WHEEL SLIP AND ENGINE TORQUE PULSE SIMULATION FOR TRANSMISSION-IN-THE-LOOP EXPERIMENTATION

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Abstract

This paper presents the development of a transmission-in-the-loop (TiL) experimentation system. In this TiL experimental setup, the input side of the transmission is controlled by a dynamometer emulating the engine, while the output sides of the transmission are controlled by two dynamometers emulating the wheels and vehicle. The models emulating these vehicle components are required to possess sufficient fidelity to simulate engine torque pulse (ETP) and wheel slip dynamics while being computationally efficient to run in real-time. While complex engine and tire models exist in the literature that accurately capture these dynamics, they are often too numerically stiff for real-time simulation. This paper presents the system level details of such a TiL setup, and the modeling concepts for the development of high fidelity real-time models of the engine and tire dynamics for use in this experiment. Parameters of the engine model are identified using experimental data. Vehicle drive cycle, torque pulse, and wheel slip test results are presented.

INTRODUCTION

A modern automotive system is typically developed using a variety of prototyping techniques. Virtual prototyping is an efficient way of supporting the initial design process, but for more accurate analysis and testing of complex systems, the need for physical prototyping still exists. Standalone physical prototyping for large and complex dynamic systems, such as modern day vehicles, can be an expensive and time consuming process as the number of iterations required before the final product could be very high. Hardware-in-the-loop (HiL) experimentation is a synergistic combination of physical and virtual prototyping. It is an approach that prototypes parts of a given system in hardware and virtually emulates the rest of the system while maintaining bidirectional information flow between these physical and virtual subsystems. Such HIL simulations are already extensively being used to evaluate the performance of complex embedded control systems, and are increasingly becoming an integral part of testing, calibration, and validation of complete automotive powertrain subsystems.

Automotive transmissions are a key component of the vehicle powertrain system, and require significant testing in various use-case scenarios, as well as for calibration and control system development as integrated systems. With the advent of alternate propulsion options, such as those in hybrid vehicles, where the dynamic load on the transmission changes in multiple ways, and engines/transmissions require more sophisticated control, testing and validations processes become even more complex. As a result, HIL experiments with transmission hardware, called transmission-in-the-loop (TiL) experiments, are becoming increasingly popular. It consists of an automotive transmission driven by an input dynamometer emulating the engine, and output dynamometer(s) emulating the vehicle. This arrangement allows the transmission to be subjected to transient loads, as seen during vehicle drive cycle operation, and enables fast and accurate testing of the transmissions under realistic in-vehicle conditions, while offering consistent, controlled repeatable evaluation scenarios.

Successfully designing a high fidelity TiL test system is usually a challenging task. TiL systems require simultaneous control of the input and output dynamometers, often at very high update rates. One of these dynamometers is usually motoring, while the other(s) is absorbing, and hence they often share the same power source through a regenerative unit. From a modeling standpoint, having high fidelity engine and vehicle models capable of accurately representing higher order dynamics such as engine torque pulses (ETP) and singularly perturbed dynamics such as that of wheel slip, while having real-time properties, is usually an arduous task. This is one of the reasons the literature is relatively deficient in highlighting the development/emulation details for such systems. Megli et al. [1] present the details of such an experimental setup at Ford Research Laboratory for subjecting the transmission to transient loads, but used a real engine to drive the transmission. Steiber et al. [2] and

Castiglione et al. [3] describe the development of TiL test system for General Motors, to study the effect of ETP simulation (ETPS) on the input side while the output is subjected to simple road loads, but a significant portion of the modeling details, of interest to a systems and controls engineer, are omitted. The objective of this work is to describe the TiL system setup developed by A&D Technology for testing front wheel drive (FWD) automatic transmissions during longitudinal motion of the vehicle, and present the modeling details for ETP emulation on the input side, and wheel slip emulation on the output side.

The proposed approach is to model the engine as a second order system producing mean torque while the combustion and inertia torque pulses, identified from experimental data in the form of a grey box model, are superimposed on this response. The engine controller is implemented in the simulation environment for idle speed control, and for torque management during gear shifts. The longitudinal tire model is adopted from the literature and transformed into a hybrid automaton using kinematic relationships at higher velocities to allow faster numerical integration, as required for real-time simulation. Dynamic models of the dynamometer and the hardware are also developed to enable testing in a pure simulation stage on a PC or a real-time system. The remainder of the paper is organized such that section 2 describes the system hardware and architecture, section 3 presents the engine emulation for the input dynamometer, and section 4 introduces the wheel slip and gross vehicle dynamics for control of the output dynamometers. Finally, section 5 concludes the paper with a summary and view of initial test data.

