

Development of a Vehicle Model Architecture to Improve Modeling Flexibility

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ABSTRACT

In this paper a dynamic, modular, 1-D vehicle model architecture is presented which seeks to enhance modelling flexibility and can be rapidly adapted to new vehicle concepts, including hybrid configurations. Interdependencies between model sub-systems are minimized. Each subsystem of the vehicle model follows a standardized signal architecture allowing subsystems to be developed, tested and validated separately from the main model and easily reintegrated. Standard dynamic equations are used to calculate the rotational speed of the desired driveline component within each subsystem i.e. dynamic calculations are carried out with respect to the component of interest. Sample simulations are presented for isolated and integrated components to demonstrate flexibility. Two vehicle test cases are presented. The application to a conventional heavy-duty vehicle demonstrates the operational capabilities of the modelling methodology, while the inclusion of electrical components to form a mild-hybrid heavy-duty vehicle demonstrates the model's potential for predicting improvements in fuel economy and performance over a specified drive cycle. Qualitative validation characteristics are presented, highlighting the ability of the model to accurately capture dynamic events and fuel consumption profiles.

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INTRODUCTION

The modelling of vehicle propulsion systems prior to any testing is common practice in most vehicle engineering applications. System modelling can drastically reduce both concept to production times and associated testing costs. A model allows the user to determine the advantages gained from any changes in vehicle topology and perform a cost-benefit analysis to determine whether the modifications are worthwhile before committing significant resources. Another advantage of simulation models is that they can be used to test and develop new and unproven components and systems to identify promising concepts or architectures removing the need for building prototype vehicles or test rigs saving both time and cost [1].

With rapid advancements in technology, coupled with increasingly stringent legislative requirements, there is a growing need for models which are not only reliable, but also extremely flexible. The main problem commonly encountered with model development is the lack of flexibility when the topology of the vehicle is changed significantly [2]. This paper proposes a vehicle model architecture (VMA) that attempts to address this problem. The proposed VMA attempts to build on those previously established [3,4] to improve the models flexibility and adaptability to new vehicle concepts.

Both the automotive and heavy duty sectors have migrated towards a modular modelling approach as it provides a range of advantages. It enables easily adaptable models capable of handling different

subsystems of variable complexity. For example, different engines or transmissions can be simulated in the same vehicle architecture to determine the effect on fuel consumption or any other system dynamics of interest. Libraries of components can be created and shared amongst different users which reduces development time. However, the downside of a modular modelling approach is that each component must conform to a standardized signal architecture to facilitate sub-system interchangeability. A robust VMA which is suitable in application to nearly all road vehicles has been previously described by Belton *et al.* [3]. The top-level subsystems were established based on a logical decomposition of the vehicle components. These are the driver, environment, electrical, auxiliaries, powerplant, transmission, driveline, chassis, braking, steering, vehicle system controller (VSC) and bus. Matlab/Simulink was used to develop the VMA, as shown in [Figure 1](#).

Each model subsystem shown in [Figure 1](#) contains various layers. The top layer is shown in [Figure 1](#) and connects each subsystem together through the main signal bus. This is denoted as Layer 1. Layer 2 is contained within each subsystem and illustrates how each of the required signals is transmitted into each component from the main bus and also shows how feedback is transmitted back to the main signal bus. Layer 3 then describes local subsystem controller and plant behavior. [Figure 2](#) shows how an IC engine and automatic transmission with a torque convertor would be implemented in this model architecture.



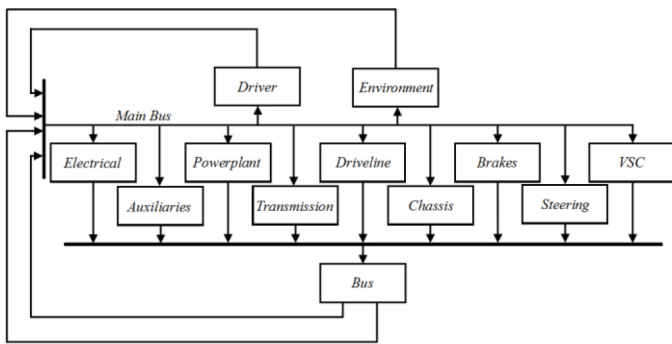


Figure 1. Top Level Vehicle Model Architecture showing critical subsystems and bus connections [3]

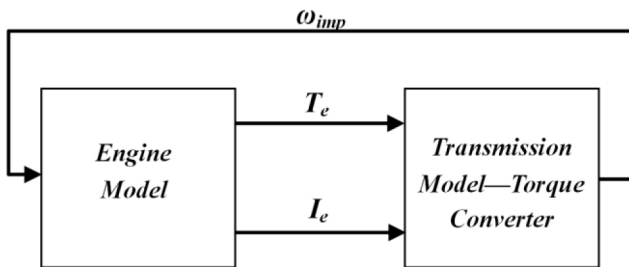


Figure 2. Engine and automatic transmission torque convertor signal paths [3]

