

Using a New Driveline Model to Define Research Engine Operating Conditions

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G. P. McTaggart-Cowan and E.J. Pallett

Wolfson School of Mechanical and Manufacturing Engineering, Loughborough University, Loughborough UK

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ABSTRACT

Steady state engine dynamometer testing provides the highest level of detail for understanding fundamental engine combustion. It can provide insight into pollutant formation mechanisms and methods for minimizing fuel consumption. However, steady-state dynamometer tests are normally carried out at test conditions far removed from the actual conditions that a vehicle engine encounters. This paper describes the application of a simple powertrain model to define steady-state engine test conditions that are more representative of real-world engine operation.

The model uses a backward-facing, modular structure. The model is validated against two powertrain configurations: a conventional powertrain equipped with a continuously variable transmission (CVT) and a parallel hybrid powertrain. Powertrain parameters and performance data for validation for both cases are supplied from the literature. The model is shown to agree well with both sets of published experimental results. These two models, along with a conventional powertrain equipped with a manual gearbox, are then used to evaluate steady-state and transient engine operation for vehicles following specified drive cycles. Steady-state test conditions are identified for all three models on the basis of both time and fuel consumption. The model results demonstrate that engine transients are typically on the order of 2-4 second duration and involve relatively modest changes in engine speed and torque.

INTRODUCTION

Dynamometer-based engine testing is central to understanding fundamental engine combustion processes and developing advanced engines that minimize fuel consumption and emissions. However, it is limited by the fact that the operating conditions encountered on an engine-dynamometer

test-bed differ significantly from the conditions encountered in a vehicle. This is especially true for steady-state engine testing, which while providing the deepest level of understanding of the engine and combustion processes, differs significantly from in-use engine conditions. Powertrain and vehicle architecture and the drive-cycle both significantly influence the operating conditions of an in-service engine. This paper describes the development of a simplified modular vehicle powertrain model and its use in defining steady-state engine test conditions for a variety of vehicle architectures and over different drive-cycles.

Powertrain models are widely used in vehicle research. Many numerical simulation models have been developed and proposed in the literature to assist the development of powertrains with a particular set of performance, emissions and economy targets [1]. The models range from simple object orientated simulation codes that use simplified models of the relevant physical processes in each of the main components in a conventional powertrain, through to complex fluid dynamic models which can, in principle, offer a detailed prediction of engine performance and emissions. These models differ substantially in terms of computational requirements and required input data.

Previous powertrain models have been developed to run transient simulations; these include models based on MATLAB/Simulink [2,3]; such models are often focused on making use of MATLAB's advanced controls capabilities. Common to these models is the use of a modular structure, which allows easy variation of powertrain geometries. AMESim (Advanced modelling environment for performing simulation of Engineering Systems) has also been used to model vehicle powertrains using a similar modular structure [4,5]. Parametric models have also been developed for specific powertrain geometries [6]. Dedicated commercially available codes also provide powertrain simulations,

including GT-Drive [1] and AVL ADVISOR. The latter is a commercialisation of the ADVISOR model developed at the National Renewable Energy Laboratory (NREL) [7]; this has been used extensively to evaluate various powertrain architectures for both light-duty [8,9] and heavy-duty vehicles [10]. These previously-developed models all have benefits and drawbacks. However, no model was identified that met the key criteria for the current project:

1. Accessible at low cost
2. Easily adaptable to the amount of information available for any given component
3. Flexible, to model various vehicle powertrains with minimum modification
4. Focused on defining engine test conditions rather than evaluating performance.

As a result, it was decided to develop a new modelling tool to meet these requirements. The first half of this paper provides an introduction to the modelling structure used and the key modules developed. The second half of the paper describes the application of the model to define engine test conditions for various powertrain architectures and vehicle drive cycles.

