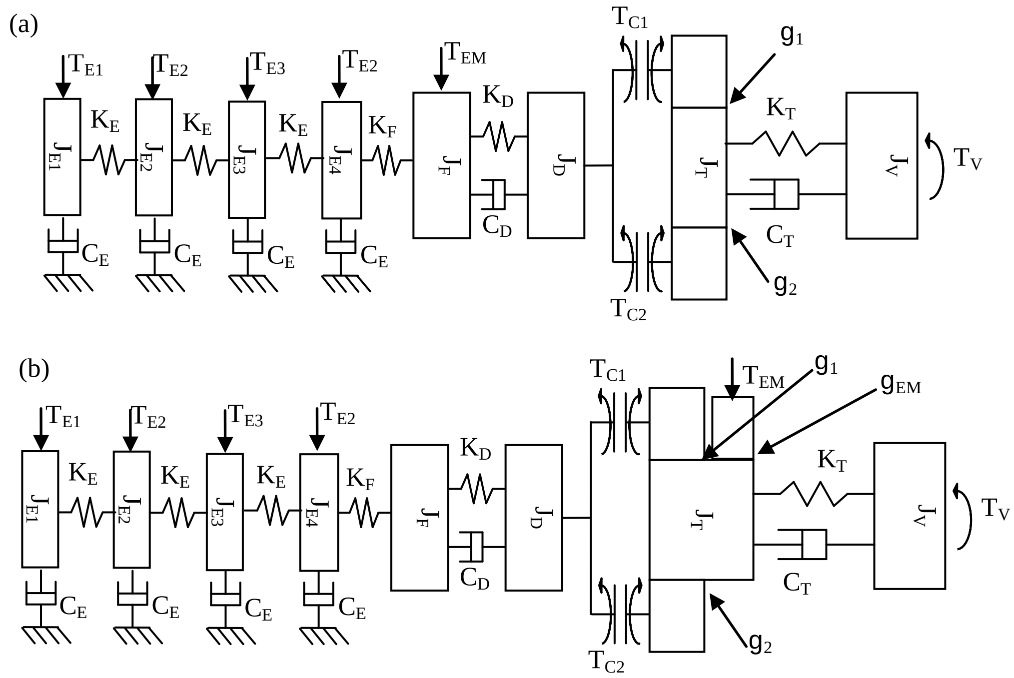

Figure 4: 8-DOF hybrid vehicle powertrain model, (a) electic machine located at the flywheel, and (b) electric machine located at the DCT output shaft

The two powertrain configurations presented in Fig. 4 are two of a range of options for locating the electric machine for hybrid vehicle powertrains. In Fig. 4 (a) the electric machine is located with the engine at the flywheel, such configurations result in the EM torque being multiplied through the currently engaged gear depending on shift requirements. However, depending on EM sizing and a range of design considerations, it is also convenient to locate the EM at the output shaft of the transmission using a constant gear ratio for power conversion; this is shown in Fig 4. (b). In this configuration torque is multiplied by a constant ratio over the entire vehicle speed range.

For both models the engine torque is derived from lookup tables of EM efficiency and output torque for a desired power and speed. Eqs. 5 and 7 must be modified to include engine torque for the models in Fig. 4 (a) and Fig. 4(b), respectively.

$$J_F\ddot{\theta}_F - C_D\left(\dot{\theta}_F - \dot{\theta}_D\right) - K_D\left(\theta_F - \theta_D\right) + K_F\left(\theta_{E4} - \theta_F\right) = T_{EM} \tag{19}$$

$$J_T\ddot{\theta}_T - C_T\left(\dot{\theta}_T - \dot{\theta}_V\right) - K_T\left(\theta_T - \theta_V\right) = \gamma_1 T_{CL1} + \gamma_2 T_{CL2} + \gamma_{EM} T_{EM} \tag{20}$$

3. Integration of active powertrain damping with up-shift control strategy

There are a number of considerations required for implementing the control strategy and its integration with gear shift control. It is well established that powertrain oscillations will be observable during transient driving conditions, such as tip-in/tip-out, clutch engagement, or in the presence of backlash excitation. For the time being, both tip-in and backlash aspects of this study are excluded as studies by Berriri [6,7] and Bruce [9] have already considered these issues. For the purposes of this paper only upshift control is considered, this is usually the most critical component of shifting as it frequently occurs under hard acceleration, and poor shifts will be more observable.

