Integration and Use of Diesel Engine, Driveline and Vehicle Dynamics Models for Heavy Duty Truck Simulation

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ABSTRACT

An integrated vehicle system simulation has been developed to take advantage of advances in physical process and component models, flexibility of graphical programming environments (such as MATLAB-SIMULINK), and ever increasing capabilities of engineering workstations. A comprehensive, transient model of the multi-cylinder engine is linked with models of the torque converter, transmission, transfer case and differentials. The engine model is based on linking the appropriate number of single-cylinder modules, with the latter being thermodynamic models of the in-cylinder processes with built-in physical sub-models and transient capabilities to ensure high fidelity predictions. Either point mass or multi-body vehicle dynamics models can be coupled with the powertrain module to produce the ground vehicle simulation. The integrated simulation can be used for predictions of dynamic response and performance of engine and driveline systems, for assessment of alternative system configurations and for integration studies in conjunction with the rest of the components of ground vehicles. Various illustrative studies are conducted for heavy-duty truck vehicles to demonstrate the capability of the simulation to predict performance and transient system response.

INTRODUCTION

Analysis, design and optimization of heavy-duty vehicles are time-intensive processes that involve costly testing of physical prototypes. The latter include engine, driveline, and vehicle components and sub-systems, as well as the complete vehicle system. The development and use of agile, predictive vehicle system simulations presents an attractive alternative to reduce the time and cost to bring new products to market.

In 1981, the US National Highway Traffic Safety Administration [1] developed a mainframe-based, DEC-10 language, heavy-duty vehicle simulation program (HEVSIM) capable of modeling a wide range of vehicles, drivetrains and driving schedules. In 1989, Phillips and Assanis [2] developed a flexible, PC-based Vehicle Powertrain Simulation (VPS) in Microsoft QUICKBASIC with a graphical user interface. The model was capable of simulating either steady-state or time-varying driver behavior using engine maps and a point mass vehicle model. Predictions of vehicle fuel economy and performance were shown to be in good agreement with vehicle test track data.

Integrating higher fidelity engine simulations with the rest of the vehicle has presented challenges. For example, Caterpillar invested significant effort into the development of ENTERPRISE, a code that married a thermodynamic diesel engine cycle simulation with DYNASTY - a dynamic system simulation solving in time domain for vehicle position, velocity, acceleration and jerk [3]. Difficulties in the integration of the engine with the rest of the vehicle were resolved by letting the engine run ahead one cycle, and using a matrix of output sensitivities to modify output values passed on to Dynasty during a given cycle. However, as stated by the author, this technique prevented the model from producing meaningful results above 15 Hz, and could even lead to instabilities.

A modular methodology, more suitable for integration of the engine with the rest of the vehicle, has emerged based on non-linear, state block diagrams and object-oriented, graphical programming environments [4,5,6,7]. Early control-type models were developed based on the so-called mean torque function. Moskwa and Hendrick [4,5] developed an automotive engine

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6 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.
modeling approach for real time control using Matrixx/System Build. Berglund [6] developed a model of turbocharged engines as dynamic system members in MATLAB/SIMULINK. Ciesla and Jennings [7] developed a powertrain model in EASY5 with the emphasis on driveline performance and control. Utilizing such mean torque models, studies have been conducted to predict driveability and shift quality as a function of clutch pressure control strategy [7], or transient fueling response to tip-in and tip-out when varying fueling control parameters [8]. However, the use of a look-up table for engine torque and brake specific fuel consumption compromises the accuracy of predictions of engine transient operation, since the table is generated through testing at discrete steady-state operating points. Furthermore, experiments have to be performed on an existing engine prior to simulation runs. Consequently, investigation of alternative designs and configurations is also severely limited. If the new design is to be assessed realistically under dynamic operating regimes, the simulation needs to include high fidelity models for the engine and its external components.

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, agile ARC simulation system to allow it to be tailored to required applications. Examples of potential uses include high fidelity modules that can fit into an existing simulation system; an overall simulation system to test a specific component or module; real-time simulation for operator-in-the loop tests; and very accurate, but slower simulations for design purposes. While the ARC develops simulations of on-road and off-road, heavy-duty vehicles, typically propelled by advanced, turbocharged, intercooled diesel engines, our approach can be applied to other classes of commercial vehicles.

As part of the ARC efforts, Munns [9] extended the flexible, modular cylinder-by-cylinder engine model and structure originally developed by Moskwa and Chen [10] in MATLAB-SIMULINK. Turbocharger map models and an automatic transmission model were developed [9] and linked with the parent model [10] to form a complete powertrain in SIMULINK [11]. Liu, et al. [12], also as part of the ARC efforts, explored techniques to decompose the engine cylinder model and to integrate process modules written in different languages within the SIMULINK framework. Their single-cylinder SIMULINK model was validated through predictions of dynamic engine behavior during cold start. Experience with model decomposition at the process level showed that there is a tradeoff between flexibility and calculation speed. It was concluded that increased internal communication associated with a large number of SIMULINK blocks can significantly increase execution time.

