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Validation and Use of SIMULINK Integrated, High Fidelity, Engine-In-Vehicle Simulation of the International Class VI Truck

Dennis Assanis[1], Zoran Filipi, Steve Gravante[2], Dan Grohnke[2], Xinqun Gui[2], Loucas Louca, Geoff Rideout, Jeffrey Stein, Yongsheng Wang

Automotive Research Center[3]

ABSTRACT

This work presents the development, validation and use of a SIMULINK integrated vehicle system simulation composed of engine, driveline and vehicle dynamics modules. The engine model links the appropriate number of single-cylinder modules, featuring thermodynamic models of the in-cylinder processes with transient capabilities to ensure high fidelity predictions. A detailed fuel injection control module is also included. The engine is coupled to the driveline, which consists of the torque converter, transmission, differential and prop shaft and drive shafts. An enhanced version of the point mass model is used to account for vehicle dynamics in the longitudinal and heave directions. A vehicle speed controller replaces the operator and allows the feed-forward simulation to follow a prescribed vehicle speed schedule. For the particular case reported here, the simulation is configured for the International 4700 series, Class VI, 4x2 delivery truck powered by a V8 turbocharged, intercooled diesel engine. The integrated vehicle simulation is validated against transient data measured on the proving ground. Comparisons of predicted and measured responses of engine and vehicle variables during vehicle acceleration from 0 to 60 mph and from 30 to 50 mph show very good agreement. The simulation is also used to study trade-offs involved in redesigning control strategies for improved performance of the vehicle system.

INTRODUCTION

Vehicle design is a costly process that is initiated with a comprehensive analysis of the vehicle system in order to determine the desired characteristics of sub-systems, followed by detailed design of sub-systems and components, and finally building and testing of prototypes. The latter include engine, driveline, and vehicle components and sub-systems, as well as the complete vehicle system. Development time and cost can be significantly reduced through the use of simulation-based design. A predictive vehicle system simulation can allow the creation and testing of the virtual vehicle for a variety of conditions, and guide component design prior to building prototypes. The availability of a comprehensive vehicle simulation with high-fidelity sub-system modules also enables concurrent design, where change in one system is reflected on the design process of related components and systems. Furthermore, design of powertrain and vehicle control strategies can be tested on the computer instead of the test cell or the proving ground. Since the latter would be necessary only for final verification, the cost of the product development process could be significantly reduced. In addition, a predictive simulation can assist in quantifying parameters associated with subjective driver feel and overall driveability that are difficult to measure and yet very important for customer acceptance.

Previous attempts to model entire vehicle systems have been characterized by a variety of approaches, differing in the fidelity of individual models as well as the methodologies used to integrate the various modules. Early attempts were based on collections of look-up tables for engine and driveline components, and simplified, point-mass vehicle dynamics models (Heavy-Duty Vehicle Simulation Program - HEVSIM [1], Vehicle Powertrain Simulation – VPS [2], NATO Reference Mobility Model - NRMM [3]). These simulations can be very valuable for quick assessments of vehicle mobility, even though they lack the ability to capture true vehicle transients. Their other limitation is the fact that, for every new component, a new look-up table needs to be generated through hardware testing, hence it is impossible to study non-existing designs.

Caterpillar was among the first to attempt to marry a thermodynamic diesel engine cycle simulation with DYNASTY [4] - a dynamic system simulation solving in time domain for vehicle position, velocity, acceleration and jerk; however, substantial challenges were reported in the integration of two independent codes. Significant improvements in the integration methodology have since

[1]Authors are listed in alphabetical order and, except if stated otherwise, are with the University of Michigan in Ann Arbor.
[2]International Transportation Corporation
[3]The ARC (http://arc.engin.umich.edu) is a U.S. Army Center of Excellence for Automotive Research at the University of Michigan, currently in partnership with the University of Alaska-Fairbanks, Clemson University, University of Iowa, Oakland University, University of Tennessee, Wayne State University, and University of Wisconsin-Madison.

emerged based on object-oriented graphical programming environments, such as EASY5 [5], MatrixX System Build and MATLAB/SIMULINK [6, 7, 8 and 9]. The utilization of this technology allows much more flexibility and is very suitable for control type of studies [5 and 9]. A fully flexible, comprehensive and predictive transient vehicle system simulation needs to take advantage of both improved models and advanced programming environments. The physically based modeling of components is the prerequisite for predictiveness, while the application of a simulation environment with block-diagram interfaces facilitates integration within a modular, dynamic structure.

Since 1994, the University of Michigan in partnership with the University of Iowa, Wayne State University, and the University of Wisconsin at Madison has established an Automotive Research Center (ARC) for the development and validation of advanced ground vehicle simulations. A hierarchy of models of varying resolution is being built into the flexible, dynamic simulation system that can be tailored for specific applications. The first generation of the high-fidelity ARC vehicle simulation was configured for the 6x6 heavy truck heavy truck [10]. The structure in MATLAB/SIMULINK, as well as the driveline model and the turbo-machinery modules, were based on the work of Munns [11]. The in-cylinder phenomenological module was based on the parent diesel engine system simulation of Assanis and Heywood [12], with dynamic extensions by Filipi and Assanis [13], and SIMULINK implementation by Zhang et. al. [14]. For the vehicle model, non-linear, detailed, 3-D multi-body kinematics and dynamics models had been developed based on the work of Sayers and Riley [16 and 17]. The application of the complete vehicle simulation demonstrated the ability to predict the effect of engine system design changes on vehicle performance, as well as the complex interaction between the powertrain and the vehicle during a hill climb on wet surface causing front wheel slippage.

The objectives of the present study are to further develop the integrated ARC vehicle simulation for enhanced fidelity and flexibility, to validate its ability to predict engine and vehicle behavior during transients, and to illustrate its use for assessing the impact of control strategy on performance. Fidelity enhancements focus on the incorporation of realistic engine fuel and vehicle speed controllers, as well as enhanced models of peripheral engine components, such as the manifolds and turbomachinery. Structural enhancements focus on clearly identifying the engine, the driveline and the vehicle dynamics modules at the top level of the SIMULINK simulation architecture.

Validation of the integrated vehicle simulation has been performed by comparing simulation predictions for an International 4x2 class VI truck against data measured on the proving ground by the manufacturer. The modules have been tailored to the needs of the particular study, i.e. simulation of on–road vehicle performance. The turbocharged engine module is configured for the V8 engine configuration with newly developed external component modules (manifolds, turbine, compressor, intercooler) based on [12]. A realistic fuel injection and timing controller is developed from data provided by International. The driveline module is configured with the 4-speed automatic transmission coupled to the engine via a torque converter. The shift logic, developed from manufacturer data, is based on vehicle speed and “throttle position”, with two sets of characteristics, one for upshifts and the other for downshifts. The vehicle dynamics module is reduced compared to [10], so as to allow more efficient calculations of vehicle acceleration performance on the straight and flat road where excitation does not generate significant pitch motion. Hence, the vehicle model is composed of two components that describe its dynamic behavior in longitudinal and heave directions. The main modules (cylinder, transmission, vehicle dynamics) are programmed in FORTRAN and C, and then converted to MEX file format in order to be able to create a SIMULINK module suitable for integration. The integration of all vehicle modules is performed simultaneously by the solver package built in SIMULINK. The integrated simulation environment is referred to as Vehicle Engine SIMulation – VESIM.

The paper is arranged as follows. The description of the physical vehicle system and the SIMULINK simulation structure are presented first. Next, details of the main modules, i.e., the engine, driveline and vehicle dynamics sub-systems are highlighted. This is followed by the description of the Intelligent Vehicle Speed (IVS) controller, developed to provide a driver demand signal for a given vehicle speed schedule. Then, the methodology for integrating the engine with the driveline and the vehicle is described. The newly configured simulation is validated against data measured on the proving ground. Experimental measurements and simulation predictions are compared during launch and acceleration from 0-60 mph and from 30 to 50 mph. Finally, the simulation is used to study the effect of varying certain control functions on the transient behavior of the vehicle system.