Key to the implementation of any minimise the requirement for additional sensor requirements. The modern automotive powertrain has a number of speed sensors, including at the engine flywheel, transmission output and wheel hubs. Respectively, these are employed for engine, shift and launch control and anti-lock braking control. Active damping of the powertrain can then utilise any of these three sensors to supress transients resulting from gear change.

The complete up shift in a DCT equipped powertrain is shown in Fig. 5 and can be divided into four steps, these are as follows:

1. Shift preparation –The synchroniser is engaged for the target gear. Then the engaging clutch hydraulic cylinder is filled with fluid to the point of contact in friction plates. At the same time the releasing clutch pressure is reduced such that the clutch static friction torque approaches the friction limit according to eq. 11.
2. Torque phase – The engaging and releasing clutch torques are manipulated to transfer driving torque load to the engaging clutch, requiring estimation of the torque driving the vehicle through the engaging clutch as a reference load. It is common in this step for engine torque to be manipulated to minimise torque hole during shift [1].
3. Inertia phase – This is when primary driving load is on the engaging clutch and slip speed in the friction plates are synchronised to complete the shift. Transmission output speed control is used to minimise vibration during gear

shift, and engine torque manipulation is used to control the duration of shift time.

4. Post shift damping – After clutch lockup powertrain oscillations are suppressed using speed sensor inputs and modifying either the engine or motor output torque, depending on the powertrain configuration.