In parallel, the ARC has been exploring the potential for integrating higher fidelity engine and vehicle models in the context of ground vehicle simulation. The engine cylinder model used in this work is based on the turbocharged, multi-cylinder, diesel engine model developed by Assanis and Heywood [13], that emphasizes advanced physical submodels for all processes. Filipi and Assanis [14] decomposed the FORTRAN-based parent simulation [13], and added equations for engine dynamics to create a non-linear, transient single cylinder code which was validated with the experimental results of Liu et al [13]. Subsequently, Zhang et al. [15] transformed the FORTRAN code into a FORTRAN-MEX file, thus demonstrating the ability to integrate this single block containing models of all in-cylinder physical processes within the MATLAB-SIMULINK graphical programming environment. The diesel engine model was validated with experimental results from the ARC DDC Series-60 engine test cell [16]. For the vehicle model, rather than focusing on point mass descriptions, non-linear, 3-Dimensional multibody kinematics and dynamics models have been developed based on the work of Sayers and Riley [18, 19]. The model includes a detailed representation of the suspension, steering, brake systems and tires, and was developed with the AutoSim [19] code generator for vehicles and other multibody systems. The resulting equations of motion were first furnished in standard C-code and then converted to C-MEX format, suitable for SIMULINK integration.

For the complete ARC wheeled vehicle simulation presented in this work, the system simulation structure developed in MATLAB-SIMULINK by Rubin et al. [11] was used. This flexible SIMULINK structure allows for marrying engine cylinder, ancillary component, driveline, and vehicle modules, at various levels of detail. In this work, linking a high fidelity engine simulation module [13] with a detailed multi-body vehicle dynamics module [18] presented challenges due to the different time steps required for the solution of the differential equations inherent in those modules. In order to resolve cyclic processes, the engine module requires a time step on the order of a crank angle degree, which is much too small for the vehicle dynamics. For computational expediency, a "wait state" algorithm has been developed to eliminate unnecessary calls to the vehicle module. The fully integrated wheeled vehicle simulation enables studies using any suitable combination of engine, driveline, and vehicle modules depending on the simulation objectives. This is particularly important for our subsequent design optimization studies using rigorous mathematical models.
The paper is arranged as follows. The ARC flexible powertrain simulation framework in the MATLAB-SIMULINK environment is presented first. Then, the main features in the modeling of the multi-cylinder engine and its external components are discussed, followed by descriptions of models for the driveline. Next, the options for representing the vehicle dynamics are presented. Following the description of the models for the various sub-systems, the methodology for integrating the engine system with the driveline (to form a "virtual" powertrain) and with the vehicle is described. Two illustrative case studies have been conducted to demonstrate the flexibility of the reconfigurable model and its ability to predict dynamic system behavior. The first study emphasizes the transient characteristics of the powertrain model in the context of a point mass model of a "virtual" vehicle. Using this model, the effect of engine system design changes on truck performance on the flat road is explored. In the second case study, a realistic articulated truck, i.e., a 6x6 tractor combined with a three-axle semi-trailer is represented using a detailed multibody vehicle dynamics model. Using this model, vehicle-powertrain interactions during extreme transients, such as a hill climb of the fully loaded truck are investigated.

**POWERTRAIN SIMULATION FRAMEWORK**

The layout of a complete truck powertrain is shown in Fig.1. The main components are the engine system, torque converter (TC), automatic transmission (Trns), transfer case (Tr-C), interaxle differential (IA-D), and propshafts connecting the interaxle differential to the front differential (D-F), front-rear differential (D-FR) and the rear differential (D-R). The last three components are providing the torque to three sets of drive shafts that have wheels with tires attached to the other end. To generate the complete powertrain system simulation, one needs to develop individual modules first, and then consider the integration methodology.

The system simulation structure developed by Rubin, et al. [20] in MATLAB-SIMULINK was used as the framework for developing a highly modular, hierarchical, reconfigurable, and user-friendly vehicle system simulation. Interfacing of SIMULINK modules written in different languages, e.g., FORTRAN, C, or MATLAB is possible - a very important feature in a situation where existing pieces of code may need to be interfaced with the newly developed routines. Consequently, SIMULINK enables graphical reconfiguration of the system without any additional programming. This feature can be utilized very effectively for the configuration of different engine and driveline layouts (e.g., in-line 6 cylinder versus V12 engine), and in connecting the powertrain model with alternative vehicle models. Before presenting the methodology for integrating the engine, drivetrain and vehicle dynamics models into one coherent dynamic system simulation using SIMULINK, it is important to define the content of the primary modules. This will be discussed in the following three sections.

**ENGINE MODULE**

The engine module is comprised of multiplexed single 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.

**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 [13]. 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 [13] to large locomotive engines [21]. The system of interest is the instantaneous contents of a cylinder. It is open to the transfer of mass, enthalpy, and energy in the form of work and heat. The cylinder contents are represented

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Fig. 1: Powertrain for the truck with automatic transmission.
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. An empirical Arrhenius expression relates the length of ignition delay to the gas temperature and pressure in the cylinder after injection. Combustion is modeled as a uniformly distributed heat release process, using Watson's correlation [22]. 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 A/F 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 [13] 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 [13, 23]. Radiative heat transfer is added during combustion [13]. 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. The heat transfer from the gas to the walls of the various system components depend on the instantaneous difference between the gas and the wall temperature. In order to predict the time-dependent temperature distribution in the combustion chamber walls, the lumped capacitance method [25] is employed.

A friction sub-model based on the Millington's and Hartles' correlation [24] 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 necessitates modification of the FORTRAN source to make it fully compatible and "open" for communication links within SIMULINK. The procedure involves development of the FORTRAN-MEX file that contains all the necessary state derivatives and gateway routines for handling input and output vectors. Hence, the single-cylinder diesel engine module becomes a SIMULINK block that can be coupled with other blocks within the SIMULINK graphical environment. More details on the conversion technique can be found in [15].