SYSTEM DESCRIPTION

This section presents the description of the TiL test setup. The TiL system architecture is shown in Figure 1. The figure shows the transmission hardware connected to high bandwidth, low inertia permanent magnet dynamometers emulating the input and outputs of the transmission and being controlled through the models running on Procyon, a high-performance, multi-core, multi-CPU real-time system developed by A&D. The speed/torque setpoints are given to the dynamometer drive from the engine/vehicle models via fiber optic communication. The drive then controls the motor to produce the desired speed and torque setpoints. The speed and torque response of the transmission to these setpoints are sensed at the dynamometer shafts using an encoder and torque meter respectively, and communicated to the model as PWM signals. The transmission control unit (TCU) is also physically present in the system, and communicates with the emulated engine controller on Procyon through CAN messaging. The real-time simulation system can also directly control transmission shift solenoids, enabling the evaluation of various transmission shift strategies without reprogramming the TCU. This system is capable of emulating the intended dynamics during the longitudinal motion of the vehicle at an update rate of 250µs in both forward and reverse modes of the transmission.

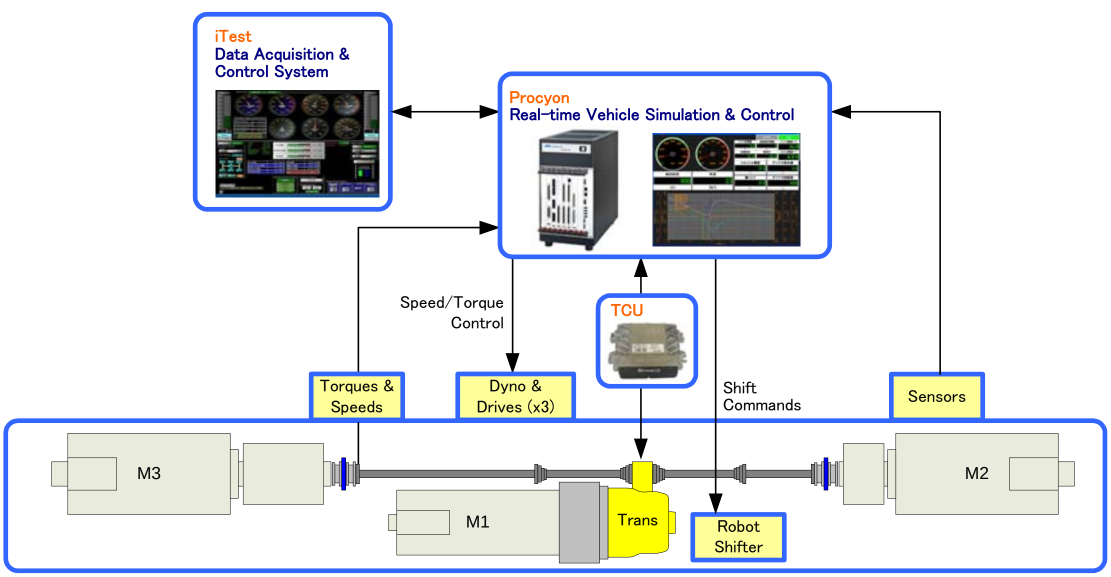

Figure 1: Transmission-in-the-Loop test system architecture

The unit under test hardware shown in Figure 1 consists of a FWD automatic transmission system and its controller (TCU). This transmission assembly includes the flywheel, torque converter, an epicyclic gear train, an open differential, and two half-shafts. This assembly also houses the transmission pump, clutches and the associated hydraulic components. The input to this transmission assembly is from an input dynamometer (M1) consisting of a 370 kW 3-phase AC permanent magnet synchronous motor and powered by a flux-vector drive inverter that is controlled to emulate the engine torque pulse. Since the objective here is to subject the transmission input to torque pulses, the engine model commands the torque setpoints to the drive, and the speed of the input dynamometer shaft is measured and communicated to the engine model as the engine speed. The drive contains open-loop torque estimators, and varies the duty cycle and switching frequency of the voltage to achieve the desired current required to produce the desired torque. The connecting shaft of this motor to the flywheel contains torque and speed sensors, and is designed to be extremely small and stiff to prevent the degradation of the system's fidelity in measuring the engine speed.