DRIVE CYCLE

The drive cycle used to evaluate a vehicle has a significant impact on its fuel consumption and emissions. This work will focus on two common drive-cycles: the first 505 seconds of the EPA urban dynamometer driving schedule (UDDS) and a single repetition of both parts 1 and 2 of the new European drive cycle (NEDC), with 195 seconds of the elementary urban cycle and the 400 seconds of the extra-urban cycle (referred to as NEDC(reduced)). Both drive cycle profiles are shown in [Figure 1](#). More information on these drive cycles, and the speed/time traces for both, can be found on the EPA website [11]. The first 505 s of the UDDS drive-cycle was used as it is the test-cycle from previous work that is used for one of the validation cases here. The second validation case used the full UDDS, which involves the speed-time trace shown in [Figure 1](#) followed by a further 900 s of lower-speed operation; however, the general profile of this second phase is less demanding for the engine than the first phase. The NEDC(reduced) was selected for comparison as it features significantly different conditions, with slower and steadier acceleration events but higher peak speeds. It should be noted that whether either of these cycles represent the driving profile of an in-use vehicle is questionable.

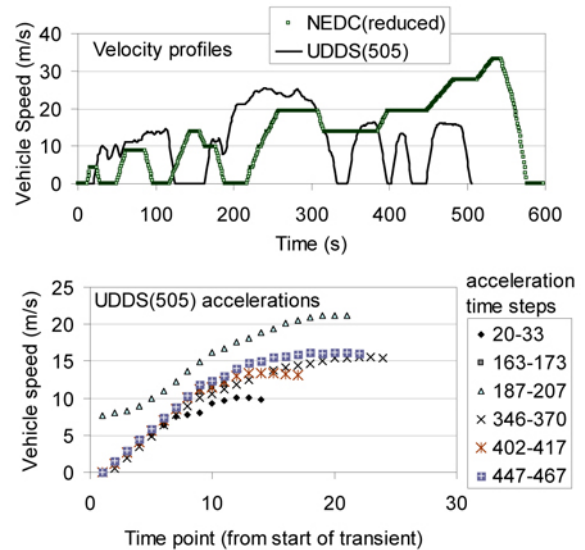


Figure 1. Driving cycles used in this work. Top - velocity profile for the NEDC(reduced) and first 505 seconds of the UDDS. Bottom - principal acceleration events in the first 505 seconds of the UDDS.

The velocity profiles in the various drive-cycles are based on a series of transient events. In developing engine test conditions representative of these events, it is important to evaluate the differences between the transients. The six principal acceleration transients (velocity changes > 10 m/s) in the UDDS(505) are also shown in [Figure 1](#). The general slope of the transients are all the same; the main differences are the maximum speed reached and (for the 187 - 207 s acceleration) the starting velocity. From an engine test point of view, this suggests that conditions derived for one of these transients should be generally representative of the others within the same test cycle. The NEDC (not shown) has similarly consistent acceleration profiles; however, there are significant differences in terms of acceleration rate and duration between the two drive cycles.

PART I - MODEL OVERVIEW AND VALIDATION

The model developed for this project uses a modular structure. This facilitates the creation of a model with a large scope, allowing for simple representations of individual components to be used, or more complex processes to be modelled if required. This also allows complex components to be split up into distinct modules if more detail is required. The model has been developed in Microsoft Visual Basic, with an MS ExcelTM interface.

The model uses a backward facing modelling approach. The process begins with a required drive cycle (vehicle speed vs. time, as discussed above) for the modelled vehicle to follow. Environmental effects (wind speed and orientation, grade)

and vehicle parameters (mass, aerodynamic drag, rolling resistance) are used to calculate the total retarding force. This value, combined with the acceleration required to follow the drive cycle, is then used to calculate the required wheel torque. The angular velocity of the wheels is also calculated based on the drive cycle and wheel diameter. The calculated wheel torque and angular velocity are then passed to the first powertrain component.

The overall structure of the model is based on a series of modules, each representative of a single powertrain component. The modules are developed such that each is independent but, they all have common inputs and outputs. As such, any desired architecture can be developed by simply adjusting the order of the modules. A new component can be developed by creating a new module and adding it to the library of existing modules, while an existing module can be modified without requiring any changes to any other component.

The values passed between modules are torque and angular velocity. These values are calculated for each module in series and the required 'input' values are passed to the next module. That is, for each physical powertrain component, the required torque and speed inputs are calculated based on the requirements of the previous component (component closer to the wheels), combined with the performance of the specific component of interest.