DESCRIPTION OF THE VEHICLE SYSTEM

The typical vehicle system is comprised of engine, drivetrain and vehicle dynamics modules. The vehicle in question is a diesel-powered 4x2 delivery truck with a 4-speed automatic transmission. The schematic of the vehicle system is given in Figure 1. The engine is connected to the torque converter (TC), whose output shaft is then coupled to the transmission (Trns), propeller shaft (PS), differential (D) and two driveshafts (DS), coupling the differential with the driven wheels. The complete vehicle system simulation is structured to directly resemble the layout of the physical system. Previous study of the integration methodology [10] indicated the key parameters that define the physical interaction between modules. For a vehicle system, these parameters include the “active” and “resistive” torques, as well as the angular speeds of key powertrain shafts. Hence, the system is configured in a way that

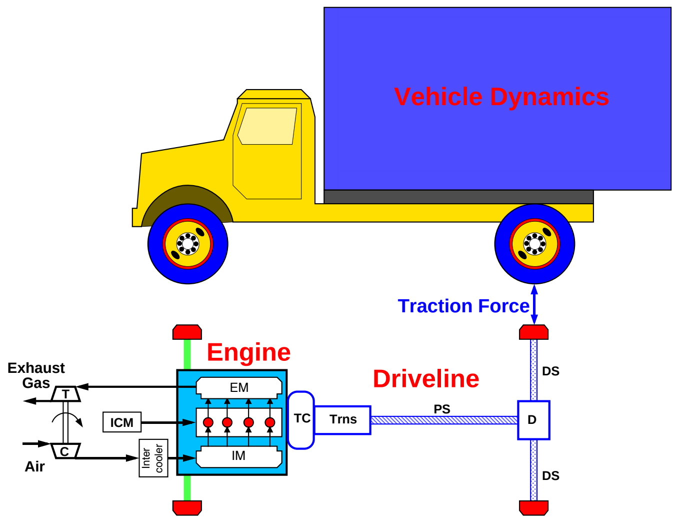


Figure 1: Schematic of the integrated vehicle system

allows a natural connection between the three main modules.

For a high degree of flexibility, the simulation structure is implemented in the MATLAB/SIMULINK graphical software environment. The top-level description of VESIM is shown in Figure 2. Besides the main modules and "buttons" used to manipulate input/output data, it shows three blocks used to control the transient simulation run: driver demand, brake and slope block. These modules allow specification of the driver action on the gas pedal and the brake pedal, as well as the road profile as a function of time. Alternatively, the IVS controller that allows the feed-forward simulation to follow a prescribed vehicle speed schedule can provide the driver demand signal. Details about the approach to system integration will be given later, in the section on integration methodology.

ENGINE MODULE

The engine module is comprised of multiple engine cylinder modules linked with external component modules, such as compressors and turbines, heat exchangers, air filters, and exhaust system elements. The engine cylinder model tracks the thermodynamic processes within the cylinder throughout a cycle as a function of crank angle. An engine dynamics module provides a link with the vehicle through the driveline. The specifications for the engine used in this work are given in Table 1 of the Appendix and correspond to the International 7.3 L V8 engine type T444E.

THERMODYNAMIC DIESEL ENGINE CYLINDER MODULE - The foundation of the diesel engine cylinder module used in this work is the physically based, thermodynamic, zero-dimensional model developed by Assanis and Heywood [12]. In the parent model, the cyclic processes in the cylinder are represented by a blend of more fundamental and phenomenological models of turbulence, combustion, and heat transfer. The parent simulation has been validated against test results from diesel engines of various sizes, ranging from highway truck engines [12 and 15] to large locomotive engines [18]. The cylinder control volume is open to the transfer of mass, enthalpy, and energy in the form of work and heat. The cylinder contents are represented as one continuous medium, uniform in pressure and temperature, characterized by an average equivalence ratio.

Quasi-steady, adiabatic, one-dimensional flow equations are used to predict mass flows past the intake and exhaust valves. The compression process is defined so as to include the ignition delay period, i.e., the time interval between the start of the injection process and the ignition point. A refined version of the Arrhenius expression relates the length of ignition delay to the gas temperature, pressure and overall fuel/air equivalence ratio in the cylinder after injection [19]. The inclusion of the equivalence ratio as the additional term has proven necessary in order to achieve the desired level of predictiveness during transients. Combustion is modeled as a uniformly distributed heat release process, using Watson's correlation [20]. The latter consists of the sum of two algebraic functions, one for premixed and one for diffusion burning and it includes ignition delay and overall fuel to air ratio terms. Hence, the correlation is able to account for the effect of engine load and speed on heat release.

Convective heat transfer in the combustion chamber is modeled using a Nusselt number correlation based on turbulent flow in pipes and the characteristic velocity concept [12] for evaluating the turbulent Reynolds number in the cylinder. The characteristic velocity and length scales required by these correlations are obtained from an "energy cascade" zero-dimensional turbulence model [12 and 21]. Radiative heat transfer is added during combustion [12]. The combustion chamber surface temperatures of the piston, cylinder head, and liner can be either specified or calculated from a specification of the wall structure.

A friction sub-model based on the Millington's and Hartles' correlation [22] is used to predict the engine friction losses and convert indicated to brake quantities. In this application the model uses the instantaneous engine speed supplied by the engine dynamics model, rather then the mean engine speed used in the traditional approach.

The diesel engine model [13] was originally coded in FORTRAN. It essentially contains the system of simultaneous, non-linear, ordinary differential equations for the cycle processes, along with a set of "utility" routines providing values for various terms in the state equations, e.g., thermodynamic and transport properties, flow rates through valves, etc. The prospect of using a single-cylinder engine code to develop a higher level multi-cylinder engine simulation necessitated modification of the FORTRAN source to make it fully compatible and "open" for communication links within SIMULINK. The procedure involves development of the

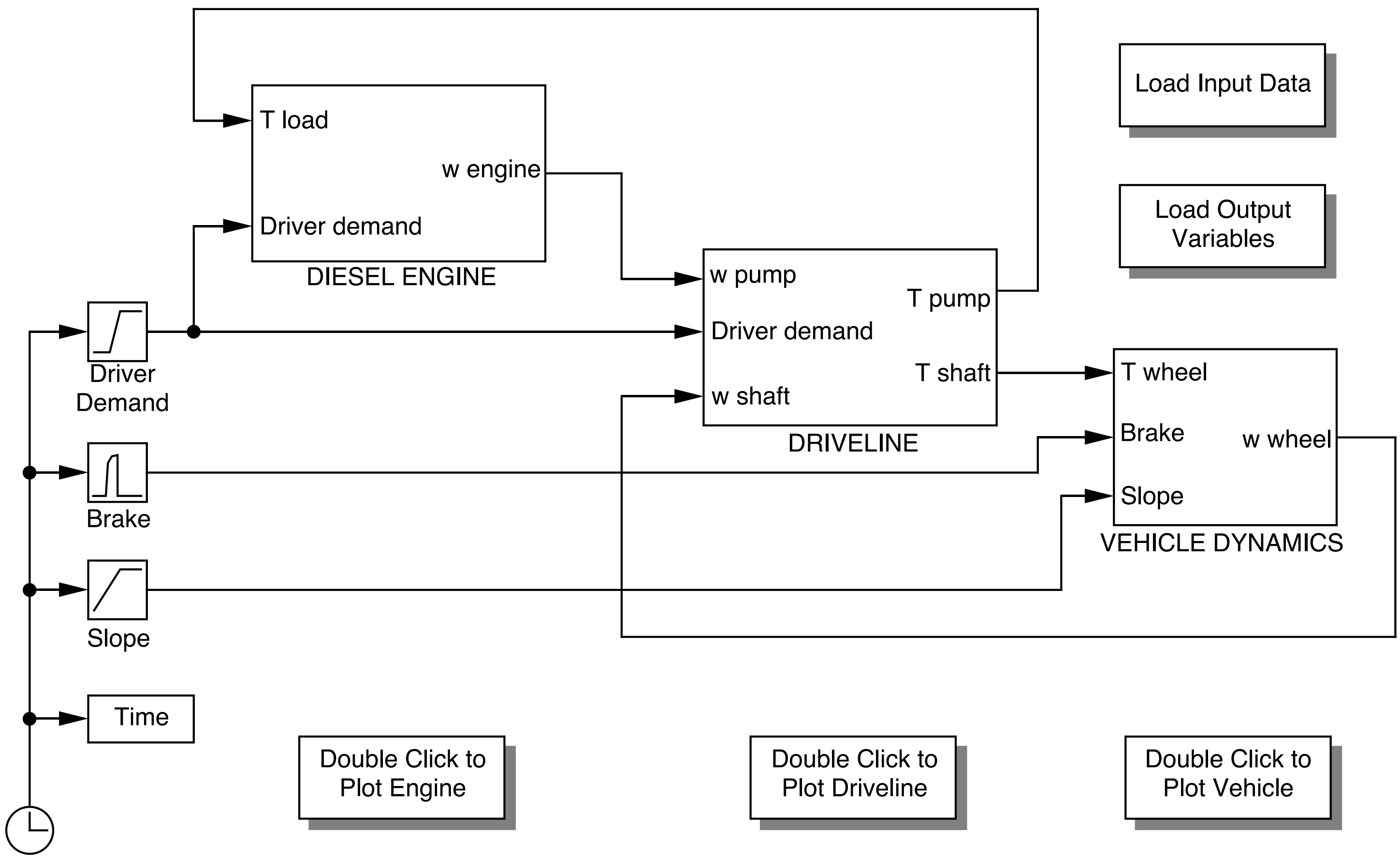


Figure 2: Vehicle system in SIMULINK – Vehicle Engine SIMulation.