The engine inertia and gross engine torque from the engine model are passed to the torque convertor model block. The impeller speed is then passed from the transmission subsystem back to the engine where it becomes the engine speed input for the engine model [3]. This inherently means that these two subsystems are somewhat coupled to one another as the engine model now has a level of dependency on the subsystem that is downstream of it. Further investigation was conducted on a similar model based on a similar architecture by Mason *et al.* [4]. Both compliant and non-compliant models for the same system were investigated. The non-compliant version, a schematic of which is shown in Figure 3, uses the approach of cumulative torque and cumulative inertia. These signals are carried to the end of the drivetrain system where the signals are terminated, at which point a speed parameter is calculated. This speed signal is then passed backwards through the drivetrain and transformed through gearing ratios accordingly to determine every other drivetrain components speed.

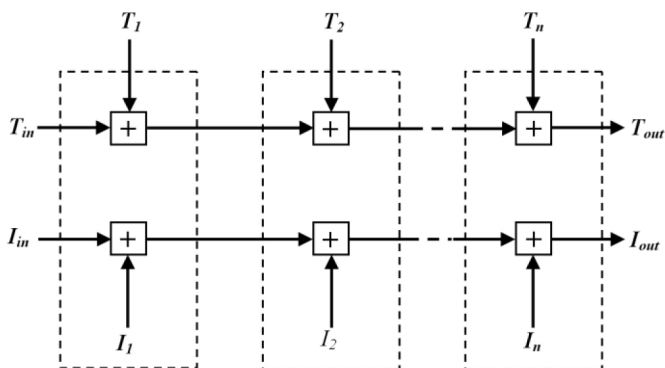


Figure 3. Modular sub models showing cumulative torque and inertia paths [4]

A similar VMA modelling approach was implemented for heavy duty vehicles by Emirler *et al.* [5]. Figure 4 shows a summarized diagram of the signal paths used in the model. In each of the model

architectures described, the subsystems involved have a level of dependency on those surrounding them. If the speed signal could be removed, allowing the speed of each component to be calculated intrinsically, this would decouple the subsystems from one another and improve the model flexibility and robustness. Other vehicle modelling packages are available, such as Advisor which uses a hybrid forward backward modelling approach. However, some users have reported limited flexibility and there can be difficulties with model reconfiguration [6]. Similarly, Mathworks uses its bespoke environment Simscape to develop its own vehicle modelling examples. However, the underlying functions of predefined components are difficult to change reducing modelling flexibility. Alternative approaches use a power flow regime, rather than torque flow, such as PSAT (Power Systems Analysis Toolbox) which is primarily used for electrical power systems [7]. This however also introduces a significant extent of coupling onto each of the subsystems in the model.

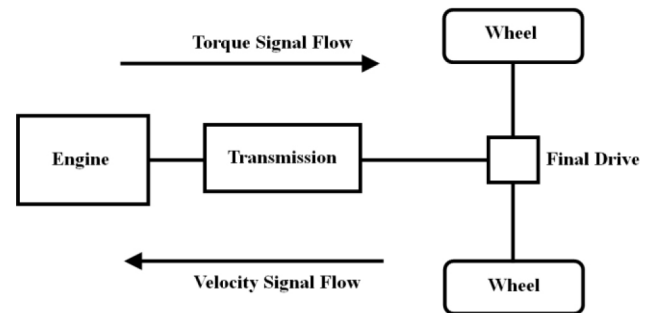


Figure 4. Rigidly connected rotational inertia system and dynamic signal flow paths as represented by modular sub models [5]

In the following sections the proposed new model architecture is established. Simulation results are then presented showing the functionality of the model, demonstrated on an engine-flywheel system, a conventional powertrain vehicle and finally a hybridized powertrain. Qualitative validation is also presented in the form of a comparison to real world driving data. This demonstrates the ability of the model to accurately capture both detailed drivetrain dynamics and fuel consumption profiles.

MODEL METHODOLOGY

The proposed VMA methodology will first be explained using simple examples for mathematical clarity. The model methodology has been approached in a dynamic, forward-facing manner to allow the implementation of existing control system models and to assist with the development of future control strategies. The intended purpose for the model developed within this current work is as a system level design tool. Thus, compliance within the system has been ignored for model simplification and all connections are assumed to be rigid, reducing the degrees of freedom of the system.