The outputs of the transmission assembly consists of two half-shafts of different lengths, as in a real vehicle, and these half-shaft ends are controlled using two output dynamometers (M2, M3), each consisting of a 400 kW 3-phase AC permanent magnet synchronous motor and a flux-vector drive inverter. The vehicle model on Procyon commands speed setpoints to these output dynamometers, and the measured torque is fed back to the model for simulation. This arrangement enables closed-loop control of the speed setpoints, which is much easier to achieve in practice than closed-loop torque control. Under certain operating conditions, the two dynamometers may measure significantly different torques, due to the presence of unequal lengths of half-shafts, highlighting the phenomenon that contributes to torque steer in vehicles. The size of the engine to be emulated is limited by the power rating of the input dynamometer and its inertia, while the effective wheel inertia of the vehicle and desired braking performance dictates the size of the output dynamometers. Although the details presented here pertain to the testing of FWD automatic transmissions, the test system is quite flexible and can easily be modified to accommodate various other propulsion configurations.

ENGINE MODEL AND TORQUE PULSE EMULATION

This section describes the engine model for torque generation and torque pulse (ETP) simulation for the input motor of the TiL system. The engine simulation can be described by classifying the model into four separate subsections: A. Mean torque based engine model, B. Inertia compensation, C. Construction of engine torque pulse, and D. Engine controller emulation. Figure 2 provides an overview of the various torque calculation and execution components.