FORTRAN-MEX file that contains all the necessary state derivatives and gateway routines for handling input and output vectors. More details on the conversion technique can be found in [14].

ENGINE DYNAMICS MODEL – An automotive engine application implies frequent, and often dramatic, variations of driver demand and external load. Hence, the thermodynamic engine simulation needs to be extended to include calculation of dynamics resulting from these varying operating conditions. The dynamic equation is derived from a two-disk system coupled with a rigid shaft. Therefore, the rate of change of crankshaft angular velocity is simply the ratio of the algebraic sum of all active and reactive torques divided by the crankshaft system inertia. The engine brake torque, calculated as the difference between the indicated and the friction torque represents the active torque. It depends on the set of instantaneous engine operating conditions, which include the action of the driver and the response of the controller. The indicated torque calculation accounts for the pressure force acting on piston as well as the inertial forces resulting from the piston and connecting rod movement [13]. The resistive torque can be either the torque converter pump torque, if the engine is coupled to a vehicle or the dynamometer torque if the engine is tested on a "virtual test stand". The engine dynamics equation is solved at each crank angle to return the new value of crankshaft speed for the next integration step.

MANIFOLDS - Intake and exhaust manifolds are treated as open thermodynamic systems. Equations for the conservation of mass, species, and energy are applied for analysis of manifold state, as well as for tracking enthalpy and fuel concentration fluxes into and out of cylinders [12]. Convective heat transfer from the working fluid to the manifold walls is included in the conservation equations. The intake manifold communicates with the intercooler, upstream, and with the cylinders, downstream. The exhaust manifold communicates with the cylinders, upstream, and with the turbine, downstream. The modular SIMULINK structure of the simulation greatly facilitates these connections for any arbitrary number of cylinders and manifolds.

TURBOCHARGER AND INTERCOOLER - The turbocharger model is based on a 2-D interpolation of digitized compressor and turbine performance maps. At any integration step, the mass flow rate and efficiency are determined from the rotor speed and the pressure ratio across the machine. Mass flow rate is an essential input into the manifold module, while the efficiency allows calculation of compressor and turbine power. These variables allow the turbocharger dynamics equation to be solved [12]. Hence, the new value of rotor speed is returned after every integration step.

The temperature drop in the intercooler is determined from the inlet air temperature, specified wall temperature and cooling efficiency [12]. The pressure drop in the intercooler is calculated using an orifice model. The orifice effective area has been calibrated against the experimental measurements obtained from International.

FUEL INJECTION CONTROL - The role of the diesel engine fuel injection control module is to provide the signal for the mass and timing of fuel injected per cycle based on driver demand, environmental conditions and current engine operating conditions. Special functions include corrections for insufficient boost pressure, cold start, high altitude, and the governing function. The correction of the amount of fuel based on the boost pressure or density in the intake manifold is especially important during full load acceleration, when turbo lag may cause the engine to operate with much lower boost pressures than normally experienced under corresponding steady-state conditions [23]. Fuel injection timing maps are designed to provide maximum fuel efficiency and satisfy the emissions constraint. The International T444E engine is equipped with the Hydraulically actuated Electronically controlled Unit Injector system (HEUI) allowing a high degree of flexibility and accuracy of control.

The selected simulation software, MATLAB/ SIMULINK, provides a very suitable environment for controller design and testing. In this work, we relied on information about the fuel control logic and data provided by the engine manufacturer to design a controller that would have all the necessary functions determining powertrain transient response. The controller is configured as a single module that accepts driver demand (or cruise control signal), engine speed and intake manifold pressure as inputs, and provides amount of fuel injected per cycle and injection timing as outputs. Hence, if there is a need to use and evaluate a proprietary controller, the user can simply cut the existing one and paste in a new module. Figure 3 illustrates the implemented control logic for the mass of fuel injected per cycle. Since the engine in question uses a drive-by-wire system, the step change in driver demand will be modulated by a tip-in function based on the desired "driver feel" upon vehicle launch. The driver demand signal varies between 0 and 1. Base calibration determines the "steady-state" amount of fuel for a given

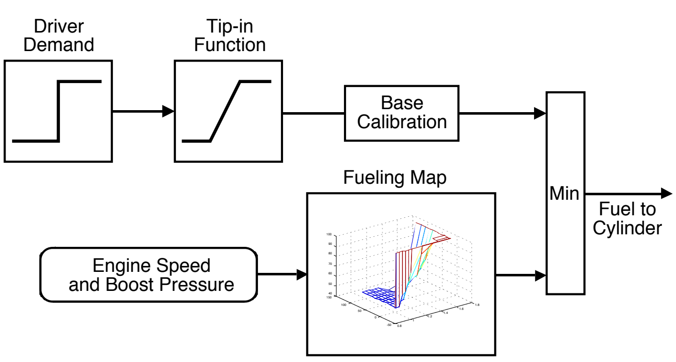


Figure 3: Control logic for the mass of fuel injected.

demand and engine speed. This value is checked against the limiting value determined from a look-up table, depending on current values of engine speed and boost. The smaller of the two values will be passed on to the actuator and determine the actual mass of fuel injected. This value can be further reduced by the governor function if engine speed exceeds the rated speed.

The algorithm for injection timing starts with the information about the engine speed and the mass of fuel per cycle as the measure of actual engine load. The 2-D look-up table determines the desired value in degrees crank angle. This is then corrected for the hydraulic delay known to exist in the International HEUI injection system before the actual value is passed on as the output.

The controller has been tested at various steady-state points before it is used for in-vehicle studies. Comparison against experimentally obtained engine test data confirmed its ability to respond correctly to any change in system operating conditions. All the subsequent in-vehicle transient simulation runs are performed with the factory calibration except the one where an alternative, more aggressive fueling calibration is contrasted to the original one.

DRIVELINE

The driveline module consists of the torque converter, transmission, propshafts, differential, and drive shafts. It provides the connection between the engine and the vehicle dynamics module (see Figure 1). The torque converter input shaft, on one end, and the wheel, on the other end, are the connecting points for the engine and the vehicle dynamics models, respectively. The state equations are generated and compiled in a SIMULINK C-MEX function. The shift logic is also coded in another C-MEX file. The two C-MEX functions are combined to form the VESIM driveline module. The module inputs are the engine speed and wheel rotational speed, and the outputs the torque to the wheels and load torque on the engine. The driveline model, excluding the shift logic, is constructed using the bond graph modeling language [24 and 25] and implemented in the 20SIM system-modeling environment [26]. The 20SIM environment is selected due to its capability to combine block diagram elements and bond graph elements. The specific elements used in the components are described below and their parameter values are given in Table 2 of the Appendix.

TORQUE CONVERTER - The torque converter is the fluid clutch by which the engine is coupled to the transmission. The typical three-element converter consists of an impeller, stator (reactor), and turbine (runner), as shown in Figure 4. The impeller is connected rigidly to the engine output shaft, and the turbine to the transmission input shaft. The stator is connected to the torque converter housing via a one-way clutch. The presence and arrangement of the stator

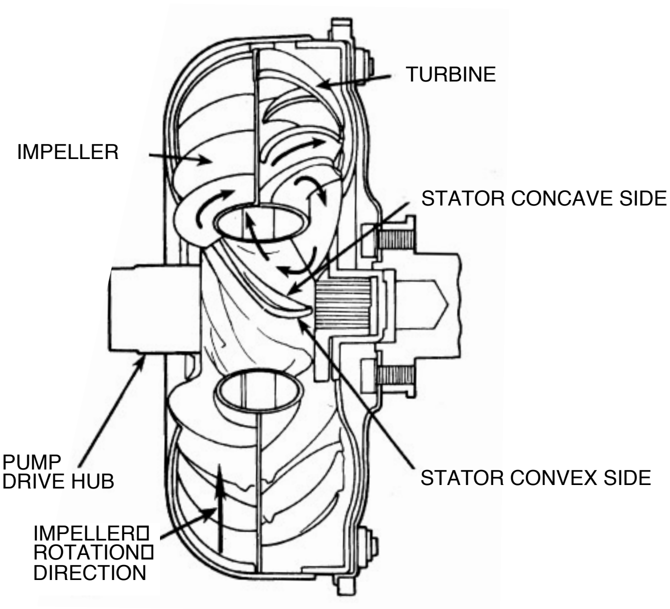


Figure 4: Torque converter schematic.