ENGINE DYNAMICS MODEL — An automotive engine experiences frequent, and often rapid, variations of driver demand and external load. Hence, the powertrain simulation requires an engine model capable of dealing with the dynamics resulting from these varying operating conditions. Fig. 2 shows a schematic of the dynamic engine system with the primary inputs coming from the human/vehicle interface, the ambient conditions and the vehicle model. The instantaneous engine speed and torque are the main outputs. The engine brake torque \( \tau_e \) is calculated based on the current set of input parameters as a difference between the indicated and the friction torque. Then, the external load torque \( \tau_L \), imposed on the engine by the vehicle or the dynamometer, is subtracted from the brake torque and the net value is passed on to the engine dynamics module. A variable crankshaft inertia \( J(\theta) \) as a function of crankshaft position \( \theta \) is included. Its value is determined at each crank-angle by considering the equivalent moment of inertia of the piston, connecting rod and crankshaft assembly [26]. The engine dynamics equation:

\[
J(\theta) \cdot \dot{\omega}_e + 0.5 \cdot \frac{\partial J(\theta)}{\partial \theta} \cdot \omega_e^2 = \tau_e - \tau_L
\]

is solved at each crank angle to return the new value of crankshaft speed \( \omega_e \).

\[
\omega_e \quad \text{crankshaft rotational speed}
\]

\[
\tau_e \quad \text{torque}
\]

Fig. 2: Block diagram of the multi-cylinder engine dynamic system.
THE TURBOCHARGER uses compressor and turbine maps to determine the instantaneous mass flow rate and turbomachinery efficiency based on the shaft speed and pressure ratio supplied by the simulation at every integration step. The turbine and compressor wheels mounted on a common rigid shaft comprise a two-disk dynamic system. The turbogenerator dynamics equation determines the rate of change of angular velocity based on the balance between the compressor torque and the turbine torque. Hence:

\[
\dot{\omega}_{TC} = \frac{\tau_T - \tau_C}{J_{TC}}
\]

where \( \omega_{TC} \) is turbocharger shaft angular velocity, \( \tau_T \) is the torque produced by the turbine, \( \tau_C \) is torque absorbed by a compressor, and \( J_{TC} \) is the rotor polar moment of inertia. Friction losses are included in the calculation of the turbine torque.

THE INTERCOOLER - The intercooler model uses specified values for wall temperature and overall heat transfer coefficient. Additional model input parameters are the volume, orifice area and discharge coefficient. The model consists of a manifold element connected to an orifice element [11]. Thermodynamic state values in the intercooler are calculated by the manifold element based on heat and mass flows in and out. These values, together with the thermodynamic state of the gas in the intake manifold are input to the orifice part of the model. The outputs are the mass flow and enthalpy flow at the intercooler outlet.

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**Fig. 3:** Fuel controller model in MATLAB-SIMULINK.

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**FUEL INJECTION AND DRIVER/VEHICLE INTERFACE** - As in an actual engine, the amount of fuel injected is determined based on driver demand, represented as "rack position", engine speed, and intake air flow rate. Engine speed is also monitored to activate fuel "cut-off" if it increases above the rated value, hence governing the maximum engine speed. The cyclic mass of fuel injected is determined by:

\[
m_{\text{inj/cyc}} = \dot{m}_{\text{air}} \cdot \phi_{\text{tg}} \cdot \left( \frac{A}{F} \right)_{s} \cdot \text{Rack} \cdot \left( \frac{60 \cdot n_R}{n_{\text{cyl}} \cdot \omega_e} \right)
\]

and

\[
\phi_{\text{tg}} = 0.45 + 8.8757 \cdot 10^{-8} \cdot (\omega_e - 2100)^2
\]

where \( m_{\text{inj/cyc}} \) is the amount of fuel injected per cycle per cylinder, \( \dot{m}_{\text{air}} \) is mass flow rate of the intake air, \( \phi_{\text{tg}} \) is the target equivalence ratio at full load as a function of the engine speed, \((A/F)s\) is the stoichiometric air fuel ratio, \((\text{Rack})\) defines driver demand varying from 0 to 1, \( n_R \) is the number of crank revolutions for each power stroke per cylinder, \( n_{\text{cyl}} \) is total number of cylinders, and \( \omega_e \) is the engine speed in rpm. The equivalence ratio typically decreases as the engine speed increases in order to reduce emission control problems and component thermal loads. The above correlation for the target equivalence ratio is extracted from experimental data acquired by the ARC using a 12.7 L heavy-duty truck diesel engine [16]. Fig. 3 demonstrates the implementation of the fuel system model in MATLAB-SIMULINK. Parameters necessary as inputs to other modules, such as mass of fuel per cycle, fuel mass flow rate and total amount of fuel consumed are saved in the MATLAB workspace.
DRIVETRAIN MODULE

The drivetrain module contains the torque converter, transmission, interaxle differential and front and rear driveline – see Fig 1. It provides the connection between the engine and the vehicle dynamics module. The torque converter input shaft on one side and the wheel on the other side are the connecting points for the engine and the vehicle dynamics models, respectively.