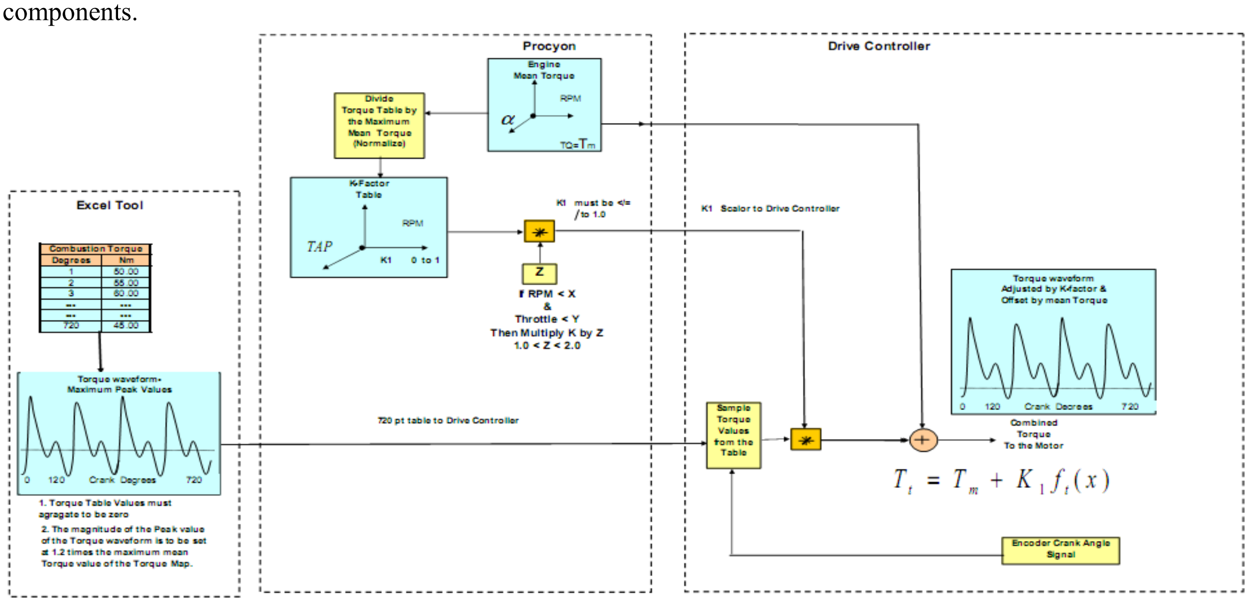

Figure 2: Engine torque simulation overview

A. Mean Torque Based Engine Model

The engine model is required to generate the torque value that the engine supplies to the transmission input shaft. The literature contains a number of engine models capturing basic engine dynamics and torque production, including torque pulses, in the form of data-driven black-box models or physics-based models. However, these models often require extensive data to be generated or detailed information about engine not usually available to the transmission engineers. Physics-based models that simulate the combustion dynamics also tend to be numerically stiff due to the time scale separation inherent in chemical processes, and hence are not suitable for real-time simulation. One of the most significant reasons for the unsuitability of the available ETP models is the bandwidth of the communication interface between the simulation system and motor drive. The drive's optical fiber link used here to communicate torque and speed setpoints has a bandwidth limitation of 4kHz. Hence the highest frequency of the digital signal that can be transmitted without aliasing is 2kHz. Based on past experience, at least 12 sample points are required to faithfully reconstruct the shape of the engine torque pulse. Thus, the maximum frequency content that the engine model can communicate to the drive for accurate reconstruction is 167Hz, which falls short of the required bandwidth to accurately capture ETP for the entire operating range of typical engines (up to 8000 rpm). The above problem is tackled by using mean torque-based models on the real-time simulation system and creating the torque pulse using the very high switching frequency of the input motor drive. The variable frequency AC motor drive can store crank angle-based torque maps that can be modulated through the fiber communication. Hence the complete torque pulse is de-constructed into a mean torque, an inertia pulse and a combustion pulse, as shown in Figure 3. The mean torque is commanded through the optical fiber, whereas inertia and combustion torque pulses are created using two crank angle-based maps directly in the drive, and the superposition of these signals is used as the required torque setpoint. The mean torque based model treats the engine as an effort source based on steady-state maps, and takes into account the engine inertia and frictional losses, modeled as a linear resistive element.

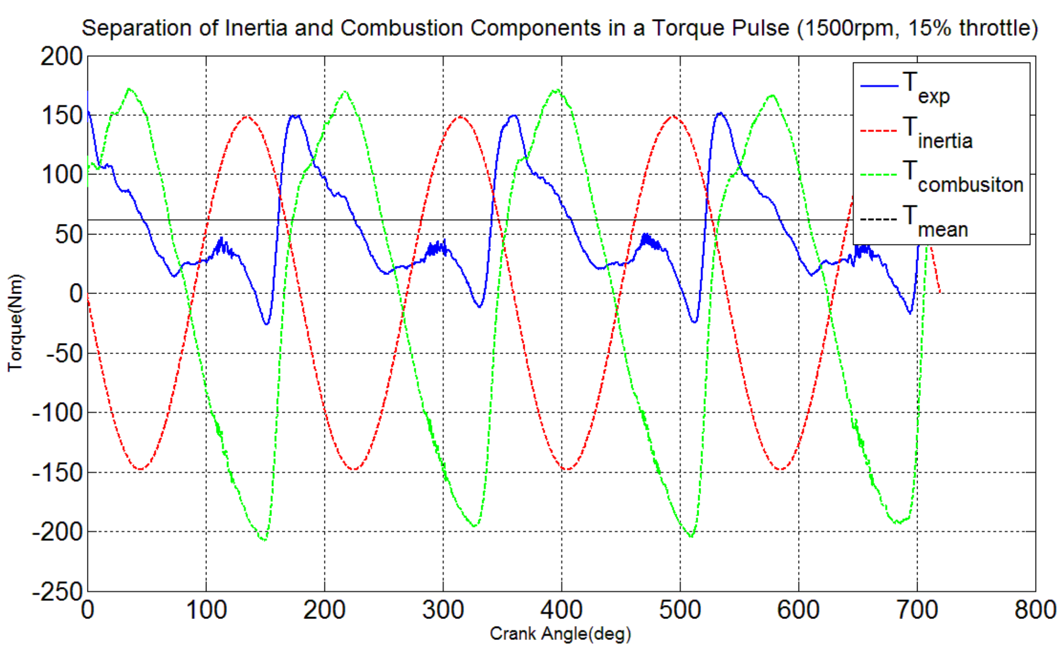

Figure 3: Engine torque composition (4-cylinder)

B. Inertia Compensation

Since the input motor has finite inertia, the effective inertia used in the engine model is calculated from acceleration considerations. In pure simulation, the load torque (engine torque available at the transmission input shaft) can be calculated as

$$T_l = T_e - C_e\omega_e - J_e \frac{d\omega_e}{dt} \tag{1}$$

where T_e is the total engine torque based on steady-state maps (see previous section), and ω_e is the engine speed. Since the input motor will emulate the engine, yet is physically a part of the system, the engine model's inertia is modified to have the same acceleration. Ignoring the torque pulse dynamics for this analysis, the load torque setpoint is then given by

$$T_l = T_e - C_e\omega_e + \left(J_{motor} - J_e\right)\frac{d\omega_e}{dt} \tag{2}$$

This inertia compensation plays a critical role in TiL system design and performance, and is used for input motor sizing based on worst-case considerations [4]. In most configurations, the input motor has a higher inertia than the engine, and hence, it becomes critical to select the appropriate torque generation capabilities and response time of the motor for accurate engine simulation.