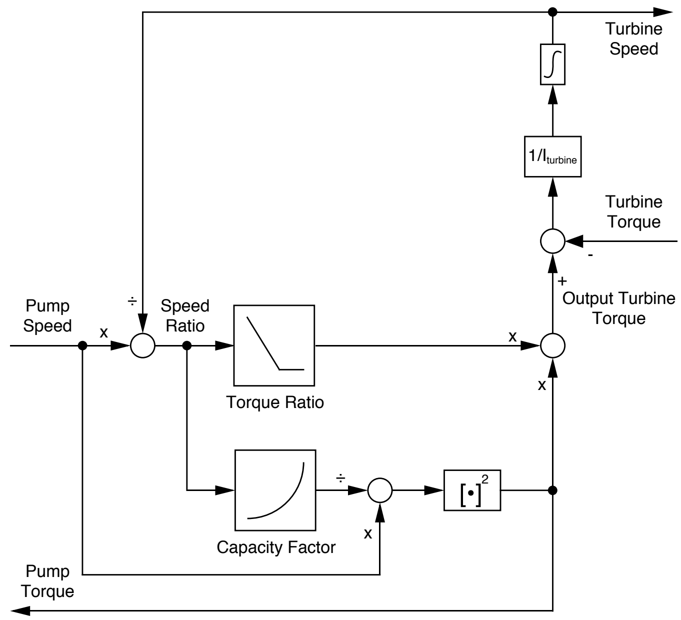


Figure 5: Block diagram of the torque converter model.

cause the torque converter to act as a torque multiplication device when operating at low speed ratios, and as an approximately direct-drive fluid coupling at higher speed ratios. The speed ratio is defined as the ratio of turbine speed to impeller speed.

The torque converter model is a quasi-steady model based on experimental data available from transmission supplier. Such a model [27, 28, 29 and 30] assumes that the torque converter is not subject to engine speed inputs with high frequency content, as would occur during the fast transients that accompany throttle steps, or fast changes in speed ratio. The torque converter model is thus a limiting factor at present, in the study of engine/transmission speeds during the short period immediately after such transients are initiated.

The TC model is shown schematically in Figure 5 as a block diagram. The inputs to the model are the pump (engine) speed and turbine (transmission) torque. Look-up tables convert the speed ratio into a torque multiplication ratio and a capacity factor, which is used in conjunction with engine speed to calculate load torque on the engine. Multiplication of the engine load torque with the torque multiplication ratio gives the output turbine torque that is applied to accelerate the turbine.

The torque converter model will also work when the transmission is driving the engine, i.e., when the speed ratio exceeds unity. Data for load torque on the engine as a function of engine and transmission speed have been converted to capacity factors for speed ratios greater than one. Piecewise curve fits for capacity factor and torque ratio versus speed ratio have been developed to match the experimental data provided by torque converter the manufacturer.

TRANSMISSION, PROPSHAFTS, DIFFERENTIAL AND DRIVE AXLE - The two primary modeling concerns for the remaining driveline components are the speed reduction / torque multiplication ratios; and the inertial, spring and damping properties. Input and output inertias, stiffnesses and damping rates for each of the transmission, propshafts, and differential/axle are determined experimentally and referenced to the input speeds thereof. While the physical properties of each individual gear and shaft are not considered explicitly, these properties are accounted for in the experimental “effective” parameters.

The central element of the transmission is a non-power-conserving transformer that models gear inefficiency and allows it to vary among different gears. The speed reduction in each gear is assumed ideal, while the torque multiplication is reduced by the appropriate inefficiency factor. The transmission torque losses due to fluid churning are modeled as a variable resistance. As per manufacturer’s recommendations, the charging pump is modeled as a constant torque loss source, regardless of gear number or transmission speed.

The two propshafts are modeled as three effective inertias and two springs in series, again with a small amount of viscous damping in parallel with each spring. The differential/axle model consists of an inertia, input stiffness, and axle cooler churning loss resistor. In a manner similar to the transmission, the differential gearing inefficiency is modeled using a non-power-conserving transformer with ideal speed reduction, but non-ideal torque multiplication.

SHIFT LOGIC - The inputs to the shift logic module are the transmission output shaft speed and the driver-demanded throttle position. A chart such as the one shown in Figure 6, based on the transmission manufacturer shift logic, determines the current gear number, and whether or not an upshift or downshift is to be initiated. The solid and dashed lines indicate upshift and downshift thresholds, respectively.

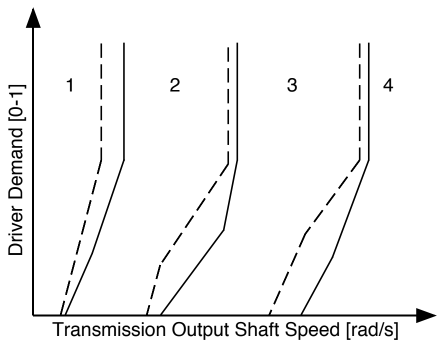


Figure 6: Gearshift logic

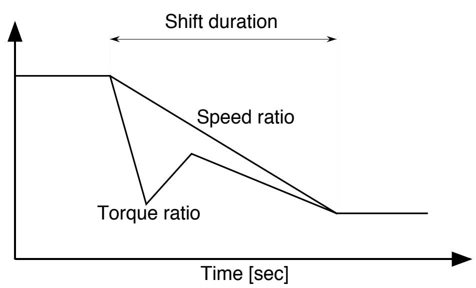


Figure 7: Blending functions for the shift event

During a gearshift, the speed reduction and torque multiplication ratios vary from the initial to the final value according to blending functions. The blending functions model the torque and speed ratio variations that occur during the shift event, as clutches and bands engage and disengage. Typical shapes of blending functions are shown in Figure 7.

VEHICLE DYNAMICS

The vehicle subsystem includes the wheels/tires, axles, suspensions and body of the vehicle, as shown in Figure 8. Vehicle dynamics describe the motion of the selected rigid bodies (wheels, axles and body), that are allowed to move in space subject to forces/moments and rigid constraints. The forces/moments are physical elements, which act at a specific point of two bodies, e.g., suspension. In addition, two bodies can be restrained to move only in specific directions by a rigid constraint, e.g., a wheel is allowed only to rotate around the axis of the axle. The connection of the vehicle with the driveline is at the driven wheels where the drive-axle is connected to the rim of the wheels.

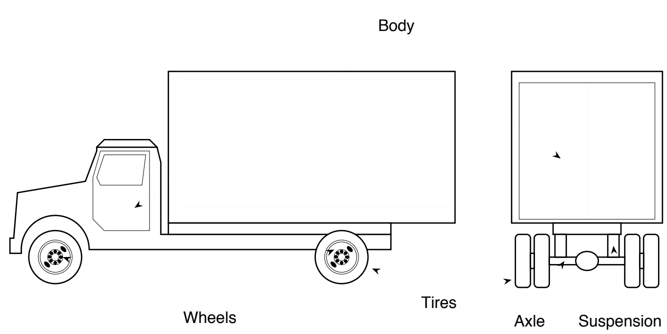


Figure 8: Vehicle and its main sub-systems.

A number of approaches can be used to model vehicle dynamics depending on the overall simulation objectives. A single Degree of Freedom (DOF), point mass model, can be selected for an initial estimate of vehicle performance as different powertrain options are explored. The model assumes that the vehicle mass is lumped at the center of gravity. Such approach can give sufficiently accurate predictions of vehicle acceleration and speed on "smooth" or flat roads. The model complexity can be enhanced with more DOFs as more severe excitations (road roughness, steering, braking, etc.) are introduced into the model. This is necessary for the investigation of vehicle-powertrain interactions during such extreme transients that induce significant pitch motion. The complexity of the model can be systematically adjusted, as proposed by Louca et. al. [31], to accommodate the needs of a specific scenario.

The studies in this work consider only the acceleration of the vehicle on a flat and straight road, where the excitation does not generate significant pitch motion. Therefore, for this "mild" scenario, the point mass model adequately predicts the interactions between the powertrain and vehicle dynamics. The enhanced point mass model is shown in Figure 9. The model is composed of two components that describe the dynamic behavior of the vehicle in the longitudinal and heave directions. The two components are coupled through the road/tire interaction.

The heave direction component is a standard quarter car model, as shown in Figure 9. The model consists of the sprung mass, namely the major mass supported by the suspension, and the unsprung mass, which includes the wheel and axle masses supported by the tire. This model represents the driven (rear) axle of

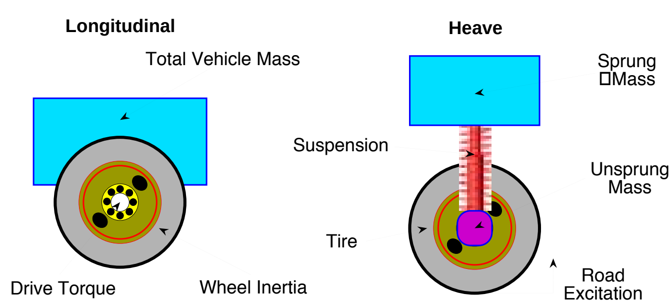


Figure 9: Schematic of vehicle dynamics.

the vehicle and the sprung mass is adjusted to account for the location of the center of gravity of the vehicle body. The suspension is modeled as a spring and a damper in parallel, and acts as a constraint force between the sprung and unsprung masses. The tire is also modeled as a spring and a damper in parallel, which transfer the road force to the unsprung mass (wheel hub). The input to the system is the road profile that prescribes a vertical velocity as a function of the longitudinal position of the vehicle.