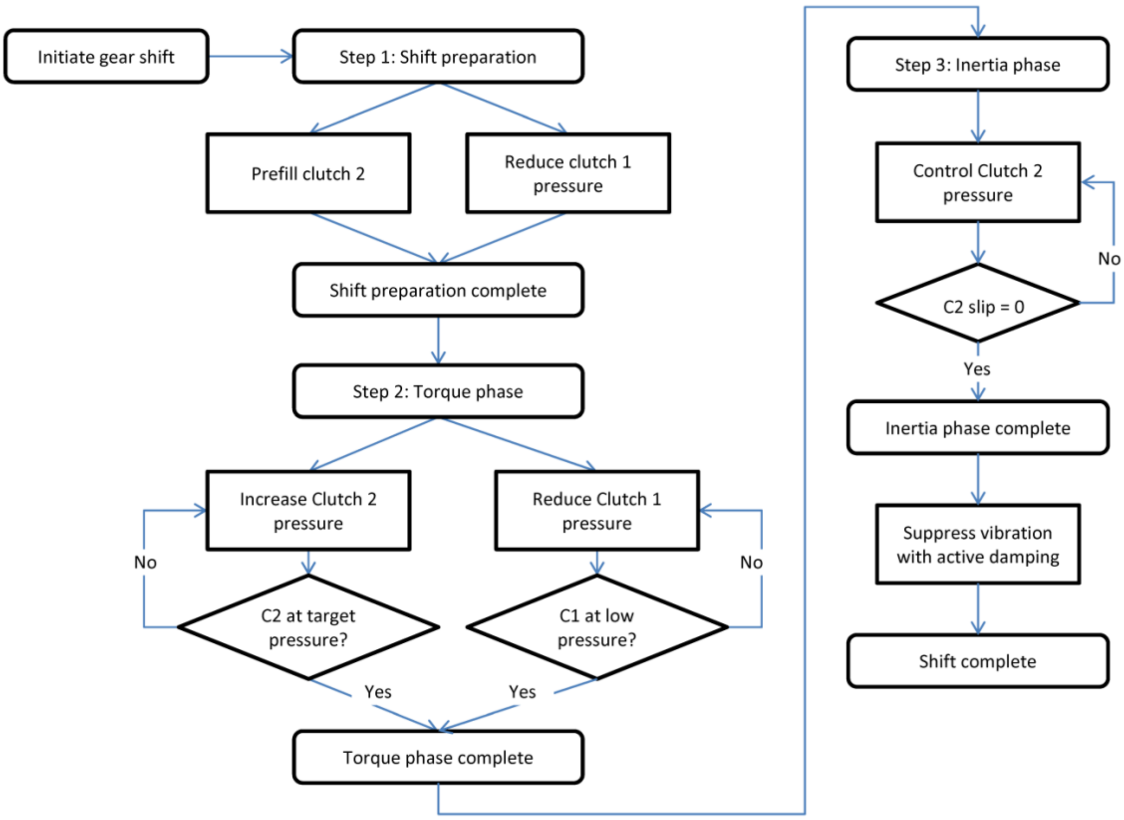

Figure 5: Flow chart of shift control for up shifting in a DCT

4. Simulations of shift control of DCT equipped powertrains

4.1. A comparison of shift control with and without slip speed control

To demonstrate the shift process in the DCT the model configuration in Fig. 2 is used in conjunction with the described shift process in steps 1-4 above, with clutch hydraulic model derived in Walker [5]. The selected gears are 3rd engaged in the transmission with 4th targeted for the up shift. Initial engine speed is 400 rad/s, equating to a vehicle speed of approximately 88 rad/s. These conditions are maintained for the remaining simulations in this paper.

The shift transient results are presented in Fig. 6 with clutch speeds in (a) and vehicle and transmission speeds in (b). Clutch fill takes approximately 100ms as C2 piston is filled to contact in the friction plates in the wet clutch, while the torque phase takes approximately 50ms. At this point the inertia phase begins and the clutch drum speed is reduced to clutch 2 speed. The results demonstrate that the slip speed control is capable of reducing vibration rapidly during the inertia phase before the clutches lockup. As indicated in Fig. 6 (b) there is still significant response in the powertrain resulting from clutch lockup, and low damping between vehicle and transmission results in limited reduction of the response. The results are fairly for typical simulated clutch-to-clutch shifting in DCTs, with the inclusion of higher frequency forced vibration from the engine model.

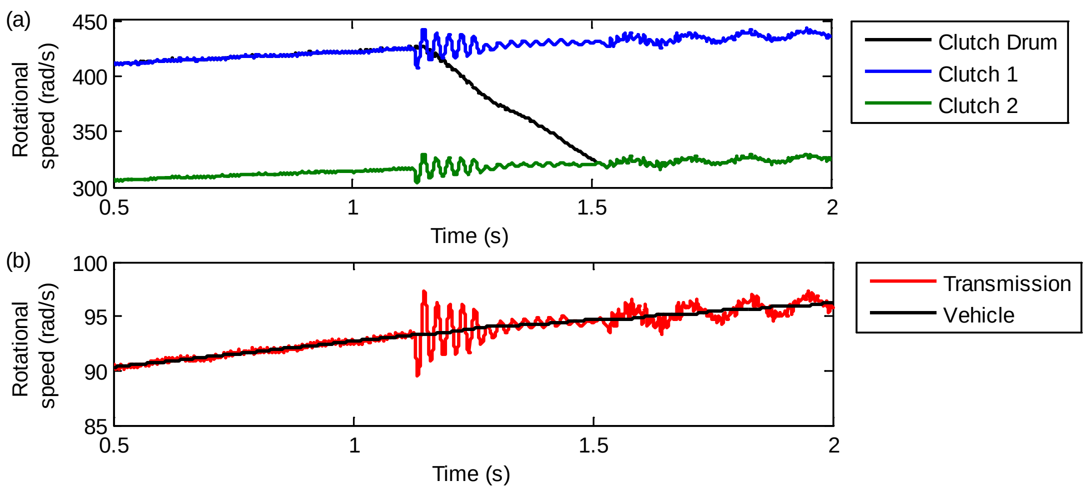

Figure 6: Shift dynamics of a DCT equipped powertrain using the shift process described in Fig. 5, (a) clutch speeds, and (b) vehicle speed