TORQUE CONVERTER - Torque is transmitted to the transmission through the torque converter, which is a critical element in the integration of the engine with the driveline. The torque converter's input shaft, i.e. TC pump shaft, is connected to the engine, while the torque converter's output shaft, i.e. TC turbine shaft is connected to the transmission. The modeling process is illustrated in Fig. 4.

\[ \text{Pump Torque} = \left( \frac{\text{Engine Speed}}{K \text{ Factor}} \right)^2 \rightarrow \text{To Engine - Load Torque} \]

![Torque ratio vs. Speed ratio Look-up Table](image)

![Turbine Torque](image) \rightarrow \text{To Transmission}

Fig. 4: Torque converter modeling methodology.

At any instant, using the pump to turbine speed ratio, a pump K factor can be determined from a look-up table. Then, the pump torque is calculated based on the instantaneous engine speed using the expression shown in Fig. 4. The turbine torque can now be determined from a look-up table using the pump-torque value and the speed ratio.

TRANSMISSION – The torque converter’s output shaft is connected to the four speed automatic transmission with planetary gears. The following assumptions have been made in modeling the transmission: all rotating links are rigid, all links have only one degree of freedom, planet gear inertia is negligible, gears exhibit no backlash, bearings have no play, and friction effects are negligible. The operating modes are defined by combinations of locked and unlocked clutches. Shifting is accomplished through simultaneous disengagement and engagement of appropriate clutches during a user-specified interval of time. Pressure profiles used for clutch action are linear, although the user has an option of applying any other profile.

A set of first order linear differential equations have been generated in state space form. These equations for each mode were derived from the effective inertia of the links and gears within the transmission and the dynamics of the rotating members. For more details on the complete transmission model, the reader is referred to [9,20]. The parent model [9] has been augmented with the gear upshift logic based on engine speed. The controller model sends the signals for activation/deactivation of clutches as soon as the engine reaches a critical angular velocity specified as input.

DRIVELINE – The transmission output torque is multiplied by the transfer case gear ratio and used by a flexible propshaft model [11] to determine the torque transmitted to the interaxle differential. The rest of the driveline is subdivided into two sub-systems: front and rear driveline. The rear driveline includes two axles complete with differentials, driveshafts, wheels and tires. The interaxle differential determines the accelerations of gears based on propshaft torques and inertias. A flexible shaft model characterized by the stiffness, damping and inertia is used for propshafts connecting the interaxle differential to three axle differentials. The driveshaft models are based on the same flexible shaft torsional module previously applied to propshafts. Hence, they calculate the torque output from the difference between instantaneous angular velocities on either end of the driveshaft as inputs. On one end, the instantaneous shaft speed is determined by the differential output. On the other end, the shaft speed is equal to the wheel speed and is effectively the result of tire-terrain interactions and vehicle dynamics calculations. Alternating of inertial devices that accept torque as inputs and pass speeds as outputs, and compliant devices that accept speeds as inputs and pass forces as outputs, ensures proper causality, thus eliminating algebraic loops and enabling efficient calculations [20].

VEHICLE DYNAMICS MODULE

Two approaches can be used to model vehicle dynamics depending on the overall simulation objectives. A point mass model can be selected for an overall 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 high fidelity predictions of vehicle acceleration and speed on flat, smooth roads. However, a detailed multibody vehicle dynamics model is necessary for the investigation of the vehicle-powertrain interactions during extreme transients associated with pitch motion, e.g. transients excited during a hill climb of the fully loaded truck on a slippery road [17].
The full multibody model of the heavy-duty truck is based on previous research [18] and the emphasis of this work has been on understanding the most important factors contributing to vehicle dynamics as excited by the powertrain. The side view schematic of the vehicle system is given in Fig. 12. The full nonlinear 3D multibody kinematics and dynamics model includes a realistic suspension model featuring a spring model with nonlinear spring rates, nonlinear dampers, kinematic roll-steer, and auxiliary roll stiffness; a realistic truck asymmetric steering system, with major compliance effects and nonlinear kinematics; a brake system with nonlinear brake torque properties, and left-right torque imbalances; and a comprehensive non-linear tire model, with user-supplied tables and road friction. A more detailed description of the vehicle dynamics model can be found in [18].

The model is composed of 8 rigid bodies, has 21 multibody DOF (Degrees of Freedom), 21 multibody coordinates, 37 auxiliary coordinates, 21 multibody speeds, 12 auxiliary speeds, and 91 active forces and 49 active moments. The eight rigid bodies are:

- Tractor with 6 DOF (3 Translations, 3 Rotations)
- Trailer with 3 DOF (3 Rotations)
- 6 Axles with 2 DOF (Roll, Jounce)

These rigid bodies are constrained by joints and forces/moments to capture the accurate behavior of the system. There are a total of 150 forces and moments:

- 6 Suspension Spring Forces (Left, Right)
- 6 Suspension Dampers Forces (Left, Right)
- 22 Tire Forces (Vertical, Longitudinal, Lateral)
- 1 Aerodynamic Drag
- 6 Auxilary Axle Moments
- 1 Hitch Moment (X, Y, Z)
- 22 Tire Aligning Moments
- 24 Tire Rolling resistance

The above model is able to predict the vehicle motion as it is excited by wheel torque, braking and steering input. However, the off-plane dynamics (Yaw and Roll) can only be excited with steering input or asymmetric road profile. In the case of zero steering input and symmetric road (left-right) the vehicle motion is reduced to the motion in the pitch plane. This simplified model has sufficient complexity to predict load transfer during the pitching motion of the vehicle and it is used in the second case study – vehicle acceleration during hill climb on slippery road. The complexity of the vehicle dynamics model is systematically adjusted, as proposed by Louca et al. [28], to accommodate the needs of a specific scenario.