The model of the longitudinal dynamics includes the total mass of the vehicle, which is the sum of the sprung and unsprung masses, and the inertia of all four wheels of the rear driven axle. In addition, energy dissipation elements are included that account for energy losses due to bearing friction, tire rolling resistance and aerodynamic drag. The torque from the driveline is applied to the wheel hub and the available traction force to accelerate the vehicle is calculated based on the wheel slip. The traction force increases linearly with the wheel slip and it saturates when it reaches a value that is equal to the tire normal force multiplied by the road/tire friction coefficient (μ). The two components (heave and longitudinal dynamics) are coupled since information from the heave dynamics is needed to calculate the longitudinal dynamics, and vice versa. The heave dynamics are coupled to the longitudinal dynamics since the tire normal force is needed to calculate the traction force. The vehicle dynamics are modeled using the bond graph language [24 and 25] and implemented in the 20SIM system-modeling environment.

The dynamic model is represented by ordinary differential equations that describe the kinematic and dynamic behavior of the real system. These equations are generated by 20SIM [26]. They are then coded in the C programming language and converted into C-MEX file. Hence, the final product is an S-function suitable for direct integration with the SIMULINK model. All the vehicle dynamics parameters correspond to the International 4700 series delivery truck and are given in Table 3 of the Appendix.

INTELLIGENT VEHICLE SPEED CONTROLLER

A delivery truck, like the International 4700 series, operates under a wide range of conditions during normal use. Traffic conditions, numerous stops, and vehicle launches initiate frequent and often rapid transients in the city. The effects of this driving cycle are compounded by the varying payload mass that affects the vehicle dynamics, and hence the resistance load felt by the engine. An additional element of driver intelligence is needed in order to achieve certain vehicle speed schedule. The driving schedule can range from maintaining constant vehicle speed on varying terrain to complex speed/load cycles, such as the ones used for emission testing. As mentioned earlier, the simulation is configured as a feed-forward system, where everything starts with the driver action and the system responds based on its abilities and external conditions. Hence,

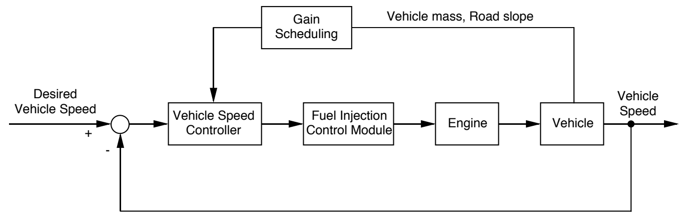


Figure 10: Intelligent Vehicle Speed controller.

the Intelligent Vehicle Speed (IVS) controller should have the ability to provide the signal representing a driver demand based on the desired driving schedule and vehicle speed feedback loop. In principle, the structure of the IVS controller includes the vehicle speed controller itself and the fuel Injection Control Module (ICM), as illustrated in Figure 10. The engine is equipped with the electronically- controlled fuel injection system (HEUI), hence ICM determines the actual amount of fuel injected and injection timing based on the IVS signal and on engine operating conditions, e.g., boost pressure, crankshaft speed and hydraulic delay in the system. Therefore, the output of the IVS controller is a signal that varies from 0 to 1 and is provided to the ICM for further processing.

The speed control algorithms used in automotive cruise controllers are often of the proportional-integral (PI) type [32 and 33]. If a closed form, mathematical model of the controlled system (vehicle system) can be formulated in the form of a transfer function or a state-space equation, the controller gain (proportional) and time constant (integral) can be readily determined from basic control theory. However, in our case, the models of the diesel engine, driveline, and vehicle are in the form of complex differential and algebraic systems of equations. This makes it infeasible to derive a simple transfer function. The Ziegler-Nichols tuning method [32] based on an open loop is selected as an alternative approach to set the controller's gain and time constant. The method relies on measurements of the dynamic response of the system when a unitary input is imposed on the system.

In addition, the vehicle system is time varying, because the characteristics of the vehicle change with the operating conditions (e.g., vehicle mass, road slope, etc.). Therefore, if an IVS controller is well tuned under one set of conditions it may exhibit sluggish behavior when the vehicle mass or road conditions change. Providing a schedule of controller gain and time constant is a practical approach to dealing with this type of non-linearity of a fully transient simulation. The controller gain and time constant are calculated using the Ziegler-Nichols method for a range of operating conditions and tabulated as a discrete set of values that depend on the vehicle mass and road slope. Consequently, when operating conditions change the IVS controller gain and time constant are modified accordingly to much the operation conditions (see Figure 10).

INTEGRATION METHODOLOGY

As illustrated in Figure 2, the vehicle system simulation (VESIM) consists of three main modules: engine, driveline and the vehicle structure. Interaction between the main modules is in the form of "active" and "resistive" torques, as well as shaft angular speeds. The engine simulation provides as output the instantaneous value of engine torque and rotational speed. The engine speed is passed on to the torque converter input shaft, i.e., torque converter pump. The torque undergoes multiple transformations as it is transmitted through the driveline. The final value at the wheel depends on the torque converter operating conditions, gear ratios in the transmission and the differential, and the flexing of the propeller and drive shafts. The torque on the wheels is converted into the tractive force, which in conjunction with other information about the vehicle and the terrain determines vehicle dynamic behavior. Hence, the vehicle dynamics module returns the instantaneous vehicle speed and the wheel angular velocity. This information is propagated back through the system, all the way to the TC output shaft, thus determining the torque converter turbine speed and the speed ratio of the TC. The latter determines the TC pump torque, which is in turn supplied to the engine module as resistive torque. The solution of the engine dynamics equations determines the engine speed value for the next integration step.

The integration is performed in the SIMULINK environment. Its graphical programming capabilities allow easy coupling of the modules, as long as each one of them has a desired set of input/output links. However it was shown [35] that the flexibility of SIMULINK comes with a certain overhead in terms of computational efficiency, the actual magnitude being strongly dependent on the level of system decomposition and the number of component modules and links. Hence, in order to enhance computational efficiency, some of the more complex modules are programmed in FORTRAN (engine cylinder) or C (driveline, vehicle dynamics), and configured as self-contained SIMULINK blocks through the use of MEX function standard. The SIMULINK solver is used to integrate the complete vehicle system; thus the classic problem in software numerical integration of "who is in charge" is avoided. More details about the integration procedure and modularization of the engine cycle simulation can be found in [10 and 14].

SIMULATION VALIDATION

Simulation validation is performed through comparisons of VESIM predictions with measurements obtained on the real vehicle on International proving grounds. The experimental data was limited to engine and vehicle speed histories. However, since these variables represent the end result of complex engine and vehicle interactions, it is felt that the agreement between these two quantities would effectively validate the behavior of the entire system.

VEHICLE LAUNCH AND ACCELERATION FROM 0 TO 60 MPH – VESIM behavior is first tested under one of the most dramatic transients – vehicle launch at full load. The vehicle is initially at stand still, with brakes applied and the engine idling at 700 rpm. The amount of fuel injected is automatically determined based on the desired idle speed and internal losses in the torque converter. At t = 1 second, the brakes are released and driver demand is linearly ramped up to the maximum value in one second. Vehicle and engine parameters are tracked until the vehicle reaches 60 mph. The comparison of predicted and measured engine and vehicle speed results is shown in Figure 11 and Figure 12, respectively. The vehicle speed results show excellent agreement. The engine speed response indicates reasonable agreement across the overall profile, with the first two gear shifts actually occurring at the same time as predicted, and only the last one occurring slightly earlier on the simulated profile. Further, the predicted drop in engine speed during gearshifts closely matches the experimental results.

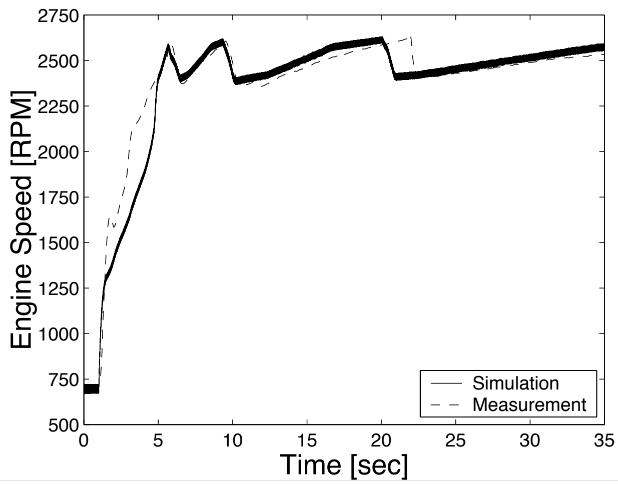


Figure 11: Comparison of predicted and measured engine speeds during the 0-60 mph acceleration test.