C. Construction of Engine Torque Pulse

Since mean torque is commanded through the torque setpoints given by the model, the superimposing waveform of the engine torque pulse should have zero mean. Utilizing the inherent periodicity in the signal, the torque pulse is decomposed into inertia torque and combustion torque, as shown in Figure 3. The inertia torque pulse takes into account the inertia load for all the cylinders during operation. A model for this purpose can be derived in the crank angle domain by analyzing the kinematics of the piston-cylinder-crank mechanism of the engine. The inertia force due to reciprocating parts is a product of piston acceleration and the total reciprocating mass. The mathematical expression for piston displacement, velocity and acceleration can be derived from the geometry [4], and this waveform is entered directly in the drive in a crank angle-based map, and is modulated with the square of the engine speed through the models on Procyon. Since the engine speed does not vary significantly during the torque pulse, modulation through the optical fiber has almost no degrading effect on the simulation fidelity. The combustion pulse, defined as the torque pulse reflecting the combustion torque and all other effects in the system, other than the inertia effects (e.g. pumping effects), is constructed from experimental data. To construct this waveform, engine torque data is recorded using a high-speed, high-accuracy digital rotational torque sensor between the desired engine (4-cylinder inline) and its flywheel, and recording data at 100kHz for different speed throttle operations. The mean torque is first subtracted from this torque pulse, to get a zero-mean waveform. The calculated inertia torque pulse for all the cylinders is added together and then subtracted to get the combustion torque pulse, as shown in Figure 3. The resulting combustion torque pulse is also stored in a crank angle-based map on the input motor drive, and modulated with engine speed and throttle through Procyon and the fiber communication link during run time. The modulation factors are developed through experimentation at different speeds and throttles, and stored on the real-time system as part of the model. The model is validated by creating the waveform using this method and comparing it with experimental data on new operating points. The model is computationally efficient, and captures the overall torque pulse reasonably well. One such validation result is presented in Figure 4. The drive is also capable of storing multiple combustion maps, and could also be used for simulating engine misfires during ETP emulation.