In a conventional powertrain the modules simulate the final drive, transmission, clutch and engine. Torques and speeds are calculated for each of these components; an outline of the structure is shown in [Figure 2](#). The final output from the model in this case is engine torque and speed. Engine fuel consumption maps allow the engine torque and speed to be converted into fuel consumption values.

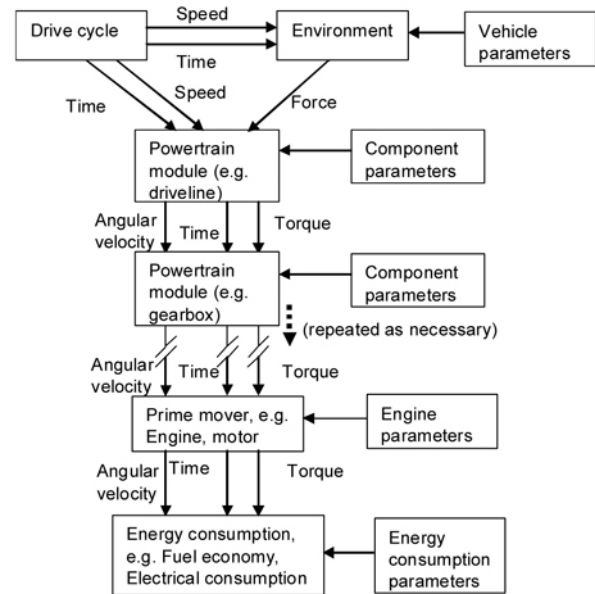


Figure 2. Overall powertrain model structure.

The torque and speed for each component of the entire powertrain is calculated for each time step in the drive cycle. Once the performance at a specific time point has been calculated, the model advances to the next time step and repeats the process. Error-checking is used in individual modules and for the overall model to ensure that no components are exceeding any stated performance limits. In cases where performance limits are exceeded (e.g., the engine torque exceeds full load), the time-step is repeated with the performance of all other modules limited by the identified factor. This may result in the model not being able to follow a prescribed drive-cycle. In such cases, each time-step is treated individually, such that the vehicle model follows the drive-cycle as closely as possible. Such a divergence from the desired drive-cycle is a limitation of the powertrain being modelled, rather than a feature of the model itself.

Individual Modules

The core of the powertrain model is the series of individual modules that represent the various physical components, as demonstrated in [Figure 2](#). The first two modules are the drive cycle, which defines the vehicle speed as a function of time, and the environment. These two modules combine to determine the force that must be exerted at the wheels to follow the prescribed drive-cycle. These modules are always the first two called, independent of what powertrain structure is being investigated. The first module in the sequence must be defined such that it converts the vehicle force input into an angular velocity and torque.

Each of the remaining modules is defined with common input and output parameters: time, angular velocity and torque. As the model is backwards-facing, the 'input' to each module is

from the preceding module (e.g. the input to the gearbox would be the torque and speed required at the driveshaft, while the output would be the torque and speed required to be transferred to the gearbox to provide that output). The complexity of the individual modules is unlimited, as long as the input and output parameters are consistent. For this work, the individual modules were constructed so that torque transfer efficiency and moments of inertia were accounted for. Angular velocity (ω) was calculated using:

$$\omega_{in} = \omega_{out} * GR, \quad (1)$$

where GR is gear ratio across the module and in and out refer to the speed physically input to the component (i.e., the unknown output from the module) and the speed out of the component (i.e., the speed calculated from the preceding module in the model), respectively. Similarly, the torque (T) at each component was calculated from:

$$T_{in} = T_{out}/(GR*\eta) + I_{comp}*\alpha, \quad (2)$$

where η is the efficiency of the component, I_{comp} is its moment of inertia, α the angular acceleration and in and out have the same definitions as described for [Eq. 1](#). This equation applies for positive torque transfer (i.e., torque transfer from the physical input to the physical output, as identified by the arrows in [Figure 2](#)). In the case of negative torque, such as encountered during regenerative braking or when driving a generator, [equation \(2\)](#) was replaced with:

$$T_{in} = (T_{out}/GR)*\eta - I_{comp}*\alpha, \quad (3)$$

where the various terms have the same meanings as in [Eq. 2](#). In and out still refer to the unknown output from and known input to the module, respectively (i.e., they are now opposite to the direction of torque transfer).