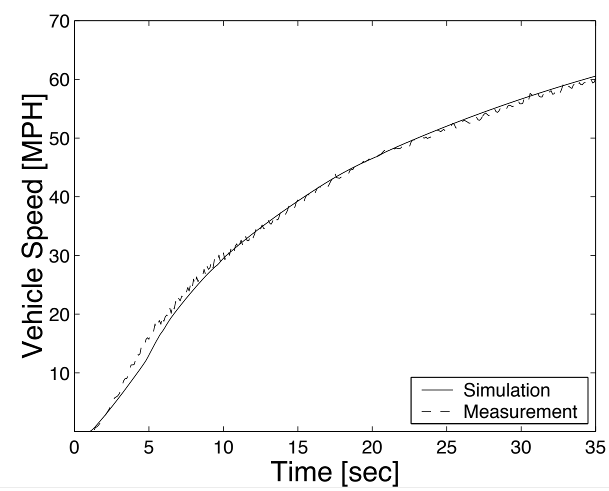


Figure 12: Comparison of predicted and measured vehicle velocities during the 0-60 mph acceleration test.

The first five seconds should be examined more closely, since that interval is influenced by turbocharger lag and system dynamics significantly more than the rest of the test. The sharp increase and the first spike on the measured engine speed profile is associated with the extreme dynamics in the torque converter and its apparent inability to provide the instantaneous response to the sudden change of engine speed. Hence, the engine overspeeds somewhat before the torque converter is able to provide feedback in the form of the increased pump torque, felt by the engine as the reaction torque. This is hypothesized based on the fact that the whole event associated with that initial spike takes place during one crankshaft revolution, and it would take some finite time to accelerate the fluid inside the torque converter. The quasi-steady TC model is not capable of capturing such a dynamic process and hence responds instantaneously, keeping the engine speed down. The comparison of the rest of the initial five second part of the profile illustrates roughly the same slope of the lines associated with the turbo lag, i.e., the gradual build up of pressure in the intake manifold and corresponding increase of engine torque are in very good agreement. A more dynamic torque converter model, which includes the fluid inertial effects, is expected to improve the simulation results.

VEHICLE ACCELERATION FROM 30 TO 50 MPH – In this test the driver maintains the 30 mph speed for few seconds, then depresses the pedal all the way and accelerates to a speed above 50 mph. This allows accurate measurement of the 30–50 mph passing time. To simulate the same transient test required the use of the Intelligent Vehicle Speed controller. The transient schedule is designed as follows: at vehicle launch, the desired speed is set at 30 mph and the system is allowed to reach steady state conditions. At a given point in time, the demanded speed is instantly changed to 60 mph, and the IVS controller adjusts the fuel delivery appropriately. The intention is to still have full fuel delivery at 50 mph and record the passing time. After a speed of 60 mph is achieved, the controller decreases fueling again, and attempts to keep the vehicle at this new quasi-steady state. The comparison between predicted and measured vehicle speed and engine speed responses is shown in Figure 13a and Figure 13b, respectively. The agreement is very good, thus providing additional evidence of VESIM's predictive capabilities. It appears that the IVS provides excellent responsiveness and stability, since the overshoot of vehicle speed above 60 mph is minimal.

Variations of the normalized mass of fuel injected during the complete schedule of events are shown in Figure 13c. Dramatic changes are evident at the initiation of the two hard acceleration sequences, as well as more subtle changes resulting from the fuel controller response to variations of boot pressure and engine speed. Predicted intake manifold pressures are also given in Figure 13c. The effect of turbo lag is especially evident at the vehicle launch (t = 5 sec). It is also interesting to note the decrease of boost pressure just before the gearshift during the 30-60 mph acceleration. This is the result of the governor function in the ICM becoming active and reducing fueling at high engine speed, despite the "full demand" by IVS. Reduced fueling leads to a decrease of exhaust enthalpy and deceleration of the turbocharger, thus the "dip" in the manifold pressure line. The instantaneous fuel/air equivalence ratio, shown in Figure 13d, is the result of dynamic interactions between the fuel and intake air systems. Two peaks of equivalence values above 0.6 predicted at the beginning of full power accelerations illustrate the ability of the simulation to identify significant departures from normal, steady state conditions, thus assisting in solving transient emissions problems.

SYSTEM RESPONSE STUDIES

The potential usefulness of VESIM as a design tool is explored through two parametric studies. In both cases the intention is to assess the possible side-effects effect associated with modifications of control strategies with the aim to improve overall vehicle performance. The first study focuses on the effect of varying the duration of the gearshift event. The second study investigates the effect of implementing a less restrictive low-boost correction in the engine fuel injection controller.

SHIFT DURATION STUDY - The duration of the shift event is varied in order to study possible effects on system response and driver comfort. As shown in the engine speed time series in Figure 11 and Figure 12, the nominal shift duration of 0.8 seconds gave good agreement with experimental data. This duration is varied by +/- 0.4 seconds and the "full-throttle" acceleration run is repeated. Changing the shift duration essentially scales the blending functions in Figure 7 along the horizontal axis. A shorter shift duration represents quicker engagement and disengagement of transmission clutches and bands, and higher slopes of the torque ratio during the speed and torque phases of the shift.

The effect on ride quality, as evidenced by maximum forward vehicle jerk (second derivative of vehicle speed) during the shift from the first gear to second, is shown in Figure 14. The latter shows that the maximum value of forward jerk increases as the shift duration decreases. Ride quality in this simple example may be considered slightly improved for longer shift duration, however, such a modification brings up questions about the clutch wear and long-term durability, as the clutch slip occurs over a longer period of time. A shorter shift duration of 0.4 seconds shows that the side effect of a crispier acceleration would be a drastically increased peak value of jerk. For a typical truck application, this would probably be unacceptable from the point of view of driver comfort. However, the ability to evaluate such a trade-off between acceleration performance and driver comfort and feel can be a valuable aid in designing a new, specialized application, e.g., commuter bus, off road vehicle, etc.