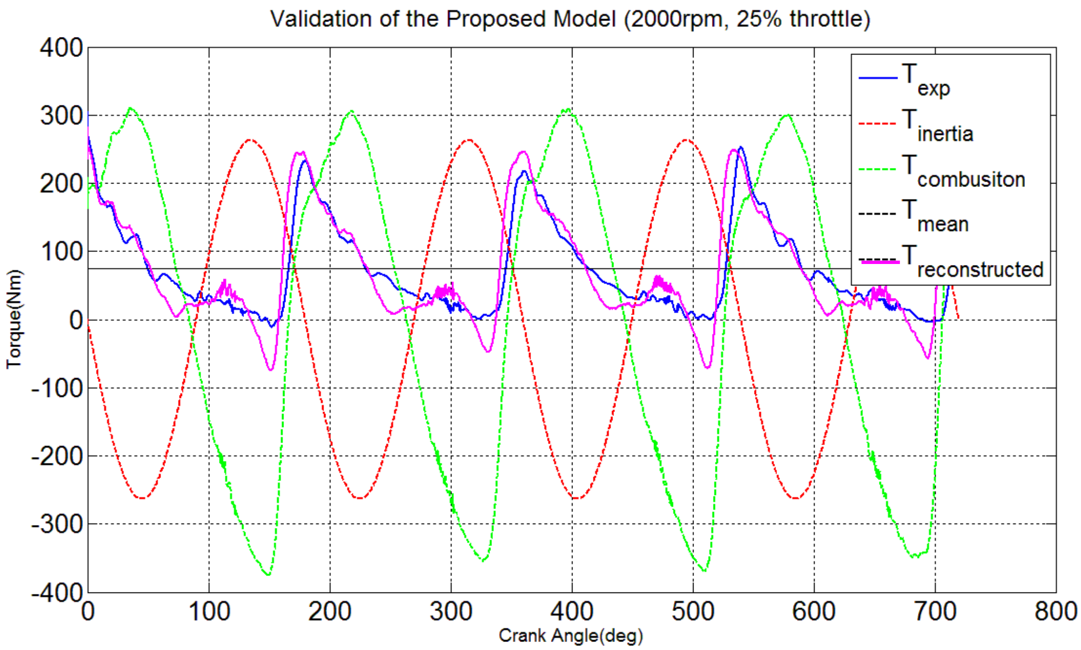

Figure 4: Engine torque composition—reconstruction

D. Engine Controller Emulation

The engine controller is implemented in the engine model to perform idle speed control, gear shifting, torque management, and CAN messaging to the transmission controller. The details of the CAN messaging are not provided in this paper, but are critical to an HiL system, in order for the TCU to operate properly. The driver model provides the accelerator pedal position to the engine model, which is converted into fuel rate for use as an input to the steady-state torque map. If the engine speed drops below the idle speed, then the idle speed controller, modeled as a closed-loop PI controller, overrides the driver pedal position and maintains the idle speed. The shift maps for the transmission are also contained in the engine controller, and gear shift events are accordingly communicated to the transmission controller. The shift commands can also come from the TCU via CAN. Figure 5 shows a schematic of engine speed and engine torque production during a gear shift event. In order to have a smooth shift in a real vehicle, the engine torque during the shift event is reduced by spark retardation which is controlled by the ECU (engine control unit). Since the engine simulator does not contain a combustion model or the actual ECU logic, the spark retardation command is converted into a torque cut command, and applied to the engine mean torque at the beginning of the event, as informed by the transmission controller. This reduces flare or tie-up in the transmission, and provides much smoother clutch-to-clutch shifts. The torque management logic is programmable to faithfully represent an OEM control strategy.

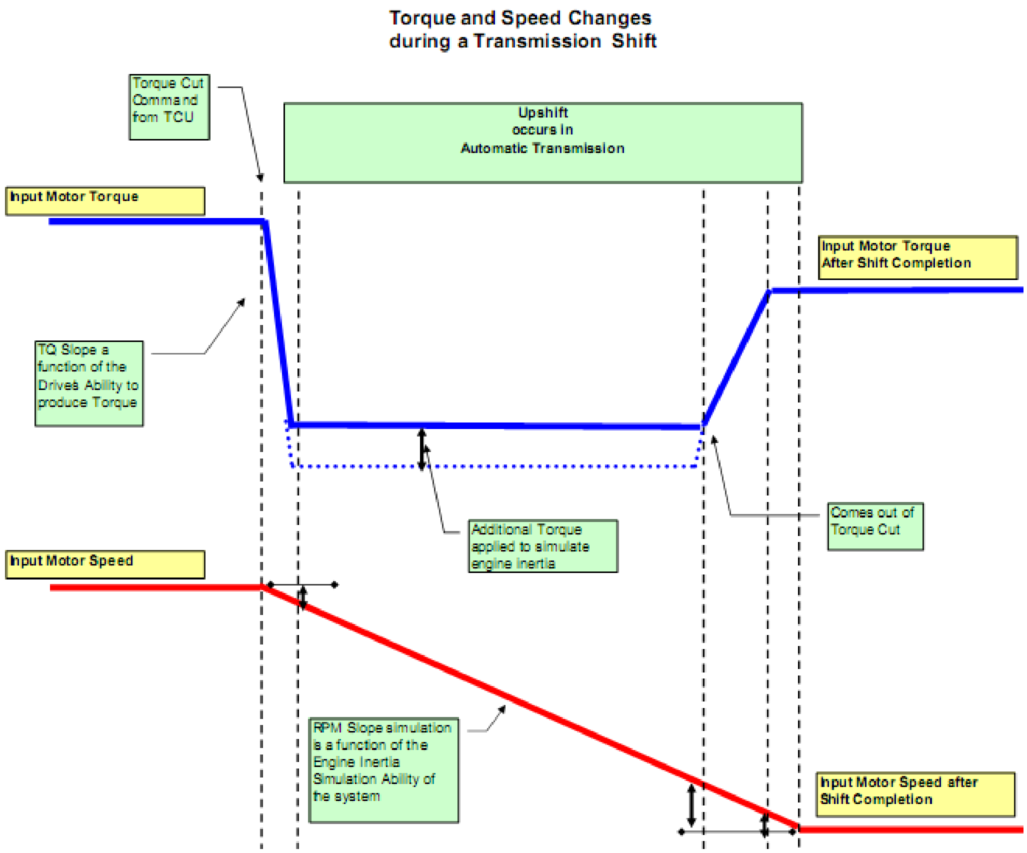

Figure 5: Engine torque management during a transmission shift event

VEHICLE MODEL AND WHEEL SLIP EMULATION

This section describes the vehicle model and wheel slip emulation for the TiL system. A detailed description of the model dynamics, including equations, can be found in [4]. The tire and vehicle model are implemented on the real-time platform and command speed setpoints to the transmission output shaft. Since the transmission assembly includes the differential and half-shafts, two separate wheel models are developed for the left and right front wheels. This also enables split-μ testing scenarios with the TiL system. All the models are developed such that transmissions can be tested in both forward and reverse modes.