To demonstrate the importance of the combination of torque and speed control in DCTs equivalent simulations to those presented in the previous section are conducted without the inclusion of slip speed control in the inertia phase. The exclusion of slip speed control is equivalent of the clutch pressure being controlled during the torque phase to hit the mean torque, and in the inertia phase, being maintained at its mean torque with no additional control to suppress vibration in the inertia phase. These results, shown in Fig. 7, demonstrate that the exclusion of slip speed control increases the powertrain vibration upon shift completion. The primary change to simulations in Fig. 6 is the continuation of vibration in the transmission at comparatively high amplitude in the transmission. As a consequence, the resulting post shift vibration is significant in the powertrain, particularly when comparing Fig 6 (b) to Fig. 7 (b).

It is worth noting here, that the additional inertia phase control to reduce transient response during the shift results in additional reduced deceleration of the clutch drum through comparison of Fig. 6 (a) to Fig. 7 (a). This results from variation in clutch 2 torque to reduce powertrain vibration. Nevertheless, the study shows that combined torque phase and inertia phase control can be used to significantly improve powertrain response during shifting.

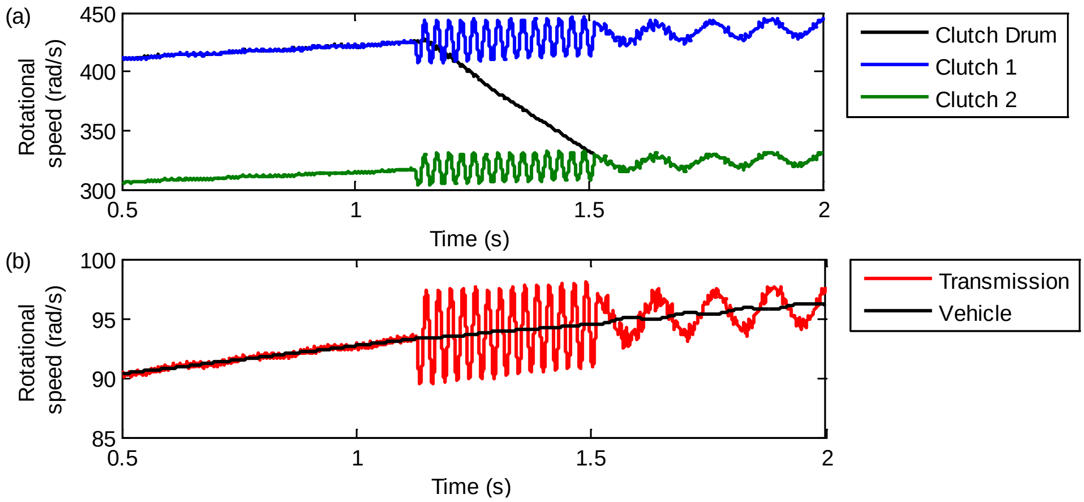

Figure 7: Shift dynamics of a DCT equipped powertrain using the shift process described in Fig. 5 with slip speed control (Fig 5 point 3A) disabled, (a) clutch speeds, and (b) vehicle speed

The combination of alternate control strategies during the inertia and torque phases also improves powertrain response. However, these results in Fig. 6 and 7 also demonstrate that torsional vibration is not completely eliminated using alternate clutch control strategies. This suggests that additional measures are required to achieve maximum possible shift quality in DCT equipped powertrains.