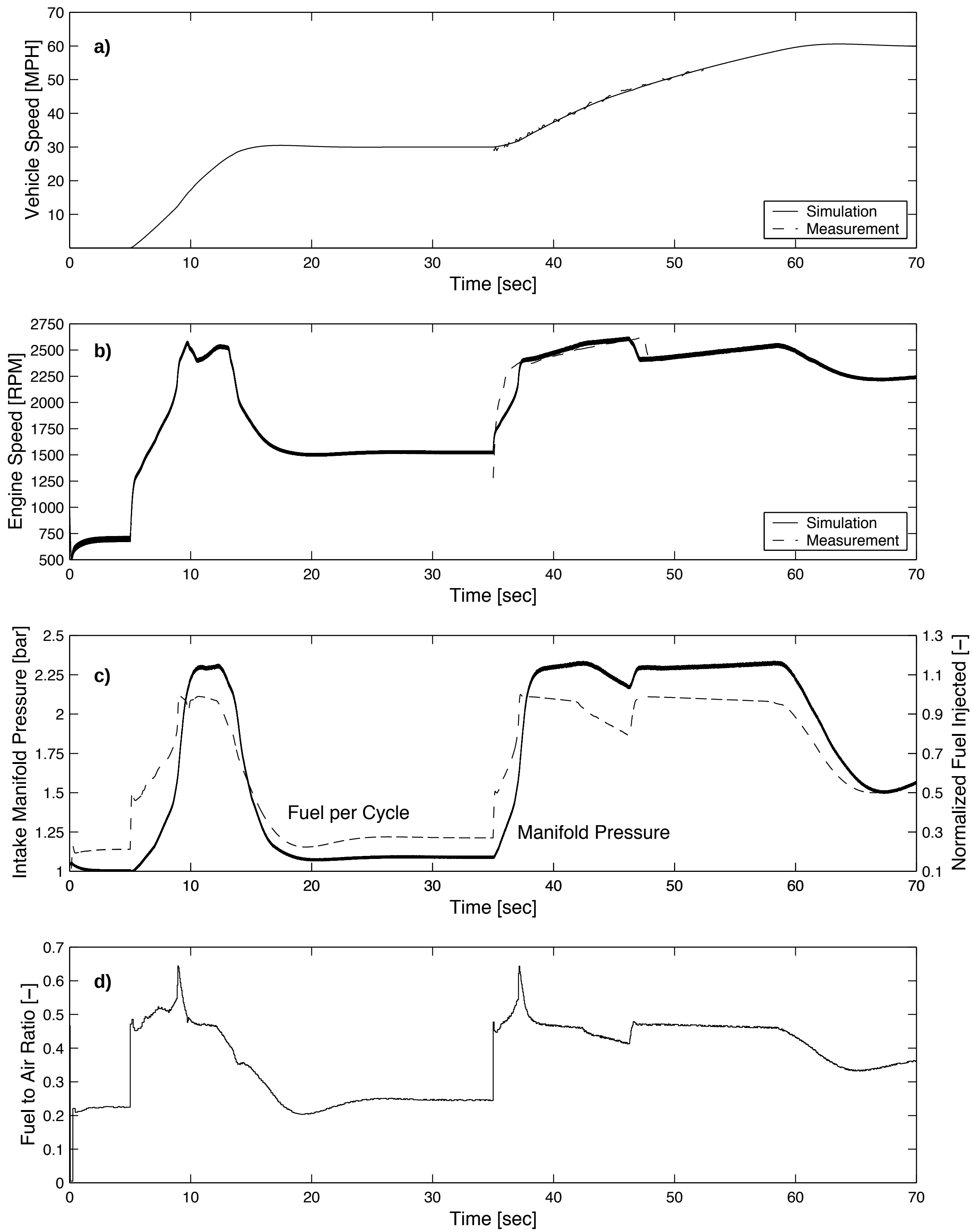


Figure 13: Sequence of engine and vehicle transients during the 30-50 mph passing time test: comparison of predicted and measured a) vehicle speeds, and b) engine speeds; predictions of c) Intake manifold pressure and Mass of Fuel Injected per cycle normalized by the maximum value, d) F/A equivalence ratio.

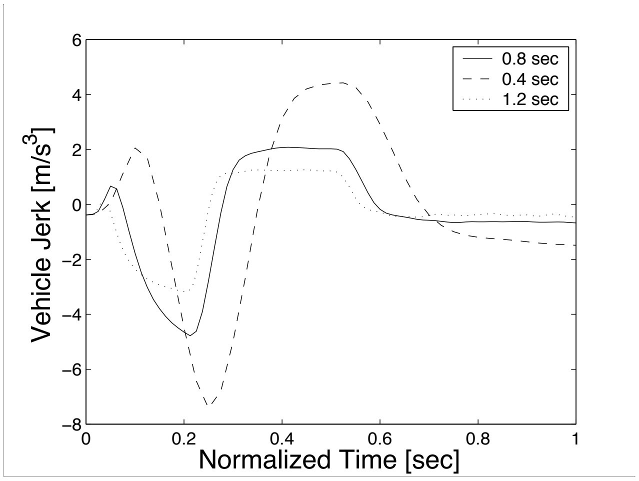
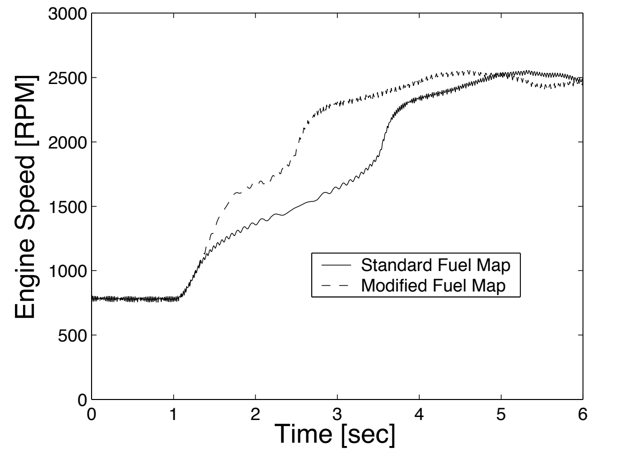


Figure 14: Variations of vehicle jerk with shift duration.

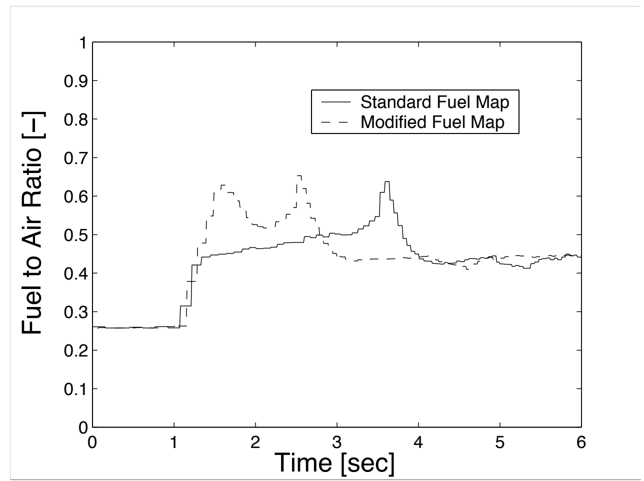
STUDY OF THE EFFECT OF FUEL CONTROL CALIBRATION ON ENGINE AND VEHICLE RESPONSE – System behavior at vehicle launch is the result of the combination of multiple effects, i.e., fuel system response to the step change of demand, turbocharger lag, fuel control correction for low boost, torque converter dynamic response, etc. The engine transient at launch can be described as rapid acceleration under load, and it is one of the most critical ones in terms of its effect on performance, particulate emissions, and powertrain vibration. In order to investigate the potential for improvements of engine transient response, a test is performed with a less restrictive low boost/low speed fueling correction than the standard one. This is accomplished by modifying the 2-D look up table in the low speed zone in order to limit the amount of fuel for a given pair of speed/boost values.

Then the vehicle launch is simulated, much like for the case described in the section on validation, and results of the two fueling calibrations are compared. Figure 15a illustrates the engine's ability to accelerate more rapidly with the new fueling map, while Figure 16a shows the corresponding increase of vehicle velocity during the first five seconds. Vehicle jerk is given in Figure 16b, and its peak values are significantly higher for the modified engine. This directly impacts the driver's subjective feel, and may prove to be too harsh for the average driver. The modified fueling map also causes more wheel slip (see Figure 16c), which in the long term may result in faster tire wear.

A closer examination of engine performance trends sheds more light onto processes taking place in the engine cylinder. The change of instantaneous in-cylinder equivalence ratio (ϕ) during vehicle launch is shown in Figure 15b. The modified, more aggressive fueling map causes two peaks of ϕ around 0.62, one at the very beginning and the other after roughly 1.7 seconds. The engine with the standard map experiences only one peak, after about 2.7 seconds. The initial high peak of ϕ caries more potential for adverse effect on engine particulate emission since it is



a)



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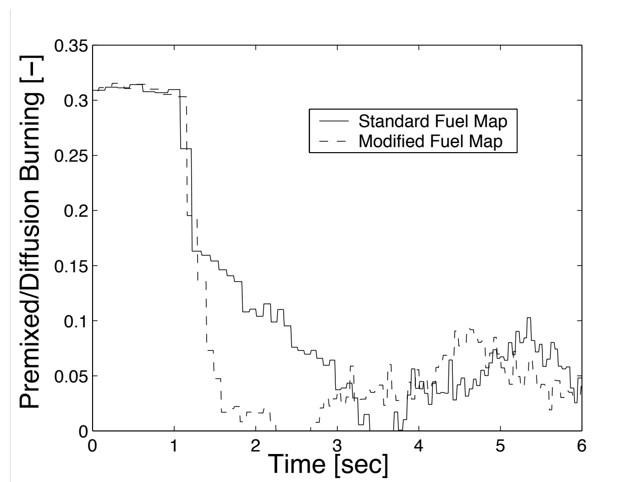


c)

Figure 15: Comparison of engine parameters during launch for two fueling strategies: a) Engine speeds, b) Fuel-to-air equivalence ratios, and c) Modes of burning.

occurring at very low engine speed, when flow and mixing in the cylinder are sluggish. Figure 15c illustrates the character of combustion through the mass ratio of fuel that burns in premixed mode versus diffusion controlled mode. For the modified fuel map, the richer mixture during the very beginning of the transient causes the fast decline of the premixed component, leading to complete domination by the diffusion burning process. This in fact can be a benefit in terms of engine NVH, since premixed burning typically translates into rapid pressure gradients after ignition and rough engine operation.