The dynamic model is mathematically represented by ordinary differential equations that describe the kinematic and dynamic behavior of the real system. These equations are produced by the AutoSim [19] multibody equation generation software and are originally coded in the C programming language. Then, to be able to integrate the dynamic model with the powertrain, the C code is converted into a C-MEX file, by applying a similar technique as the one used to create the FORTRAN-MEX single-cylinder engine module. Hence, the final product is an S-function suitable for direct integration with the powertrain SIMULINK model.

INTEGRATION METHODOLOGY

In our approach, we consider three main sub-systems/modules: engine, drivetrain and the vehicle dynamics. The first step in the integration methodology is identification of key parameters that define the physical interaction between modules. In case of a vehicle system, those are the “active” and “resistive” torques, as well as the angular speeds of key powertrain component shafts. The schematic in Fig. 5 illustrates three sub-systems and links defining their interaction. The transient turbocharged engine simulation provides as output the instantaneous value of engine torque and rotational speed. The engine speed is passed on to the

Fig. 5: Powertrain system integration methodology.
torque converter input shaft, i.e. torque converter pump. The pump/turbine speed ratio in the torque converter will determine the multiplication of torque in the TC; hence, the calculated TC turbine torque will be available at the transmission input shaft. Further torque multiplication depends on gear ratios in the driveline components between the TC and the drive shafts connected to the wheels. The torque on the wheels is now translated into the tractive force determining vehicle dynamic behavior, in conjunction with other information about the vehicle and the terrain. Thus, the instantaneous vehicle speed and the wheel angular velocity will be the output of the vehicle dynamics module. This information is propagated back through the system, all the way to the TC output shaft, thus determining the TC turbine wheel speed.

At the same time, the torque converter pump speed and the speed ratio between the torque converter pump and turbine rotors, allows calculation of the pump torque which is in turn used as the resistance torque in the engine dynamics equation. As long as the instantaneous engine torque is greater then the pump torque, the engine will accelerate; and vice versa, any deficiency in engine torque will cause the engine to decelerate. The newly calculated engine speed and torque, together with new values of vehicle wheel and torque converter turbine speeds are used as input in the next integration step.

The coupling of the modules is accomplished by creating graphical links between SIMULINK blocks. The classic problem in software integration of "who is in charge" is avoided by using the common solver from the MATLAB-SIMULINK library for the complete vehicle system. The engine part is typically more demanding in terms of the ability of the integrator to handle stiff systems and to adjust the integration step appropriately. Therefore, the performance of the multi-cylinder engine module was investigated off-line prior to integration [15]. Results showed that the RK-3 integrator behaves the best, i.e. produces output closest to the predictor-corrector routine ODERT of Shampine and Gordon [27], previously optimized in FORTRAN. Thus, the RK-3 integrator was selected for further studies of the vehicle system. However, the step size required for the high fidelity cycle simulation is usually much too small for the vehicle dynamics code and unnecessary calls of the vehicle module can significantly slow down the overall calculation. This problem was alleviated by introducing wait states into the vehicle dynamics block. Thus, vehicle dynamics calculations are effectively performed with a larger integration step than the one specified for the engine computations.

VEHICLE PERFORMANCE PREDICTIONS

Two illustrative case studies have been conducted to demonstrate the flexibility of the reconfigurable model and its ability to predict dynamic system behavior.

CASE STUDY 1: ACCELERATION ON FLAT ROAD
One of the primary criteria for evaluating powertrain performance and truck response is based on ability to accelerate after a sudden increase of driver demand. Hence, the first case study emphasizes the transient characteristics of the powertrain model in the context of a point mass vehicle model. Using this model, the effect of engine system design changes on truck performance on a flat road is explored. The vehicle is placed on the flat road and it starts accelerating from stand still with the sudden increase of driver demand to 100%. The fuel control strategy is based on the target fuel/air equivalence ratio, as described previously in the engine module section. Hence, the amount of fuel injected in every cycle depends on the driver demand, instantaneous engine speed and mass flow rate of air through the engine.

For this case study, a virtual vehicle, representative of a super-heavy truck hauling oversize loads, is simulated. The gross vehicle weight was specified to be 70 tons. This vehicle requires a sizeable powerplant: a V12 turbocharged, intercooled engine with total displacement of 37.7 liters. The design compression ratio was 15, and two turbocharger-intercooler sets (one set per 6 cylinders) were used to provide sufficient airflow through the engine. The flexibility of the SIMULINK powertrain system simulation was utilized to its fullest, i.e. previously developed engine cylinder modules and blocks containing component models, such as manifolds, restrictions, turbochargers, etc. were "resized" and assembled to generate a hypothetical 1350 HP (1007 kW) diesel engine system. Look up tables for the torque converter models were modified to allow it to handle the large amounts of torque produced by a V12 engine. The polar moments of inertia of the transmission components were modified accordingly. The final gear ratio in the differentials was adjusted to allow good match between the engine and the vehicle, i.e. a good trade-off between acceleration, maximum speed and fuel economy.