4.2. Vibration suppression with engine control for a conventional powertrain

After the shift completes there is obvious response in the powertrain, which is caused by the change in several powertrain variables, e.g., primarily engine speed and inertia in the transmission, but also potentially results from inaccuracy in output torque estimation for clutch control. To suppress the powertrain response it is suggested here that the variation of engine torque can be used to suppress this response.

Fig. 8 represents the suggested control strategy for manipulating engine output torque to suppress the powertrain vibration. It makes use of the difference between vehicle and transmission speeds to modify the driver demand signal and control engine output torque. Relative speeds are chosen at this point as the oscillation between transmission and vehicle inertias is indicative of poor powertrain response that can be observed by the driver. To compensate for the relative speed in the powertrain the PID control supplies a signal output that is superimposed with driver demand, while driver demand is limited to 85% throttle such that the controller can make use of increasing and decreasing engine torque to actively reduce vibration without significantly flaring throttle, resulting in abrupt variation in engine speed.

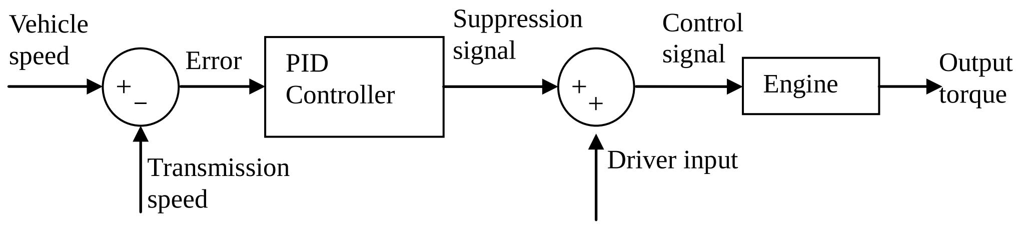

Figure 8: Engine control strategy for DCT vibration suppression

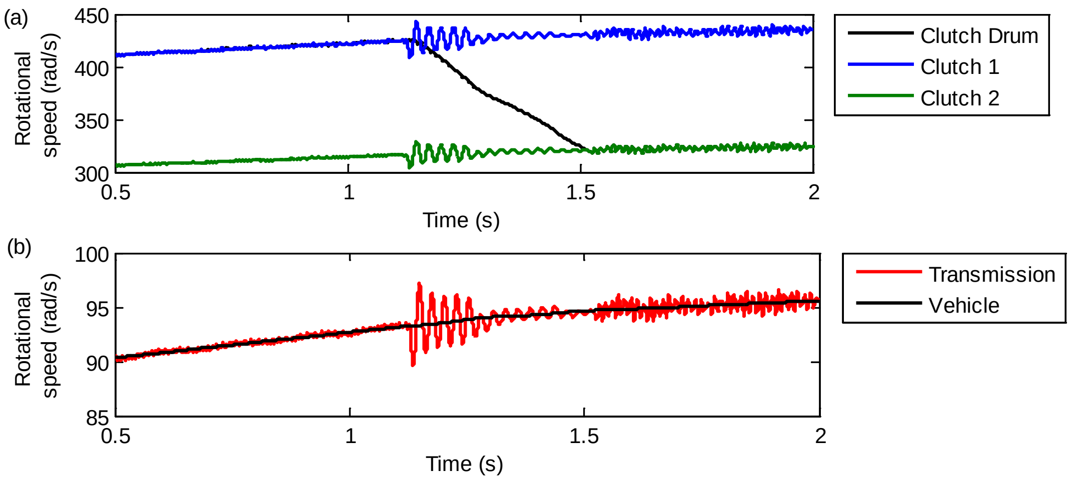

Figure 9: Shift dynamics of a DCT equipped powertrain using the shift process described in Fig. 5 using vibration suppression in the engine, (a) clutch speeds, and (b) vehicle speed