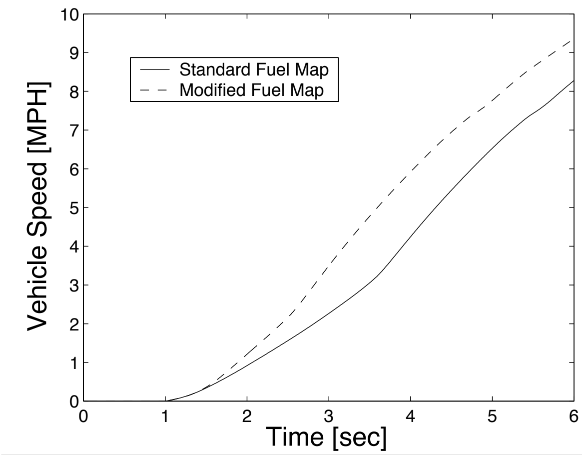
In summary, the engine and vehicle simulation with a comprehensive fuel control module allows evaluation of trade-offs associated with the fueling strategy modifications aimed at performance improvements. The final decision depends on careful evaluation of subjective (reaction to jerk, roughness) and objective criteria (soot emission, engine noise) for a specific application, i.e., vehicle configuration.

SUMMARY AND CONCLUSIONS

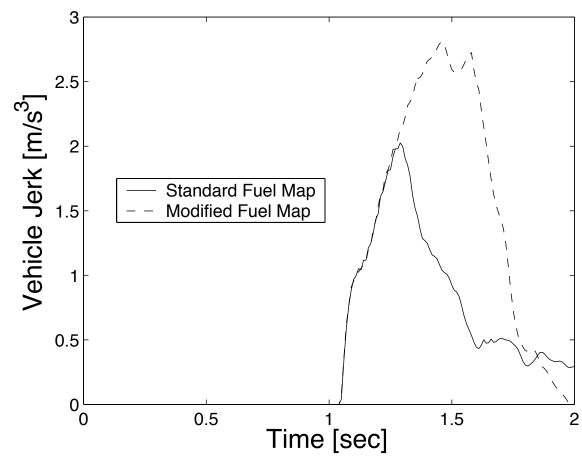
This work has reported further development, validation and use of a SIMULINK integrated vehicle system simulation. Our approach has relied upon integrating a transient thermodynamic engine module with driveline and vehicle dynamics modules of proper complexity for the simulation objectives. Fidelity enhancements included the incorporation of realistic engine fuel and vehicle speed controllers, as well as enhanced models of peripheral engine components, such as the manifolds and turbomachinery. The complete vehicle simulation structure was implemented in SIMULINK. Some of the main modules were programmed as SIMULINK MEX functions in order to enhance the computational efficiency of the integrated system. For validation purposes, the simulation has been configured to predict the dynamic behavior of an International 4700 4x2 truck, powered by the T444E V8, turbocharged, intercooled engine, and equipped with a four speed automatic transmission.

The following conclusions have been drawn from our work:

- The SIMULINK system framework allows easy integration of the vehicle system and addition or substitution of modules depending on goals of the specific study.
- The SIMULINK software environment also facilitated implementation of the Intelligent Vehicle Speed controller, which provided a replacement for the operator in the loop and allowed a feed-forward simulation to follow a prescribed vehicle speed schedule.
- For studies of vehicle acceleration performance on a straight and flat road, where excitation does not generate significant pitch motion, an enhanced point-mass model composed of two components (longitudinal and heave direction dynamics) is adequate.



a)



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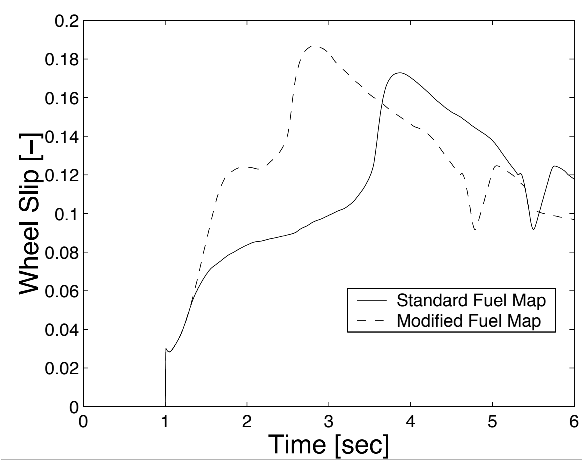

c)

Figure 16: Comparison of vehicle parameters during launch for two fueling strategies: a) Vehicle speeds, b) Vehicle jerk, and c) Wheel slip.

The simulation was applied to a series of transient runs with two main goals. One was to validate the complete engine/vehicle simulation under transient conditions, while the other was to study the effect of selected control parameters on critical aspects of system response under severe dynamic conditions. Validation was performed through comparisons of predictions and measurements with following results:

- Vehicle launch from stand still, and subsequent full load acceleration to 60 mph, showed very good agreement between predictions and measurements. The vehicle speed profile was predicted with great accuracy. The overall engine speed history showed very good agreement, with the exception of the very beginning of the transient. Discrepancies are attributed to the insufficient fidelity of the torque converter model, and its inability to handle the highly dynamic conditions, from idle to full load, within one revolution of the rotor. The rest of the engine speed profile during the first gear acceleration sufficiently captured turbo lag effects.

- Predictions of 30-50 mph passing time showed good agreement with measurements. Step changes of engine load, and consequent response of the torque converter, driveline and vehicle were somewhat less severe than in the case of vehicle launch. All modules were able to handle dynamic conditions experienced during this run. This and the previous test effectively validate the adequacy of predictions of engine and vehicle transient performance, as well as transmission behavior, including shift logic.

System response studies indicated the following:

- The duration of the shift event was shown to have significant effect on the vehicle performance. A shorter shift duration causes increased vehicle jerk, which can be a measure of driver comfort. However, the shorter shift duration may improve transmission clutch wear.

- Comparison of results obtained for the standard and modified fueling strategy enabled quantification of the effect of such modification on all aspects of the vehicle system response. More aggressive fueling strategy resulted in improved vehicle acceleration; significantly increased jerk felt by the driver; more wheel slip; more pronounced peaks of the in-cylinder F/A equivalence ratio; and predominance of diffusion burning process during vehicle launch.

These studies demonstrated the potential of the simulation to be a useful engineering tool for evaluation of vehicle design trade-offs, such as those associated with increasing performance, while at the same time managing vehicle NVH, emissions, driveability, driver feel, comfort, and durability.

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APPENDIX

Table 1: DI Diesel Engine Specifications

Configuration	V8, Turbocharged, Intercooled
Displacement [L]	7.3
Bore [cm]	10.44
Stroke [cm]	10.62
Connecting Rod Length [cm]	18.11
Compression Ratio [-]	17.4
Rated Power [HP]	210 @ 2400 rpm

Table 2: Driveline Specifications

Torque converter - Turbine inertia [kg*m^2]	0.068
Transmission - 1st gear churning losses coeff. R11	0.0192
Transmission - 2nd gear churning losses coeff. R12	0.015
Transmission - 3rd gear churning losses coeff.R13	0.031
Transmission - 4th gear churning losses coeff.R14	0.0367
Transmission - 1st gear churning losses coeff.R21	1.361 10^{-5}
Transmission - 2nd gear churning losses coeff.R22	5.719 10^{-6}
Transmission - 3rd gear churning losses coeff.R23	-3.189 10^{-5}
Transmission - 4th gear churning losses coeff.R24	-4.177 10^{-5}
Transmission - 1st gear ratio [-]	3.45
Transmission - 2nd gear ratio [-]	2.24
Transmission - 3rd gear ratio [-]	1.41
Transmission - 4th gear ratio [-]	1.00
Transmission - Fluid charging pump loss [N*m]	-6.12
Transmission - 1st Gear efficiency [-]	0.9893
Transmission - 2nd Gear efficiency [-]	0.966
Transmission - 3rd Gear efficiency [-]	0.9957
Transmission - 4th Gear efficiency [-]	1.0
Propshafts/Differential - Axle churning loss coeff.R0	8.34
Propshafts/Differential - Axle churning loss coeff.R1	0.04087
Propshafts/Differential - Differential drive ratio [-]	3.21
Propshafts/Differential - Differential efficiency [-]	0.96

Table 3: Vehicle Specifications

CG location from front axle [-]	0.61875
Sprung mass [kg]	6581.6
Unsprung mass rear [kg]	430.9
Unsprung mass front [kg]	244.9
Longitudinal - Wheel inertia [kg*m^2]	18.755
Longitudinal – Break viscous damping [N*m*s/rad]	100.0
Longitudinal – Break coulomb damping [N*m]	0.0
Longitudinal - Wheel radius [m]	0.4131
Longitudinal - Tire pressure [psi]	115.0
Longitudinal - Number of tires on rear axle [-]	4.0
Longitudinal - Wheel bearing damping [N*m*s/rad]	3.0
Longitudinal - Road/tire friction coefficient [-]	0.7
Longitudinal- Aerodynamic drag = 0.5*Cd*ρ*Area	2.081
Vertical - Rear suspension compliance [m/N]	6.34461 10^{-7}
Vertical - Rear tire compliance [m/N]	2.97403 10^{-7}
Vertical - Rear suspension damping [N*s/m]	7000.0
Vertical - Rear tire damping [N*s/m]	2000.0