Newton's second law is used to calculate the velocity trajectory for the vehicle. The velocity of the vehicle at any instant can be found through an integration of the acceleration as shown in Equation 1:

$$v = \int \frac{\sum F_x}{m} dt$$

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where v is the velocity in ms^{-1} , F_x is the force in N, and m is the mass in kg. In this case, the mass includes the effect of the inertia of all the powertrain components. The sum of the forces includes the applied force at the wheels, which is a function of the torque that was produced at the engine, and all of the resistive forces on the vehicle. Due to the modular modelling approach adopted, each component subsystem has its own inertia/mass programmed within it. A parameter referred to as the reduced mass moment of inertia for the system was calculated at each point, with respect to that point. This means that not only can each component model operate independently but they can also function as an integrated system without any modifications to the subsystem structure or input/output (I/O) signals. Dynamic equation parameters change depending on what is upstream or downstream of the component in terms of torque flow in the model (upstream and downstream directions being defined by torque flow from the engine to the wheels). Speed is not inferred onto any component, but intrinsically calculated. The rotational speed of any rotating component is derived by means of an integration of the resultant torque applied to the component and the reduced mass moment of inertia for the system with respect to that component as shown in Equation 2:

$$\omega = \int \frac{\sum T}{I} dt \quad (2)$$

where T is torque in Nm, I is the rotational inertia in kgm^2 and ω is the rotational speed of the component in rads^{-1} . For a single rotating mass, there is only one inertia term in the dynamic equation to consider; the inertia of the component itself. However, if two masses (I_1 and I_2) are rotating together at identical speed, then the dynamic equation which describes the motion of the system can be written as shown in Equation 3:

$$T_{app} - T_{res} = I_1 \dot{\omega} + I_2 \dot{\omega} \quad (3)$$

Where T_{app} is the applied torque on the component in Nm, T_{res} is the resistive torque on the component in Nm. Equation 3 gives a simple explanation of the term reduced mass moment of inertia if the two inertia terms are grouped, as in Equation 4:

$$T_{app} - T_{res} = (I_1 + I_2) \dot{\omega} \quad (4)$$

The term $(I_1 + I_2)$ represents the reduced mass moment of inertia for the system. For systems which contain gearing ratios between rotating components, the inertia of upstream components must be multiplied by the square of the gearing ratio when being passed downstream, and vice versa, the inertia of downstream components must be divided by the square of the gear ratio when being passed upstream. An example of a simple gear train system is shown for clarification in Figure 5.

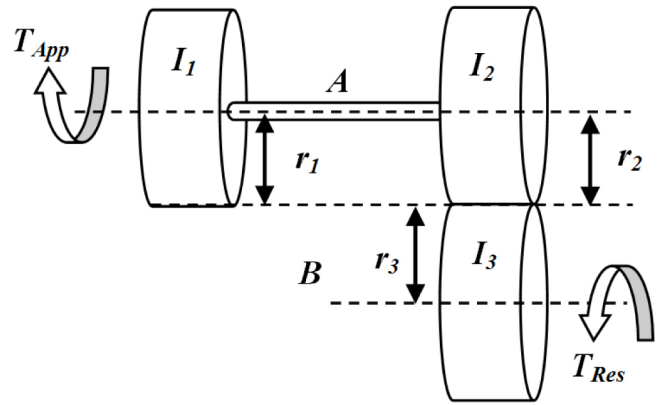


Figure 5. Simple gear train diagram showing rotating masses, rigid connections, critical dimensions and applied and resistive torques

Using this methodology, the system shown in Figure 5 can be described using two equations. One is formulated with respect to the rotating shaft A (Equation 5), and the other is written with respect to rotating shaft B (Equation 6). Note that the torque terms are also transformed when passing through gearing ratios accordingly as shown in equations 5 and 6:

$$T_{app} - \frac{T_{res}}{i} = (I_1 + I_2 + \frac{I_3}{i^2}) \dot{\omega}_A \quad (5)$$

$$T_{app} i - T_{res} = ((I_1 + I_2) i^2 + I_3) \dot{\omega}_B \quad (6)$$

Where i is the gearing ratio between the two shafts, calculated as $\frac{r_3}{r_2}$. This method allows for independent determination of the rotational speeds of both shafts via integrations of the angular accelerations. The next step is to determine how to account for the vehicle mass in the rotational inertia terms (and vice versa). Referring to the previous example, assume that the gear train shown in Figure 5 is the powertrain of a vehicle which must accelerate the vehicle mass (Inertia I_3 is effectively acting as a wheel). It can be shown that the vehicle mass is accounted for in the powertrain by multiplying by the square of the wheel radius when passing the mass upstream. Subsequently, this is combined with any other algebraic gearing ratio transformations as previously described for the system which results in a set of equations as follows (Equations 7, 8 and 9). Also the powertrain inertia must be divided by the square of the wheel radius when being passed to the vehicle model.

$$T_{app} - \frac{F_{res} r_3}{i} = \left(\left(\frac{m r_3^2 + I_3}{i^2} \right) + I_1 + I_2 \right) \dot{\omega}_A \quad (7)$$

$$T_{app} i - F_{res} r_3 = (I_3 + m r_3^2 + (I_1 + I_2) i^2) \dot{\omega}_B \quad (8)$$

$$\frac{T_{app} i}{r_3} - F_{res} = \left(\left(\frac{((I_1 + I_2) i^2 + I_3)}{r_3^2} \right) + m \right) \dot{\omega}_A \quad (9)$$