Simulation results shown in Fig. 9 and Fig. 10 demonstrate the application of engine control to suppress torsional vibration in the vehicle powertrain. Here the powertrain is modelled according to Eqs. 1-8, and the standard shift control strategy are used with the addition of engine throttle manipulation, the goal being the rapid reduction of vibration in the vehicle powertrain. The results are quite promising; with the rapid reduction of vibration in the vehicle speed after gear shift is completed. In Fig. 9 (a) results show comparable shift transients to Fig. 6 (a), however by the completion of the first period of oscillation after shifting is complete there is significant suppression of vibration at this clutch. This result continues through to the transmission and vehicle speeds, with vibration in the vehicle speed rapidly suppressed. Complementing these results is the instantaneous piston torques and throttle control results in Fig. 10 (a) and (b), respectively. It is important to note here that at the completion of shift it takes two piston firings in the engine before output torque increases, suggesting that time delay is an important factor in engine control for DCTs. These results suggest that the application of engine throttle control can be used to rapidly suppress powertrain vibration post gear shift in DCT equipped powertrain.

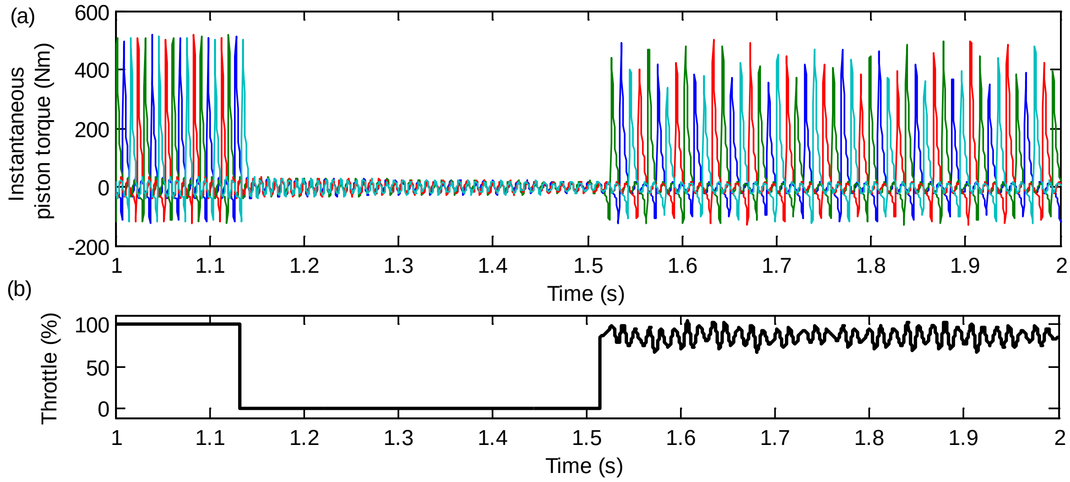

Figure 10: Engine output and control corresponding to Fig. 10. (a) Gas and inertia torque for individual pistons, and (b) throttle angle manipulation for vibration suppression

For comparative purposes engine control is also conducted using clutch control without inertia phase control, here clutch torque is maintained at a mean torque, similar to the simulation in Fig. 7. While the transient period is very similar, the post shift response is significantly improved, within two vibration periods the oscillations are suppressed. Giving the amplitude of transmission vibration in Fig. 7 (b) this is a very important result, as significantly higher oscillations are suppressed in a similar duration to those of a much more successful shift. Thus these simulation results in Figs. 9 to 10 demonstrate that is the application of engine throttle manipulation for suppression of powertrain response can be used to improve the shift quality of a lightly damped powertrain, actively controlling vibration response in a lightly damped powertrain.