Most of the powertrain modules follow this same basic format. However, added complexity occurs in cases where component parameters are a function of physical input (i.e., unknown) conditions, such as in the case of the continuously variable transmission (CVT) module used in Validation Case 1. In this case, the efficiency depends on the torque and speed input from the engine. To overcome this, an iterative process is used: an efficiency is assumed and the torque input to the CVT is then calculated using [Eq. 2](#). From this calculated value, the efficiency is re-estimated using a look-up table based on input conditions and the input torque is recalculated. This process repeats until the results converge.

The model presented in [Figure 2](#) relates only to a conventional series-organized drive-train. Parallel drive-trains (such as the hybrid vehicle discussed in Validation

Case 2) involve torque being divided between two separate ‘arms’, which may be operating at different rotational speeds. This split can be managed by developing a new module that determines how much torque should be being delivered to or from each arm of the driveline. A specific example of the torque transfer between two drivetrain arms is presented in more detail in Validation Case 2. Beyond this module, the two arms are treated independently.

For the model results reported here, the final output is fuel consumption. This was determined from an engine torque-speed fuel consumption map. Once the engine speed and torque were calculated for each time-step, a two-way linear interpolation was used to determine fuel consumption at those conditions from the externally-prepared lookup table. The fuel consumptions at each time point were then summed to give the drive-cycle fuel consumption. Similar methods could be used to evaluate other parameters, such as criteria pollutant emissions, if sufficient data was available.

VALIDATION

The model structure was validated using two published powertrain architectures: (1) a conventional gasoline-engine vehicle equipped with a CVT transmission [8], and (2) a parallel hybrid vehicle equipped with a manual transmission [9]. These two architectures were selected as they provided enough test data and powertrain parameter information to validate the model. They also provide further insight into the model and the design and performance of individual modules. The powertrain parameters used are given in the [appendix](#).

Validation Case 1 - CVT

The first validation case involves a 2001 Toyota Opa equipped with a direct injection gasoline engine and a CVT. Experimental results included fuel flow rate and engine torque and speed; powertrain component efficiencies and an engine fuel consumption map were also provided [8]. The model structure for this case is as shown in [Figure 2](#), with the following powertrain components: driveline; CVT; clutch; engine. The results in [8] were acquired over the ‘hot-505’ cycle, which is equivalent to the UDDS(505) discussed above.

One feature of this vehicle is the CVT. It operates to maximize the time the engine is at its peak efficiency point (low speed, moderate load). The module developed to represent the CVT used an algorithm where the CVT held the engine speed at a low level, resulting in increased torque at higher power, until the line of best efficiency was reached. As power demand increased further, the gear ratio was adjusted so that both engine speed and torque followed the line of best engine efficiency. The efficiency of the CVT itself depended on engine power output; as discussed previously, an iterative process was used to determine CVT efficiency for each time-step. The CVT gear ratio over the drive cycle is shown in