To demonstrate the simulation's ability to predict the effects of engine transients on vehicle acceleration and to show its usefulness as a design tool, the following three design options were investigated. First, a run was performed with the baseline engine described above. Next, the inertia of the turbocharger was reduced to half of the baseline value. Finally, the vehicle performance was explored with the same engine as the baseline, but without the intercooler. The results are discussed below.
Acceleration with the baseline engine - Fig. 6 shows the time history of angular velocity for various shafts in the system during the first 35 seconds of the truck acceleration, while Fig. 7 illustrates the variations of torque in the system during the same period of time. Engine speed increases sharply right after the start. This immediately initiates a transient in the torque converter, since a very large slip ratio occurs between the pump wheel, connected to the engine, and the turbine wheel, connected to the transmission. The large slip in the converter produces significant multiplication of engine torque felt at the TC turbine shaft, and propagated through the transmission towards the wheels, during the first three seconds of the transient (see Fig. 7). This torque multiplication is the essence of the TC role during the start from the stand still. At the same time, the TC pump torque is felt by the engine as an external load. The large TC speed ratio produces an increase in the pump torque. Thus, engine speed shows hesitation during the first second of the run, before the engine is able to equilibrate and produce enough brake torque to start rapid acceleration. During the second half of operation in first gear, the speed ratio in the torque converter diminishes, thus the engine torque becomes very close to the TC output. The first 8 seconds are characterized by a fairly gradual increase of engine torque, as boost pressure builds up (see Fig. 8).

![Graph showing rotational speed histories for various shafts in the baseline powertrain system.](image)

**Fig. 6:** Rotational speed histories for various shafts in the baseline powertrain system.

![Graph showing torque histories for various shafts in the baseline powertrain system.](image)

**Fig. 7:** Torque histories for various shafts in the baseline powertrain system.

![Graph showing variation of pressures in the inlet and exhaust manifold of the baseline engine system.](image)

**Fig. 8:** Variation of pressures in the inlet and exhaust manifold of the baseline engine system.

As illustrated in Fig. 8, variations of intake and exhaust manifold pressures with time show that a significant turbo lag occurs at the beginning and effectively persists until the first gear shift. Since the fuel controller limits the amount of fuel per cycle at low boost pressures, compressor lag is directly responsible for the engine brake torque response shown in Fig. 7. In summary, while the behavior of the powertrain system during the first 3 seconds is largely affected by the engine/torque converter interaction, the rest of the run in the first gear depends primarily on turbocharger dynamics and fuel control.

The gearshift creates a very sharp fluctuation in shaft angular velocities and torques. As a result of the gear ratio change, the "transmission in" speed (TC out)
moves closer to "transmission out" speed, almost instantaneously. This also introduces a new departure between the flywheel speed and the TC turbine (TC in) speed, thus creating large slip in the torque converter and multiplication of torque. Therefore, a "saw tooth" can be identified on the torque converter out torque history (see Fig. 7) after every shift. Increased slip in the TC produces an increase of pump torque loading the engine, and thus a sharp decrease of engine speed is observed immediately after the shift. This reduces mass and enthalpy flow rates through the engine and causes a decrease in turbocharger speed and exhaust pressure. However, both the engine and turbocharger speeds remain at fairly high levels (see Fig. 9 and Fig. 10), even at the end of gearshift event, i.e. they never drop down to levels seen at start-up. Consequently, after the shift, the engine accelerates again with practically no effect of turbo lag, and it maintains nearly constant output torque throughout the rest of the run.

Reducing the overall gear ratio during the next two shifts produces a similar transient behavior, with reduced amplitude of speed and torque spikes during the shift. The torque levels on the driveshaft show a general decreasing trend with increasing vehicle speed and decreasing overall gear ratio, as expected. Spikes associated with the shift are not visible on the driveshaft speed line since the driveshaft is modeled as flexible, and thus able to absorb sudden fluctuations.

**Acceleration with the low inertia turbocharger** – The second run was performed with the same vehicle and the engine, but with a low inertia turbocharger. It was assumed that the turbocharger polar moment of inertia is reduced by a factor of two through application of new lightweight materials and advanced rotor design. All input parameters remained the same, including the fuel control logic. The run started as before with the sudden increase of driver demand to 100%. The main features of the transient run remained the same, e.g. initial engine - torque converter interaction and fluctuations during gearshifts are quite similar to those of the conventional powertrain. However, the low inertia turbocharger version shows better vehicle acceleration than the base one.

The engine with lower inertia turbocharger accelerates at a faster rate during the first half of the first gear transient, as shown in Fig. 9. This is a direct consequence of a shorter turbo lag illustrated by the comparison of turbocharger speed histories in Fig. 10. After the turbocharger speed stabilizes at about 75,000 rpm, the benefit of the lower inertia turbocharger becomes insignificant. Therefore, it can be concluded that the low inertia turbocharger can significantly improve the powertrain response, but only during rapid transients spanning a wide range of engine speeds or loads. So, in this case the overall vehicle acceleration improvement is basically the result of the better system response in first gear. Another typical situation where the vehicle dynamic response would improve with the reduction of rotor inertia is overtaking, i.e. sudden acceleration after cruising at moderate vehicle speeds.