4.3. Vibration suppression with an electric machine for a hybrid vehicle

The two hybrid vehicle powertrain configurations presented previously provide alternative methods for active controlling of the powertrain response. The application of an electric machine to active vibration suppression has significant advantage in terms of less time delay and more controlling torque than that of an internal combustion engine in which throttle response limits the capability for output torque variation. The EM control method for vibration suppression is presented in Fig. 11, here vibration between vehicle and transmission is detected and torque suppression requirements are requested from the EM. Unlike vibration suppression with the engine, engine control remains with the driver and EM torque independently suppresses vibration. Thus the driver is less likely to notice the effect of vibration suppression on vehicle performance than would be the case with engine throttle manipulation.

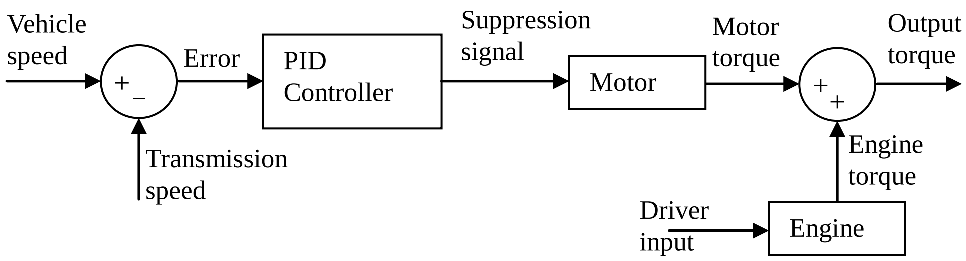

Figure 11: EM control strategy for DCT vibration suppression

To demonstrate the effectiveness of the use of the EM for vibration suppression, simulations are presented in the following two figures. Fig. 12 presents results for vibration suppression using the hybrid vehicle configuration presented in Fig. 4 (a), while Fig. 13 presents the results for Fig. 4 (b). For these simulations the controller used has inertia phase control deactivated to present a worst case scenario, and initial EM and vehicle speeds are 400 rad/s and 88 rad/s respectively.

The simulation results for the hybrid vehicle configurations demonstrate a capability to rapidly suppress transient response in the powertrain after gearshift completes. The combination of slip speed control and vibration suppression is to rapidly reduce undesirable transient vibration in the powertrain. In Fig. 12 (b) the transmission speed drops below the nominal vehicle speed after the completion of shifting, and the EM rapidly counters the introduced vibration by supplying torque to accelerate the transmission to the nominal speed and suppress vibration. These results are further improved in Fig. 13 (b) with the location of the EM capable of directly impacting on powertrain response.

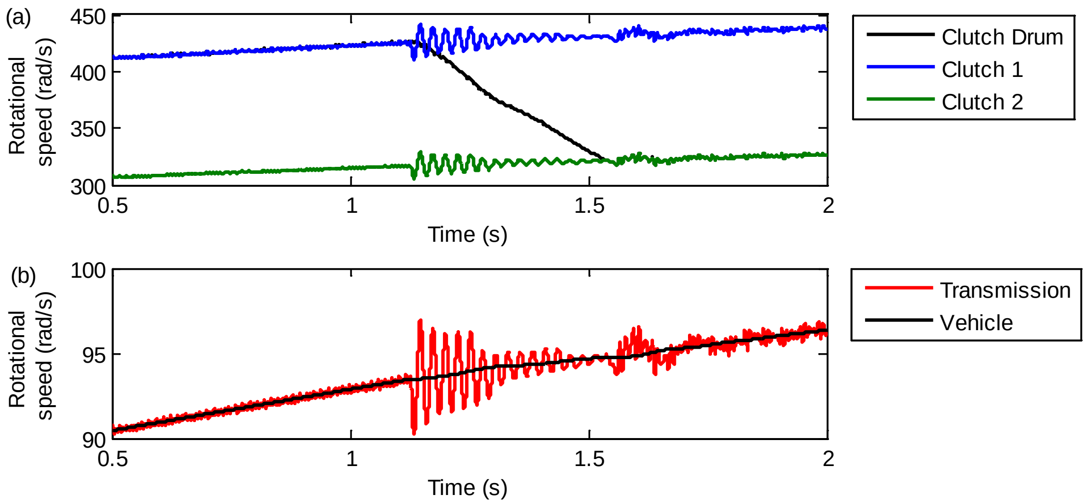

Figure 12: Response of a hybrid DCT equipped powertrain for the configuration in Fig 4. (a), using an electric machine for vibration control, (a) clutch speeds, and (b) vehicle speed