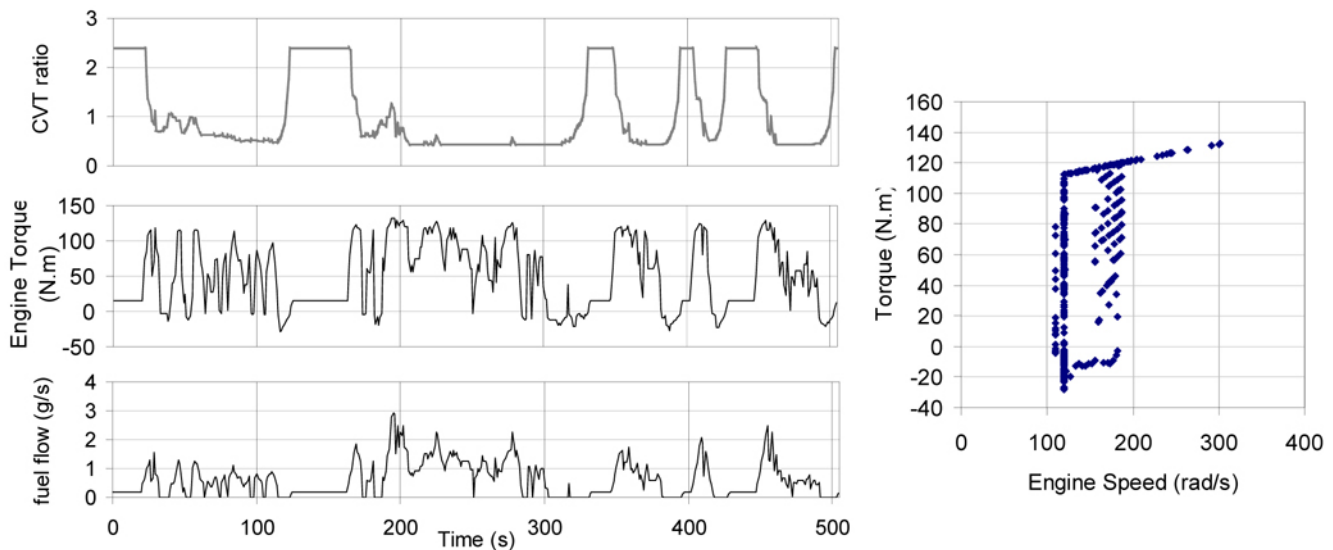


Figure 3. Model output for CVT equipped vehicle validation case.

Figure 3, along with engine torque and fuel flow rate at each time step. The speed-torque operating points for the engine are also plotted in Figure 3. These results indicate that, as desired, the CVT control is maintaining the engine at its lower speed or peak-efficiency points. Qualitatively, the model results shown in Figure 3 agree well with those from the reference case [8].

<figure 3 here>

The total fuel consumption over the drive cycle can be used to provide a quantitative validation of the model, as shown in Table 1. The results indicate that the model over-predicted fuel consumption by 3.6%; no tuning parameters were used to achieve these results. The difference shown in Table 1 is considered acceptable given the uncertainty in having derived the engine fuel consumption and CVT efficiency results from graphical sources in [8]. Other sources of uncertainty include the CVT control algorithm; this is likely more complex than was used in the model. The good agreement shown in Table 1 indicates that such a simplified model can give a reasonable prediction of vehicle performance.

Table 1. CVT model result validation

	Test	Model
Fuel consumed (g)	303.7	314.8
Mileage (MPG)	32.99	31.82
Difference:		3.6%

Validation Case 2 - Hybrid

The second validation case poses a greater challenge for the model structure as it involved a parallel-hybrid powertrain. The validation data, generated by NREL, was for a 2000

Honda Insight parallel-hybrid vehicle with a 1.0 L 50 kW gasoline engine [9]. Supplemental power was provided by a 10 kW electric motor which also acted as a generator for regenerative braking; energy storage was in a 144V, 6.5 Ah NiMh battery pack. The net power from the motor and engine was connected to the powertrain through a torque coupler and a five speed manual gearbox. Validation data was extracted using NREL's on-board data acquisition tool [9] when the vehicle was operated over the UDDS drive cycle. Vehicle and component parameters are given in the appendix.

This hybrid system is more complex to model, as it involves parallel energy transfer to and from the electric motor/generator and the engine. A new module, the torque coupler, was developed to split the power transfers between the two modules; however, the exact operation of the vehicle torque coupler is uncertain [9]. As a result, the module assumes a control strategies based on four general operating conditions, as defined in [9]:

1. Engine start/stop - if the battery state of charge (SOC) is sufficiently high, the engine turns off when the vehicle is stationary.
2. Acceleration - the electric motor provides supplemental torque ('motor assist'). The motor assist depends on the battery SOC; the higher the SOC, the more supplemental torque from the motor (up to a maximum of 50 N.m). The electric motor never operates independently from the engine.
3. Cruising (small variations in speed) - engine torque is low, so the motor acts as a generator, increasing the engine torque into a more efficient range and storing the extra energy in the battery. The torque absorption depends on SOC. Charging stops at 70% SOC, to allow for energy storage from regenerative braking.