**Acceleration performance without intercooling** – The next scenario was chosen to demonstrate the use of the simulation as a decision-making tool for assessing early-design trade-offs between performance and component cost. As an illustration, the intercooler was removed.
from the engine system. In general, the dramatic reduction in charge density due to the removal of intercooling reduces the intake mass flow substantially compared to the baseline, intercooled engine, thus leading to lower power levels. However, Fig. 9 and Fig. 10 reveal that the difference between the two versions is surprisingly small at the beginning until about 6 seconds. To better understand this phenomenon we should examine the intake air temperature histories of the two engines given in Fig. 11. At the beginning of the transient, when the boost pressure is still low, air temperatures are relatively low even without intercooling, thus the difference between the two is not significant. At the same time, the dynamic response of the non-intercooled engine benefits somewhat from the fact that one restriction is removed from the intake air path and one less volume needs to be filled before the air enters the cylinder. When the turbocharger reaches high speeds and boost increases, the effect of intercooling becomes obvious, and vehicle performance starts to suffer due to lower air flow and power output produced by the non-intercooled engine. It is worth noting that if an engine torque look-up table was used instead of the high fidelity engine model to predict vehicle acceleration, the difference between the intercooled and non-intercooled engines might be exaggerated due to the lack of capability to accurately capture turbolag.

A comparison of the vehicle performance predicted for the three engine system configurations is given in Table 1. Clearly, the vehicle with the low inertia turbocharger reaches higher speed and further travel distance after 35 seconds. These effects can be simulated only with a code that has the capability of predicting the transient behavior of the realistic engine system.

**Table 1: Vehicle performance after 35 seconds.**

<table>
<thead>
<tr>
<th></th>
<th>Baseline</th>
<th>Low Inertia Turbocharger</th>
<th>No Intercooling</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Vehicle Speed [km/h]</strong></td>
<td>89.04</td>
<td>90.00</td>
<td>87.13</td>
</tr>
<tr>
<td><strong>Traveled Distance [m]</strong></td>
<td>528.9</td>
<td>543.2</td>
<td>519.6</td>
</tr>
</tbody>
</table>

**CASE STUDY 2: HILL CLIMBING** – In the second case study, a detailed pitch plane multibody vehicle dynamics model is used to investigate the vehicle-powertrain interactions during extreme transients, such as a hill climb of the fully loaded truck [17]. For this study, a complete vehicle simulation was configured for the M916A1/M870A2 tractor/semitrailer combination. The propulsion system is based on a 12.7 L turbocharged, intercooled, six-cylinder DI diesel engine (DDC Series 60), and a 6x6 driveline that includes a torque converter and a four speed automatic transmission (Allison HT 740), transfer case, interaxle-differential, one front and two rear differentials/axles. The tractor-semi was placed on an uphill road with a 6% grade, as shown in Fig. 12. The initial conditions are that of a coast at 10 mph, i.e. the vehicle is traveling at 10 mph, but it is slowing down because there is no drive torque at the wheels to overcome the grade, aerodynamic drag and rolling resistance. At t=0 sec, the driver suddenly presses the pedal all the way down and demands maximum torque from the engine in order to accelerate the vehicle. Again, the actual mass of fuel injected at any instant is determined by the controller based driver signal, calculated mass flow rate of air and engine speed. The truck was fully loaded (Gross Curb Weight is 126,000 lbf – 57 tons) and the road surface was assumed to be wet (tire/ground friction coefficient, $\mu = 0.4$).

![Articulated truck schematic on grade.](image-url)
Vehicle behavior — Fig. 13 shows how the vehicle speed and acceleration change during the first 8 seconds of the hill climb. The vehicle speed decreases slightly during the first 0.5 seconds, but then it starts to increase at a fairly high rate. After about 5 seconds, the slope of the line tapers off. Initially, the vertical load (Fz) on the front axle is much lower than on each of the rear axles because of the position of the tractor's center of gravity (CG) and the pitch of the tractor on the inclined road (see Fig. 14). After the engine torque is transmitted to the wheels, the tractor starts to pitch more and the weight distribution changes. More specifically, the vertical load on the front axle decreases, while the load on the rear axles increases. Fluctuations observed on the Fz response are the result of oscillations in the suspension system initiated with the change of tractor pitch. These transients of the vertical load affect the ability of the wheels to maintain traction under slippery road conditions. Consequently, as shown in Fig. 15, the rear wheel speed is similar to the vehicle speed, while the front wheel speed departs dramatically from the expected trend after only a fraction of a second. Correspondingly, the wheel tread speed increases from around 10 mph to 24.5 mph in about 0.5 seconds. This is a clear indication of the front wheel slip and it explains the loss of vehicle performance during the very first part of the transient.

![Graph of vehicle velocity and acceleration](image)

**Fig. 13:** Vehicle velocity and acceleration during hill climbing.

![Graph of wheel loads](image)

**Fig. 14:** Wheel loads during hill climbing.

![Graph of wheel speed-tread](image)

**Fig. 15:** Front and rear wheel speeds indicating front wheel slippage during hill climbing.

In summary, the simulation of the vehicle and powertrain dynamic behavior during the full power hill climb reveals a very dramatic departure from normal system behavior during the first few seconds due to the slip of the front wheels. This results in very rapid engine acceleration, since wheel slip is felt as the instantaneous unloading of the engine system. The high rate of engine acceleration leads to the even more pronounced turbo-lag and creates conditions very different from any steady-state operating point. Results like these demonstrate the importance of considering extreme transient conditions when designing driveline and engine system components and when analyzing system control aspects.

Critical Engine System Transients — Beyond using the high fidelity powertrain simulation to study vehicle response, the simulation can identify the repercussions of rapid transients on engine processes. Critical engine transients will produce conditions much different than any steady-state operating point. It is these critical
transients that typically prove to be most challenging when it comes to combustion optimization and tradeoffs between efficiency and emissions. Simulation results can point out what are the worst conditions an engine cylinder may "see" during dynamic engine operation. After a design or a control strategy has been modified to improve certain aspects of engine transient behavior, the simulation may be used to run the virtual vehicle again, according to a prescribed driving schedule, and thus evaluate the overall effect of these modifications.