Since torque is measured at the end of the half-shafts and the motors are subjected to speed control, inertia compensation on the output dynos is not required. The vehicle brake models are based on friction models from Margolis's fixed causality stick-slip model [5]. When the vehicle speed is high and the brakes are applied, the brakes act as non-linear resistors; at smaller velocities the brakes act as a spring and damper. Such a model has good numerical performance and eliminates the causality switch inherent in stick-slip friction. The tire friction model also needs to include road friction, but most hysteresis-based friction models found in the literature are not conducive to real-time simulation. Hence, a simpler tire friction model was created, where the tire is modeled as an elastic band in the longitudinal direction. The friction force of the tire, contributing to traction during acceleration and braking force during deceleration, is dependent upon the vehicle's longitudinal velocity and tire's rotational velocity, specifically a calculated dynamic difference between the two, represented in the slip coefficient. The traction/braking force is calculated assuming pure adhesion for low slips, while for higher slip values the friction force is produced by both adhesion and sliding (again, see [4] for details). Of note, the tire/road surface coefficient for friction itself increases with slip until it reaches some peak, and then decreases with slip/skid, almost linearly. Such a map can be developed based on the Stribeck friction curve of the tire [6].

The overall vehicle dynamics—mass, road slope, aerodynamic effects, etc.—are modeled using the traditional ABC-type equation:

$$m\frac{dv}{dt} = F - \frac{1}{2}\rho C_d A v^2 \operatorname{sgn}(v) - mg\sin(\theta) \tag{12}$$

where F is the traction and braking forces described earlier. In addition, a closed-loop driver model with preview functionality has been developed as part of the system model to enable speed tracking and provide throttle and brake commands to other portions of the model.

RESULTS AND SUMMARY

This paper presented an overview of the components of a new, high-fidelity TiL test system. Figures 6 and 7 present actual data and highlight both the engine torque pulse simulation and wheel slip (spin) simulation capabilities of the system. The extensive functionality of the TiL test system enables both supplementing and replacing in-vehicle transmission calibration and test activities in order to shorten transmission system development time and increase overall transmission product quality.