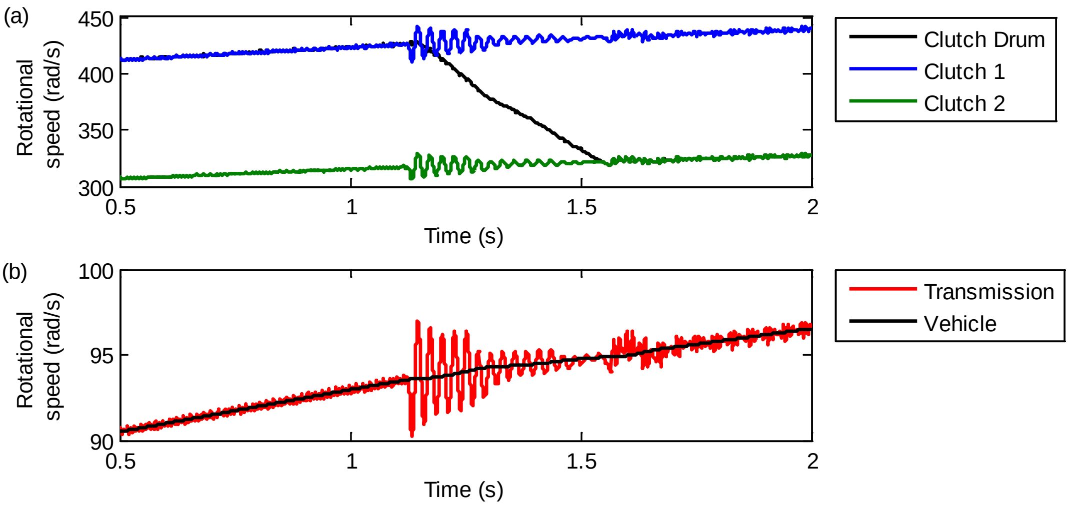

Figure 13: Response of a hybrid DCT equipped powertrain for the configuration in Fig 4. (b), using an electric machine for vibration control, (a) clutch speeds, and (b) vehicle speed

5. *Conclusion*

The control of shifting in dual clutch transmission equipped powertrains has been studied in this paper. To conduct this research a powertrain model is presented with detailed engine model, such that the engine torque variation resulting from piston firing is simulated, while a more compact look-up table is used for motor torque. These strategies are utilised to demonstrate the variation in delay between internal combustion engines and electric motors, with the EM being capable of faster response to variation in torque demands. Simulations of shift control with and without inertia phase control of the clutches for a conventional powertrain demonstrated the importance of inertia phase on minimising powertrain vibration during and after shifting. Implying that variations in the quality of gear shift can be significant. The obtained results show that it was not possible to completely eliminate the transient vibration in the powertrain, even when using inertia phase control.

To suppress the post shifting powertrain vibration, which is introduced at the completion of clutch changeover, active vibration control using the engine for conventional powertrains, and an electric motor for hybrid vehicle powertrains is suggested. This vibration suppression makes use of variation in engine or EM output torque to suppress powertrain response after gearshift. The results of engine control demonstrate the capability to successfully suppress these vibrations rapidly; however control is limited by delay in piston firing and the ability to supply high torque variation while maintaining vehicle speed. Application of the same strategy to hybrid vehicles was also successful regardless to the location of electric machine. The higher torque available in the EM combined with significantly less time delay also contributes to the improved control. These results have demonstrated that active engine or EM control can be used to successfully suppress vibration in vehicle powertrains where there is traditionally insufficient damping to provide rapid passive vibration suppression.

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