The results of Fig. 16 provide more insight into one of the critical transients observed during hill climbing. It shows pressure histories in the cylinder, intake manifold and exhaust manifold during start up, i.e. during the first 0.8 seconds of sudden acceleration. The first important feature is the fact that the pressure in the exhaust manifold increases almost instantaneously due to the sudden increase of exhaust enthalpy following the step change of fuel input. However, the inertia of the turbocharger and the gas dynamics of filling the intake manifold cause a substantial lag on the intake side. Consequently, there is a large negative pressure differential (up to 0.5 bar) between the intake and exhaust sides. This is not seen during steady-state operation, where the turbocharging system was shown to produce a positive pressure differential that benefits both scavenging of the combustion chamber and system efficiency. Hence, during such a transient, the residual fraction in the cylinder is much higher than normal, causing degradation in combustion and system efficiency. Peak in-cylinder pressures roughly follow the gradual increase in intake manifold pressure. However, the fact that scavenging is very different than at steady-state for the same level of intake pressure, emphasizes the possible pitfalls in case the fuel injection control is based solely on manifold pressure. Clearly, the amount of fuel injected should be reduced and the injection timing should be adjusted if conditions such as the above are detected, otherwise particulate emissions may become excessive.

Another critical transient is illustrated in Fig. 17. The same quantities are shown during a gearshift interval on the flat road. Initially, the intake manifold pressure is above the exhaust manifold pressure line. These conditions could be considered quasi-steady due to the fact that the rate of change of engine variables has significantly decreased after 4.5 seconds of acceleration. However, the shift sequence initiates very sharp fluctuations of engine speed, thus there are equally dramatic fluctuations of manifold pressure, especially on the exhaust side. A speed increase when the transmission clutch starts releasing causes an increase in exhaust pressure above intake. This is followed by a large drop in exhaust pressure due to the drop in engine speed after the higher gear clutch starts engaging. Finally, exhaust pressure will exceed the intake again, before the shifting event is over. This is due to a light downward fluctuation in engine speed caused by inertial effects before the system is stabilized again in the higher gear. Details of this type of a transient are very critical for optimization of the shift quality, and for possible optimization of the engine controller to assist in a smoother shift process. However, such "wild" fluctuations of the pressure drop across the engine will also produce conditions in the cylinder that are far away from regular operation and may adversely affect combustion and emissions.
SUMMARY AND CONCLUSIONS

A complete wheeled vehicle system simulation is reported in this work. A high fidelity, powertrain system simulation has been developed based on the integration of a physically based diesel engine model with models of the external components and the driveline, within the hierarchical system simulation structure in MATLAB-SIMULINK. The same software environment is utilized to integrate the powertrain with the vehicle dynamics model. The flexibility of the new tool has been utilized to create virtual in-line 6 cylinder and V-12 turbocharged diesel systems, with or without intercooling. Three alternative vehicle dynamics descriptions, a 2 state point mass model, a 39 state 2D pitch plane model and a 91 state, 3D multi-body dynamics model, have been integrated with the complete powertrain model. It has been shown that integrating high fidelity powertrain and vehicle dynamics simulation models can be invaluable for studying a variety of issues related to transient system performance. The following issues were specifically addressed in this paper:

1) Assessment of the effect of external engine component characteristics, such as turbocharger and intercooler, on vehicle acceleration. More specifically:

- Results show that lowering the turbocharger inertia can significantly improve engine response during acceleration in first gear. The time between start-up and the first gearshift is reduced by roughly one second. After the first gear shift, the turbocharger operates continuously at high speed and engine performance and response do not differ much for the two turbocharger designs.

- Lack of intercooling is almost not felt at all during the turbo-lag period. This is the result of the small effect of intercooling when the boost pressure and temperature are relatively low. Also, the lack of intercooling appears to be offset by the fact that there is no pressure drop across the intercooler, and the volume of the intake system that needs to be filled with compressed air is smaller. In the "high boost" phase of the acceleration transient, i.e., after the first gear shift, the non-intercooled engine starts to suffer and falls behind due to lower air flow rate and peak power output.

2) Assessment of dynamic interactions between the powertrain and the vehicle dynamics during an extreme transient, such as full power hill climbing maneuver, i.e.:

- The vertical load on the front tires is reduced due to increased tractor pitch that causes front wheel slippage. This results in very rapid engine acceleration, since wheel slip is propagated back as unloading of the engine system. The high rate of engine acceleration leads to even more pronounced turbo lag, unseen at other conditions.

3) Detailed investigation of two critical engine/driveline transients during rapid accelerations, i.e.:

- There is a large negative pressure differential (up to 0.5 bar) between the intake and exhaust sides at start-up. After the step change of fuel input, the pressure in the exhaust manifold increases due to the sudden increase of exhaust enthalpy, while the inertia of the turbocharger and the gas dynamics of filling the intake manifold cause a substantial lag on the intake side. This is in sharp contrast with quasi-steady operation characterized by a positive pressure differential.

- The shift sequence initiates very sharp fluctuations of engine speed and equally dramatic fluctuations of manifold pressure, especially on the exhaust side. The pressure differential between the intake and exhaust sides changes from a large positive value to a negative one, within less than a second of engine operation. Details of this type are very critical for optimization of shift quality, and for stable combustion with reduced emissions.

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