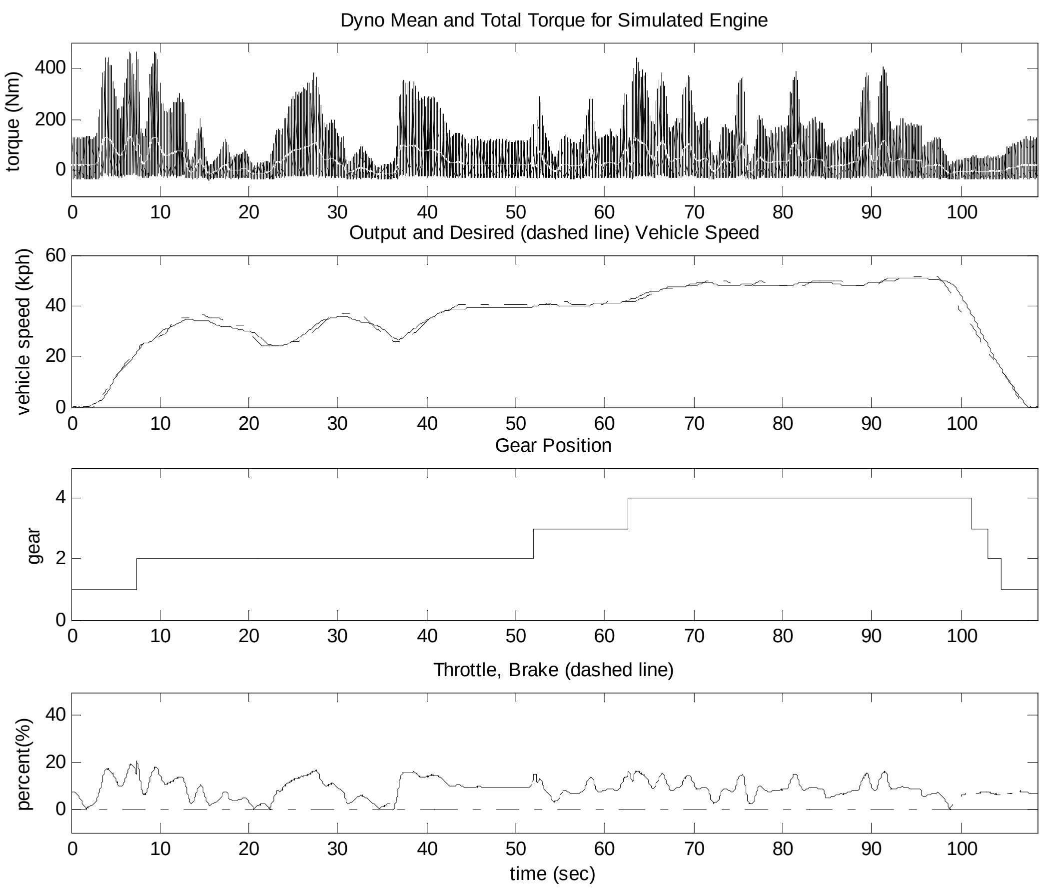

Figure 6: Test data for part of the FTP75 drive cycle highlighting engine torque pulse simulation

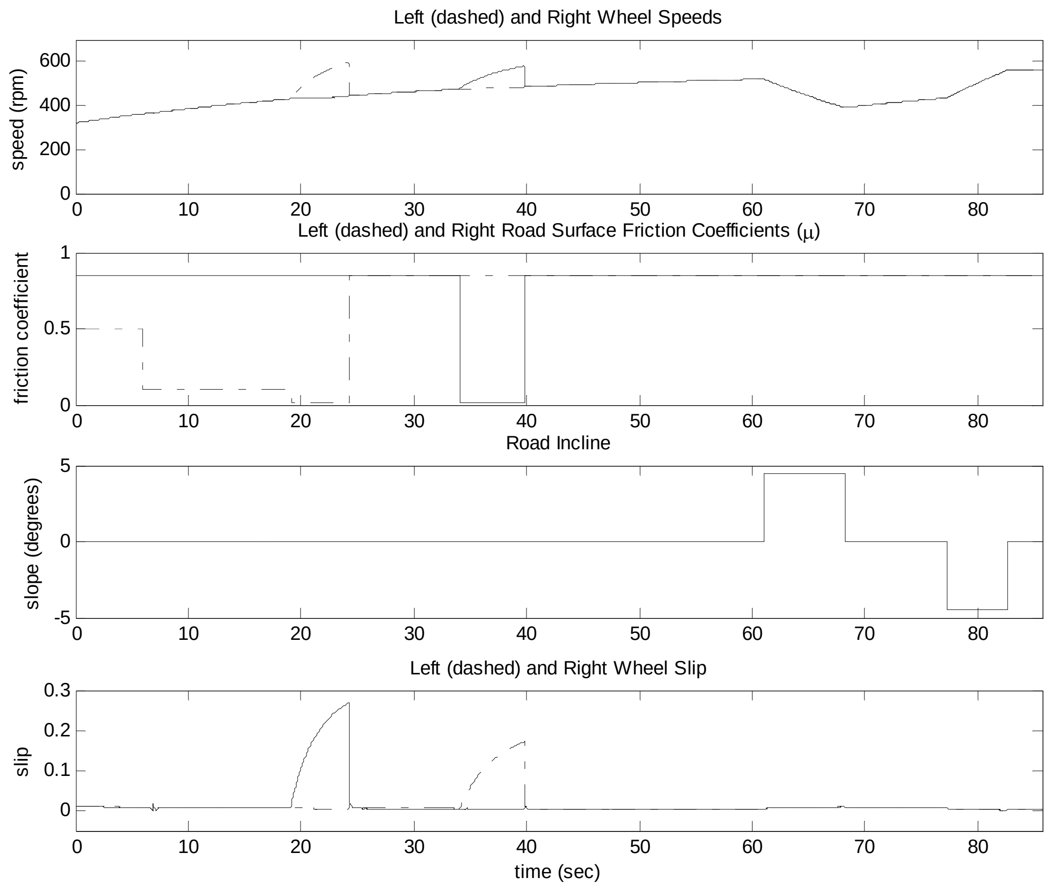

Figure 7: Test data for wheel slip (spin) simulation and road incline variation

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